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# Condensation of R-134a inside the Vertical Smooth and Dimpled Helically Coiled Tubes

**Anand Kumar Solanki[1], Ravi Kumar[2]**
[1]Department of Mechanical Engineering
Gayatri Vidya Parishad College of Engineering
Visakhapatnam (A), 530048, India
anandsolanki@gvpce.ac.in
[2]Department of Mechanical and Industrial Engineering
Indian Institute of Technology Roorkee
Roorkee, 247667, Uttarakhand, India
ravi@me.iitr.ac.in

***Abstract*** – An experimental study has been carried out to study the frictional pressure drops during condensation of R-134a vapours while flowing inside the vertically positioned smooth and dimpled helically coiled tubes. The tests were conducted at the mass flux of 115 to 190 $kg.m^{-2}.s^{-1}$. The saturation temperature of the R-134a condensing vapour remained 35 ºC. The effect of the mass flux and vapour quality on frictional pressure drops was investigated. In addition, a comparison has also been made for the frictional pressure drop between the vertical smooth helically coiled tube and the dimpled helically coiled tube. The experimental results showed that for the same mass flux, the dimpled helical coiled tube has a higher frictional pressure drop compared to the smooth helically coiled tube. Moreover, a comparison of the experimental frictional pressure drop data for vertical smooth helically coiled tubes with that of renowned correlations has also been made. The correlations predicted the experimental data in an error band of ± 30%.

***Keywords*****:** Condensation, frictional pressure drop, R-134a, vertical dimpled helically coiled tube, mass flux.

## 1. Introduction

Condensation of refrigerants in the helically coiled tube is widely used in industries such as air conditioning, refrigeration, nuclear and chemical industries. In order to achieve the effective design of the condenser, the frictional pressure drop inside the tube should be low because it affects the pumping power. Some research work had been performed by Kang [1], Yu et al. [2], Han et al. [3], Wongwises and Polsongkram [4] to determine the pressure drop characteristics inside helically coiled tubes. Gupta et al. [5] reported the effect of mass flux, vapour quality and saturation temperature on the heat transfer coefficients and pressure drop of R-134a inside a horizontal helical coiled tube-in-shell type heat exchanger. Lin and Ebadian [6] reported the effect of inclination angle ($0^o$, $45^o$ and $90^o$) on condensation heat transfers and pressure drops of R-134a flowing inside the annular section of the tube. Mozafari et al. [7] conducted the experimental studies of heat transfer and pressure drop of R-600a inside helical tube-in-tube heat exchanger at different inclination angles of zero, $+30^o$, $+60^o$ and $+90^o$ with different mass flux ranging from 155 to 265.5 $kg.m^{-2}.s^{-1}$ at saturation temperature 38.5-47ºC. The results showed that the pressure drop was slightly affected by the inclination angle. Solanki et al. [8,9] carried out the experimental frictional pressure drop data during the condensation of R-134a inside the horizontal position of helically coiled tubes. As per the literature, the pressure drop of refrigerant depends on the orientation of helically coiled tubes. In the previous work, the pressure drop of refrigerant was found in the horizontal position for the dimpled helically coiled tube. In this work, the vertical position of a dimpled helically coiled tube is taken. In this study, the effects of mass flux and vapour quality on the frictional pressure drops were determined inside the vertically positioned smooth and dimpled helically coiled tubes at the same operating condition. Moreover, the comparison of the frictional pressure drops for the smooth and dimpled helical coiled tube at the same operating condition is also presented.

## 2. Experimental Apparatus and Procedure

The simplified schematic diagram of the experimental set-up is depicted in Fig 1. The main components of the experimental set-up were composed of a pre-heater, test-condenser, post-condenser, three refrigerant magnetic gear pumps, a Coriolis mass flow meter and a data acquisition system. The test-section was connected to the post condenser, where two-phase quality of refrigerant coming from the test-section was sub-cooled to a liquid state by extracting heat to cold water circulated from the condensing unit. The municipal water was employed for cooling purposes, which was circulated inside the shell of the test-section. A slight glass after the post-condenser was placed to ensure the continuous liquidity of refrigerant before entering the three magnetic gear pumps connected side by side. The flow rate of refrigerant was monitored by operating the speed of the magnetic gear pump through the frequency inverter. The Coriolis mass flow meter, followed by a magnetic gear pump, was accommodated to measure the refrigerant flow rate. In order to control the vapour quality at the inlet of the test condenser, an appropriate pre-heater was designed from a 6 m long U-bend stainless steel tube. By supplying high current and low voltage through a step-down autotransformer, liquid refrigerant was evaporated by heating the tube. A standard filter-drier was incorporated following the pre-heater to remove any foreign particles and moisture in the refrigerant loop. A number of hand shut-off valves were facilitated to provide facile installation of a component in the apparatus.

