CFD Letters

CFD Analysis of Dual-Phase Flows Inside Helically Coiled Tubes in Vapour Compression Micro-Refrigerator

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Parashurama Siddegowda[1,*], Govindegowda Mundur Sannappagowda[2], Srinath Mandya Sridharamuthy[3], Ramesha Devarahalli Kempegowda[4]

[1] Mechanical Engineering Department, JSSATE, Bengaluru, 560 060, India
[2] Principal, Vivekananda college of Engineering and Technology, Puttur, D.K, India
[3] Professor, Aeronautical and Automobile Engineering Department, Manipal Institute of Technology, Manipal Academy of Higher Education , Manipal, 576104, India
[4] Professor, Department of Industrial and Production Engineering, MCE, Hassan, 573 202, India

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ABSTRACT

This study focus on experimental exploration of heat transfer by helical coil. Helical pipe of circular cross section is used widely in variety of applications due to inexpensive and easiness in production. Enhancement of heat transfer by helical pipe is more prominent than in straight pipe, which has supported by many literatures. Controlling pressure drop (which is also a significant factor) to enhance heat transfer in the helical pipe. Modeling of dual phase, especially fluid – vapor flow under adiabatic conditions inside a flat helical tube utilizing CFD examination is difficult with the accessible dual phase models in fluent due to constantly changing flow patterns. In the present investigation, CFD examination of two phase flow of refrigerants inside a level helical pipe of inward diameter across, 20 mm and 100mm length is completed utilizing homogeneous model in adiabatic conditions. The refrigerants considered are R290 and R134a. The investigation is performed at saturation temperatures with fluid flow rates to assess the nearby frictional weight drop. Flow characteristics, velocity, pressure drop, turbulent kinetic energy, and temperature contours along the total length of the coil are to be analysed. Velocity, pressure drop, turbulent kinetic energy, and temperature distribution of multi-phase flow inside the pipe also to be analyzed. By utilizing homogeneous model, normal properties are acquired for each of the refrigerants that are considered as multi-stage pseudo liquid.

1. Introduction

It has been generally described in literature that heat transfer in helical pipe are higher when contrasted with that in straight tubes. Due to the reduced structure and high heat transfer coefficient, helical pipe heat exchangers are broadly utilized as a part of mechanical applications, for example, control era, atomic industry, process plants, heat recovery system, refrigeration,

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* Corresponding author.
E-mail address: parashurams@rediffmail.com (Parashurama Siddegowda)

sustenance industry, and so on [1]. Heat transfer rate of helically coiled heat exchanges is significantly larger because of the secondary flow pattern in planes normal to the main flow than in straight pipes. Modification of flow is due to the centrifugal forces caused by the curvature of the tube. Many studies have been conducted to analyse the heat transfer rate of coiled heat exchangers in laminar and turbulent flow regimes. Basically two different concepts to increase the rate of heat transferred are active and passive method. Active method is mostly for straight pipes, in case of passive techniques heat transfer enhancement by chaotic mixing in helical pipes has great importance and investigated by Kumar and Nigam [2]. Experimental investigations have been conducted by CFD analysis in helical pipe tube possibly increases the heat transfer rate because of the developed swirling motion. Basic objective of this study is to investigate the impact of CFD analysis with dual-phase flow of the corrugation for the inner side heat transfer rate in case of helical coil.

1.1 Basic Components of a vapor compression system

a) Compressor: It is motor driven; it sucks vapor refrigerant from evaporator and compresses.
b) Condenser: High pressure vapor refrigerant is condensed into liquid form in the condenser using cooling medium such as water.
c) Expansion Valve: High pressure refrigerant is throttled down to evaporator pressure; rate of flow is metered
d) Evaporator: A cooling chamber in which products are placed; how pressure liquid refrigerant flows in the coils of evaporator and absorbs heat from products; the refrigerant vaporizes and leaves for compressor.

1.2 Selection of condenser for a VCR system

It is main component of the system, it dissipates heat absorbed by the refrigerant during evaporation (refrigeration effect) and compression (heat of compression).
a) Air cooled
b) Water cooled
c) Evaporative type

These are the types of condensers on the basis of cooling used to dissipate heat.

1.3 Characteristics of helical coil

In the present investigation, we consider helical pipe which are vertically situated, i.e., where the pipe axis is vertical. Figure 1 stretches the representation of the helical pipe. The coil has an internal diameter across 2r. The coil distance across (measured between the focuses of the channels) is spoken to by 2Rc. The separation between two adjoining turns is called pitch, H. The coil width is additionally called as pitch circle distance across (PCD). The proportion of pipe measurement to coil width (r/Rc) is called curvature ratio. The proportion of pitch to created length of one turn (H/2πRc) is named non-dimensional pitch, λ. Consider the projection of the loop on a plane going through the hub of the coil. The angle, with projection of one turn of the loop makes with a plane opposite to the hub, is known as the helix angle, α. For any cross-segment of the pipe, made by a plane going through the helical coil axis, the side of pipe divider closest to the loop hub is named as internal side and the most remote side is named as external side. Using Dean Number, the fluid flow in helical pipe can be distinguished, similar to Reynolds Number. The Dean number, De is well-defined as [1]

