Journal of Engineering and Sustainable Development
http://jeasd.uomustansiriyah.edu.iq
https://doi.org/10.31272/jeasd.25.1.1

Vol. 25, No. 01, January 2021
ISSN 2520-0917

EXPERIMENTAL AND NUMERICAL STUDY OF INSERTING AN INTERNAL HOLLOW CORE TO FINNED HELICAL COIL TUBE-SHELL HEAT EXCHANGER

*Bahaulddin K. Roomi[1] Maathe A. Theeb[1]

1) Mechanical Engineering Department, Mustansiriyah University, Baghdad, Iraq

Received 3/5/2020 ***Accepted in revised form 19/7/2020*** ***Published 1/1/2021***

Abstract: In the present study, an experimental study and a numerical method are used to simulate the effect of utilizing finned helical coil heat exchanger horizontally oriented with and without internal hollow core inside the shell and monitoring the heat transfer intensification behavior and performance enhancement, by using ANSYS FLUENT 2019R3 package. The working fluid in the inlet side of the coil will be superheated vapor refrigerant (R410a) after the compression stage of the air conditioner (1TR) split type, the working fluid in the shell side is water with various flow rates (1,2 and 3)LPM. Found that when using a heat exchanger with internal hollow core reduces the time needed to reach the steady-state outlet temperature and directly proportional to the size of the core by 20% and slightly increasing the water outlet temperature by 5%.

Keywords: *finned helical coil, tube-shell heat exchanger, internal hollow core*

1. Introduction

Helical coil heat exchangers are widely used in manufacturing processes and industries. A higher heat coefficient is created due to the centrifugal strength provided by tube curvatures compared to the fluid flow and heat transfer in straight tubes. The centrifugal force causes a secondary flow and increases heat transfer intensities. A. Ameen et al. [1] analyzed the three stages of refrigerant R-134a passage through refrigerator's wire and tube condenser (de-super-heating, condensation, and sub-cooling) under normal conditions of convection with finite element method. By using the temperature of ambient effecting for finding the tube's length required to achieve refrigerant's phase changing. They found that for a definite refrigerant flow rate, the subcooled stage can't occur at higher temperatures, consequently presenting the heat exchanger's inefficiencies. They predicted a design tool developing program (VISUAL FORTRAN 6.1) to check the tubes' numbers in wire and tube condenser under extreme ambient situations. Q. Mahdi et al.[2] carried out a numerical and experimental evaluation of heat exchanger (shell-helical coil tube) with inserting wire coil. They used a tubular guide to get a huge increase in heat transferred and friction loss in the experimental results. Simulation of heat transfer and flow in the helical coil was carried out by numerical analysis using ANSYS FLUENT package 14.0. They have been predicting new empirical correlations Nusselt number has been developed and good agreement shows the result of comparing between the experimental results and the numerical solution. M. Zareh et al. [3] studied the action of specific parameters such as

**Corresponding Author*: bahaa.albedairi@gmail.com

inlet temperature, inlet pressure, geometric dimensions and subcooling degree on phase change of various refrigerants (R12, R22 and R134a) in a helical tube under the same conditions. Their results show that for the identical length and parameters, mass flux inside helical coil tube sizes (40, 80 and 100mm) 11%, 30% and 36% respectively, less the flow inside the straight tube.

H. Saffari and R. Moosavi [4] investigated a numerical solution for (single and two-phase) fluid flowing inside helically coiled pipe. Asymmetrical velocity profile and shifted heavier fluids towards the pipe's outer section because of the curvature's centrifugal force in coiled pipes. From the numerical simulation, they concluded that the coefficient of friction increase directly in coil curvature, a generation of Coriolis force due to the increase of coil pitch. Increasing in void fraction due to decreasing of the mixture of two-phase density, cause to more reduction of shear stress, pressure and friction coefficient. Alhamdo et, al[5] carried out an experiment to use fins with different profiles (circular, longitudinal, spiral) attached to water-phase change material (PCM), double tube system had developed and tested in a different orientation, to increase the PCM's thermal conductivity using the rejected heat from the air conditioning system, significant improvement performance has been shown in the results for the spiral and circular fins in the vertical orientation and the longitudinal, without fins in the horizontal orientation.

S. Nada et al [6] conducted experimental investigation characteristics and enhance the performance of heat transfer for shell-helical coil heat exchanger cooler with radial external fins and shells with different diameters. Enhance the performance of the water cooler by improving the compactness of the coil. The cooling fluid is refrigerant inside the coil and the water needed to be cooled in the shell side. The findings showed an improvement in performance and compactness by adding external radial fins and increasing the ratio shell diameter to helical coil diameter. F. Bagherzadeh et al. [7] presented a numerically investigates study of heat transfer of (steady state, laminar, viscous and incompressible) flow of nanofluids in helical coil tubes. They applied two boundary conditions to the walls of the coil; constant heat flux and constant temperature. By using the model of four equation numerical solution (mass, momentum, energy and continuity) equations, proposed a simplified model compared to the two phase model. In the coil, the secondary flow was augmented for various curvature ratios and Reynolds numbers. They found that the method of solution was less complicated than the method of two phases, and the four equation model can be considered as efficient method for study heat transferred of nanofluids in helically coils tubes, mainly in fractions have higher volume.

A. Hussain and A. Golam [8] introduced experimental and numerical research for heat exchanger (helically coiled double pipe) using a counter and parallel flow arrangements, testing, and water was used in the inner and annulus tube sides. Various cold water (20-25˚C) flow rate and constant for the hot water (40-70˚C). The results show that the distribution of axial temperature for the heat exchanger affected by the mass flow rate ratio, by increasing the ratio of mass flow rate causes to decrease the efficiency and effectiveness. Comparisons have been made between the experimental and numerical analysis by using ANSYS FLUENT 14.0 package. E.Abed & A. Abdulkadhim [9] were studied refrigerant's phase change of flowing in a capillary tube numerically by using ANSYS16.1 by utilizing homogeneous model flow from phase to another Eulerian-Eulerian multiphase along with phase change thermally found on the mass fraction transfer appliance by employing simulation the phenomenon of phase changing for the refrigerant in the capillary tube below the conditions of adiabatic states. V. Hameed and A. Al-khafaj [10] studied helical coil tube-shell heat exchanger experimentally and numerically, water was the working fluid in both shell and tube. They used for the shell a perplex tube 1000,150 and 2mm length, diameter, and thickness respectively, for the helical a copper tube with 12.7 and 0.1mm diameter and thickness respectively. The numerical simulation was made by ANSYS fluent 16. Three cases of coil pitch were investigated and compared the results with a straight tube. The heat transfer rate increases (4.5, 5.8 and 7.2) times for (5.27, 4.27, and 3.27) cm coil pitch comparing with a straight tube. The results were found that decreasing the coil pitch will increase the heat exchanger performance due to the secondary flow.