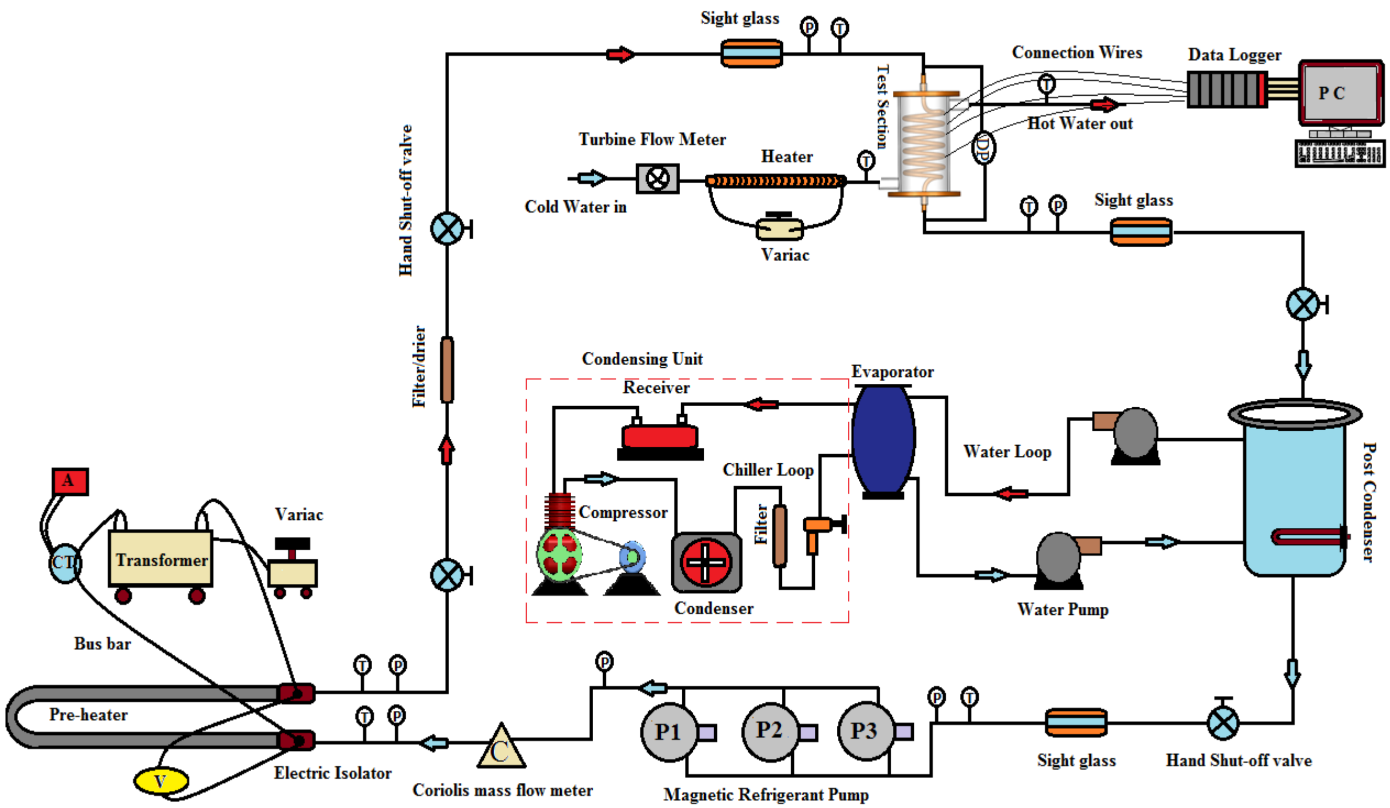


Fig.1. Schematic diagram of the experimental set-up

The test section was a counter flow vertical dimpled helically coiled tube in a shell type of heat exchanger in which the refrigerant flowing inside the inner tube extracts the heat to the cooling water flowing the opposite direction to refrigerant flow inside the shell. There were two types of tubes taken as inner tube, one smooth helical coiled tube and another one dimple helically coiled tube. The inner helically coiled tube was made by wrapping the straight tube over the wooden pattern. As a next step, the helically coiled tube was processed to make the projections on its surface by using a round tip tool, and thus, a dimple helically coiled tube was obtained. The shape of dimples on the inner surface of the helically coiled tube was spherical. The Schematic diagram of the inner tubes is shown in Fig 2. Details of the inner tubes are given in Table 1. The T-type thermocouples were located to measure the refrigerant temperature at the inlet of the pre-heater, inlet and outlet of the test-section. The outer wall temperature of the helically coiled tube was also measured with T-type thermocouples at six positions. In each position, four thermocouples were placed 90$^{o}$ apart over the top, bottom and two sides. The thermocouples were calibrated in the operating temperature range of -10 °C to 70 °C with an accuracy of ±0.1°C of full scale. The pressure transducers with an accuracy 0.25% were utilised to measure the refrigerant pressure at the inlet and outlet of the test-section and at the entrance of the pre-heater. Also, the pressure drop