A review on the two-phase pressure drop characteristics in helically coiled tubes

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Abstract

Due to their compact design, ease of manufacture and enhanced heat transfer and fluid mixing properties, helically coiled tubes are widely used in a variety of industries and applications. In fact, helical tubes are the most popular from the family of coiled tube heat exchangers. This review summarises and critically reviews the studies reported in the pertinent literature on the pressure drop characteristics of two-phase flow in helically coiled tubes. The main findings and correlations for the frictional two-phase pressure drops due to: steam-water flow boiling, R-134a evaporation and condensation, air-water two-phase flow and nanofluid flows are reviewed. Therefore, the purpose of this study is to provide researchers in academia and industry with a practical summary of the relevant correlations and supporting theory for the calculation of the two-phase pressure drop in helically coiled tubes. A significant scope for further research was also identified in the fields of: air-water bubbly flow and nanofluid two phase and three-phase flows in helically coiled tubes.

Keywords Two-phase flow; curved tubes; frictional pressure drop; flow boiling; nanofluids

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The Editor,
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8th February 2016

Dear Prof. Rose,

I would like to submit the attached manuscript entitled: *‘*A review on the two-phase pressure drop characteristics in helically coiled tubes*’* for publication in the International Journal of Heat and Mass Transfer. This research complements our earlier paper entitled: ‘A review on the two-phase heat transfer characteristics in helically coiled tube heat exchangers’. For the purposes of the current submission, I have selected the Elsevier ‘Your Paper Your Way’ submission process.

I thank you for taking the time to consider this request and I look forward to hearing from you in due course.

Yours sincerely,

Dr. Andrew M. Fsadni
PhD, MSc, MBA, BEng (Hons), EUR ING, FHEA, CEng, MIMechE
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Author Declaration

I wish to confirm that there are no known conflicts of interest associated with this publication and there has been no significant financial support for this work that could have influenced its outcome.

I confirm that Dr Justin P.M. Whitty and the undersigned are the sole authors of the manuscript and therefore, there are no other persons who satisfied the criteria for authorship but are not listed.

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Dr Andrew M. Fsadni

08th February 2016

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Highlights

- Detailed review on the two-phase pressure drop characteristics and correlations
- Impact of curvature on the flow boiling frictional pressure drop is not significant
- Nanofluids' impact on the pressure drop is significant
- There is a significant gap in literature in the field of frictional drag reduction

A review on the two-phase pressure drop characteristics in helically coiled tubes

Abstract

Due to their compact design, ease of manufacture and enhanced heat transfer and fluid mixing properties, helically coiled tubes are widely used in a variety of industries and applications. In fact, helical tubes are the most popular from the family of coiled tube heat exchangers. This review summarises and critically reviews the studies reported in the pertinent literature on the pressure drop characteristics of two-phase flow in helically coiled tubes. The main findings and correlations for the frictional two-phase pressure drops due to: steam-water flow boiling, R-134a evaporation and condensation, air-water two-phase flow and nanofluid flows are reviewed. Therefore, the purpose of this study is to provide researchers in academia and industry with a practical summary of the relevant correlations and supporting theory for the calculation of the two-phase pressure drop in helically coiled tubes. A significant scope for further research was also identified in the fields of: air-water bubbly flow and nanofluid two phase and three-phase flows in helically coiled tubes.

Keywords: Two-phase flow; curved tubes; frictional pressure drop; flow boiling; nanofluids

1. Introduction

Due to their compact design, ease of manufacture and high efficiency in heat and mass transfer, helically coiled tubes are widely used in a number of industries and processes such as in the food, nuclear, aerospace and power generation industries and in heat recovery, refrigeration, space heating and air-conditioning processes. Due to the formation of a secondary flow, which inherently enhances the mixing of the fluid, helically coiled tube heat exchangers are known to yield improved heat transfer characteristics when compared to straight tube heat exchangers. The secondary flow is perpendicular to the axial fluid direction and reduces the thickness of the thermal boundary layer. Goering et al. [1] estimated the secondary flow to account for circa 16-20% of the mean fluid flow velocity. This phenomenon finds its origins in the centrifugal force due to the curvature of the coil structure and is more evident with laminar flow due to the limited fluid mixing in straight tube laminar flow [2,3]. However, for single and two-phase flows, the secondary flow could also result in an undesirable increase in the frictional pressure drop over that of straight tubes. For air-water two-phase flow in helically coiled tubes, Akagawa et al. [4] reported frictional pressure drops in the range of 1.1 to 1.5 times greater than those in straight tubes, ceteris paribus. Therefore, the performance of helical coils is also a function of the geometry and design parameters such as the tube diameter and the pitch (Fig. 1) as well as the resultant pressure drop. Through their study on the investigation of the heat transfer characteristics with the addition of multi-walled carbon nanotubes nanoparticles to oil, Fakoor-Pakdaman et al. [5] reported their results in terms of the Performance Index (PI), given in Eq. (1). This captures the simultaneous effects of heat transfer and two-phase pressure drop with the use of nanofluids and helical tubes on the overall performance of the heat exchanger. When the performance index is greater than unity, the PI implies that the benefits gained through enhanced heat transfer coefficients outweigh the effects of larger pressure drops as a result of the nanoparticles and helical tubes.

$$\eta = \frac{\frac{h^*}{h_{st}}}{\frac{\Delta P^*}{\Delta P_{st}}} \quad \textbf{(1)}$$

where h* is the mean heat transfer coefficient after the application of enhancement techniques, h_{st} is the mean heat transfer coefficient in a straight tube with the base fluid only, ΔP^* is the mean pressure drop after the application of enhancement techniques and ΔP_{st} is the mean pressure drop inside a straight tube with the base fluid only.

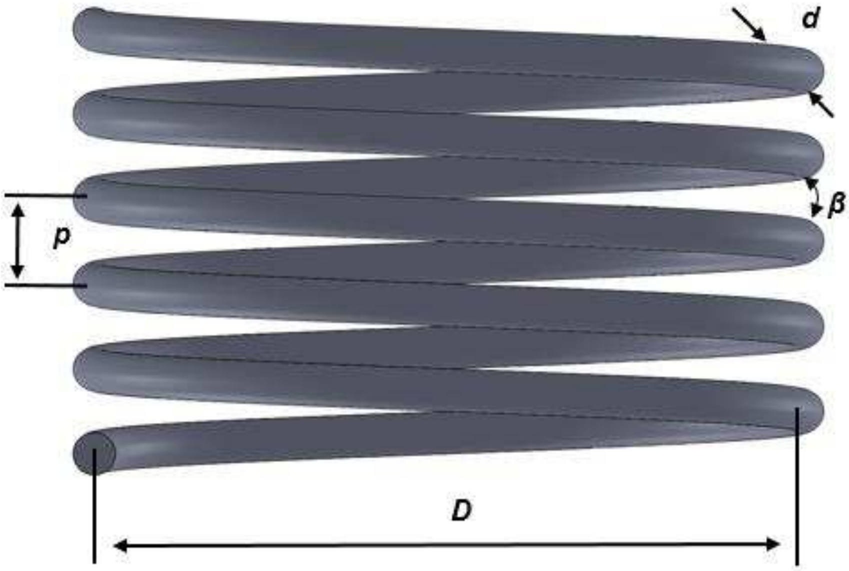


Figure 1: Schematic representation of helical pipe characteristics

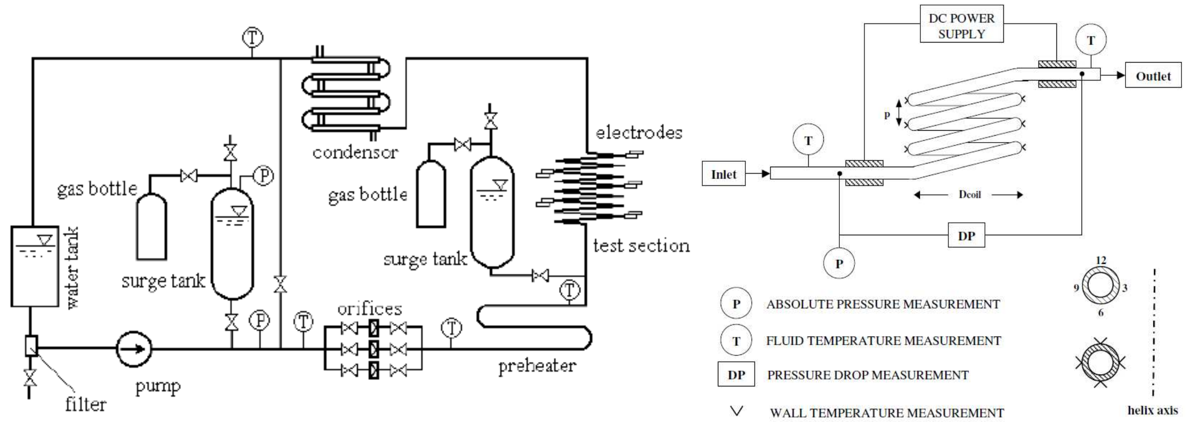
The pertinent literature, presents a considerable number of widely cited studies on the pressure drop for single-phase flow in helically coiled tubes [6,7]. A lesser number of studies have investigated the two-phase pressure drop characteristics in helically coiled tubes. Whilst being more relevant to real-life engineering systems, when compared to single-phase flow, two-phase flow is significantly more complex due to the combination of the three forces governing the flow regime, these being the: inertia, liquid gravity and centrifugal forces [8]. Numerous studies investigated the two-phase frictional pressure drop with steam-water flow boiling [9,10], R-134a refrigerant flows [11,12] and air-water flows [4] whilst more recently, a number of authors investigated the application of nanofluids [13,14] in helically coiled tubes through experimental and computational studies. Mandal and Das [15] and Murai et al. [16] reported that the phase with the lower density is subjected to a smaller centrifugal force which forces the lighter phase to shift towards the inner side of the coil's wall. However, Saffari et al. [17] reported that for bubbly flows at elevated Reynolds numbers and characterised by small bubble diameters (b<0.5mm), the enhanced fluid mixing could result in a quasi-homogenous distribution of the secondary phase. This draws an analogy to similar investigations with nanofluids where no significant phase separation was reported [18].

A recent development in the field of bubbly air-water two-phase flow has resulted in the injection of microbubbles in the flow to achieve a reduction in the system frictional pressure drop. Hitherto, this research has focused on the injection of air bubbles over flat plates and in straight tubes with a minimal consideration for the investigation of the pressure drop reduction in coiled tubes. When investigating the drag reduction inside a channel Nouri et al. [19] reported that bubble injection can be used to decrease the flow transfer costs. In fact, they reported a 35% reduction in the pressure drop in turbulent upward pipe flow with the maximum experimental volumetric void fraction of 9%. This is attributed to the congregation of the larger bubbles at the pipe wall. To the best of the authors' knowledge, the sole investigation with coiled tubes was done by Saffari et al. [17] who reported an increase in the magnitude of drag reduction with increasing volumetric void fraction and decreasing Reynolds and Dean Numbers. These conclusions contrast to the findings reported by the majority of investigations on air-water bubbly flows, where two-phase pressure drop multipliers in excess of unity were reported [20, 8]. The pertinent literature also presents some controversy through conflicting results on the impact of nanoparticles on the frictional pressure drop in helically coiled tubes. In fact, whereas the majority of investigations reported a rise in the pressure drop with the particle concentration [21, 22], some investigations concluded that the opposite effect could occur [23].

Naphon and Wongwises [24] briefly reviewed the single and two-phase flow and pressure drop characteristics in curved tubes. However¸ their review was principally focused on the single-phase flow characteristics and hence they failed to adequately review the pertinent literature for two-phase flow. Therefore, to the best of the authors' knowledge the open literature does not present comprehensive reviews on the pressure drop characteristics of two-phase flow in helically coiled tube heat exchangers. The current study will therefore present a review of the pertinent literature on the two-phase frictional pressure drop characteristics and correlations in helically coiled tubes. It is the authors' hope that this review will be useful to both academics and industry based engineers through the provision of a comprehensive report on the relevant current knowledge and controversies in literature. The present study will also identify areas for further research.