$$D_e = R_e \sqrt{r/R_c} \tag{1}$$

where, Re is the Reynolds number = $\dfrac{2rU_{av}\rho}{\mu}$

Numerous scientists have distinguished that a perplexing stream flow pattern exists in a helical pipe because of which heat transfer enhancement is attained. The curvature of coil represents the gal force while the pitch (or helix point) impacts the torsion to which the liquid is subjected to. The radiating power brings about improvement of secondary flow stream as indicated by Darvid *et al.,* [3]. Due to curvature impact, the liquid flow streams in the external side of pipe moves faster than the liquid streams in the inward side of pipe (as shown in Figure 2). The distinction in velocity sets-in auxiliary flow, whose structural pattern changes with the Dean number of the stream.
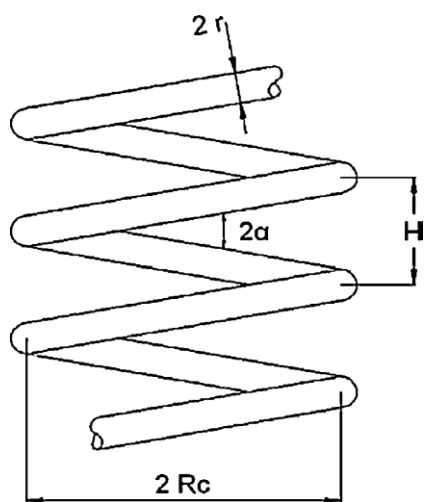
Fig. 1. (a) Schematic figure of the helical coiled tube heat exchanger with the geometric parameters
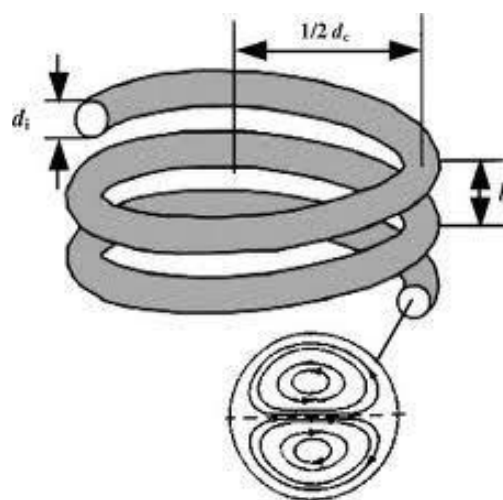
Fig. 2. Secondary flow in helical coil

The transformation of laminar system to turbulent system takes place at higher Reynolds number than compared with straight pipe.

1.4 Design of helical condenser

Table 1
Helical condenser coil parameters

Diameter of design coil D (mm)	200
Diameter of tube W (mm)	20
Coil pitch (mm)	30
turns	3
Height H (mm)	100
Wall thickness (mm)	0.81

1.5 The Volume of Fluids (VOF) Model

In the volume of fluid (VOF) model (as shown in Figure 3), a solo set of conservation equations is combined with the phases and the volume fraction of every phase is followed in all computational cells through the domain.
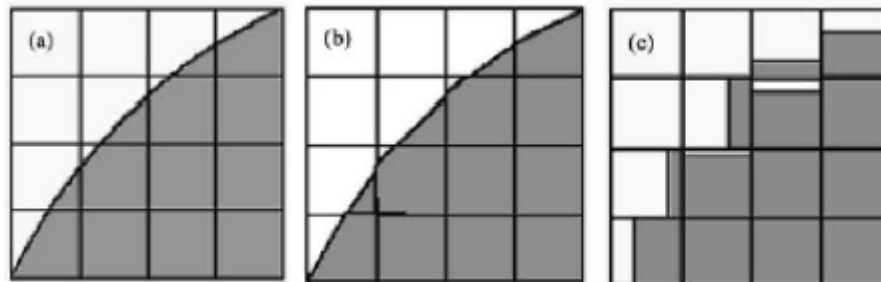
Fig. 3. VOF interface reconstruction methods (a) Actual interface method (b) Piecewise Linear Interface Calculation (PLIC) method (c) Simple Line Interface Calculation (SLIC)

Essential one dimensional model of single segment dual phase flow is created for the stratified flow stream where each stage is in contact with the channel and has a typical interface. The subsequent momentum and energy equations are additionally understood for pressure drop utilizing distinctive models like Homogeneous flow stream model and Separated stream demonstrate.

Expectation of dual phase pressure drop inside a tube is of fundamental significance to the plan and advancement of refrigeration, air-conditioning and heat pump system. From the isolated flow display, the frictional pressure drop for dual phase, two segment, isothermal stream in horizontal tubes was at first created by Lockhart and Martinelli in 1944 [4] utilizing the dual phase multiplier. A later augmentation of their work to cover the accelerative segment come about into sure understanding Martinelli-Nelson connection for the expectation of pressure drop for forced flow movement steaming and condensation. Afterwards, the computation system for two stage friction multiplier was created by was created by Thom [5], Baroczy [6] and Chisholm [7]. Also, the accompanying suggestions were made:

$\left(\mu_l / \mu_g \right) < 1000 \; and \; G < 2000 \; kg/m^2 \; s$, Friedel relationship would be used.

$\left(\mu_l / \mu_g \right) > 1000 \; and \; G > 100 \; kg/m^2 \; s$, Chisholm relationship would be used.

$\left(\mu_l / \mu_g \right) > 1000 \; and \; G < 100 \; kg/m^2 \; s$, Lockhart – Martinelli relationship would be used.

Kattan *et al.,* [8] isolated the information by flow regimes utilizing the flow structure delineate the authors found that prescient techniques work diversely with shifting the flow regime, since the models are not ready to catch totally the impacts of the varieties in flow structure. Recently, Moreno Quiben and Thome [9,10] distributed a work in which they made an extensive review to run precise tests. At that point utilizing another flow pattern delineates Wojtan *et al.,* [11], they assembled a flow pattern based model for anticipating pressure drops. However, there is very little information revealed in the paper on design and analysis of dual phase flow using CFD analysis. Akeel Mohammed Ali Morad [12] presented an analytical model for the calculation of two-phase pressure drops for homogeneous separated flow for circular tube condensers.

By using CFD analysis for the dual-phase flows inside helically coiled tubes in VC micro refrigerator the heat transfer rate can be easily derived. The movement of the fluid particles can be easily traced by the CFD analysis.