A. Alimoradi et.al [11] investigate numerically the intensity of heat transfer in thirteen designs of heat exchangers type shell-helical coiled tube via annular fins installing in the outside of helical coil with different fin's height and number. They have researched three cases of different Reynolds number (7500, 15000, 30000) in the shell side. The study has been two methods, the first method used experimental correlations for both shell and coil side for Nusselt number. The coefficient of heat transfer was compared with experimental results from previous studies in the second method. Results declared that, in the range of $7500 \leq Re_{sh} \leq 30000$, the rate of transferred heat increased up to 44.11%. T. Ariwibowo et al. [12] discussed the condensation of refrigerant R-134a at saturated temperature 40˚C and its flow regimes (annular bubble, stratified, wavy and intermittent) numerically. They found that 60-70% heat exchanger's length includes annular flow. The phase change pressure drop inside the tube increased with increasing of mass flux (500, 800 and 1100 Kg/m^2.sec) and 0.9 vapor fraction. The interface of shear stress on the walls decreased with the thickness of condensation film decreasing. M. Sepehr et al.[13] used Numerical work on the heat transfer, the pressure drop, and entropy in helically bent tube heat exchangers. Through mounting the annular fins on the coiled tube's outer surface, the heat transfer rate is increased. By using hot water inside the coil with temperature 70˚C and 1 m/Sec speed and the shell side working fluid are dry, cold air with temperature 10˚C and speed ranged between 1-4 m/Sec. An empirical correlation was the basis of predicted values and the models' validities compared with numerical values of Nusselt No. and made a designer guide of these heat exchangers types. J. Wang et al.[14] established a numerical method to survey the fin shape, geometry and the flow rate at the shell inlet effects on the loss of exergy in a finned helical coiled tube heat exchanger with an internal hollow core at the shell inside. The indication of the results is with the inlet flow rate of the shell side, fins dimensions and number, NTU increases due to Exergy loss increases about 23.4% of the rate of the heat transferred, and obtained two correlations for proposing optimum operational and geometrical values of the heat exchanger characteristics.

R. Lopez et al. [15] presented a numerical model of water cooled condenser in the heat pump cycle and refrigerant R410a as a working fluid inside the cycle. The real behavior of the flow was described by k-ε turbulence model. In the shell side the water pressure drop about 25.9 kPa, the increase of water temperature was of 3˚C and outlet pipe velocity 3 m/sec. The simulation results that the COP of the pump 7.6

and 7.32 from experimental, both results were below the equipment's manufacturing value. M. Kassim & S. Lahij [16] were studied an experimental and numerical investigation of the friction factor and the heat transfer features in double pipe counter flow heat exchanger without and with Nanofluid (15-20 nm Nanoparticle size). A turbulent flows of fully developed ranges were performed. The results show that Nanofluid volume concentration increasing leads to increasing in heat transferred rate and Reynolds number. Simulation of heat transferred and flow of the fluid have been made by using ANSYS FLUENT 2015 package and gives acceptable results with the experimental work

In this paper, Analyzing the effect of using finned helical coil tube-shell heat exchanger (horizontally oriented) with and without an internal hollow core in the center of the shell. The heat exchanger worked with hot refrigerant vapor after the compression stage of an air conditioner (1TR) split type, on the tube side and water as heat transfer fluid in the shell side. The energy, discharging simulates numerically dependence on experimental work. The results discussed the difference between the using of internal hollow core from not using it in terms of water outlet temperatures, velocity inside the shell and the refrigerant mass fraction (dryness factor) inside the helical coil.

2. Numerical Simulation

The commercial version of ANSYS fluent 2019 R3 package was used to compute the waste heat recovery in the heat exchanger to the water in the shell side and predict the phase change in refrigerant. Three dimensions *(r, θ, Z)* of the shell were created by geometry software and engaged in the geometric modeling meshing software. Zone regions and boundary layers were clarified. The governing equations of the model are based upon the equations of mass, energy and momentum conservation, of the fluid.

For the refrigerant phase changing flow model inside the helical coil, the mixed density of (vapor-liquid) flow can be expected. [17]

$$\rho_{tp} = \sum_{i=1}^{i=n} x_i \rho_i = \rho_g + (1-x)\rho_f \tag{1}$$

For phase changing flow model, the formula of two-phase (vapor-liquid) viscosity is calculated as an empirical equation as a dryness fraction function of the dryness factor (x) done in ANSYS fluent 2019 R3

$$\mu_{tp} = \sum_{i=1}^{i=n} x_i \mu_i = \mu_g + (1-x)\mu_f \tag{2}$$

Continuity equation

$$\rho\left(\frac{1}{r}\frac{\partial(r v_r)}{\partial r} + \frac{1}{r}\frac{\partial(r v_\theta)}{\partial \theta} + \frac{\partial v_z}{\partial z}\right) = 0 \tag{3}$$

Momentum equation

In the r-component:

$$\rho\frac{\partial v_r}{\partial t} + \rho\left(v_r\frac{\partial v_r}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_r}{\partial \theta} - \frac{v_r^2}{r} + v_z\frac{\partial v_r}{\partial z}\right) = \mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial(r v_r)}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 v_r}{\partial \theta^2} - \frac{2}{r^2}\frac{\partial v_\theta}{\partial \theta} + \frac{\partial^2 v_r}{\partial z^2}\right) - \frac{\partial P}{\partial r} + \rho g_r \tag{4}$$