1.1 Research Methods

Experimental and numerical methods were used to investigate the pressure drop characteristics in helically coiled tubes. Fig. 2 presents a schematic diagram of the test facility developed by Guo et al. [25] and Cioncolini et al. [26] for the investigation of the steam-water flow boiling pressure drop in helically coiled tubes at varying operating system parameters such as



Test Rig **Test Section**

Figure 2: Schematic diagram of the typical experimental test rig for the investigation of the flow boiling two-phase pressure drop in helically coiled tubes (Guo et al. (2001b) [25] Fig. 1) and the typical test section (Cioncolini et al. [26], Fig. 2)

the pressure, heat and mass fluxes. This setup is typical for most experimental studies in this field of study. An experimental uncertainty of 2.5% was reported by Cioncolini et al. for their two-phase pressure drop measurements.

As illustrated in Fig. 2, the typical experimental setup for the investigation of the flow boiling two-phase pressure drop characteristics in helically coiled tubes included a centrifugal pump for maintaining the system mass flow rate. Before entering the test section, the working fluid was heated to a subcooled state through the use of the pre-heater. The system bulk fluid flow rates were typically controlled by the system circulation pump. Stainless steel [25,27, 28] was used for the test section, which was thermally insulated to minimise the heat losses to the environment. The majority of the studies reviewed in this paper used the electrical direct heating method to heat the test section whilst, armoured K-type thermocouples were typically used to measure the bulk fluid temperature along the test section. K-type thermocouples, welded to the outside surface of the tube, were also used to measure the tube's wall temperature. These thermocouples were electrically insulated in order to avoid the effects of the heating electrical currents on it. Pressure sensors, installed at the return and flow ends of the helically coiled tube, measured the total two-phase pressure drop whilst a water cooled condenser condensed the steam or refrigerant vapour after the test section. The signals from the various measuring sensors were channelled to a data acquisition system for data monitoring and processing purposes.

All the numerical investigations reviewed in the current study were developed through the use of a commercially available computational fluid dynamics package, namely ANSYS Fluent [22, 29]. The majority of authors validated their experimental and numerical methods through the comparison of the single-phase frictional pressure drop data with widely cited single-phase correlations for helically coiled tubes, such as those given by Ito [30] and Mishra and Gupta [31].

2. Flow boiling heat transfer coefficient

2.1. Steam and Water

A number of correlations are presented in the open literature for the calculation of the flow boiling pressure drop multiplier in helically coiled tubes for a wide range of system parameters. The reviewed correlations are summarised in Table 1 according to the key parameters governing their applications. The total two-phase pressure drop can be broken down into three component pressure drops these being the frictional, gravitational and the momentum pressure drops (Eqs.2-5) [32]. Many researchers have presented the two-phase frictional pressure drop as a function of the pressure drop multiplier and the single-phase frictional pressure drop as given in Eq. (3).

$$\Delta P_{total,TP} = \Delta P_{f,TP} + \Delta P_{grav} + \Delta P_{acc} \quad \textbf{(2)}$$

$$\Delta P_{f,TP} = \Delta P_l \emptyset_l^2 \quad \textbf{(3)}$$

where $\Delta P_{f,TP}$ is the two-phase flow frictional pressure drop of helical coils, and ΔP_l is the frictional pressure drop of the single-phase fluid flowing through the tube with the assumption that only liquid flows through the tube. Many authors have used the single-phase friction factor numerical model given by Ito [6] to calculate the latter pressure drop.

$$\Delta P_{grav} = \left[\frac{gH}{x_{exit}-x_{inlet}}\right]\left[\frac{ln\left(1+x\left(\frac{\rho_l}{\rho_g}-1\right)\right)}{\left(\frac{1}{\rho_g}-\frac{1}{\rho_l}\right)}\right]_{exit} - \left[\frac{gH}{x_{exit}-x_{inlet}}\right]\left[\frac{ln\left(1+x\left(\frac{\rho_l}{\rho_g}-1\right)\right)}{\left(\frac{1}{\rho_g}-\frac{1}{\rho_l}\right)}\right]_{in} \quad \textbf{(4)}$$

$$\Delta P_{acc,TP} = G^2\left\{\left[\frac{1-x}{\rho_l}+\frac{x}{\rho_g}\right]_{exit} - \left[\frac{1-x}{\rho_l}+\frac{x}{\rho_v}\right]_{in}\right\} \quad \textbf{(5)}$$

There appears to be a general agreement amongst the pertinent studies reviewed that the two-phase flow boiling frictional pressure drop increases with the vapour quality and mass flux whilst it decreases with higher system pressures. The curvature ratio does not appear to have a significant influence on the two-phase flow boiling frictional pressure drop multiplier whilst there is some controversy surrounding the influence of the coil orientation and heat flux. Over the past 50 years, the application of numerical models to predict the flow boiling frictional pressure drop in coiled tubes has highlighted the general difficulty in predicting the flow characteristics of two-phase flow. Therefore, many authors have presented their own empirical or semi-empirical models, or correlated existing models to fit their experimental data. The earliest investigations on the flow boiling frictional pressure drop in helically coiled tubes [10, 28, 33] correlated the experimental data with well-known numerical models for the two-phase frictional pressure drop multiplier for straight tubes as given by Lockhart and Martinelli [34], Martinelli and Nelson [35] and Chen [36]. The latter are typically a function of the Lockhart and Martinelli parameter, which, in turn, is a function of the vapour quality and the densities and viscosities of the liquid and gas phases.

Kozeki et al. [28] reported that at the flow boiling region, the frictional pressure drop was circa 70 percent larger than that predicted by the Martinelli and Nelson numerical model for two-phase flow in straight tubes. The higher frictional pressure drop was attributed to the secondary flow phenomenon in the vapour core region where the largest influence was recorded at low pressures and high Reynolds numbers. Such results are in agreement with more recent studies reported by Guo et al. [37] and Santini et al. [38] who concluded that the frictional pressure drop decreases with higher system pressures (Fig.3). This is due to the resultant lower specific volume which in turn yields a lower mixture velocity. Nariai et al. [10] also reported that the effects of the flow boiling phenomena on the frictional pressure drop are not distinct in the fluid conditions.

The influence of the vapour quality on the frictional pressure drop does not appear to be uniform over the complete vapour quality range. Guo et al. [37] and Zhao et al. [27] reported that at vapour qualities below 0.3, the frictional pressure drop increased significantly with the vapour quality whilst at higher qualities this increase was less significant. Santini et al. also reported that the increase in the frictional pressure drop stopped at a vapour quality of 0.8 and subsequently decreased as the quality approached unity. They attributed this phenomenon to the annular flow regime where the liquid film becomes too thin to maintain the interface waves. No other authors have reported similar results for helically coiled tubes and therefore, the latter results can be classified as indeterminate and hence, present ample scope for further investigations.

Bi et al. [32] and Zhao et al. [27] are the sole authors to report that the heat flux does not have a significant impact on the frictional pressure drop. However, more recently, Cioncolini et al. [26] reported that the heating effects resulted in an influence on the frictional pressure drop and hence, their correlation for the frictional pressure drop multiplier is also a function of the system heat flux. They attributed this influence to the interface between the liquid film and the vapour core being dependent on the evaporation and nucleation processes.

Bi et al. [32] and Guo et al. [37] are the sole authors who investigated the flow boiling frictional pressure drop as a function of the coil orientation. However, whilst the former reported that the coil orientation had no significant impact on the two-phase frictional pressure drop, the latter reported distinctly different results. Guo et al. reported that the horizontal coils resulted in the smallest frictional pressure drop whilst the 45 degree, downwards inclined coils resulted in the largest measured pressure drop (70% higher than that measured for the horizontal orientation). The frictional pressure drop for the vertical coil was between that measured for the horizontal and the inclined orientations. Guo et al. attributed these results to the variation in the secondary flow regime with the tube orientation. The authors of the present study cannot adequately address the differences in these two results as the system parameters for both studies were distinctly similar. However, drawing on the conclusions reported by Santini et al. [38] regarding the influence of the system pressure on the pressure drop, the significantly higher system pressure used in Bi et al.'s investigation could suggest that at high system pressures, the flow boiling frictional pressure drop is quasi-independent of the coil orientation.

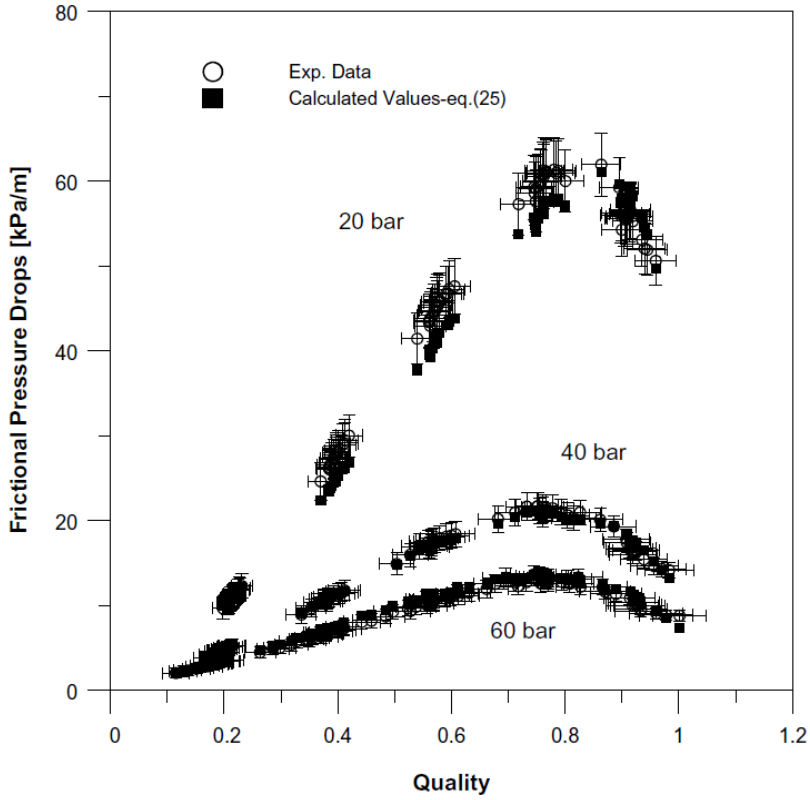


Figure 3: Experimental and predicted (Equation for $\Delta P_{f,TP}$ in Table 1) two-phase flow frictional pressure drop with system pressure and vapour quality at a constant mass flux of 600kg/m^{2}s (Santini et al. [38], Fig. 7)

Correlations derived from the widely used two-phase flow pressure drop correlations for straight tubes (P<3.5MPa & d≥12mm)				
Authors	**Helical coil design parameters**	**Principal experimental parameters**	**Steam quality**	**Main conclusions, proposed correlation and mean error**
Owhadi et al. (1968) [33]	15.9mm OD 12.5mm ID 250<D<527 Vertical	0.024<δ<0.05 P=0.1MPa 60<q< 256 0.0097<$\dot{m}$<0.039 80<G<315	0.5<x<1	Data has resulted in a considerable scatter. In general, it agreed with the Lockhart and Martinelli [34] equation for a straight tubes $\phi_{l,tt}^2 = 1 + \frac{C}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}$ where C is a constant dependent on the gas and liquid Reynolds numbers $\chi_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$
Kozeki (1970) [28]	21.7mm OD 628<D<682mm	0.032<δ<0.035 0.5<P<2.1MPa 151<q<348 161<G<486	0<x<1	Pressure drop is greater than that for a straight tube and it increases with vapour quality and mass flux..