1.6 Nature of turbulent flow and heat transfer in helical coils

Heat transfer enhancement analysis of refrigerant passes through inside the helical coil is carried out using CFD package FLUENT version 15.0. As a typical case, coil of PCD = 200mm and coil pitch of 30mm is obtainable from discussion. Inside diameter of the pipe used in the coil is 20mm.The design structure model (as shown in Figure 4) and the mesh will be created by mesh modeler of the FLUENT package. Boundary mesh was made for the pipe fluid volume. The grid used for this analysis is given in Figure 5.
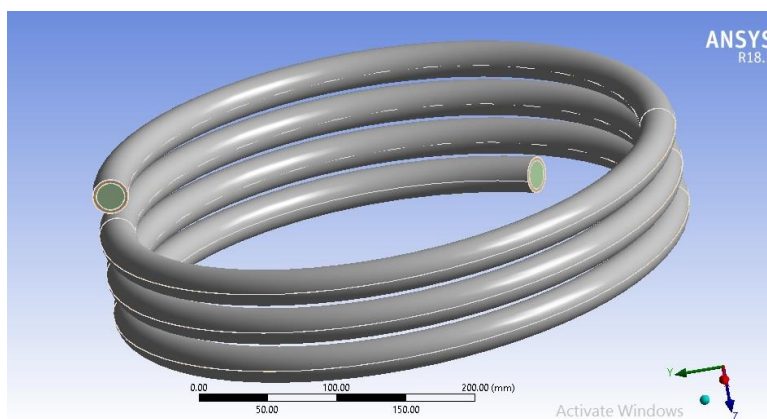

Fig. 4. design structure model

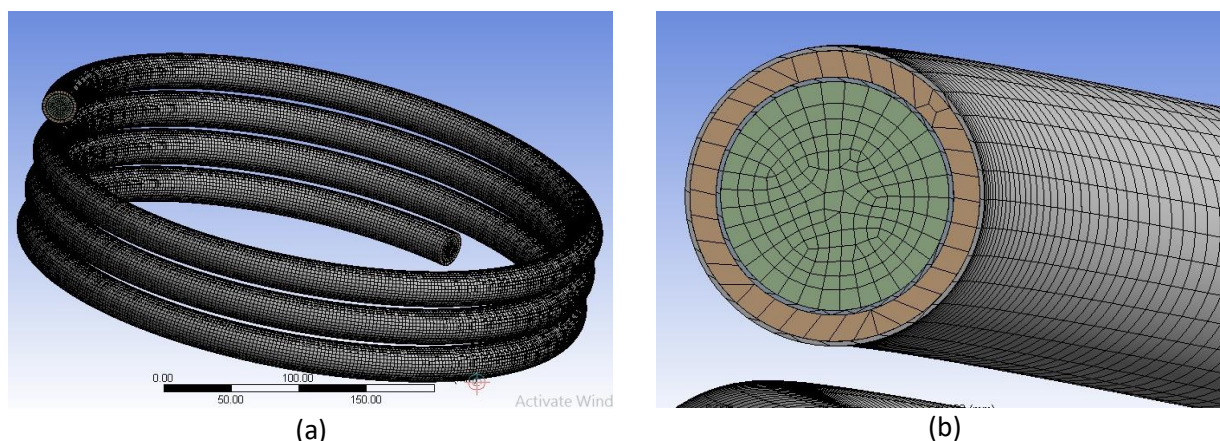|	
(a)	(b)

Fig. 5. Mesh structure (a) Grid of the helical pipe used for analysis (b) The grid at any cross-section of the helical pipe

Pressure velocity connection was done by SIMPLEC scheme. Momentum equations were solved by QUICK scheme. The attainable k–ε turbulence model is used in solving computation. This scheme is perfect for flows concerning rotation, boundary layers under strong opposing pressure gradients, separation, and recirculation.

Power law order of discretization is applied for dissipation rate equations and turbulent kinetic energy. Convergence criterion used was 1.0×10^{-5} for continuity, velocities, k, and ε. Temperature dependent properties as polynomial functions were used for refrigerant. Characteristics of the computational design equations are obtainable in Jayakumar *et al.,* [13]. For third order energy

equation, implementation of the QUICK discretization scheme was employed. Convergence criterion for energy balance was 1.0 x 10^{-07}.

2. Separated Flow Model

The fundamental conditions for the separated flow model are not subject to the specific flow arrangement. It is accepted that the velocity of each phases are consistent. Pressure drop amid In-Tube build up can be acquired from the two stage stream force condition or condensation condition in view of the separated flow demonstrates. It is accepted that velocity of each phase are consistent, in any given cross segment inside the zone involved by the phases. From the dual flow momentum condition, the weight drop condition for build up inside horizontal tube is produced as:

$$-\left(\frac{dp}{dz}\right) = -\left(\frac{dp}{dz}\right)_f - \left(\frac{dp}{dz}\right)_z - \left(\frac{dp}{dz}\right)_z \tag{2}$$

$$-\left(\frac{dp}{dz}\right)_z = g\left[\varepsilon\rho_g + (1-\varepsilon)\rho_l\right] \tag{3}$$

$$-\left(\frac{dp}{dz}\right)_a = G^2 d\left\{\left(\frac{x^2}{\rho_g \varepsilon}\right) + \frac{(1-x)^2}{(\rho_l(1-\varepsilon))}\right\}/dz \tag{4}$$

Where,

$-\left(\frac{dp}{dz}\right)_z$ is the gravity pressure drop component,

$-\left(\frac{dp}{dz}\right)_a$ is the acceleration pressure gradient and $-\left(\frac{dp}{dz}\right)_f$ is the friction factor gradient.

The gravity pressure gradient is pertinent just for long vertical tubes, while the momentum pressure drop brings about an expansion in the pressure at the exit than at the inlet port, with respect to consolidating flows, the kinetic energy of active flow is smaller than that of approaching flow. Hence, it is regular practice to disregard the momentum recuperation as just some of it might really be acknowledged in the flow and overlooking it gives some conservatism in the geometry.