In the θ component:

$$\rho\frac{\partial v_\theta}{\partial t} + \rho\left(v_r\frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_\theta}{\partial \theta} - \frac{v_r v_\theta}{r} + v_z\frac{\partial v_\theta}{\partial z}\right) = \mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial(r v_\theta)}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 v_\theta}{\partial \theta^2} + \frac{2}{r^2}\frac{\partial v_r}{\partial \theta} + \frac{\partial^2 v_\theta}{\partial z^2}\right) - \frac{1}{r}\frac{\partial P}{\partial \theta} + \rho g_\theta \tag{5}$$

In the z- component:

$$\rho\frac{\partial v_z}{\partial t}+\rho\left(v_r\frac{\partial v_z}{\partial r}+\frac{v_\theta}{r}\frac{\partial v_z}{\partial \theta}+v_z\frac{\partial v_z}{\partial z}\right)=$$
$$\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v_z}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 v_z}{\partial \theta^2}+\frac{\partial^2 v_z}{\partial z^2}\right)-\frac{\partial P}{\partial z}+\rho g_z \quad (6)$$

Energy equation:

$$\rho Cp\frac{\partial T}{\partial t}+\rho Cp\left(v_r\frac{\partial T}{\partial r}+\frac{v_\theta}{r}\frac{\partial T}{\partial \theta}+v_z\frac{\partial T}{\partial z}\right)=$$
$$k\left[\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}\right]+\mu\emptyset \quad (7)$$

$\emptyset$ Is dissipation (internal friction) function [18], the term $\mu\emptyset$ is the viscous dissipation representation per unit volume. For mixture of a multi-component, the transportation of energy sometimes caused by the diffusion and should be accounted for the conservation of mixture energy.

The total surface area of (helical coil + fins) is equal to 0.5m^2, the total shell volume is 18 liters and the volume of the internal hollow core is 5 liters.

2.1. Geometry Creation

The geometry of the heat exchanger classified as the model consists of finned helical coil heat exchanger horizontal oriented with cylindrical shell 200 mm diameter supposed to be insulated from the outside and no heat transfer through the shell, having one inlet 15mm diameter and one outlet 15mm diameter portions of water with three water flow rates (1, 2 and 3 LPM) flow in the shell side. A R410A refrigerant flow through finned helical copper coil tube 9.53mm diameter with inserting annular circular fins. The second case was placed internal hollow core 100 mm diameters in the shell center to reduce the shell volume by 25%

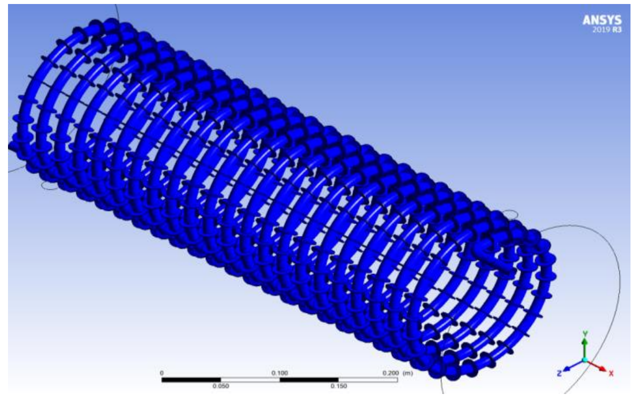
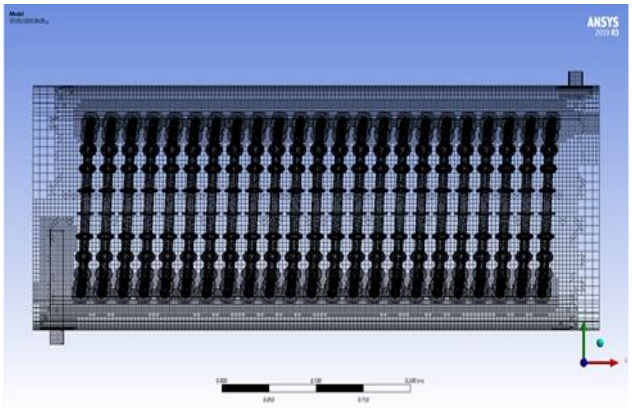


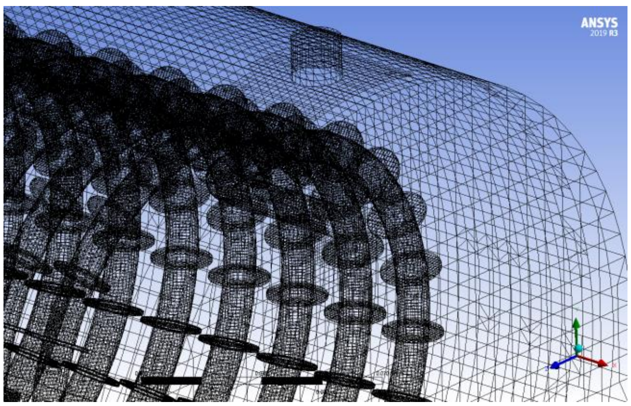
Figure 1. geometry of finned helical coil heat exchanger frame

2.2. Mesh Generation

A preferable solution for meshing process is to separate the heat exchanger in all cases to two parts, shell side (water), coil side (refrigerant). The partial differential of fluid flow equations cannot analyze except in very simple cases. Therefore, the flow domain is subdivided into (elements cells) and the collection of all elements is called the mesh to analyze fluid flow. Within each of these domain sections, the governing equations are resolved. In these models on the heat exchanger shell side a mesh of tetrahedrons was used (cut-cell mesh) method to take the advantages of quick mesh generating, mostly having the results with high quality hexahedral elements and concentrate the elements near the flow regime and high heat transfer surfaces as a default program generates and assuming no heat transferred from the outer shell. The calibration of different grid sizes and transported into applications was performed for the optimum results of the mesh comparing the results with the ideal grid scale of the liquid portion of the measured grids.