	Vertical			Numerical model based on the Martinelli and Nelson prediction for two-phase flow in straight tubes $\phi_{g,tt}{}^{2} = 0.895 + (\chi_{tt} + 0.076)^{0.875} + 1.21 * 10^{-0.334(log\chi_{tt}+0.668)^{2}}$ where: $\phi_{ltt} = \frac{\phi_{g,tt}}{\chi_{tt}^{0.875}}$
Nariai et al. (1982) [10]	14.3&20mm ID D=595mm Vertical	$0.024<\delta<0.034$ 2<P<3.5MPa 0.7E5<q<1.8E5 $150<G<850$	0.1<x<0.9	Pressure drop increases with mass flux and vapour quality. Martinelli and Nelson [35] prediction for straight tubes: $\Delta P_{f,TP} = R_{MN}\Delta P_{l}$ $R_{MN} = (1-x)^{1.75}\phi_{l,tt}^{2} = \phi_{l}^{2}(P,x)$ Experimental values for ϕ_{l}^{2}were given in table as a function of the system pressure, P and quality, x. Kozeki [28] prediction (better fit) $\phi_{g,tt}{}^{2} = 0.895 + (\chi_{tt} + 0.076)^{0.875} + 1.21 * 10^{-0.334(log\chi_{tt}+0.668)^{2}}$ where: $\phi_{ltt} = \frac{\phi_{g,tt}}{\chi_{tt}^{g,tt}}$ (30%)
Guo et al. (2001) [37]	10&11mm ID D=132&256 mm Horizontal/Vertical and Inclined	$0.043<\delta<0.076$ 3<P<3.5MPa 0<q<540 $150<G<1760$	-0.01<x<1.2	The coil orientation has a significant influence on the frictional pressure drop. Pressure drop is also a function of the system pressure and mass quality. Based on Chen's [36] correlation for straight tubes: $\phi_{l}{}^{2} = \psi_{1}\psi\left[1 + x\left(\frac{\rho_{l}}{\rho_{g}} - 1\right)\right]$ where: for G≤1000 $\psi = 1 + \frac{x(1-x)\left(\frac{1000}{G} - 1\right)\left(\frac{\rho_{l}}{\rho_{g}}\right)}{1 + x\left(\frac{\rho_{l}}{\rho_{g}} - 1\right)}$ for G>1000 $\psi = 1 + \frac{x(1-x)\left(\frac{1000}{G} - 1\right)\left(\frac{\rho_{l}}{\rho_{g}}\right)}{1 + (1-x)\left(\frac{\rho_{l}}{\rho_{g}} - 1\right)}$ $\psi_{1} = 142.2\left(\frac{P}{P_{crit}}\right)^{0.62}\delta^{1.04}$ (±12%)
Correlations for high system pressures based (P>3.5MPa)				
Ruffell (1974) [39]	10.7<ID<18.6 mm	$0.0054<\delta<0.16$ 6<P<18MPa 41<q<731 300G<1800	0<x<1	$\phi_{l}{}^{2} = (1+F)\frac{v_{m}}{v_{l}}$ where: $F = \sin\left(\frac{1.16G}{1000}\right)\left\{0.875 - 0.314y - \frac{0.74G}{1000}(0.152 - 0.07y) - x\left(\frac{0.155G}{1000} + 0.7 - 0.19y\right)\right\}\{1 - 12(x - 0.3)(x-0.4)(x-0.5)(x-0.6)\}$

				$$y = \frac{D}{100d}$$
Unal et al. (1981) [40]	18 mm ID 700&1500mm =D Vertical	$0.0054<\delta<0.022$ 14.7<P<20.2MPa 41<q<731 112<G<1829	0.08<x<1	$$\Delta P_{f,TP} = \frac{2(1+b_1b_2)f_lG^2}{d\rho_l}$$ where: $$b_1 = 3850x^{0.01}Pr^{-1.515}Re_l^{-0.758}$$ $$b_2 = 1 + Re_l^{0.1}(3.67-3.04P_b)^{((-0.014\delta^{-1})-(2\delta^{-1}))}$$ where; $$P_b = \frac{P}{P_{crit}}$$ $f_l = 0.076Re^{-0.25} + 0.00725\delta^{0.5}$ [6] (±20%)
Chen and Zhou (1981) [41]	18 mm ID 235, 446,907 mm=D Vertical	$0.02<\delta<0.076$ 4.2<P<22MPa 400<G<2000	0<x<1	$$\Delta P_{f,TP} = \xi\Delta P_{st}$$ where: $$\xi = 2.06\delta^{0.05}Re_{TP}^{-0.025}\left[1+VF\left(\frac{\rho_g}{\rho_l}-1\right)\right]^{0.8}\left[1+x\left(\frac{\rho_l}{\rho_g}-1\right)\right]^{1.8}\left[1+VF\left(\frac{\mu_g}{\mu_l}-1\right)\right]^{0.2}$$
Santini et al. (2008) [38]	12.53 mm ID D=1000mm Vertical	δ=0.019 1.1<P<6.3MPa 50<q<200 192<G<824	0<x<1	Frictional pressure drop increases with the vapour quality and mass flux whilst it decreases with the system pressure. $$\Delta P_{f,TP} = K\left(\frac{G^{1.91}v_m}{d^{1.2}}\right)\Delta z$$ where: $$K(x) = -0.0373x^3 + 0.0387x^2 - 0.00479x + 0.0108$$ (RMS = 6.2)
Correlations for large tube diameters (d≥12mm)				
Guo et al. (1994) [42]	20 mm ID 240, 480,960 mm=D Horizontal	$0.021<\delta<0.083$ 1.5 <P<3MPa 150<G<1400	0<x<0.8	$$\phi_l^2 = 1 + (4.25 - 2.55x^{1.5})G^{0.34}$$
Correlations for small tube and helix diameters (d<12mm)				
Kubair (1986) [43]	6.4&6.5mm ID 110<D<177 Laminar & Turbulent Vertical	$0.037<\delta<0.056$ 8<P<16kPa 6<q<80 $0.0028<\dot{m}<0.016$ 1300<Re<5200	0.2<x<0.8	Frictional pressure drop is larger than that for straight tubes. No correlation provided.
Bi et al. (1994) [32]	10&12mm ID D=115mm Horizontal& Vertical	$0.087< \delta <0.104$ 4<P<14MPa 0<q<750 400<G<2000	0<x<1	Coil orientation has no significant effect on the two-phase frictional pressure drop. The two-phase frictional pressure drop was not influenced by the conditions of the thermodynamic system i.e. adiabatic or electrically heated tubes. $$\phi_l{}^2 = 1 + \left[\frac{\rho_l}{\rho_g} - 1\right][C + x^2]$$ where:

				$C = 0.14691x^{1.3297}(1-x)^{0.59884}\delta^{-1.2864}$ (±15%)
Ju et al. (2001) [44]	18mm OD D=112mm Turbulent	δ =0.161 P=3MPa 2500<Re <23000	0<x<1	$\Delta P_{f,TP} = f\left(\frac{L}{d}\right)\left(\frac{\rho V^2}{2}\right)\left[1 + x\left(\frac{\rho'}{(\rho''-1)}\right)\right]\psi$ where: $\psi = (1.29 + A_n x^n)\left[1 + x\left(\left(\frac{\mu''}{\mu'}\right)^{0.25} - 1\right)\right]$ A_1=2.19, A_2=-3.61, A_3=7.35, A_4=-5.93
Zhao et al. (2003) [27]	9mm ID D=292mm Laminar Horizontal	δ=0.031 0.5<P<3.5 MPa 0<q<900 236<G<943 10000<Re <80000	0.1<x<0.2	Frictional pressure drop is a function of the mass flux, vapour quality and the system pressure. Heat flux has no effect on the pressure drop. $\phi_l^2 = 1 + \left[\frac{\rho_l}{\rho_g} - 1\right][0.303x^{1.63}(1-x)^{0.885}Re_l^{0.282} + x^2]$ (±12%)
Cioncolini et al. (2008) [26]	4.03&4.98mm ID 130<D<376 Turbulent Vertical Saturated flow boiling	0.011<δ <0.038 120<P<660kPa 50<q<440 290<G<690 10000<Re <60000 2<Fr<14	0<x<0.9	Minimal effect of the coil curvature on the frictional pressure drop. Lockhart and Martinelli correlation for straight tubes corrected for heating effects [45]: $\phi_l^2 = \left[1 + \frac{C}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}\right]\left[1 + 0.0044\left(\frac{q}{G}\right)^{0.7}\right]$ (16.7%) Zhao et al. [27] $\phi_l^2 = 1 + \left[\frac{\rho_l}{\rho_g} - 1\right][0.303x^{1.63}(1-x)^{0.885}Re_l^{0.282} + x^2]$ (16.3%)

Table 1: Review of the experimental studies on the flow boiling frictional pressure drop characteristics of steam-water in helically coiled tubes

2.2. R-134a

The pressure drop characteristics and relevant correlations for flow boiling and condensation of R-134a in helically coiled tube heat exchangers are summarised in Table 2. In contrast to the conclusions made by a number of investigations on steam and water flow boiling, the curvature ratio appears to have some impact on the resultant frictional pressure drop for R-134a flow in non-miniature helically coiled tubes. The pertinent investigations have also concluded that the frictional pressure drop increases with higher vapour qualities and refrigerant mass fluxes, whilst the tube orientation has no significant impact on the pressure drop. The total two-phase pressure drop for the flow boiling of R-134a in micro-finned helically coiled tubes is given in Eq. (6) [46] whilst the two-phase frictional pressure drop was calculated through the use of the pressure drop multiplier as in Eq. (3).

$$\Delta P_{total,TP} = \Delta P_{f,TP} + \Delta P_{grav} + \Delta P_{mom,TP} \qquad \textbf{(6)}$$

where:

$$\Delta P_{grav} = g\rho_l tan\beta(1 - VF) \qquad \textbf{(7)}$$

$$\Delta P_{mom,TP} = G^2 \left\{ \left[\frac{(1-x)^2}{\rho_l(1-VF)} + \frac{x^2}{\rho_v VF} \right]_{out} - \left[\frac{(1-x)^2}{\rho_l(1-VF)} + \frac{x^2}{\rho_v VF} \right]_{in} \right\} \quad \textbf{(8)}$$

Aria et al. [47]:

$$VF = \frac{x}{\rho_v} \left[\left(1 + 0.12(1-x)\right) \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right) + \frac{1.18(1-x)[g\sigma(\rho_l - \rho_v)]^{0.25}}{G^2 \rho_l^{0.5}} \right]^{-1} \quad \textbf{(9)}$$

Cui et al. [46]

$$VF = \frac{1}{1 + 0.49\chi_{tt}^{0.8036}} \quad \textbf{(10)}$$

Elsayed et al. [48]

$$VF = \frac{1}{\left[1 + 0.79\left(\frac{(1-x)}{x}\right)^{0.78} \left(\frac{\rho_v}{\rho_l}\right)^{0.58} \right]} \quad \textbf{(11)}$$

Wongwises and Polsongkram [49]

$$VF = \frac{1}{1 + S\left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}} \quad \textbf{(12)}$$

Laohalertdecha and Wongwises [50]

$$VF = \left[1 + \frac{(1-x)}{x} \left(\frac{\rho_v}{\rho_l} \right)^{2/3} \right]^{-1} \quad \textbf{(13)}$$

The numerical models for the R-134a refrigerant frictional pressure drop in vertical helically coiled tubes as reported in the pertinent studies are a function of the Lockhart and Martinelli parameter, whilst the sole correlation for horizontal tubes is based on a numerical model by Kim et al. [51] for R-22 flow in coiled tubes. There is a general agreement that the higher mass fluxes and the vapour qualities increase the frictional pressure drop. These results are attributed to the higher vapour velocities which increase the shear stress at the interface of the vapour and the liquid. Furthermore, higher vapour qualities result in increased magnitudes of secondary flow which will result in higher degrees of entrainment and droplet redeposition, thus yielding greater flow turbulences [49]. Moreover, Lin and Ebadian [52] reported that when compared to the flow in the inner tube, the effects of the mass flux on the pressure drop were more significant in the annular section of the coil. These findings were attributed to the larger velocity and turbulence fluctuations of the refrigerant flowing in the annular section.

The effects of the coil geometry on the two-phase refrigerant frictional pressure drop were investigated by Scott Downing and Kojasoy [53] and Elsayed et al. [48]. For miniature diameter tubes, Downing and Kojasoy reported that when compared to single-phase flow, the curvature effects had a minimal impact on the frictional pressure drop. However, for small diameter tubes, Elsayed et al. reported that the frictional pressure drop is mainly a function of the tube diameter, with the pressure drop increasing with smaller tube diameters. The effect of the coil diameter was reported to be less significant. Elsayed et al. focused their study on the heat transfer characteristics and hence failed to provide a comprehensive analysis of their reported results.

When investigating the frictional pressure drop as a function of the heat flux, Wongwises and Polsongkram [54] reported that the heat flux had a minimal effect on the condensation frictional pressure drop. However, the evaporation frictional pressure drop was reported to be a strong function of the heat flux [49]. This was attributed to the increase in the number of active nucleation sites on the tube wall which yielded higher bubble generation rates. The latter agitated the liquid film thus increasing the turbulence. Furthermore, the breaking of the bubbles at the liquid film surface induced the entrainment and redeposition of droplets which increased the shear stress. Kang et al. [11] and Wongwises and Polsongkram [49, 54] reported a decrease in the frictional pressure drop with higher wall temperatures. These results were attributed to the lower refrigerant viscosity and specific volume, which in turn resulted in a lower vapour velocity and shear stress between the vapour and liquid interface.