In addition, assessment of momentum and gravity pressure drop involves in the void fraction data. In condenser, because of high vapor density which is the consequence of maximum pressure on condenser side, at a given mass flux and quality, the vapor velocity is slower than that of evaporator. Bringing down the velocity conveys the flow nearer to stratified floe regime. In this zone, void fraction expectations by any of the regular void fraction models are off base coming about into in correct pressure drop forecasts. Henceforth in the present review, just frictional pressure drop is assessed using CFD investigation and contrasted and the connections in view of isolated flow demonstrate accessible in the literature.

2.1. CFD Modeling

Heat transfer from the hot fluid flowing intimate the helical pipe to the cold fluid flowing through the tank that has been modelled using FLUENT. Flow velocity through the pipe considered were 0.8 m/s in steps of 0.25m/s. This gives a range of D_e from 4000 to 11,000. Inlet was at 355K and the cold water inlet was at 300 K. The Realisable k–ε turbulence with standard wall functions was used in this analysis.

Pressure-velocity coupling was resolved using the SIMPLE algorithm with skewness correction factor 1. For pressure, linear discretization was used. For momentum, turbulent kinetic energy, and turbulent dissipation rate the Power law scheme was used. For the energy equation, second order upwind was employed. A convergence criterion of 1.0×10^{-05} was used for continuity and x, y and z velocities. The convergence criterion for energy equation was 1.0×10^{-08}, while that for the k and ε was 1.0×10^{-04}.

3. Mathematical Formulation

A homogeneous flow model is a case of isolated flow examination in which vapor and fluid velocity are ought to be consistent and break even with. In this model, two-phase flow is dealt with as single phase pseudo liquid with reasonably arrived at the midpoint of properties of fluid and vapor stage. The constant homogeneous flow demonstrates the essential conditions for build up inside a horizontal tube are decreased to the accompanying structure:

Continuity Equation: $\dot{m} = \bar{\rho}\bar{u}A$ (5)

Momentum Equation: $-Adp - d\bar{F} - A\bar{\rho}gdz = \dot{m}d\bar{u}$ (6)

Where the overall wall shear force, dF in terms of wall shear stress, τ_w acting through inside area of the tube can be expressed as:

$$d\bar{F} = \tau_w \, (Pdz)$$
(7)

This frictional pressure drop equation for the dual phase friction factor can be considered through the Blasius equation expending the average properties.

$$f_{TP} = 0.079 \left[\frac{Gd}{\bar{\mu}}\right]^{-0.25}$$
(8)

The normal belongings for homogeneous pseudo fluid are established from the fundamentals as mentioned elaborately in Collier [14].

The average fluid density is given by:

$$\frac{1}{\bar{\rho}} = \left[\frac{x}{\rho_g} + \left(\frac{1-x}{\rho_l}\right)\right]$$
(9)

Possible forms of relationships for mean two phase viscosity, $\bar{\mu}$ based on limiting condition, at x=0, $\mu_l = \bar{\mu}$ and at x=1, $\mu_g = \bar{\mu}$, are [3]:

$$\frac{1}{\bar{\mu}} = \left[\frac{x}{\mu_g} + \left(\frac{1-x}{\mu_l}\right)\right] \text{(McAdams)}$$
(10)

$$\bar{\mu} = x\mu_g + (1-x)\mu_l \text{ (Cicchitti)}$$
(11)

$$\bar{\mu} = \bar{\rho}\left[\frac{x\mu_g}{\rho_g} + \left(\frac{(1-x)\mu_l}{\rho_l}\right)\right] \text{(Duker)}$$
(12)

Based on the exceeding normal properties, dual phase frictional pressure drop for horizontal tube of internal diameter, d is calculated as [7]

$$\Delta P = \left. 2f_{TP}G^2L \middle/ \bar{\rho}d \right. \tag{13}$$

3.1. Heat transfer in a helically coiled tube for constant wall boundary condition

So as to study the impact of fluid properties on the modeling of heat exchange, the instance of heat exchange to liquid flowing inside a helical tube, which is heated to a consistent wall temperature, is broke down. The grid matrix utilized as a part of investigation is appeared in Figure 5. A steady wall temperature of 300K was determined as the limited boundary condition. Refrigerant at a temperature of 355K is entering the helical coil at the top (velocity inlet boundary condition) and leaving at the base (pressure outlet limit condition). In the principal arrangement of examinations, the properties of refrigerant were kept consistent comparing to the fluid inlet temperature and pressure (360K temperature, 1 atm pressure). Second arrangement of investigations were finished utilizing the temperature subordinate properties of water; as given by Eq. (14) − (17). These conditions were customized in FLUENT as polynomial capacities to process the properties. In both the arrangements of examination, shell conduction through the pipe mass of 2.7mm thickness was considered.

At the pipe wall, for the energy equation, a Dirichlet boundary constrains and for momentum and pressure equations homogenous Neumann boundary constrains is indicated. At the inlet condition, turbulent intensity of 4% and hydraulic diameter of the largest size eddy, which is occupied as 0.3 times pipe inner diameter, are quantified. At the outlet condition, a pressure outlet constrains is required

$$\mu\,(T) = 2.1897e-11T^4 - 3.055e-8T^3 + 1.6028e-5T^2 - 0.0037524T + 0.33158 \tag{14}$$
$$\rho(T) = -1.5629e-5T^3 + 0.011778T^2 - 3.0726T + 1227.8 \tag{15}$$
$$k(T) = 1.5362e-8T^3 - 2.261e-05T^2 + 0.010879T - 1.0294 \tag{16}$$
$$Cp(T) = 1.1105e-5T^3 - 0.0031078T^2 - 1.478T + 4631.9 \tag{17}$$

These relationships were obtained by regression analysis using MATLAB. In the above relationships, temperature is specified in K. It might be distinguished that the pressure of the fluids which does not change substantially and also since the pressure addiction of the properties of an incompressible fluid is negligibly very small; only the moderate dependency was taken into account in the studies.