a) Complete mesh view



b) Zoomed mesh view

Figure 2. Heat Exchanger Mesh

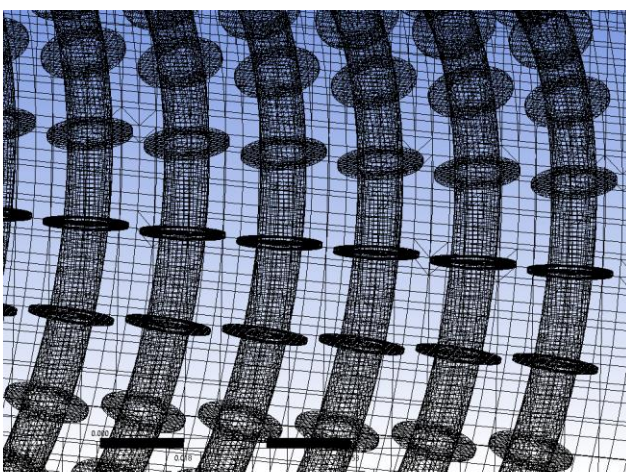


Figure 3. Annular Fins Mesh

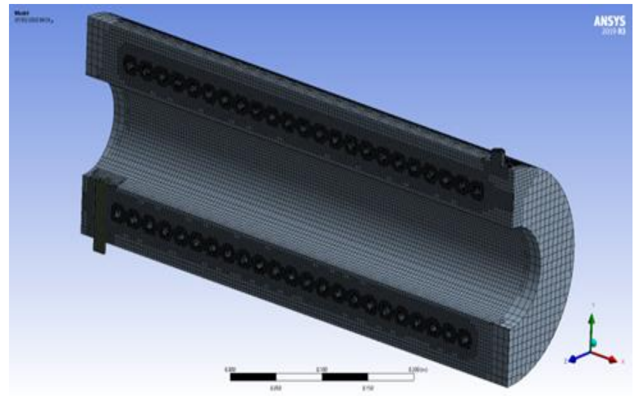


Figure 4. Heat Exchanger Section with Internal Hollow

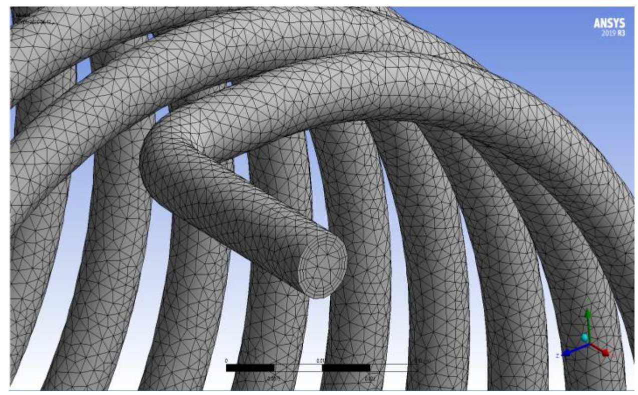


Figure 5. Refrigerants inside the Helical Coil Mesh

Table 1. Number of nodes and elements

Case	Fluid	Nodes No.	Elements No.
Heat Ex.	Water	5,181,998	4,868,140
Heat Ex. With internal hollow core	Water	5,130,756	4,818,985
Helical coil	Refrigerant	423,442	1,077,864

2.3. Boundary conditions and assumptions

Boundary conditions are used according to the need of the model as shown in figure (6). The assumptions have been evaluated according to the refrigeration cycle readings in the experimental section

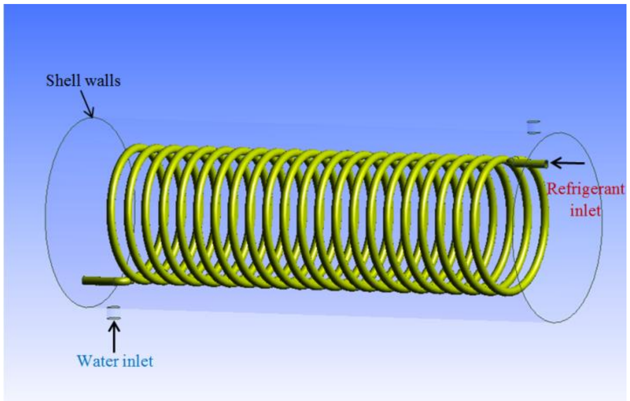


Figure 6. Heat Exchanger Boundary Conditions

a) Inlet water

The water inlet by 15mm diameter defined by inlet temperature 28±1˚C and velocity according to the water mass flow rate (During charging process) and water heat transfer coefficient for each case.

b) Inlet refrigerant

The inlet superheated vapor refrigerant passing through the helical coil copper tube is defined by the amount of flow rate ($\dot{m}_r$) of 0.018 kg/sec and constant velocity 3.6 m/sec. The average temperature of the refrigerant at the inlet is 55±1˚C of the heat transfer fluid (During the discharging process) and the pressure as considered the outlet from the compression stage in the refrigeration cycle and refrigerant heat transfer coefficient for each case.

c) Walls

The outside shells of the heat exchanger wall were supposed to be well insulated and no heat transferred through the walls. And no-slip condition is considered for all walls.

3. Experimental work

To achieve experimental study, the test rig is designed and constructed as photographed and described in figure (7) to transact with all concerning measurements the storage system of thermal energy. The effort had taken to construct the test rig (finned helical coil heat exchanger) part of the refrigeration cycle of air conditioner split type with nominal cooling capacity 1TR, (R410a) refrigerant as a working fluid and placed between the compressor and the condenser in the outdoor unit.

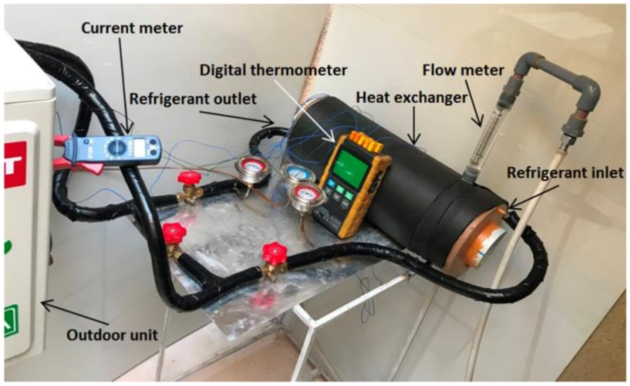


Figure 7. Photograph of the Test Rig

Creating turbulent flow in the water side and expanding the helical coil surface area by inserting annular copper fins with a uniform cross section (inner diameter 10mm, outer diameter 20mm and thickness 1mm). Using copper welding to fix and fill any possible gap between the annular fin and the helical coil to ensure total contact as shown in figure (8)

Figure 8. Annular Fins in the Helical Coil

Placing internal hollow core with diameter 100mm in the shell's center for reducing the water volume inside the heat exchanger and the time required for reaching the steady state temperature of the outlet water from the heat exchanger by as shown in figure (9) and.