The sole study that investigated the frictional pressure drop of R-134a as a function of the coil orientation was reported by Lin and Ebadian [52] who concluded that the coil orientation resulted in an insignificant impact on the frictional pressure drop. The applications of micro-finned or corrugated helically coiled tubes were investigated by Cui et al. [46] and Laohalertdecha and Wongwises [50]. Both investigations reported correlations for the pressure drop multiplier based on the Lockhart and Martinelli numerical model for straight tubes. Moreover, both authors reported a significant increase in the frictional pressure drop (up to 70%) over that of a smooth tube. Laohalertdecha and Wongwises attributed these results to the increased drag forces, the flow blockage due to the reduction in the tube cross-sectional area, the turbulence augmentation and the enhanced rotational flow reduction. The impact of the vapour quality and mass flux on the frictional pressure drop was similar to that reported for smooth tubes.

Authors	**Helical coil design parameters**	**Principal experimental parameters**	**Quality**	**Main conclusions, proposed correlation and mean error**
Miniature (d_{it}<1mm)				
Scott Downing and Kojasoy (2002) [53]	234<ID<881μm 2.80<D<7.94mm	0.075<δ_{it}<0.3 0.62<P<1.4MPa 0<q<25 750<G<6330 500<Re<8000	0<x<0.9 Evaporation	Curvature effects have a minimal effect on the frictional pressure drop when compared to single-phase flow. $\phi_{l,tt}^2 = 1 + \frac{C}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}$ where; $C = 3.598\left(\frac{1}{\chi_{tt}}\right)^{0.012}$ (±15%)
Vertical orientation & Smooth tubes				
Kang et al. (2000) [11]	Tube-in-tube 12.7mm ID_{it} 21.2mm ID_{ot} D=177.8mm Laminar & Turbulent	δ_{it}=0.075 100<G<400 T=33^{0}C 1500<Re<9000	0<x<1 Condensation	Very slow increase in the pressure drop with an increase in the mass flux. Pressure drop is a function of the cooling wall temperature, with a decrease in the pressure drop with an increase in the wall temperature. For $T_{wall} = 12$°C, $\Delta P_{TP} = 14.2 m_{ref}^{0.093}$ For $T_{wall} = 22$°C, $\Delta P_{TP} = 4.2 m_{ref}^{0.26}$ (-37.3% to 35.7%)
Han et al. (2005) [55]	Tube-in-tube 9.4mm ID_{it} 12.7mm OD_{it} 21.2mm ID_{ot} D=177.8mm	δ_{it}=0.053 100<G<420 T_{sat}=35,40,46^{0}C 1500<Re<9000	0<x<1 Condensation	Pressure drop increases with refrigerant mass flux. Frictional pressure drop is higher than that in a straight tube, whilst the effect of the mass flux on the pressure drop is more significant in straight tubes. No correlation provided
Wongwises and Polsong	Tube-in-tube 7.2mm ID_{it} 9.52mm OD_{it} 21.2mm ID_{ot}	δ_{it}=0.025 5<q<10 400<G<800	0.0<x<1 Evaporation	Increase in the frictional pressure drop with increasing quality, mass flux and heat flux. Marginal decrease with increasing saturation temperature.

kram (2006a) [49]	23.2mm OD_{ot} D=305mm	$10<T_{sat}<20^0C$		$$\phi_l^2 = 1 + \frac{13.37}{\chi_{tt}^{1.492}}$$ Used Ito's [6] correlation for the single-phase friction factor (±20%)
Wongwises and Polsongkram (2006b) [54]	Tube-in-tube 8.3mm ID_{it} 9.52mm OD_{it} 21.2mm ID_{ot} 23.2mm OD_{ot} D=305mm	δ_{it}=0.025 5<q<10 $400<G<800$ $40<T_{sat}<50^0C$	0.01<x<1 Condensation	Frictional pressure drop increases with average vapour quality and mass flux and decreases with increasing saturation temperature of condensation. Heat flux has a minimal effect on the pressure drop. $$\phi_l^2 = 1 + \frac{5.569}{\chi_{tt}^{1.492}} + \frac{1}{\chi_{tt}^2}$$ Used Ito's [6] correlation for the single-phase friction factor (±20%)
El-Sayed Mossad et al. (2009) [56]	Tube-in-tube 7.39mm ID_{it} 9.54mm OD_{it} 16.92mm ID_{ot} 19.05mm OD_{ot} D=216mm	δ_{it}=0.03 810<P<820kPa 2.5<q<12 $95<G<710$ 1000<Re<14000	0.0<x<1 Condensation	Increase in the frictional pressure drop with the refrigerant mass flux. Pressure drop is significantly higher than in a straight tube. Used Han et al.'s [55] correlation
Aria et al. (2012) [47]	Tube-in-tube 8.9mm ID_{it} 9.52mm OD_{it} 29mm ID_{ot} D=305mm	δ_{it} =0.031 $112<G<152$	0.1<x<0.8 Evaporation	Pressure drop increases with higher inlet vapour quality and refrigerant mass flow rate. 150-220% higher than pressure drop in straight tubes. Used Wongwises and Polsongkram's [49] correlation for helical tubes (-73% to +39%)
Micro-finned or corrugated tube				
Cui et al. (2008) [46]	Micro-finned 11.2mm ID 12.7mm OD D=185mm Vertical	δ=0.061 0.5<P<0.58MPa 2.0<q<21.8 $65<G<315$	0.05<x<0.92 Evaporation	Two-phase pressure drop is greater than that in a straight pipe. Micro-fins also increase the pressure drop as does increasing mass flux and vapour exit quality. For Stratified flow: $$\phi_l^2 = 1 + \frac{48.2}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}$$ For Annular flow: $$\phi_l^2 = 1 + \frac{59.8}{\chi_{tt}} + \frac{3.5}{\chi_{tt}^2}$$ Ito's [6] correlation for the single-phase friction factor was used (±20%)
Laohalertdecha and Wongwises (2010) [50]	Corrugated ID_{it} =8.7mm OD_{it} =9.52mm ID_{ot} =21.2mm E=1.5mm Horizontal	5<q<10 $200<G<700$ T_{sat}=40,45,50^{0}C	0.01<x<0.9 Condensation	Frictional pressure drop increases with refrigerant mass flux and quality. 70% increase in the frictional pressure drop over that of smooth tubes

				$$\phi_{l,tt}^2 = 1 + \frac{10}{\chi_{tt}} + \frac{1}{\chi_{tt}^2}$$ (±30%)
Horizontal orientation & Smooth tubes				
Elsayed et al. (2012) [48]	1.1<ID<2.8mm 1.47<OD<4mm 30<D<60mm	$0.037<\delta<0.047$ 0.35<P<0.6MPa 2.5<q<12 $100<G<450$	0.2<x<0.9 Evaporation	Frictional pressure drop is a strong function of the inner tube diameter. The coil diameter has a marginal effect on the pressure drop. Kim et al.'s [51] correlation for R-22: $$\Delta P_{f,TP} = 2\frac{f_{TP}G_{ref}^2}{\rho_l d_{it}}\left[1 + x\frac{\left(\frac{1}{\rho_v} - \frac{1}{\rho_l}\right)}{\frac{1}{\rho_v}}\right]$$ where: $$f_{TP} = 0.079\left(\frac{G_{ref}d_{it}}{\mu_{TP}}\right)^{-0.25}$$ $$\mu_{TP} = \rho_{TP}\left[\frac{\chi\mu_v}{\rho_v} + \frac{(1-\chi)\mu_l}{\rho_l}\right]$$ $$\rho_{TP} = \rho_v VF + \rho_l(1 - VF)$$
Various orientations and smooth tube				
Lin and Ebadian (2007) [52]	Tube-in-tube 9.4mm ID_{it} 12.7mm OD_{it} 21.2mm ID_{ot} D=177.8mm Horizontal/45°/Vertical	$\delta_{it}=0.053$ $60<Re_{ref}<200$ $3600<Re_{wt}<22000$ $30<T_{ref}<35$ $16<T_{wt}<24$	Condensation	The effects of the tube orientation on the pressure drop were not significant whilst the effects of the refrigerant mass flow rate on the pressure drop were more significant in the annular section of the pipe when compared to the inner tube. $$\phi_l^2 = 1 - \frac{1.271}{\chi_{tt}^{1.492}} + \frac{1}{\chi_{tt}^2}$$ (±6%)

Table 2: Review of experimental studies on the flow boiling/condensing frictional pressure drop characteristics of R-134a in helically coiled tubes

3. Gas-Water

In contrast to the paucity of studies on the gas-water two-phase flow heat transfer characteristics in helically coiled tubes [57], the open literature presents numerous studies on the two-phase gas-water pressure drop characteristics. Table 3 summarises the experimental studies and correlations for the frictional pressure drop with gas and water two-phase flow as presented in the pertinent literature. The majority of studies reviewed in this section have demonstrated a reasonable agreement with the original and modified Lockhart and Martinelli correlations. Some investigators have also reported the helix angle and the curvature ratio to have some impact on the frictional pressure drop whilst other investigators reported the frictional pressure drop to be independent of the latter design parameters. The effects of the air volumetric void fraction remain indeterminate due to conflicting results. As in the case of steam-water flow, the total two-phase pressure drop with air-water systems is calculated through Eq. (2) whilst the two-phase frictional pressure drop is calculated through the application of the pressure drop multiplier as in Eq. (3).

Most studies on the two-phase air-water flow in helically coiled tubes were developed for vertically orientated tubes. The earliest study was reported by Rippel at al. [58] who investigated annular, bubbly, slug, and stratified flows. In agreement with some studies reported for steam-water, they reported that the Lockhart and Martinelli correlation for horizontal straight tubes predicted their data with reasonable accuracy. These results were attributed to the fact that the Lockhart and Martinelli parameters are essentially ratios, while the geometry of the tube does not impact on the ratio of the two-phase to single-phase pressure drop given in Eq. (3). However, they also reported that the latter methodology also results in a number of limitations, principally due to the fact that some pertinent factors that affect two-phase flows are neglected. In view of this, Rippel et al. presented three empirical correlations for the calculation of the two-phase flow pressure drop for annular, bubbly and stratified flows. These correlations are based on the two-phase drag coefficient. Banerjee et al. [59] reported similar results with the Lockhart and Martinelli correlation and presented modified equations for the gas and liquid pressure drop multipliers, and the Lockhart and Martinelli parameter. Banerjee were the first investigators to report that the helix angle did not have a significant impact on the frictional pressure drop. Agakawa et al. [4] also reported a good agreement with the Lockhart and Martinelli correlation whilst the frictional pressure drop was reported to be independent of the coil curvature. They also presented two key empirical correlations to calculate the ratios of the two-phase frictional pressure drop in the coil to those in a straight tube and coil with liquid flow only. Xin et al. [8] presented a further development of the Lockhart and Martinelli correlation whereby they included the effects of the three main forces affecting the pressure drop these being: the inertia, liquid gravity and centrifugal forces. In fact, they reported that the helix angle, coil diameter and pipe diameter had some effect on the frictional pressure drop. Awwad et al. [20, 60] investigated the two-phase frictional pressure drop in horizontal helically coiled tubes. Their conclusions and correlations are similar to those presented by Xin et al. [8] for vertical tubes. Therefore, whilst being based on the original Lockhart and Martinelli model, their correlations for horizontal coils included the effects of the three principal forces affecting two-phase flow in coiled tubes.

Xin et al. [61] investigated the two-phase flow frictional pressure drop in annular helicoidal pipes. As done in their earlier study [8] on vertical coils, they presented a correlation for the calculation of the pressure drop multiplier which is a function of the Lockhart and Martinelli parameter, as well as the Froude number. However, for the case of the annular tubes, the latter is also a function of the inner and outer tube diameters. Vashisth and Nigam [62] were the sole authors to investigate the frictional pressure drop in a coiled flow inverter. The pressure drop was reported to be significantly higher than that for a straight helix. This result was attributed to the higher recirculation rates and the complete flow inversion as a result of the sudden shift in the flow direction. Vashisth and Nigam also reported that the Lockhart and Martinelli correlation, both in its original and modified form, predicted their data for a very limited range of flow rates. Therefore, they presented their own correlation for the two-phase friction factor in a coiled flow inverter which is a strong function of the number of bends and the curvature ratio. The latter was included due to their conclusions that smaller coil diameters resulted in higher intensity secondary flows which consequently increased the two-phase frictional pressure drop.