## 4. Results and Discussions
4.1. CFD Modeling

Heat transfer from the hot fluid flowing intimate the helical pipe to the cold fluid flowing through the tank that has been modelled using FLUENT. Flow velocity through the pipe considered were 0.8 ms−1 in steps of 0.25ms−1. This gives a range of De from 4000 to 11,000. Inlet was at 355K and the cold water inlet was at 300 K. The Realisable k–ε turbulence with standard wall functions was used in this analysis.

Pressure–velocity coupling was resolved using SIMPLEC algorithm with skewness correction factor 1. For pressure, linear discretization was used. A convergence criterion of 1.0×10^{-05} is used for continuity and x, y and z velocities. Convergence criterion for energy equation is 1.0×10^{-08}, while that for the k and ε was 1.0×10^{-04}.

Figure 6(a) displays an overview of velocity contours at various sections along length of coil, while details at a few cross sections are available in Figure 6(b). The planes are identified by angle (θ) which that plane makes with plane passing through inlet pipe. In Figure 6(a), first plane represents on top is at 60° from inlet (i.e., θ =60°) and subsequent planes are 10° apart.

Up to an angle of θ=90°, velocity profile at cross-section is found to be symmetric. Consequently, this identical velocity structure changes to a high velocity region situated at outer side of coil. This behaviour is seen predominantly by θ =135° and continues to develop. It can be visualized that by θ = 225°, the high velocity region is present only in outer half cross-section. Part of high velocity region, further decreases as the flow gets established and covers approximately 2/3rd of flow area by θ = 270°. No significant change is identified in flow structure in downstream.
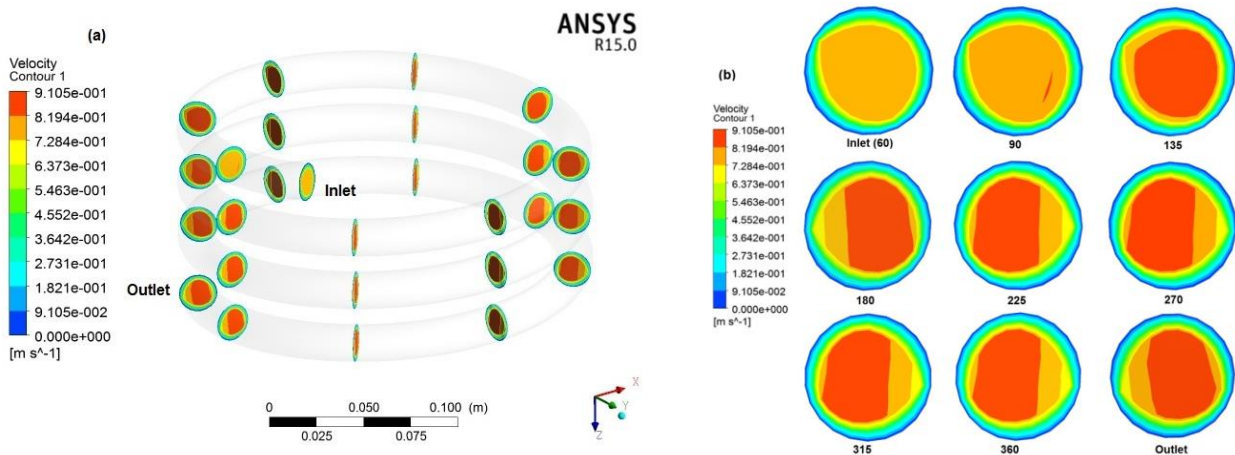

Fig. 6. Velocity (ms⁻¹) contours at (a) various planes along the length of the coil. (b) selected planes along the length of the coil.

The wall velocity will be zero due to the viscous region of the pipe wall. The inlet and outlet velocity will plot between the velocity magnitude vs the pipe axis (as shown in Figure 7). The velocity plot will show the velocity distribution over the pipe. At inlet of the pipe will uniform and outlet velocity distribution will various up to 0.9ms⁻¹.
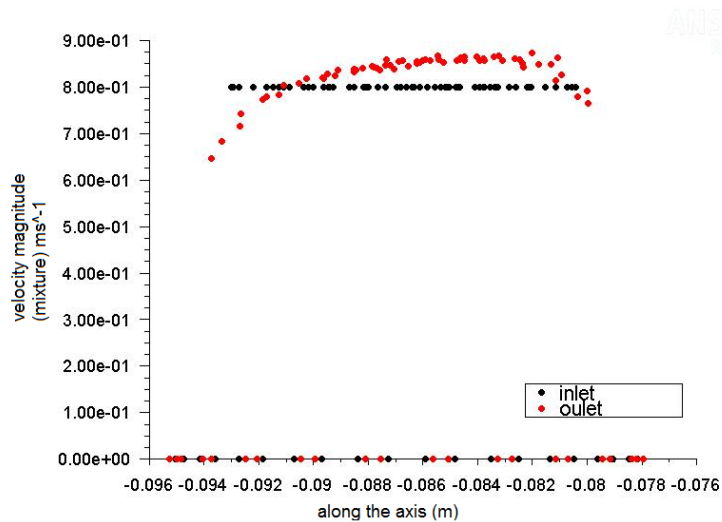