across the test-section was measured by a differential pressure drop transducer. The Coriolis flow meter, with an accuracy of 0.1%, was employed to measure the refrigerant mass flow rate. The turbine flow meter has an accuracy of 1% of the full scale. All thermocouple and pressure transducer signals were transferred to the multichannel data acquisition system with PXI controller. The steady-state condition was assumed when the reading of temperatures, pressure and mass flow rate remain constant for at least 20 min. The details of the experimental condition are shown in Table 2. The methodology suggested by Klein and McClintock [10] was applied to find the uncertainties in the experimental results, as shown in Table 3.

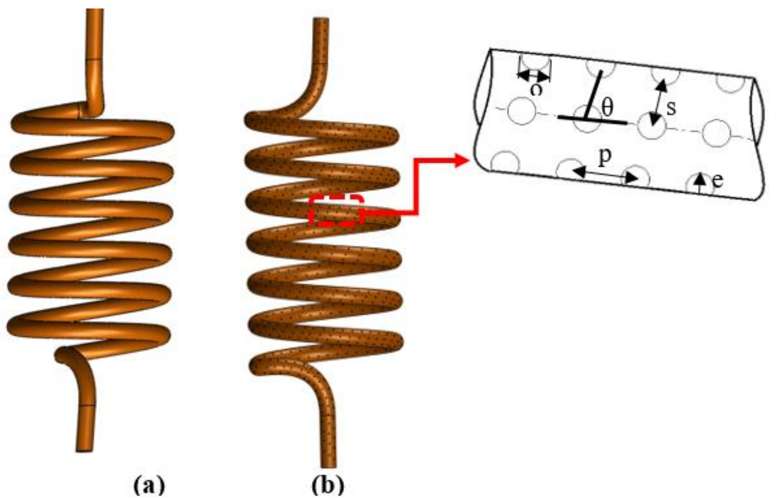


Fig. 2. (a) Vertical smooth helically coiled tube and (b) Vertical dimpled helically coiled tube

Table 1: Dimension of inner tubes

| *Parameter* | *Smooth helically coiled tube* | *Dimpled helically coiled tube* |
|---|---|---|
| Inner diameter of tube, mm | 8.92 | 8.92 |
| Outer diameter of tube, mm | 9.52 | 9.52 |
| length of tube, mm | 2200 | 2200 |
| Inside tube area, $cm^2$ | 575.4 | 610.4 |
| Coil mean diameter, mm | 110 | 110 |
| Coil pitch, mm | 25 | 25 |
| Number of turns | 6 | 6 |
| Dimple pitch (s), mm | - | 8.5 |
| Helical pitch (p), mm | - | 10 |
| Dimpled depth (e), mm | - | 1 |
| Helix angle (θ), degree | - | 49 |
| Dimpled diameter (o), mm | - | 2 |

Table 2: Experimental conditions

| *Parameter* | *Range* |
|---|---|
| Refrigerant | R-134a |
| Average saturation temperature (ºC) | 35 |
| Mass flux (kg $m^{-2}s^{-1}$) | 115,156,191 |
| Inner tube material | copper |