Chen and Guo [63] investigated the three phase oil-air-water flow in helically coiled tubes. The frictional pressure drop was reported to be independent of the coil diameter whilst, due to increased mixture viscosities, higher oil fractions resulted in higher pressure drops. A correlation which is essentially a modified Chisholm correlation for straight tubes was also presented. Chisholm's correlation was also used to correlate the data for sulphur hexafluoride-water flow in helically coiled tubes [64]. The sole study in the pertinent literature that investigated the gas-non-Newtonian pressure drop in helically coiled tubes was reported by Biswas and Das [65]. They reported a large deviation with the Lockhart and Martinelli correlation which was attributed to the non-Newtonian fluid properties. Therefore, they presented an empirical correlation for the calculation of the friction faction factor which is a function of the fluid and gas Reynolds number, the curvature ratio and fluid properties. In agreement to pertinent conclusions made for gas-water flow [20, 59], the impact of the helix angle on the frictional pressure drop was also found to be negligible.

A recent study reported by Saffari et al. [17] investigated the frictional drag reduction through the use of two-phase bubbly flow in vertical helically coiled tubes. This study is based on earlier initiatives developed for straight tubes whereby two-phase bubbly flow resulted in a drag reduction over the corresponding single-phase flow [66, 67]. For turbulent flow with an air volumetric void fraction of 0.09, Saffari et al. reported a maximum drag reduction of 25% over that of single-phase flow, ceteris paribus (Fig. 4). The latter reduction was at its highest at the lower end of the turbulent flow Reynolds numbers. Saffari et al. attributed these results to the impact of the centrifugal force on the lighter phase, this being air, whereby due to their lighter density, bubbles accumulate on the tube inner wall in the flow boundary layer. At lower Reynolds numbers these bubbles are widely spread on the tube wall and consequently result in a significant reduction of the turbulent Reynolds stresses. Such results are in contrast to the findings reported in this section where all investigators reported frictional pressure drop multipliers in excess of 1.

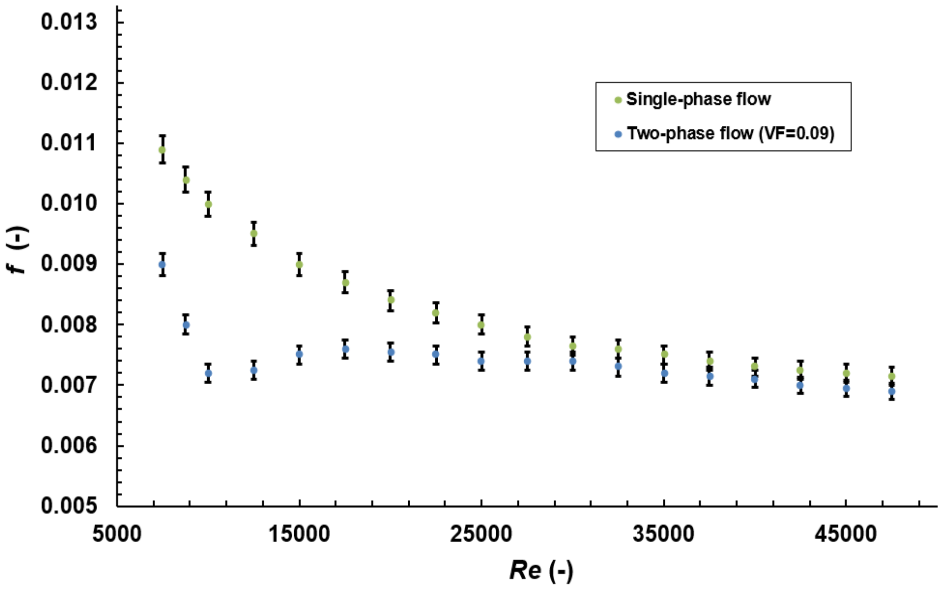


Figure 4: Comparison of the friction factor for single-phase and two-phase flow in helically coiled tubes at a volumetric void fraction of 0.09 (Saffari et al. [17], Fig. 4)

Authors	**Helical coil design parameters**	**Principal experimental parameters**	**Main conclusions, proposed correlation and mean error**
		Vertical orientation – Air-Water	
Rippel et al. (1966) [58]	d=6.35mm D=203mm	δ=0.031 100<Re<15000 Bubbly&Slug/Annular/Stratified	Data fitted the Lockhart-Martinelli parameter. Developed correlations based on the two-phase drag coefficient. Annular Flow: $\left(\frac{\Delta P}{\Delta L}\right)_{f,TP} = \left(\frac{\Delta P}{\Delta L}\right)_g + 4.44\varepsilon^{0.86}\left(\frac{\rho_g U^2}{gd}\right)$ Bubble and Slug Flow: $\left(\frac{\Delta P}{\Delta L}\right)_{f,TP} = \left(\frac{\Delta P}{\Delta L}\right)_g + 31.3\varepsilon^{1.25}\left(\frac{\rho_g U^2}{gd}\right)$ Stratified Flow: $\left(\frac{\Delta P}{\Delta L}\right)_{f,TP} = \left(\frac{\Delta P}{\Delta L}\right)_g + 3.2\varepsilon^{0.875}\left(\frac{\rho_g U^2}{gd}\right)$

Banerjee et al. (1969) [59]	15.34<d<54.8mm 152<D<610mm	0.108<δ<0.090 500<Re<40000	Helix angle had no significant effect on the frictional pressure drop. Data correlated well with the Lockhart and Martinelli correlation using the modified $\emptyset_l, \emptyset_g$ and χ_{tt}; $\emptyset_l^2 = SF^{n-2}\left(\frac{d}{HD_l}\right)^{5-n}$ $\emptyset_g^2 = SF^{m-2}\left(\frac{d}{HD_g}\right)^{5-m}$ $\chi_{tt}^2 = \frac{\left(\frac{\Delta P}{\Delta z}\right)_l}{\left(\frac{\Delta P}{\Delta z}\right)_g}$ (±30%)
Akagawa et al. (1971) [4]	d=9.92mm D=109, 225 mm	0.044<δ<0.091 0<U_g<5m/s 0.35<U_l<1.16m/s Bubbly&Slug	Frictional pressure drop was measured as 1.1 to 1.5 times greater than that in straight tubes, ceteris paribus. Pressure drop is not a function of the curvature. Data fitted the Lockhart and Martinelli correlation. Empirical equations were also provided: $\frac{\Delta P_{f,TP,c}}{\Delta P_{f,TP,s}} = \frac{(1+144\delta^{1.61})}{Re_{TP}^{1.4\delta}}$ where: $Re_{TP} = \frac{U_l d}{(1-VF_g)\mu_l}$ $\frac{\Delta P_{f,TP,c}}{\Delta P_{f,l,c}} = (1-VF_g)^{\left[\left(\frac{2.3d}{D}\right)-1.4\right]}$ (±35%)
Kasturi and Stepanek (1971) [68]	d=12.5mm D=665 mm	δ=0.019 1E+3<De<1E+6 Stratified&Wavy	Data fitted the Lockhart-Martinelli parameter.
Whalley (1980) [69]	d=20.2mm D=1000mm B=6	δ=0.019 Stratified& Annular	Frictional pressure drop is the dominant pressure drop over the acceleration and gravity pressure drops. No correlation provided.
Rangacharyulu and Davies (1984) [70]	d=11,13mm 1.52<β<2.69	0.0427<δ<0.0541 1<vf_g<10 m^3/h 0.04<vf_l<0.75 m^3/h	Correlation also valid for air in glycerol and isobutyl alcohol solutions. $\emptyset_g - 1 = 0.05 Re_l \delta^{0.5}\left(\frac{U_{TP}}{CS}\right)^{-0.68}\left(\frac{\mu_l^4 g}{\rho_l \sigma_l^3}\right)^{0.18}\delta^{3.66}$
Xin et al. (1996) [8]	d=12.7,19.1,25.4,38.1mm D=305,609 mm	0.02<δ<0.125 0.008<U_w<2.2 0.2<U_g<50 Bubbly flow	The helix angle, coil and pipe diameters have a marginal effect on the frictional pressure drop. For $F_d > 0.1$ $\phi_l = \left[1+\frac{\chi}{434.6F_d^{1.7}}\right]\left[1+\frac{20}{\chi}+\frac{1}{\chi^2}\right]^{0.5}$ For $F_d \leq 0.1$ $\phi_l = \left[1+\frac{\chi}{65.45F_d^{0.6}}\right]\left[1+\frac{20}{\chi}+\frac{1}{\chi^2}\right]^{0.5}$ where; $F_d = Fr\left(\frac{d}{D}\right)^{0.5}(1+tan\beta)^{0.2}$ $Fr = \frac{U_l^2}{gd}$

			(±35%)
Mandal and Das (2003) [15]	d=10,13mm 131<D<2222mm 0<β<12	0.046<δ<0.095 1.5<vf_g<52.5E-5 3.65<vf_l<14.2E-5 28<T_{mean}<32°C	The helix angle has no effect on the pressure drop. Empirical correlation was developed to calculate the two-phase friction factor. $f_{TP,l} = 5.8853 Re_l^{-1.1829\pm0.0215} Re_g^{0.952\pm0.0142} \left(\frac{\mu_l^4 g}{\rho_l \sigma_l^3}\right)^{0.022\pm0.0086} \delta^{-0.282\pm0.0369}$
Murai et al. (2006) [16]	d=20mm D=540,750mm	0.027<δ<0.04 P=0.101MPa 1.76<U<5.28 15<T<17°C Re>10^4 Bubbly/Plug/Slug flow	$\Delta P_{f,l} = f \rho_l (1 - VF) \frac{L}{d} \frac{U^2}{2}$ Used Ito's [6] correlation for the single-phase friction factor
Saffari et al. [17]	d=12,19mm D=200mm	0.06<δ<0.095 P=0.101MPa 10000<Re<50000 0.03<VF<0.09 Bubbly flow	25% reduction in the frictional pressure drop with VF=0.09 of air over that of single-phase flow, ceteris paribus. No correlation provided.
Horizontal orientation – Air-Water			
Awwad et al. (1995) [20]	12.7<d<38.1mm 330<D<670mm 1<β<20	0.04<δ<0.057 0.2<U_g<50m/s 0.008<U_l<2.2m/s Bubbly flow	Frictional pressure drop is a function of the flow rate of air and water and the Lockhart-Martinelli parameter. The helix angle has almost no effect on the frictional pressure drop whilst the tube and coil diameters have some effects which diminish at higher fluid flow rates. $\phi_l = \left[1 + \frac{\chi}{C(F_d)^{n1}}\right]\left[1 + \frac{12}{\chi} + \frac{1}{\chi^2}\right]^{0.5}$ where: $F_d \leq 0.3$, C=7.79 & n1=0.576 $F_d > 0.3$, C=13.56 & n1=1.3 (±32%)
Awwad et al. (1995) [60]	d=25.4mm D=350, 660mm 1<β<20	0.04<δ<0.073 0.2<U_g<50m/s 0.008<U_l<2.2m/s	Same conclusions as in Awwad et al. [20] $\phi_l = \left[\frac{\chi}{9.63 F_d^{0.61}}\right]\left[1 + \frac{12}{\chi} + \frac{1}{\chi^2}\right]^{0.5}$ $F_d = Fr\delta^{0.1}$ (±35%)
Coiled flow inverter – Air-Water			
Vashisth and Nigam (2007) [62]	5<d<15mm	0.05<δ<0.149 8.33<vf_g<100E-5 3.33<vf_l<1000E-6	Pressure drop increases by a factor of 1.2-2.5 more than that of a straight helix. Smaller coil diameters result in higher frictional pressure drops. $f_{TP} = \frac{29.4 N^{0.16} \left(\frac{D}{d}\right)^{0.19} Re_g^{0.06}}{Re_l^{0.94}} \quad 400 < Re_l \leq 9000$ $f_{TP} = \frac{0.065 N^{0.003} Re_g^{0.001}}{\left(\frac{D}{d}\right)^{0.003} Re_l^{0.13}} \quad Re_l \geq 10200$ (±15%)