Fig. 7. Velocity magnitude of multi-phase flow inside the pipe
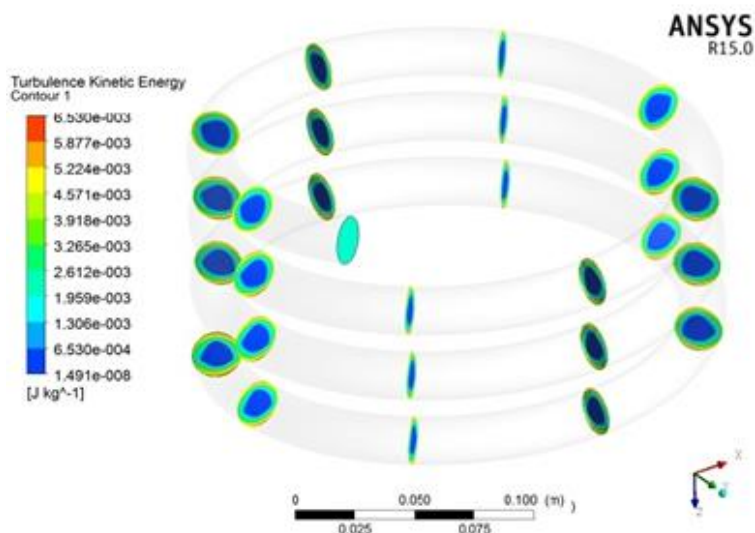
Fig. 8. Turbulent kinetic energy (m^2 s^{-2}) contours at various planes along the length coil.

Turbulent kinetic energy at various planes alongside the span of the coil is presented in Figure 8. In the fully established area, the turbulent kinetic energy has minor values at the top side of the coil as compared to the values on the middle region. The higher value of turbulent kinetic energy is 6.53×10^{-3} and lower value is 1.49×10^{-8}.

Figure 9 shows the turbulent kinetic energy plot along the pipe, the top and bottom inside of the pipe having higher turbulent kinetic energy up to 6.53×10^{-3} as showed in graph. At the middle region of the inside pipe have no turbulent, as generated it will be clearly showed in graph.
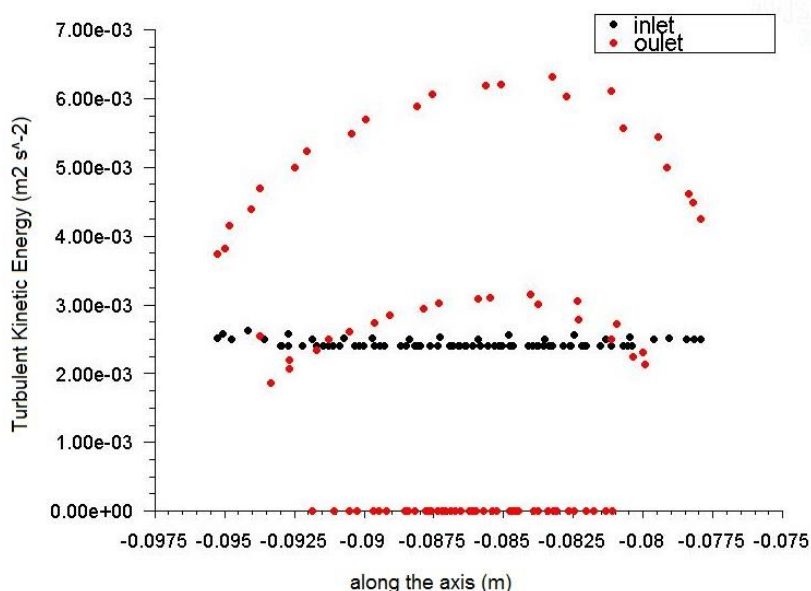
Fig. 9. Turbulent kinetic energy of multi-phase flow inside the pipe

Distribution of temperature at various planes along length of coil is displayed in Figure 10(a) and 10(b), the former represents a global picture while latter represents the details at selected planes. At the inlet, temperature is uniform across the cross-section. Since the wall is preserved at a lower temperature, the refrigerant cools down as it flows through coil. Up to an angle of 70°, heat transfer

is uniform along the periphery. In contrast to heat transfer in a straight tube, high temperature regions are seen on the mid axis of the coil. This occurrence is predominant from the plane at angle θ=90°. This trend continues to develop and by 130°, clearly three regions viz., high temperature (353–320 K) of the coil, intermediate temperature (321–310 K) at the centre and low temperature (309–300 K) on the inner side wall of the coil, are visible. When fluid flows downwards along the pipe, this temperature profile is generated and the high temperature zone region decreases in the outlet region, a fully developed temperature profile is obtained and the fluid gradually loose heat by lower wall temperature.