Table 3: Uncertainties Analysis

| **Measurement** | **Uncertainty** |
|---|---|
| Temperature (ºC) | 0.1 |
| Refrigerant mass flow rate (kg $s^{-1}$) | $0.050\times10^{-3}$ |
| Water mass flow rate (kg $s^{-1}$) | $1.41 \times 10^{-3}$ |
| Pressure (kPa) | 5.201 |
| Pressure drop (kPa) | 0.125 |
| Average vapor quality | 4.5% |
| Heat transfer rate at pre-heater | 7.2% |
| Heat transfer rate at test-section | 12% |

## 3. Data Reduction

The total pressure gradient $(dP/dz)_T$ is directly measured by a differential pressure transducer at the inlet and outlet of the test section. The frictional pressure gradient $(dP/dz)_{tp,f}$ is calculated by subtracting the acceleration pressure gradient$(dP/dz)_{tp,a}$ and the gravitational pressure gradient $(dP/dz)_{tp,g}$ from the total pressure drop $\left(\frac{dP}{dz}\right)_T$, as

$$\left(\frac{dP}{dz}\right)_{tp,f} = \left(\frac{dP}{dz}\right)_T - \left(\frac{dP}{dz}\right)_{tp,a} - \left(\frac{dP}{dz}\right)_{tp,g} \quad (1)$$

The acceleration pressure gradient is defined by the following equation:

$$\left(\frac{dP}{dz}\right)_{tp,a} = G^2 \frac{d}{dl}\left\{\left(\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_l(1-\alpha)}\right)\right\} \quad (2)$$

Where, G is the mass flux, $\rho_g$ is the vapor density, $\rho_l$ is the liquid density of the refrigerant, and α is the void fraction which is calculated from Abdul-Razzak et al. correlation [11]:

$$\alpha = (1 + 0.49\, \chi_{tt}^{0.3036})^{-1} \quad (3)$$

Where, the Martinelli parameter $\chi_{tt}$ is expressed as the following equation:

$$\chi_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \quad (4)$$

The two-phase frictional pressure gradient multiplier, $\phi_l^2$ is expressed as the following equation:

$$\phi_l^2 = \frac{\left({}^{dP}/{}_{dz}\right)_{tp,f}}{\left({}^{dP}/{}_{dz}\right)_{l,f}} \quad (5)$$

## 4. Result and Discussion

The frictional pressure drop was determined inside the vertical smooth and dimpled helically coiled tubes at the same operating condition. Fig. 3 shows the effect of mass flux and vapor quality on the frictional pressure drops for both tubes. It is found that the frictional pressure drop is increased with the rise of mass flux and vapor quality of refrigerant. In the case of a vertical smooth helically coiled tube, the frictional pressure drops at the mass flux of 115 kg.$m^{-2}$.$s^{-1}$are 0.136–0.848 kPa.$m^{-1}$, corresponding to vapor quality 0.1 and 0.8, respectively. As the mass flux increases from 115 to 190 kg.$m^{-2}$.$s^{-1}$, the frictional pressure drops also increased to 0.283–1.918 kPa.$m^{-1}$ for the vapour quality 0.1 and 0.7, respectively. For the dimpled helically coiled tube, when the mass flux enhances from the 156 to 190 kg.$m^{-2}$.$s^{-1}$, the frictional pressure drops increase from 0.408 $kPa.m^{-1}$ to 0.599 $kPa.m^{-1}$ for vapor quality 0.1 and, from 3.639 $kPa.m^{-1}$ to 4.367 $kPa.m^{-1}$ for the vapor quality 0.6, respectively. This can be explained as, due to higher vapour qualities, the velocity of vapor phase enhanced, and the intensity of secondary flow inside the helically coiled tube also increased. As a result, higher shear stress exists between the vapour-liquid interfaces. Therefore, frictional pressure drop increases as the vapour quality of refrigerant increases. Moreover, the frictional pressure drop rises with the increase in the mass flow rate of refrigerant. This is because the amount of secondary flow also increases as the flow velocity of refrigerant enhances. Therefore, more entrainment of droplets is established as well as higher interfacial shear stress between vapour-liquid mixtures is also raised. As a result, the frictional pressure drop increases with the rise of the mass flux of refrigerant. Also, Fig. 3 presents the comparison of the frictional pressure drops for the smooth and dimpled helical coiled tube at the same operating condition.