Annular helicoidal tubes – Air-Water			
Xin et al. (1997) [61]	Tube-in-tube OD_{it}=6.35,9.525,12.7mm ID_{ot}=10.21,15.748,21.18mm D=114.3,177.8,196.85mm Vertical and Horizontal orientations	$30<Re_g<30000$ $210<Re_w<23000$	Frictional pressure drop is a function of the flow rate of air and water and the Lockhart-Martinelli parameter, whilst the flow rate effect diminishes with an increase in the tube diameter. $$\phi_l = \left[1+\frac{0.0435\chi^{1.5}}{F}\right]\left[1+\frac{10.646}{\chi}+\frac{1}{\chi^2}\right]^{0.5}$$ where; $$F = Fr^{0.9106}e^{0.0458(lnFr)^2}$$ $$Fr = \frac{U_l^2}{g(ID_{ot}-OD_{it})}$$
Three-phase: Oil-Air-Water			
Chen and Guo (1999) [63]	d=39mm D= 265, 522.5mm $1<\beta<20$	$0.45<U_g<19.02$m/s $0.018<U_{wt}<1.85$m/s $0.0141<U_o<0.91$ m/s $15<T_{mean}<20^oC$ 0.1<P<0.5MPa $VC_{oil}<30\%$ $VC_{wt}>70\%$ Stratified/Oil-Droplet Stratified/Oil-Droplet/Annular Oil flow	The frictional pressure drop increases with the oil fraction in the mixture. Coil diameter has no effect on the frictional pressure drop. $$\phi_l^2 = f(\theta)\left[1-\frac{0.603}{\chi}+\frac{1}{\chi^2}\right]$$ where: $$f(\theta) = R^{0.0172}\left(\frac{1526}{G}\right)^{1.596}(\delta)^{0.175}\left(\frac{\mu_g\rho_o}{\mu_o\rho_o}\right)^{-1.238}\left(\frac{\mu_{wt}\rho_o}{\mu_o\rho_{wt}}\right)$$ $$R = \frac{U_g}{U_l}$$ (±30%)
Gas-Non-Newtonian Fluid			
Biswas and Das (2008) [65]	9.3<d<12mm 176.2<D<266.7mm $0<\beta<12$ Vertical orientation	$0.035<\delta<0.09$ $0.44<vf_g<42.03$ E-5 $3.334<vf_l<15.003$E-5 $28<T_{mean}<32^oC$ 0.2<MC<0.8	The effect of the helix angle on the pressure drop was negligible. Empirical correlation was developed to calculate the two-phase friction factor. $$f_{TP,l} = 0.4Re_g^{0.757\pm0.025}Re_l^{-1.437\pm0.059}\left(\frac{\mu_{eff}^4 g}{\rho_l\sigma_l^3}\right)^{-0.348\pm0.017}\delta^{0.721\pm0.076}$$ (RE 8%)
SF6-Water			
Czop et al. (1994) [64]	d=19.8mm D=1170mm β=7.27	δ=0.017 0.1<P<1.35MPa 26000<Re<50000 500<G<3000 Slug & Bubbly flow	Significant differences with the Lockhart-Martinelli correlation. Fairly good agreement with the Chisholm correlation [71]: $$\phi_{l,tt}^2 = 1+\frac{C}{\chi_{tt}}+\frac{1}{\chi_{tt}^2}$$ where: $$\chi_{tt} = \frac{1-x}{x}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}$$ $$C = 1.5\left[\left(\frac{\rho_g}{\rho_l}\right)^{0.5}+\left(\frac{\rho_l}{\rho_g}\right)^{0.5}\right]$$

Table 3: Review of experimental studies on the air-water frictional pressure drop characteristics in helically coiled tubes

4. Nanofluids

4.1. Experimental studies

There is a significant paucity of studies on the pressure drop characteristics of nanofluids in helically coiled or curved tube heat exchangers. Fakoor-Pakdaman et al. [72] reported that the few studies reported on the investigation of nanofluid flow in helically coiled tubes were mainly focused at investigating the heat transfer characteristics with the system parameters. In fact, their study, published in 2013, was the first study to comprehensively investigate the isothermal pressure drop with nanofluids in helically coiled tubes. As reported in Section 1 of the present study, a number of authors have considered the ratio of the resultant pressure drop with nanofluids in a helically coiled tube to that in a straight tube with the base fluid only, to calculate the performance index given in Eq. (1). This is principally used to appraise the application of heat transfer enhancement techniques as a function of the ratios of the heat transfer coefficients and the pressure drops. The latter is particularly relevant to the calculation of the heat exchanger performance, as the use of helical coils and nanofluids could result in a significant increase in the pressure drop (circa 3.5 times) over that of the base fluid in straight tubes [72].

Table 4 summarises the pertinent experimental studies reviewed, categorised according to the nanoparticles and the base fluids investigated. The principal nanoparticles which have been reported by researchers are the oxides of copper and aluminium whilst the base fluids are water and oil.

Most of the researches reviewed in the present study have reported an increase in the nanofluid pressure drop with the nanoparticle concentration and the Reynolds number. This is mainly attributed to the resultant higher relative mixture densities and viscosities [3, 73, 74]. However, most researchers agreed that at low fluid velocities the rate of increase in the pressure drop with the nanoparticle volume concentration was smaller than that at higher fluid velocities. Mukesh Kumar et al. [74] attributed this result to the dominance of the viscosity effects at low Dean numbers. Furthermore, Hashemi and Akhavan-Behabadi [73] reported that the higher rate of chaotic motion and migration of the nanoparticles at increased Reynolds numbers could be the reason for the different rates of pressure drop increases. There are no experimental studies which investigated the pressure drop characteristics of the principle nanofluids, these being the oxides of aluminium and copper dispersed in water, at identical system parameters. However, Hashemi and Akhavan-Behabadi reported that due to the spherical properties of CuO nanoparticles, reduced levels of friction could result when compared to other nanofluids. This is due to the rolling effect (instead of sliding) between the oil and solid phases.

There is an agreement amongst authors [18, 14] that the transitional velocity, and hence the critical Reynolds number of nanofluids will be higher than that of the base fluid. This is due to the higher viscosity of the former. As reported in our review on the two-phase heat transfer characteristics in helically coiled tubes [57] some controversies characterise the studies on nanofluid flow in these tubes. The majority of investigations reviewed in the present study reported a significant appreciation in the pressure drop with nanofluids over that of the base fluid only. Furthermore, the increment in the pressure drop for helical tubes was reported to be higher than that for straight pipes. In view of this, Suresh et al., Fakoor-Pakdaman et al. and Kahani et al. [13, 72, 75] presented correlations for the calculation of the friction factor and pressure drop with nanofluids. These correlations are principally a function of the coil geometry, Dean or Reynolds numbers and the nanoparticle concentration. With a 2% weight concentration of CuO nanoparticles in oil, flowing through a helically coiled tube, Hashemi and Akhavan-Behabadi [73] reported an increase in the pressure drop of 20.3% over that of the base fluid only whilst for a straight tube, this was measured as 13.2%. Similarly, for 0.2% volume concentration of CuO in water, Kannadasan et al. [18] reported that the friction factor, when compared to water flow only, increased by 24% and 23% for horizontal and vertical orientations respectively. However, in contrast to these findings, Suresh et al. and Wu et al. [75, 14] reported that the resultant pressure drop increment with a wide range of nanoparticle concentrations was marginal when compared to that of the base fluid alone. In fact, Wu et al.'s pressure drop results were reasonably predicted by the Ito [6] (laminar) and Seban and McLaughlin [76] (turbulent) equations for single-phase flow in helically coiled tubes. Suresh et al. attributed these results to the nanoscale size of the additive nanoparticles. Furthermore, whilst Wu et al. [14] reported that, due to their higher viscosity and density, nanofluids resulted in a mitigation of the secondary flow, Mukesh Kumar et al. [3] reported contradictory results. The latter results were attributed to the random motion of the nanoparticles which did not impede the formation of the secondary flow.

The nanofluid pressure drop as a function of the coil geometry was investigated by Kahani et al. [77] and Fakoor-Pakdaman et al. [72] who both reported lower pressure drops with a decrease in the curvature ratio. The pressure drop was also independent of the coil pitch. The former was principally attributed to the weaker centrifugal forces, hence minimising the effects of the secondary flow on the system pressure drop. The sole study which investigated the nanofluid pressure drop as a function of the helical coil orientation was reported by Kannadasan et al. [18]. They reported that the nanofluids in a vertical coil resulted in marginally lower pressure drop increments (over that of pure water) when compared to horizontal coils, ceteris paribus (Fig. 5). However, they failed to critically analyse these results.

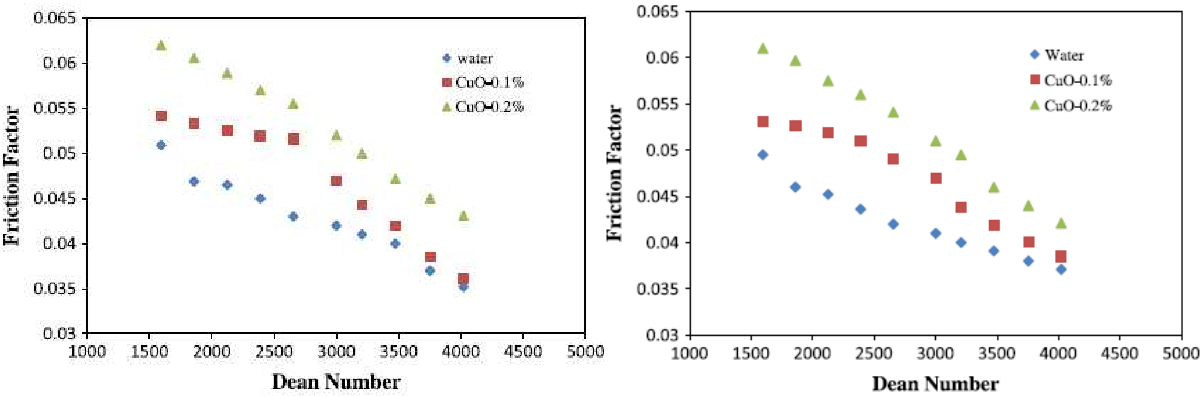


Figure 5: CFD simulation of the CuO nanoparticles in water, friction factor for: (a) Horizontal orientation (b) Vertical orientation (Kannadasan et al. [18], Figs. 7&8)

Authors	Heat exchanger type/Flow regime	Nanofluid	Volume or weight conc.	Main conclusions, proposed correlation and mean error
		Copper & Copper oxide nanoparticles & Water		
Akbaridoust et al. (2003) [78]	Laminar 200<Re<1000	Cu/H_2O	0.1-0.2% (VF)	The pressure drop increased with increasing particle volume concentration and mass flow rate. No correlation.
Suresh et al. (2011) [75]	Horizontal with smooth and dimpled surface Turbulent ID=4.85mm OD=6.3mm 2500<Re<6000	CuO/ H_2O	0.1-0.3% (VF)	Quasi no increase in the pressure drop with nanofluids over that with distilled water. $f = 0.1648Re^{0.97}(1+VC)^{107.89}\left(1+\frac{PR}{d}\right)^{-4.463}$ (±20%)

Kannadasan et al. (2012) [18]	Horizontal & vertical Turbulent ID= 9mm OD=10.5mm D=124mm 1600<De<4000	CuO/ H_2O	0.1-0.2% (VC)	For both horizontal and vertical coils, an increase in the friction factor was measured with higher nanoparticle volume concentrations. Higher Dean numbers decreased the friction factor. For 0.2% volume concentration, the friction factor, when compared to water flow only, increased by 24% and 23% for horizontal and vertical orientations respectively. No correlation.
Copper oxide nanoparticles & Oil				
Hashemi and Akhavan-Behabadi (2012) [73]	Horizontal Laminar ID=14.37mm D=324mm Re<125 700<Pr<2050	CuO/Oil	0.5-2% (WC)	The pressure drop increased with increasing particle volume concentration and Reynolds numbers. For 2% WC, the pressure drop, when compared to oil flow only, increased by 20.3%. For a straight tube this was measured as 13.2% No correlation.
Multi-Walled Carbon NanoTubes nanoparticles & Oil				
Fakoor-Pakdaman et al. (2012) [5]	Vertical Laminar ID=15.6mm 220<D<320mm 100<Re<1800	Multi-Walled Carbon NanoTubes/Oil	0.1-0.4% (WC)	Performance index, increases with higher nanoparticle weight concentrations. No correlation.
Fakoor-Pakdaman et al. (2013) [72]	Vertical Laminar ID=15.6mm 220<D<320mm 10<Re<2000	Multi-Walled Carbon NanoTubes/Oil	0.1-0.4% (WC)	The pressure drop increased with increasing particle volume concentration and mass flow rate. 31% pressure drop increase over the base fluid at the highest concentration. Pressure drop is independent of the coil pitch whilst a decrease in the curvature ratio results in a lower pressure drop. Pressure drop in the coiled tube is up to 2.5 times higher than that in a straight tube. $\frac{f_{TP}}{f_{bf}} = [1 + 0.031(logDn_m)^4](1 + 10WC)^{4.9}$ where: $Dn_m = Re\left\{\delta^{-1}\left[1 + \left(\frac{p}{\pi D}\right)^2\right]\right\}^{-0.5}$ (±20%)
Aluminium oxide & titanium dioxide nanoparticles & Water				
Kahani et al. (2013a) [13]	Horizontal Laminar d=7mm D=70,140mm 500<Re<4500 5.89<Pr<8.95 115.3<He<1311.4	Al_2O_3/H_2O TiO_2/H_2O	0.25-1.0% (VC)	The pressure drop increased with increasing particle volume concentration and mass flow rate. $\Delta P_{TP} = 5.584He^{1.36}VF^{0.446}d^{0.163}RA^2$ where: $He = De\left[1 + \left(\frac{p}{2\pi D}\right)^2\right]^{0.5}$
Kahani et al. (2013b) [77]	Horizontal Laminar d=7mm D=70,140mm 500<Re<4500 5.89<Pr<8.95 115.3<He<1311.4	Al_2O_3/H_2O	0.25-1.0% (VC)	The pressure drop increased with increasing particle volume concentration and mass flow rate. A decrease in the curvature ratio results in a lower pressure drop, whilst the coil pitch had a minimal effect on the pressure drop. No correlation.
Mukesh Kumar et al. (2013)	Laminar 5100<Re<8700 ID=9mm	Al_2O_3/H_2O	0.1-0.8% (VC)	Generally, the pressure drop increased with increasing particle volume concentration and mass