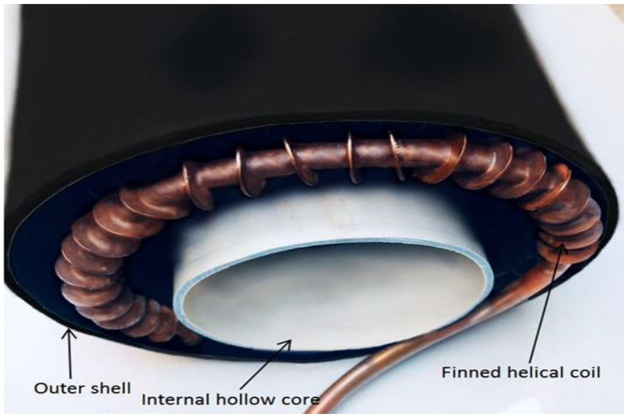


Figure 9. Finned Heat Exchanger with Internal Hollow Core

4. Results and Discussions

All the results of the cases were classified according to the water inlet flow rate in the shell side (1, 2 and 3 Liter per Minute).

4.1 Experimental results

Figures (10, 11 and 12) show that reducing of the water volume in the shell side need to be heated by the refrigerant effect by using internal hollow core and for reducing the reaching time to the steady state condition, and also the increasing of the outlet temperature leaving the heat exchanger for the three water flow rates due to increasing of heat transfer rates and coefficient.

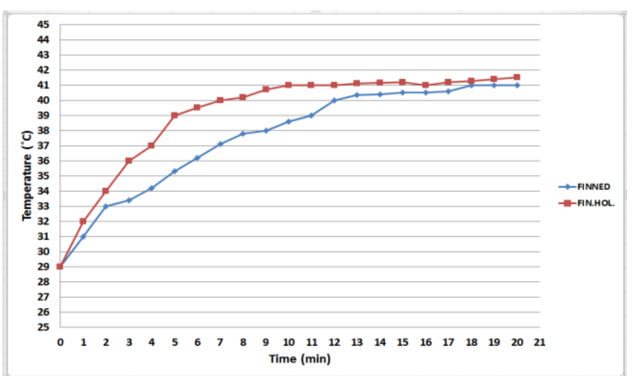


Figure 10. Temperature augmentation vs Time for finned and Finned with Hollow heat exchanger (1LPM water flow)

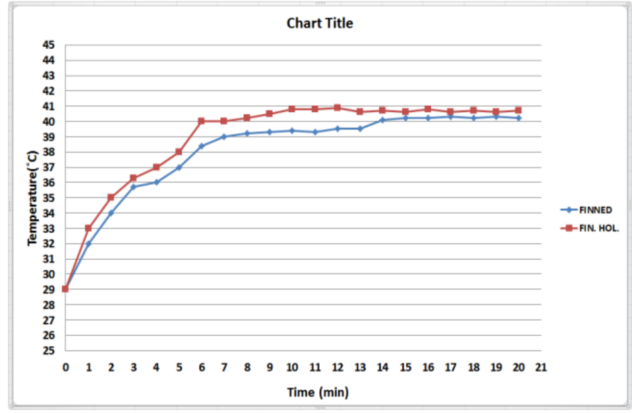


Figure 11. Temperature augmentation vs Time for finned and Finned with Hollow heat exchanger (2LPM water flow)

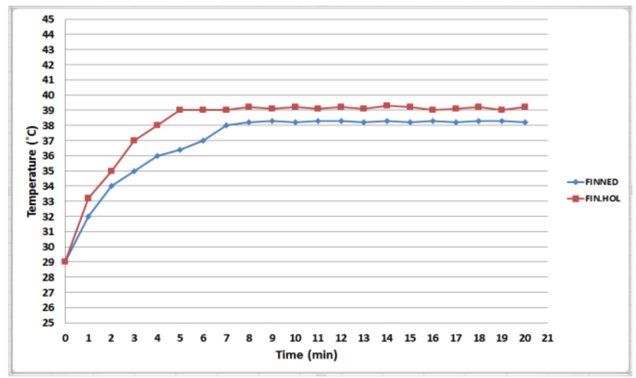
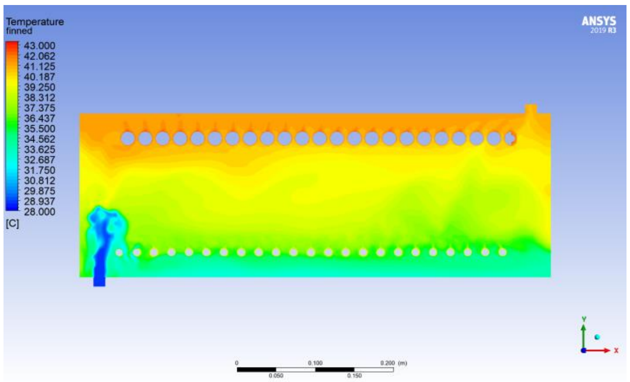


Figure 12. Temperature augmentation vs Time for finned and Finned with Hollow heat exchanger (3LPM water flow)

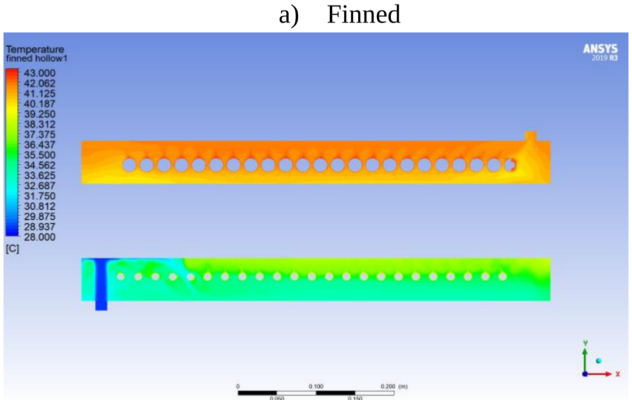
The cost of materials' purchase, construction of the heat exchanger was about (204$) and The additional cost of adding the internal core (UPVC) 150mm pipe and its heavy duty glue, Very little financial value (10$).