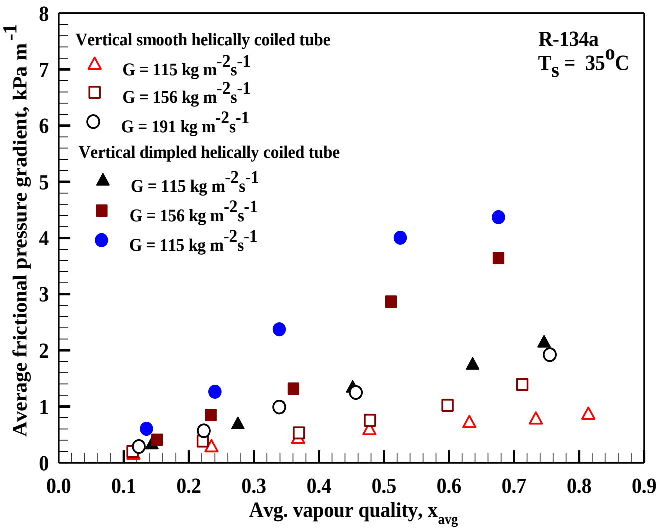


Fig. 3. Comparison of the frictional pressure drop for tubes

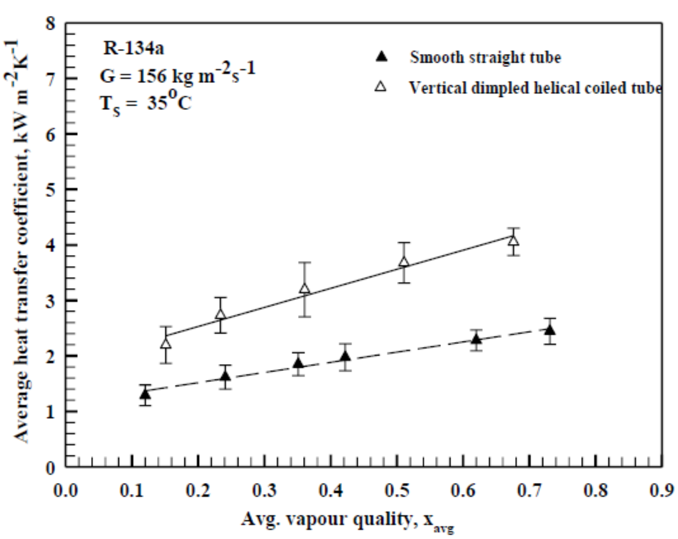


Fig. 4. Comparison of the heat transfer coefficient [12]

It can be seen that, at the same saturation temperature, a dimpled helical coiled tube produces a higher frictional pressure drop compared to the smooth helically coiled tube in the percentage of 130-180% for 115 $kg.m^{-2}.s^{-1}$and 106-205% for 156 kg.m$^{-2}$.s$^{-1}$, respectively. This occurs because the protrusions provided inside the dimpled helically coiled tube which increase the intensity of turbulence on the vapor-liquid mixture flow. Moreover, the secondary flow becomes stronger compared to that of a smooth, helically coiled tube. As a result, higher frictional pressure drops are achieved in the vertical dimpled helically coiled tube. Besides this, it has been reported that the dimpled helically coiled tube at the vertical position showed a higher heat transfer coefficient compared to a smooth straight tube at the same operating condition, as shown in Fig. 4 [12].

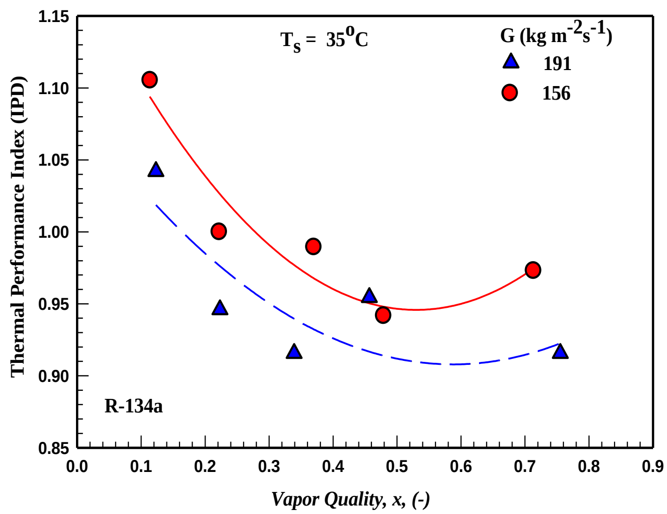

Fig 5: The effect vapor quality and mass flux on Thermal performance index

The thermal performance index is defined as the Equation (6)

$$\text{Thermal performace index (TPI)} = \left(\frac{Nu_{DHLT}}{Nu_{SHLT}}\right) \Big/ \left(\frac{\Delta P_{DHLT}}{\Delta P_{SHLT}}\right)^{1/3} \qquad (6)$$