(a)   (b)
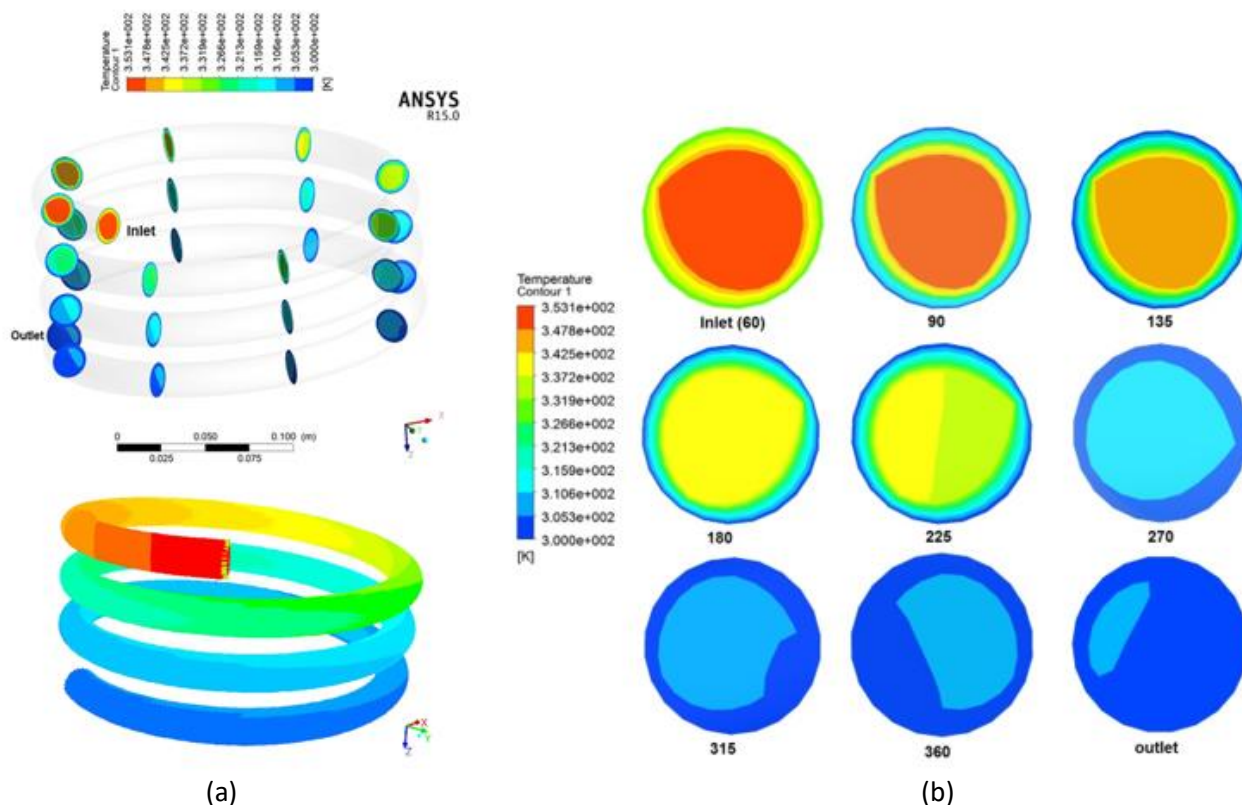
Fig. 10. Temperature (K) contours at (a) Various planes along the length of the coil. (b) Selected various planes along the length of the coil.

Figure 11 shows the temperature distribution over the helical coil, due to convection heat transfer the temperature will be decreased because of the exit of the helical coil as it is shown in graph. X axis will plot as along the helical coil axis and y axis will be temperature distribution. At maximum temperature at inlet has 353.6K and exit of the coil outlet temperature has 300K.

Equally the fluid flows over the helical coil, the fluid substances undertake rotational motion. The fluid particles also experience movement from internal side of coil to external side and vice versa. Figure 12 displays particle trace for fluid particles which are positioned along a line parallel to X-axis at the pipe inlet.