4.2 Numerical results

Figures (13, 14 and 15) show that the transient temperature solution for the water side until reaching the steady state temperature distribution for the front section inside the heat exchanger shell side of water flow (1, 2 and 3 LPM) and the maximum temperature reached in the 1LPM flow in the heat exchanger with internal hollow core. Comparing with conventional heat exchanger, the heat transfer rates accomplished the higher water outlet temperature because of the expansion of the surface area and the shape of the fins effected on the water flow inside the shell to produce larger numbers of the boundary layers from every annular fin inserted to the helical coil. For the same effect of inserting an internal hollow core to the heat exchanger to reduce the time required to reach the steady state and increasing the outlet temperature. With increasing of the water flow of the heat exchanger with the hollow core cause of high heat transfer coefficient due to the difference in the shell side's hydraulic diameter.

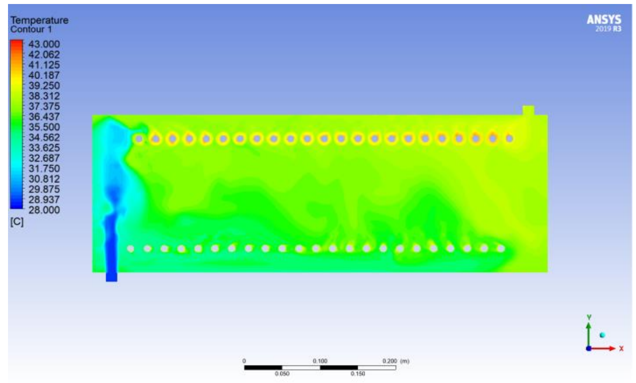


a) Finned

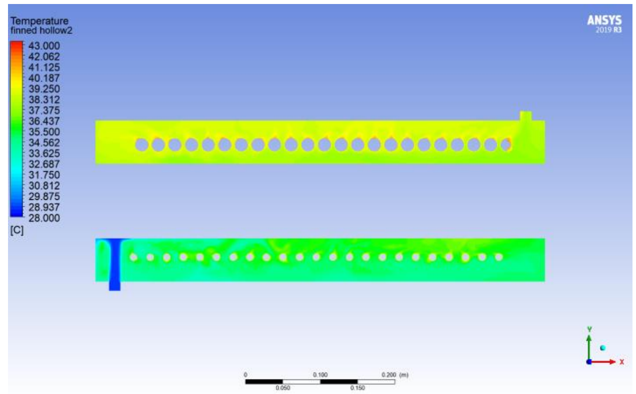


b) Finned hollow

Figure 13. Water temperatures distribution inside shell for 1LPM frame

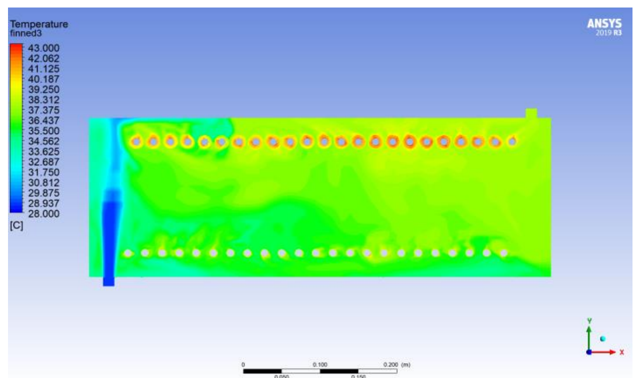


a) Finned

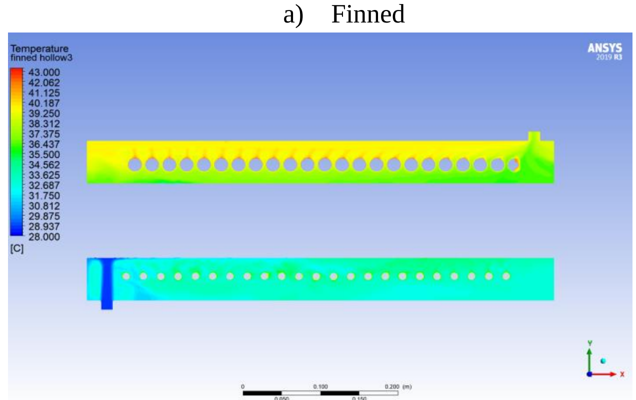


b) Finned hollow

Figure 14 Water temperatures distribution inside shell for 2LPM frame



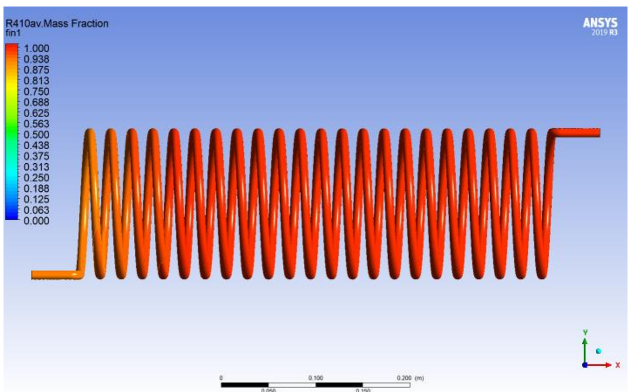
a) Finned



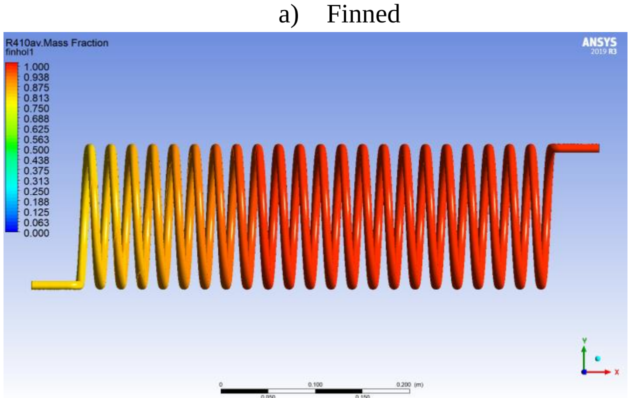
b) Finned hollow

Figure 15. Water temperatures distribution inside shell for 3LPM frame