The effects of vapor quality and mass flux on the thermal performance index are shown in Fig 5. It is found that the thermal performance index is decreased as the vapor quality and mass flux are increased.

The correlations used to predict the frictional pressure drop of R-134a inside the smooth helical coiled tube are suggested by Wongwise and Polosongkram [4] and Gupta et al. [5]**.** The following correlations for smooth helically coiled tubes are listed in Table 4.

Table 4: Wongwise and Polsongkram [4] and Gupta et al. [5] correlations for smooth helically coiled tube

| Author | Correlations |
|---|---|
| Wongwise and Polsongkram [4] | The two-phase frictional pressure gradient multiplier $(\phi_l^2)$ <br> $\left(\frac{\Delta P_{tp,f}}{\Delta P_{l,f}}\right) = (\phi_l^2) = \left(1 + \frac{5.569}{\chi_{tt}^{1.496}} + \frac{1}{\chi_{tt}^2}\right)$ |
| Gupta et al. [5] | The two-phase frictional pressure gradient multiplier $(\phi_l^2)$ <br> $\left(\frac{\Delta P_{tp,f}}{\Delta P_{l,f}}\right) = (\phi_l^2) = 2.76\left(1 + \frac{7.094}{\chi_{tt}^{1.378}} + \frac{1}{\chi_{tt}^2}\right) p_r^{0.7}$ |

Fig. 6 (a) and (b) present the comparison of the experimental frictional pressure drop for vertical smooth helically coiled tube with that of the renowned correlation given by the Wongwise and Polosongkram [4] and Gupta et al. [5]. It is evident that Wongwise and Polosongkram correlation [4] underestimated most of the experimental data points of R-134a.

However, all experimental data point falls within an error band of +10% to -30%. Similarly, the experimental results were predicted by the Gupta et al. correlation [5] with an error band of ±30%.

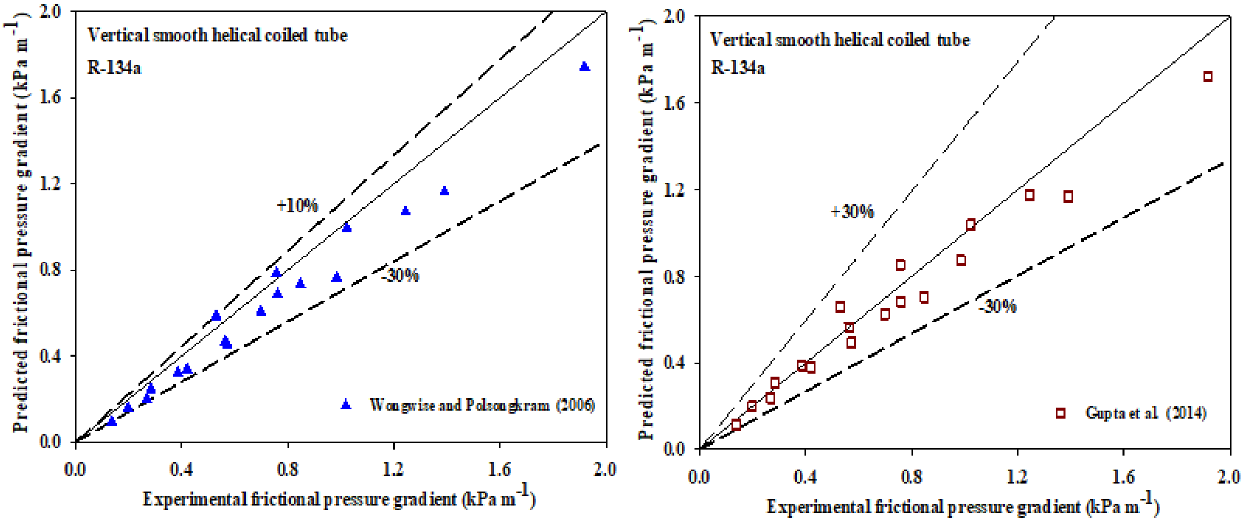


Fig. 6. Comparison of experimental frictional pressure drop for vertical smooth helical tube with existing correlation.