[3]	OD=10.5mm D=93mm			flow rate. Rate of pressure drop increase was higher when the Dean number increased. No correlation.
Wu et al. (2013) [14]	Double pipe Laminar & Turbulent ID_{it} =13.28mm ID_{ot} =26mm D=254mm 800<Re<10000	Al_2O_3 /H_2O	0.78-7.04% (WC)	Mitigation of secondary flow with nanofluids. Friction factor decreases with higher Reynolds numbers for laminar flow while it increases slowly for higher Reynolds numbers in turbulent flow. Their friction factor was predicted through the use of the Ito [6] (laminar) and Seban and McLaughlin [76] (turbulent) equations for single-phase flow. (±30%)
Mukesh Kumar et al. (2014) [74]	Laminar $0.03<\dot{m}<0.05$ 1600<De<2700 ID=10mm OD=11.5mm D=93mm	Al_2O_3 /H_2O	0.1-0.8% (VC)	Generally, the pressure drop increased with increasing particle volume concentration and mass flow rate. Rate of pressure drop increase was higher when the Dean number increased. Hence, no significant increase in the pressured drop with 0.1% and 0.4% nanofluid particle volume concentration. No correlation.

Table 4: Review of experimental studies of the pressure drop characteristics of nanofluids in helically coiled tubes

4.2 Numerical Studies

Research on the pressure drop and the general thermo-physical properties of nanofluids in helically coiled heat exchangers is a relatively new development. In fact, the earliest research in the pertinent literature was reported by Sasmito et al. [23] in 2011. The ANSYS Fluent commercial software package was used in all of the studies reviewed and summarised in this section. Therefore, the fluid flow and heat transfer governing equations, given in Eqs. (14-16), were solved to measure the pressure drop and temperature distribution along the helically coiled tubes.

Continuity:

$$\frac{\partial \rho}{\partial T} + \nabla.(\rho V) = 0 \tag{14}$$

Momentum:

$$\rho\frac{\delta V}{\delta T} + \nabla.\tau_{ij} - \nabla P + \rho FB = 0 \tag{15}$$

Energy:

$$\rho\frac{De}{\delta T} + \rho(\nabla.V) = \frac{\partial Q}{\partial T} - \nabla.Q + \phi_d \tag{16}$$

where V is the fluid velocity, FB are the body forces, ϕ_d is the energy dissipation term and Q is the heat transfer by conduction. The numerical analysis studies reviewed in this section assumed that the nanofluid flow through the tubes is incompressible, single-phase and fully developed, both hydrodynamically and thermally. The SIMPLEC algorithm was used by a number of studies to solve the flow field [21, 31], whilst for turbulent flow modelling, the Standard Turbulence k-ε model as proposed by Launder and Spalding, was used [79]. The thermo-physical properties of the nanofluids were obtained using the equations given in Eqs. (17-28) [21, 31].

Density:

$$\rho_{nf} = (1 - VF)\rho_{bf} + VF\rho_{np} \quad \textbf{(17)}$$

Heat capacity:

$$(\rho C_p)_{nf} = (1 - VF)\left(\rho C_p\right)_{bf} + VF\left(\rho C_p\right)_{np} \quad \textbf{(18)}$$

Effective thermal conductivity:

$$k_{eff} = k_{static} + k_{Brownian} \quad \textbf{(19)}$$

Static thermal conductivity:

$$k_{static} = k_{bf}\left[\frac{k_{np} + 2k_{bf} - 2(k_{bf} - k_{np})VF}{k_{np} + 2k_{bf} + (k_{bf} - k_{np})VF}\right] \quad \textbf{(20)}$$

Brownian thermal conductivity:

$$k_{Brownian} = 5E4\beta VF\rho_{bf}C_{p,bf}\sqrt{\frac{\kappa T}{2\rho_{np} r d_{np}} f(T, VF)} \quad \textbf{(21)}$$

where the Boltzmann constant, κ = 1.3807E-23 J/K

Modelling function for CuO, 1%≤VF≤6%, β:

$$\beta = 9.881(100VF)^{-0.9446} \quad \textbf{(22)}$$

Modelling function for Al_2O_3, 1%≤VF≤10%, β:

$$\beta = 8.4407(100VF)^{-1.07304} \quad \textbf{(23)}$$

Modelling function for ZnO, 1%≤VF≤7%, β:

$$\beta = 8.4407(100VF)^{-1.07304} \quad \textbf{(24)}$$

Modelling function for SiO_2, 1%≤VF≤10%, β:

$$\beta = 1.9526(100VF)^{-1.4594} \quad \textbf{(25)}$$

Modelling function, f (T,VF):

$$f(T, VF) = (2.8217E - 2)VF + (3.917E - 3)\left(\frac{T}{T_o}\right) + \left(VF(-3.0699E - 2) - (3.91123E - 3)\right) \quad \textbf{(26)}$$

Dynamic viscosity:

$$\frac{\mu_{eff}}{\mu_{bf}} = \frac{1}{1-34.87\left(\frac{di_{np}}{di_{bf}}\right)^{-0.3} VF^{1.03}} \quad \textbf{(27)}$$

where the equivalent diameter of the base fluid molecule is:

$$di_{bf} = \left(\frac{6M}{N\pi\rho_{bf}}\right) \quad \textbf{(28)}$$

Table 5 summarises the numerical studies on the pressure drop characteristics with nanofluid laminar and turbulent flow in helically coiled tubes. The studies summarised in this section are in reasonable agreement with the data reported in the experimental studies reviewed in Section 4.1. Furthermore, some controversy also characterises the reviewed numerical studies. Hence, whilst a number of authors [21, 22, 80] reported an increase in the frictional pressure drop with higher nanoparticle volume concentrations as well as higher nanofluid pressure drops over that of the base fluid only, Sasmito et al. [23] reported that at a nanoparticle volume concentration of 1%, the calculated pressure drop was lower than that for pure water. Sasmito et al. attributed these results to the fact that at low volume concentrations, the nanoparticles have a minimal effect on the fluid viscosity, whereas the temperature effects on the nanofluid thermo-physical properties are more significant. Intriguingly, Aly [30] also reported that the aluminium oxide nanoparticles with a maximum volume concentration of 2% did not result in an increase to the resultant pressure drop. Therefore, their data was in reasonable agreement with the single-phase friction factor correlations by Mishra and Gupta and Ito [7, 6]. Similar results were also reported by Suresh et al. and Wu et al. [75, 14] through their experimental investigations. One of the advantages of numerical simulations is the minimal cost incurred for each simulation. Hence, Narrein and Mohammed [21] were able to investigate the flow characteristics of a combination of oil, ethylene glycol and water based nanofluids. They reported that due to the high viscosities of oil based nanofluids, the latter resulted in the highest calculated pressure drop when compared to ethylene glycol and water based nanofluids. Narrein and Mohammed also reported higher pressure drops with decreasing nanoparticle diameters which were more intense at higher fluid velocities. These results were attributed to the resultant increase in the fluid viscosity which could result in higher wall shear stresses.

Through their numerical investigations, Elsayed et al. [29] and Moraveji and Hejazian [80] presented correlations for the prediction of the friction factor. The former's correlation takes the form of a ratio of the nanofluid flow pressure drop in a helically coiled tube to that in a straight tube, ceteris paribus. This correlation is a function of the fluid properties, represented through a modified Reynolds number and the tube geometry, represented through the curvature ratio. The correlation presented by the latter authors is markedly different as it is not a function of the coil geometry. In fact, it is a sole function of the fluid properties, represented through the nanoparticle volume concentration and the Reynolds number.

Mohammed and Narrein [31] and Aly [30] investigated the nanofluid pressure drop characteristics with the coil geometry. In agreement with the experimental results reported by Kahani et al. [13] and Fakoor-Pakdaman et al. [5], an increase in the nanofluid frictional pressure drop was reported with a reduction in the coil diameter, whilst the pressure drop decreased with larger tube diameters. As reported in Section 4.1, these results can be attributed to the reduction in the centrifugal forces with larger helix diameters. As illustrated in Fig. 6, the pressure drop was also reported to be independent of the helix pitch.

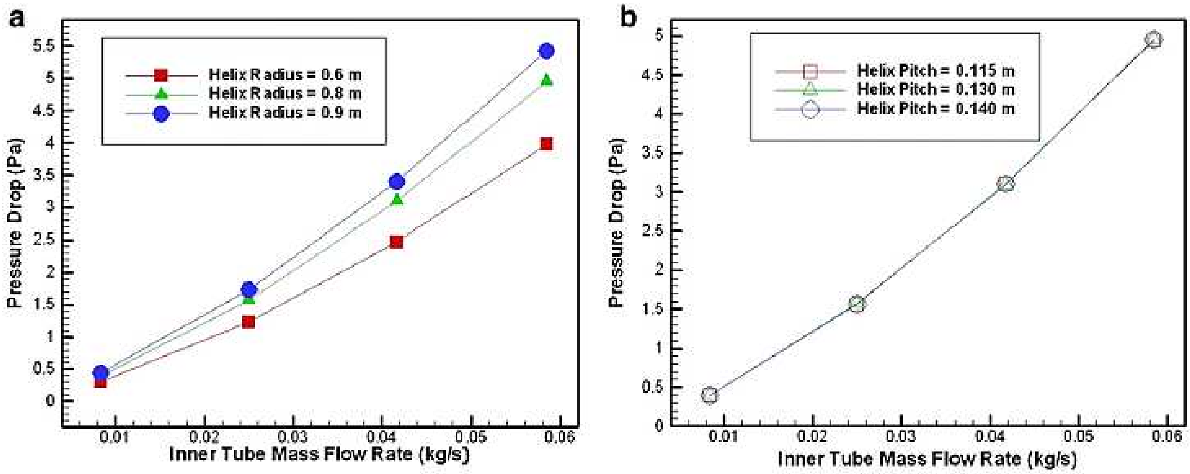


Figure 6: CFD simulation of the CuO nanoparticles in water, pressure drop for: different helix radii (a), pitches (b) (Mohammed and Narrein [31], Fig. 7a&b)

Authors	**Heat exchanger type / Flow regime**	**Nanofluid**	**Volume or Weight Concentration**	**Main conclusions, proposed correlation and mean error**
Sasmito et al. (2011) [23]	Square tubes Laminar	Al_2O_3/H_2O CuO/H_2O	0-1% (VC)	Helical coil resulted in the highest pressure drop when compared to straight, conical and in-plane coiled tubes. At 1% nanoparticle concentration, the pressure drop was lower than that for water. No correlation.
Jamshidi et al. (2012) [22]	Laminar 1700<Re<2500	Al_2O_3/H_2O	1-3% (VC)	Friction factor increased with higher nanoparticle volume concentrations and lower Reynolds numbers. No correlation.
Mohammed and Narrein (2012) [31]	Laminar $0.01<\dot{m}<0.06$ d=32,42,52mm D=600,800,900 mm	CuO/H_2O	4% (VC)	Pressure drop increased with a reduction in the coil diameter and decreased with larger tube diameters. No correlation.
Narrein and Mohammed (2013) [21]	Laminar $0.01<\dot{m}<0.06$	CuO/H_2O/Engine Oil/Ethylene glycol Al_2O_3/H_2O/Engine Oil/Ethylene glycol ZnO/H_2O/Engine Oil/Ethylene glycol SiO_2/H_2O/ Engine Oil/Ethylene glycol	1-4% (VC)	Due to the different densities, SiO_2 had the highest pressure drop followed by Al_2O_3, ZnO, CuO. Due to higher viscosities, pressure drop increased with higher nanoparticle concentrations and decreasing nanoparticle diameters. Due to higher wall shear stresses this effect is more intense at higher fluid velocities. Pressure drop for oil based nanofluids resulted in the highest pressure drop followed by ethylene glycol and water based nanofluids. No correlation.