Figures (16, 17 and 18) show the mass fraction of the refrigerant phase change inside the helical coil and obviously clear that the refrigerant behavior depends directly on the heat observed in the water in the shell and phase change happen and start earlier with more effectively and increase with the increasing of the heat transfer rate. The numerical simulation conducted, show that the phase change appeared on the refrigerant and the vapor to liquid mass fraction. The mass fraction percentage refers to the refrigerant condensation inside the helical coil and the heat transferred from the refrigerant in the tube side to the water on the shell side.

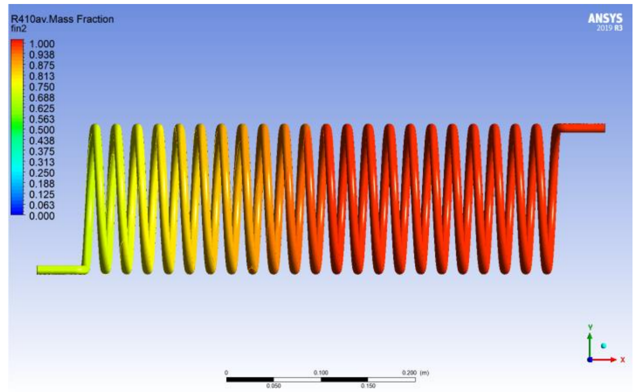


a) Finned

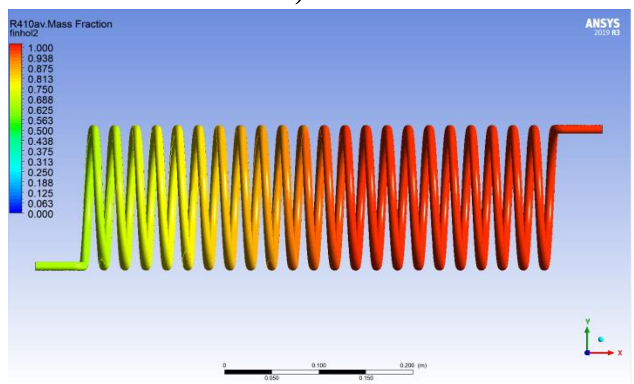


b) Finned hollow

Figure 16. Refrigerant Mass Fractions for 1 LPM

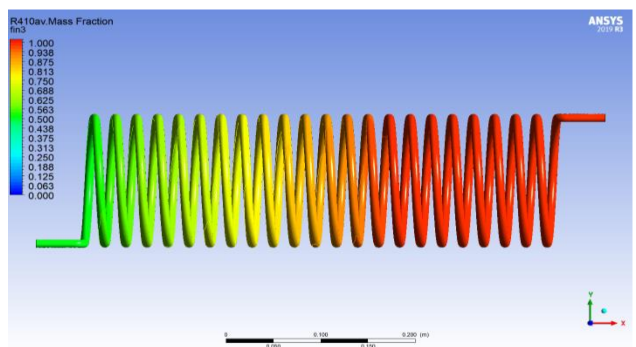


a) Finned

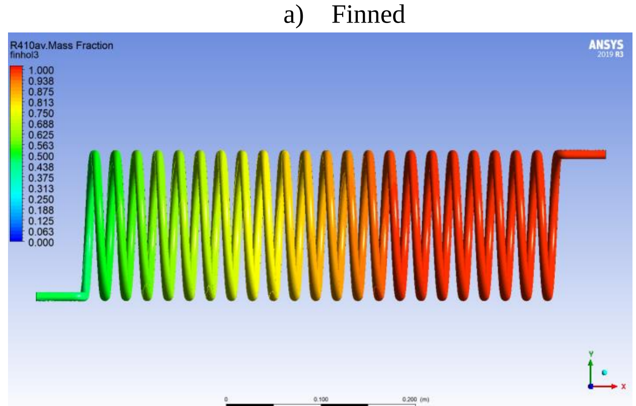


b) Finned hollow

Figure 17. Refrigerant Mass Fractions for 2



a) Finned



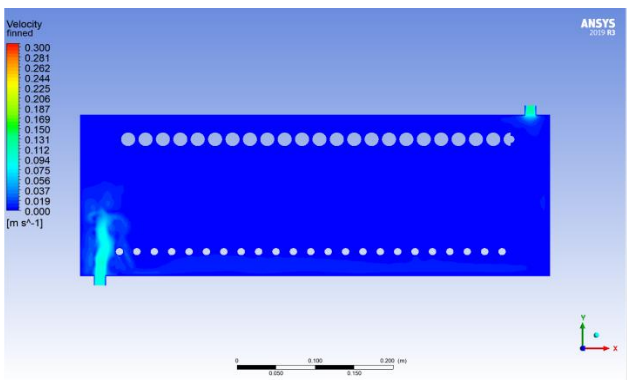
b) Finned hollow

Figure 18. Refrigerant Mass Fractions for 2 LPM

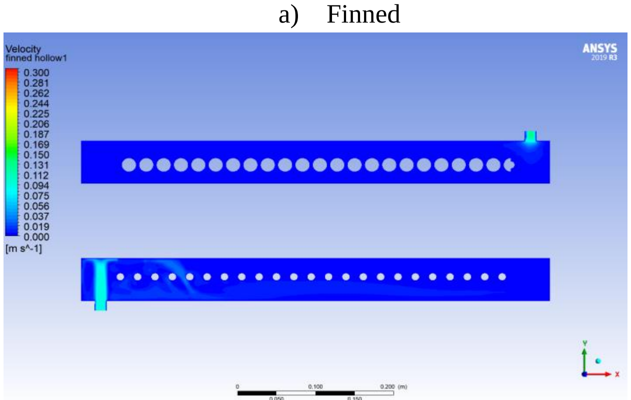
Table 2. Refrigerant mass fraction, dryness factor (X) at the outlet of helical coil

Case	1LPM	2LPM	3LPM
Finned helical coil heat exchanger	0.926	0.734	0.532
Finned helical coil heat exchanger with internal hollow	0.867	0.665	0.489

Figures (19, 20 and 21) show The corresponding values of velocities at different water flow rates are as flows (1LPM= 0.09m/sec, 2LPM= 0.18m/sec, 3LPM= 0.27m/sec) and the direct effect of inserting a hollow core inside the heat exchanger cause of reducing the hydraulic diameter and the cross sectional area of the shell with the same water flow rates to increase the heat transfer coefficients and rates.