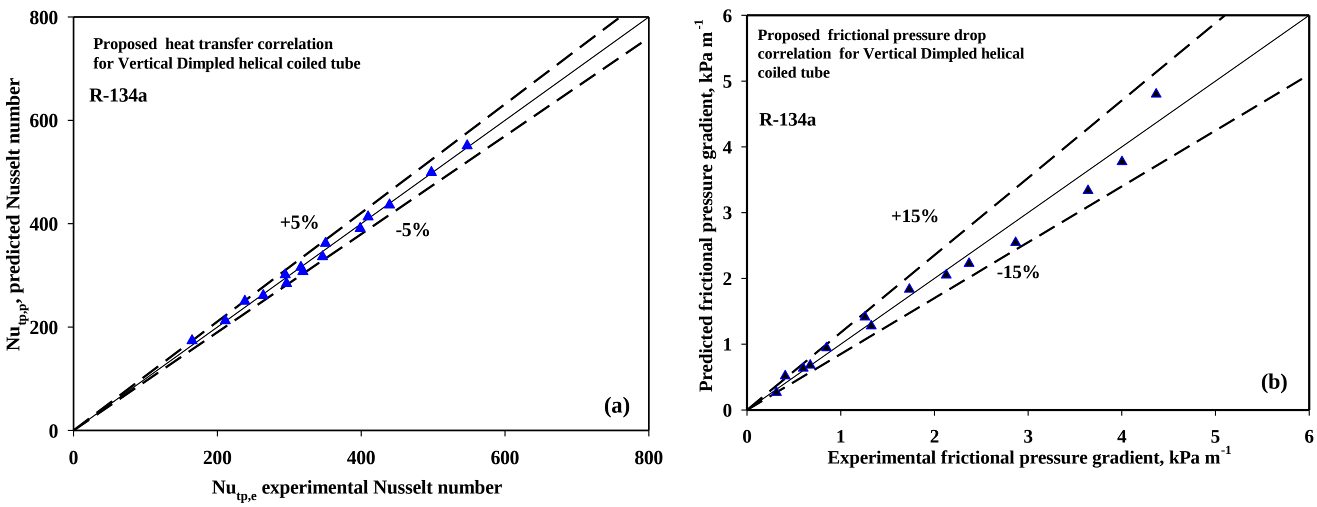


Fig 7: Experimental data compared with proposed correlation for (a) Nusselts Number (b) Frictional pressure drop

Fig. 7 shows the comparison of experimental results with newly developed correlation for dimpled helically coiled tube. The developed correlations are given below:

For Nusselt number:

$$Nu_{DHLT} = 0.02095\,(De_{Eq})^{1.1504}(\chi_{tt})^{0.1909} \tag{7}$$

For frictional pressure drop:

$$\frac{\Delta P_{tp,f}}{\Delta P_{l,f}} = \phi_l^2 = 0.1009\left(1 + \frac{162.05}{\chi_{tt}^{1.623}} + \frac{1}{\chi_{tt}^2}\right) \tag{8}$$

It can be seen from Fig. 7 that the experimental results were well predicted with a developed correlation for heat transfer and frictional pressure drop within the range of ±5% and ±15%, respectively.

## 5. Conclusion

1. The frictional pressure drop inside the vertical smooth helical coiled tube and the dimpled helical coiled tube is increased with the rise of mass flux and vapor quality of refrigerant.
2. At the same saturation temperature, the dimpled helical coiled tube produces a higher frictional pressure drop compared to the smooth helical coiled tube in the percentage of 130-180% for 115 kg $m^{-2}s^{-1}$ and 106-205% for 156 $kg.m^{-2}.s^{-1}$, respectively.
3. For vertical smooth helical coiled tubes, Wongwise and Polsongkram correlation [4] underestimated most of the experimental data points of R-134a. However, all experimental data point falls within an error band of +10% to -30%. Similarly, the experimental results were predicted by the Gupta et al. correlation [5] with an error band of ±30%.
4. The thermal performance index is improved with a decrease in mass flux and vapor quality.

## 6. References


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