Elsayed et al. (2014) [29]	Turbulent 20000<Re<50000	Al_2O_3 /H_2O	0-3% (VC)	Pressure drop in coils with nanofluids was measured as a ratio to that of water in a straight tube. $\frac{f_{nf,c}}{f_{bf,s}} = \frac{f_{nf,c}}{f_{nf,s}} = \frac{4(0.08Re_{nf}^{-0.25} + 0.012\delta^{0.5})}{0.316Re_{nf}^{-0.25}}$ (±5%)
Aly (2014) [30]	Tube-in-tube Turbulent D=180,240,300mm $2<vf_{it}<5$LPM $10<vf_{it}<25$LPM	Al_2O_3 /H_2O	05-2% (VC)	Single-phase friction factor correlations by Ito and Mishra and Gupta are also valid for nanofluids. Friction factor increased with curvature ratios. No pressure drop increase with nanoparticle volume concentration. No correlation.
Moraveji and Hejazian (2014) [80]	Laminar d=14.4mm D=324mm	CuO/Oil	0.5-2% (WC)	Pressure drop with 2% nanoparticles is 11% higher than that for the base fluid, ceteris paribus. Pressure drop for the base fluid in the helical coil was 3 times higher than that in a straight tube. $f = 1.9254Re^{-1.223}(1+VC)^{0.00781}$ (±15%)

Table 5: Review of numerical studies on the pressure drop characteristics of nanofluids in helically coiled tubes

5. Scope for further research

As discussed in Section 3, recent studies have suggested that the introduction of small air bubbles (b<0.5mm) in turbulent flow could result in a substantial reduction of the frictional pressure drop over that of pure water. A number of studies have investigated this concept for two-phase flow in straight tubes [66, 67] whilst Saffari et al. [17] presented the sole study for helically coiled tubes. Saffari et al.'s conclusions are in substantial disagreement with the findings reported by the majority of investigations on air-water two-phase flow, where the introduction of the second phase was reported to enhance the frictional pressure drop. In fact, the pressure drop multiplier in Eq. (3), as originally defined by Lockhart and Martinelli, was typically reported to be in excess of unity. Such evident controversies should be addressed through further research on the frictional pressure drop due to bubbly air-water two-phase flows in coiled tubes.

As reported in a study published by one of the authors of the present study [81], air-water two-phase flow through helically coiled tube heat exchangers characterises many modern condensing sealed heating systems. Bubbly flow finds its origins in the supersaturated conditions at the heat exchanger wall. Whilst numerous studies investigated the air-water bubbly flow pressure drop, these studies were developed through the insertion of artificial bubbles. This presents significant scope for further research on the bubbly flow two-phase frictional pressure drops, where bubbles nucleate and detach at the tube wall and therefore, the volumetric void fraction at the return and flow ends of the heat exchanger would be dissimilar. Moreover, in view of the fact that numerous studies have suggested enhanced heat transfer coefficients with the addition of nanoparticles to water [75, 18], there is scope for further research on the three-phase air-water-nanoparticles frictional pressure drop in helically coiled tube heat exchangers.

Due to the recent development of nanofluids as a means for the enhancement of the fluid heat transfer characteristics, there is ample scope for further research in this field of study. The majority of the pertinent studies available in the open literature have focused their investigations on the resultant heat transfer characteristics. This is evidenced by the paucity of correlations presented for the calculation of the frictional pressure drop when compared to those available for the heat transfer coefficient [57]. Further investigations should be developed to address the conflicting results for the impact of the nanoparticle concentration on the frictional pressure drop as outlined in Section 4. Studies should also be developed for the purpose of investigating the frictional pressure drop as a sole function of the type of nanoparticles. Such studies are deemed necessary in view of the conclusions made by Hashemi and Akhavan-Behabadi [73] who reported that due to their typical spherical shape, copper oxide nanoparticles could yield lower frictional pressure drops. The open literature presents a single study on the two-phase frictional pressure drop as a function of the coil orientation [18], where horizontal coils were reported to yield marginally higher pressure drops. However, the authors failed to provide a detailed appraisal for the latter results. Furthermore, this study was developed with copper oxide nanoparticles in water and hence, further studies are required to investigate the impact of the coil orientation with widely used nanoparticles and base fluids, such as aluminium oxide and oil respectively. Moreover, the pertinent literature failed to comprehensively investigate the distribution of the secondary phase (nanoparticles) in coiled tubes. Therefore, whilst Wu et al. [14] reported that nanofluid flow in coiled tubes did not yield a significant phase separation, no other relevant studies investigated this pertinent flow characteristic. Such avenues for future fundamental research will complement and facilitate the research and development of high efficiency heat exchangers as well as open new opportunities for industry-led heat exchanger tube design initiatives, whereby the distribution of the secondary phase could be manipulated for optimised system efficiencies.

6. Conclusions

This paper has provided a review on all the investigations available in the pertinent literature on the two-phase pressure drop characteristics in helically coiled tubes. Therefore the relevant investigations on steam-water flow boiling, R-134a evaporation and condensation, air-water flow and nanofluids have been critically reviewed. The correlations for the calculation of the frictional two-phase pressure drop were also tabulated with the corresponding system parameters. Whilst being more complex than single-phase flow, two-phase flow is more relevant to numerous engineering applications. Therefore a comprehensive understanding of the two-phase pressure drop is necessary to ensure that no excessive, energy consuming, pumping power is required. The pertinent conclusions outlined in the current study can be summarised through the following points:

- For steam-water flow boiling, the frictional pressure drop increases with the vapour quality and mass flux whilst it decreases with higher system pressures. The appreciation of the frictional pressure drop with the vapour quality is more significant at qualities below 0.3. The curvature ratio does not appear to have a significant influence on the two-phase flow boiling frictional pressure drop multiplier whilst there is some controversy surrounding the influence of the coil orientation and heat flux. Some studies have correlated their data to widely cited correlations for straight tubes such as those given by: Lockhart and Martinelli, Martinelli and Nelson and Chen.
- For R-134-a evaporation and condensation in helically coiled tubes, the curvature ratio appears to have some impact on the resultant frictional pressure drop for R-134a flow in non-miniature helically coiled tubes (d>1mm). The pertinent investigations have also concluded that the frictional pressure drop increases with higher vapour qualities and refrigerant mass fluxes, whilst the tube orientation has no significant impact on the

pressure drop. The majority of the correlations presented are a function of the Lockhart and Martinelli parameter.

- The pertinent literature presents numerous correlations for the prediction of the two-phase frictional pressure drop with air-water bubbly flow. The majority of investigations have correlated their data using the original or modified Lockhart and Martinelli correlation for straight tubes, whilst other authors presented their own empirical correlations. The early investigations reported the two-phase pressure drop to be independent of the coil design parameters such as the curvature ratio and the helix angle, whilst more recent studies have suggested a marginal impact on the two-phase pressure drop by the latter parameters. The frictional pressure drop as a function of the air volumetric void fraction remains indeterminate due to conflicting results.
- Few correlations are available to calculate the frictional pressure drop with nanofluids. The majority of experimental and numerical investigations on nanofluids flowing in helically coiled tubes have reported a significant increment (up to 3.5 times) in the frictional two-phase pressure drop over that of pure water in straight tubes. Such conclusions were mainly attributed to the higher relative mixtures and densities as well as the secondary flow formed in curved tubes. Due to the dominance of the viscosity effects at low fluid velocities, the impact of the nanoparticle concentration on the two-phase frictional pressure drop is stronger at higher Reynolds numbers. The frictional pressure drop was also reported to be a function of the curvature ratio and the coil orientation with marginally larger pressure drops for horizontal coils. Controversy surrounds the impact of nanofluids on the frictional pressure drop, where some investigations reported a decrease in the resultant pressure drop while other studies reported the pressure drop to be quasi-identical to that with pure water in coiled tubes.

This paper has also outlined areas for further research, principally in the fields of air-water and nanofluids two-phase flows and three-phase air-water-nanoparticles flow. Such studies could take the form of fundamental research as well as industrial research and development initiatives with the aim of enhancing the system efficiencies through the reduction of the two-phase pressure drop.

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Notation List

A	Heat transfer area (m^2)
b	Bubble diameter (m)
bf	Base fluid (-)
c_p	Specific heat (J/kgK)
C	Constant depending on the flow condition of the vapour and liquid i.e. 5 for laminar and 20 for turbulent flows (-)
CS	Stratified speed of sound (m/h)
d	Tube diameter (m)
D	Helix diameter (m)
De	Dean number (-)
Di	Diameter of nanoparticle (m)

E	Corrugation depth (m)
f	Friction factor (-)
Fr	Froude number (-)
FB	Body forces (N)
g	Acceleration due to gravity (m/s)
G	Mass flux (kg/m^2s)
h*	Mean heat transfer coefficient after applying enhancement techniques (Nanoparticles and helical coils) (W/m^2K)
h_{st}	Mean heat transfer coefficient inside a straight tube with base fluid only
He	Helical coil number (-)
H	Coil vertical height (m)
HD	Hydraulic diameter (m)
ID	Inner tube diameter (m)
k	Thermal conductivity (W/mK)
L	Length (m)
$\dot{m}$	Mass flow rate (kg/s)
M	Molecular weight (mol/g)
MC	Mass concentration (kg/m^3)
N	Number of bends (-)
OD	Outside tube diameter (m)
p	Pitch (m)
P	System pressure (-)
Pr	Prandtl number (-)
PI	Performance index (-)
PR	Pitch ratio (-)
ΔP^*	Mean pressure drop after applying enhancement techniques (Nanoparticles and helical coils) (Pa)
ΔP_{st}	Mean pressure drop inside a straight tube with base fluid only (Pa)
ΔP_{TP}	Two-phase frictional pressure drop (Pa)
q	Heat flux (kW/m^2)
Q	Heating power (kW)
rd	Radius of nanoparticle (m)
Re	Reynolds number (-)
RA	Adjusted correlation coefficient (-)
RE	Average relative error (%)
RMS	Root mean square (-)
S	Slip ratio (-)
SF	Shape factors (-)
T	Temperature (^0C)
U	Superficial velocity (m/s)
v	Specific volume (m^3/kg)
vf	Volume flow rate (m^3/s)
V	Flow velocity (m/s)
VC	Volume concentration (-)
VF	Void fraction (-)
WC	Mass concentration as fraction (-)
x	Steam quality (-)
z	Vertical elevation (m)

Greek symbols

β Helix angle (^{o})
γ Friction factor multiplier (-)
δ Curvature ratio: Internal tube radius d_i/mean coil radius D (-)
ε Volumetric quality, liquid volume flow rate to total volume flow rate (-)
η Performance index (-)
κ Boltzmann constant (J/K)
μ Dynamic viscosity (Pa/s)
ρ Density (kg/m^3)
σ Surface tension (N/m)
τ Sheer stress (N/m^2)
ϕ_d Dissipation term (m^2/s^3)
ϕ_l Two-phase multiplier (-)
χ Lockhart-Martinelli parameter (-) $\chi = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$
ψ The unevenness correction factor (-)

Subscripts

Acc Acceleration
bf Base fluid
c Coil
crit Critical
eff Effective
f Frictional
g Gas properties/flow
grav Gravity
g,tt Gas phase turbulent flow
it Inner tube
l Liquid properties/flow
l,tt Liquid phase turbulent flow
m Mixture
n nth
nf Nanofluid
np Nanoparticle
NS No slip conditions
o Oil
ot Outer tube
ref Refrigerant side
s Straight tube
sat Saturation conditions
st Single-phase conditions
TP Two-phase conditions
tt Turbulent liquid and vapour flow
v Vapour
v,tt Vapour phase turbulent flow
wt Water

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