It can be well-known that fluid particles are enchanting several trajectories and also move with dissimilar velocities. The particles, which were forming a line to begin with, have found to be totally distributed at the pipe exit. It can be obvious that the great velocity region fluctuates as the fluid flows lengthwise of the helical pipe. This reasons varies in the values of Nusselt number [8].
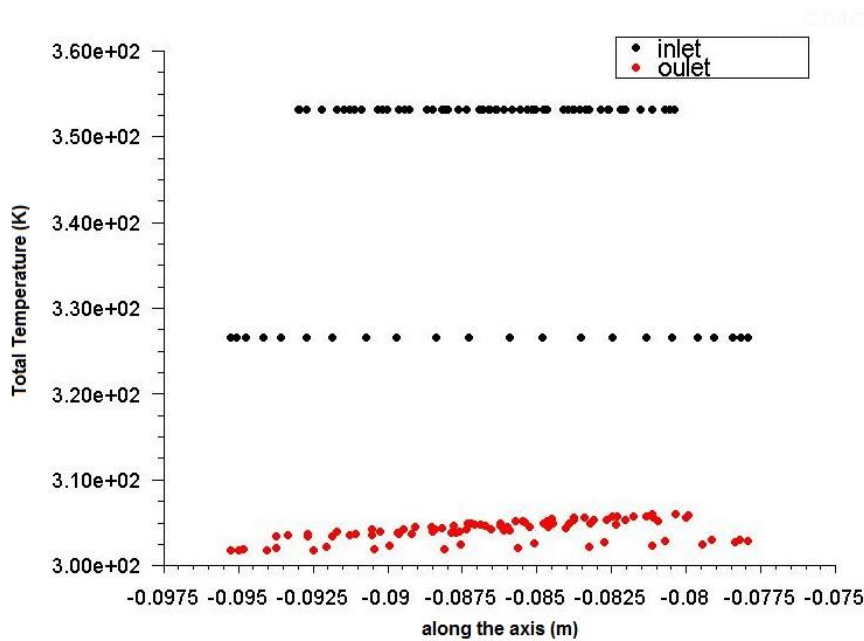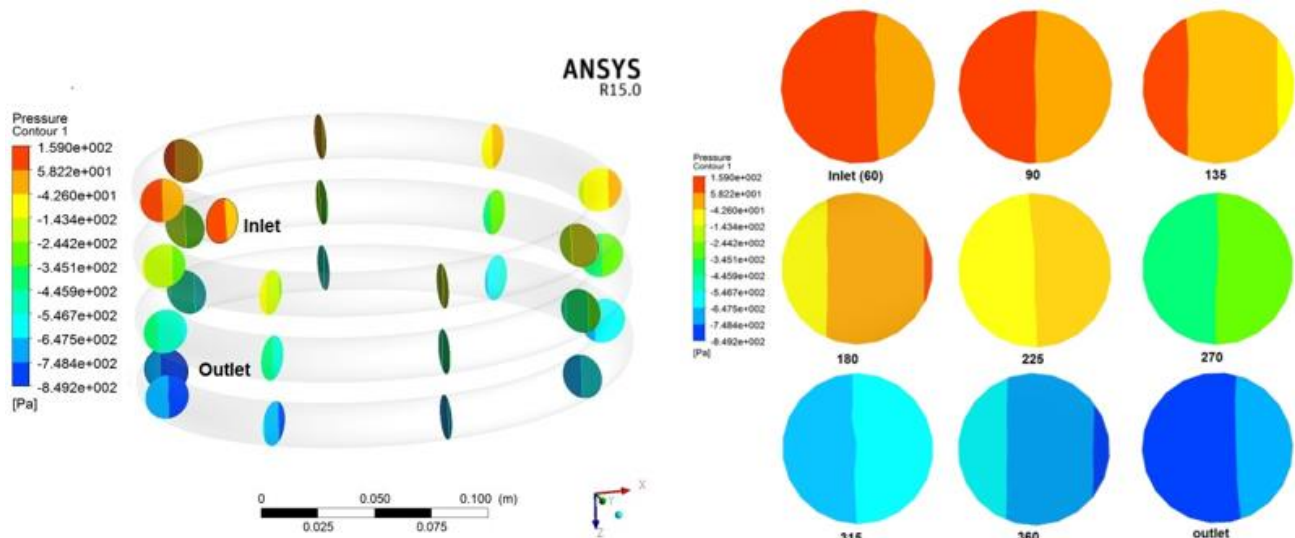
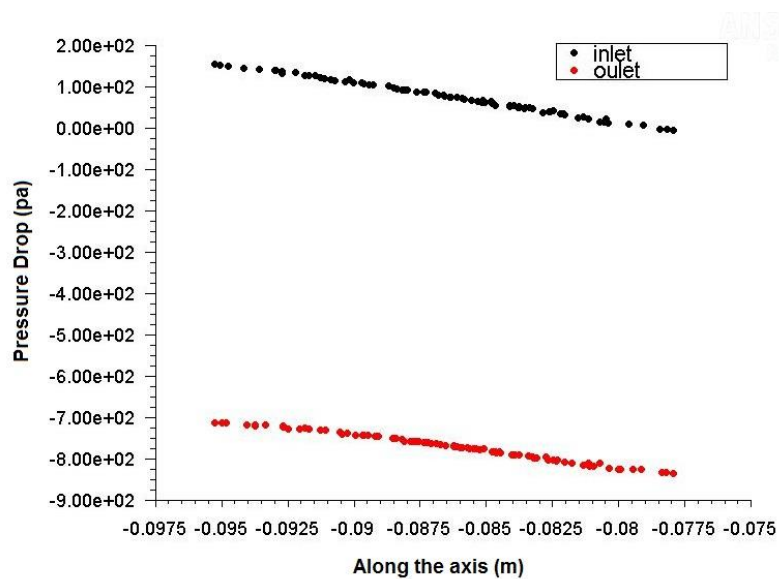**Fig. 11.** Temperature distribution of multi-phase flow inside the pipe
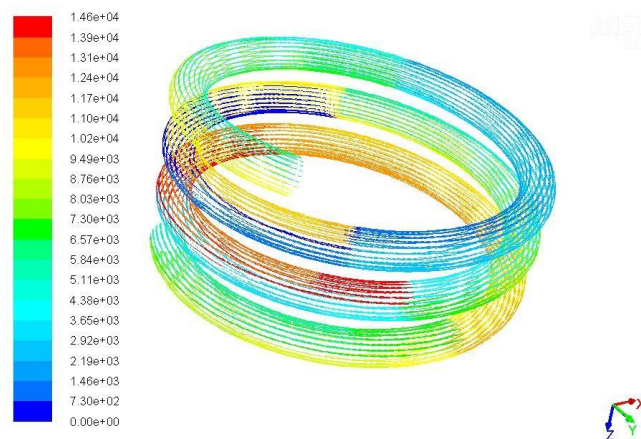
Fig. 12. Flow characteristic of helical coil

Pressure drop distribution at various planes along length of coil is projected in Figure 13(a) and 13(b), former represents a global picture while latter represents the details at selected planes. At the inlet, pressure is uniform across cross-section. This occurrence is predominant from the plane at angle θ=90◦. This trend endures to develop and by 180◦, pressure drop will occur because of friction factor of helical pipe. Pressure drop occurring in helical coil is 70% due to friction coefficient. As the fluid flows down the pipe, right side of the coil having high pressure and another side have low pressure because of dual flow inside the coil.

Figure 14 shows the pressure drop distribution over the helical coil, the frictional law is occurred in the helical coil where the pressure drop will be obtained. At x plot the helical coil axis and y plot will be the pressure drop distribution in the helical coil segment. The maximum pressure at inlet is 200Pa and exit region, the pressure will decrease by -700pa.

Fig. 13. Pressure drop contours at (a) Various planes along the length of the coil. (b) Selected various planes along the length of the coil.

Fig. 14. Pressure drop distribution of multi-phase flow inside the pipe

5. Conclusion

Based on investigation with computational analysis for analyzing the performance, the succeeding conclusions were made: Liquid vapor flow of low and high pressure refrigerants, viz.R134a, and R290 is exhibited by Volume of Fraction (VOF) model from existing CFD software, FLUENT. Transient analysis is implemented to trajectory the helical geometry of interface with Realisable k–ε turbulence with standard wall functions was used in this analysis and hence attains flow regimes at working conditions for low, medium and high mass fluxes. The flow regimes achieved by plotting the contours of mixture density, showed that the VOF model reproduced all the flow regimes including flow regime transitions. The numerically attained overall heat transfer coefficient for numerous values of flow rate in inner-coil region was reported. It was inferred that overall heat

transfer coefficient increases with increase in the inner-coil flow rate in the helical coil region. To improve the heat transfer of refrigerant and pressure drop that occurs in helical coil. These regimes represent mixing type flow without a clear interface between vapor and liquid. Since the VOF model implemented in CFD analysis, depends on interface flows, larger deviations are observed at these flow conditions.

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