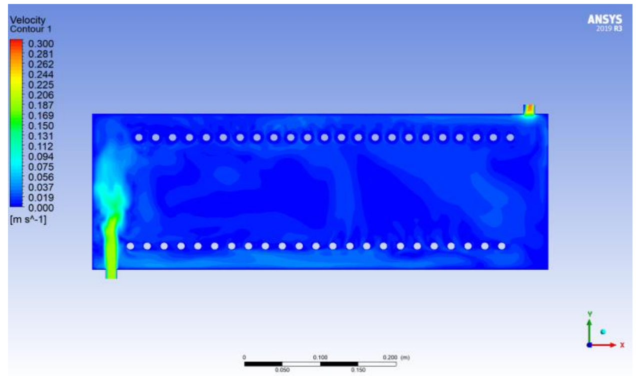


a) Finned

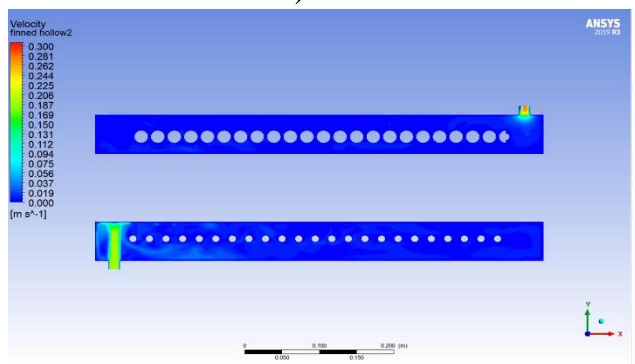


b) Finned hollow

Figure 19. Water velocity distributions inside shell for 1LPM

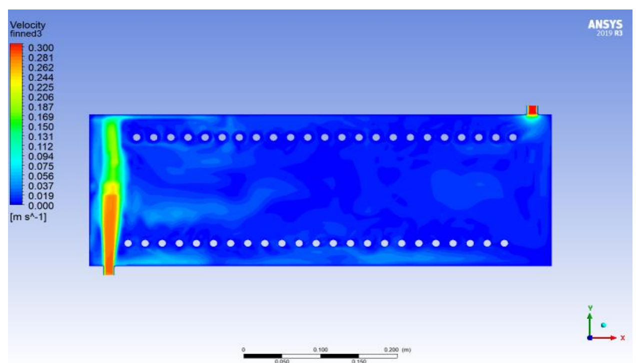


a) Finned

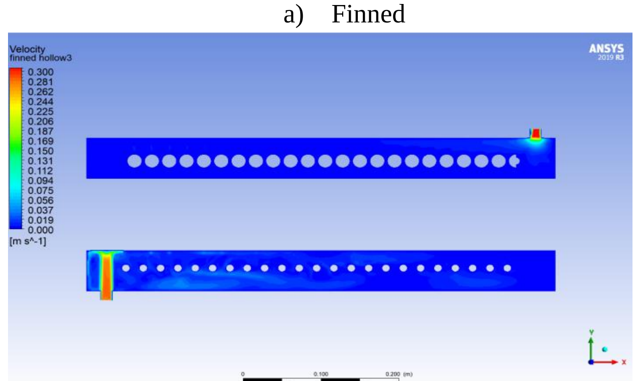


b) Finned hollow

Figure 20. Water velocity distributions inside shell for 2LPM



a) Finned



b) Finned hollow

Figure 21. Water velocity distributions inside shell for 3LPM

4.3 Experimental and Numerical Results Rapprochement

For reaching logical results, it is very important to confirm the previous work. After, constructing the model, the results were compared between the experimental and numerical data. Figure (22, 23) demonstrates the difference between the results of the water outlet temperature from the heat exchanger shell side for 2LPM flow to the experimental two models (finned helical coil with and without internal core) with the results obtained in the numerical process. An acceptable matching has been appearing between the results for the three models. As can be observed, experimental and numerical water outlet temperature histories during the charging processes reaching the steady state temperatures are satisfactory matched; thus, the numerical solution and its boundary conditions can be considered. The slight non-exact corresponding was not reached between the results due to the losses in the heat transfer from the shell and the location of thermocouples in the unit. The maximum uncertainty achieved of the water outlet temperatures results was 2.66% in the case without internal core, 2.16% for the finned heat exchanger with internal core.

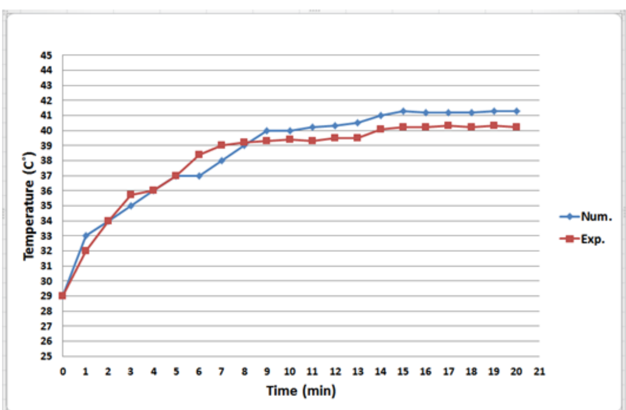


Figure 22. Comparison between experimental and numerical results for finned helical coil

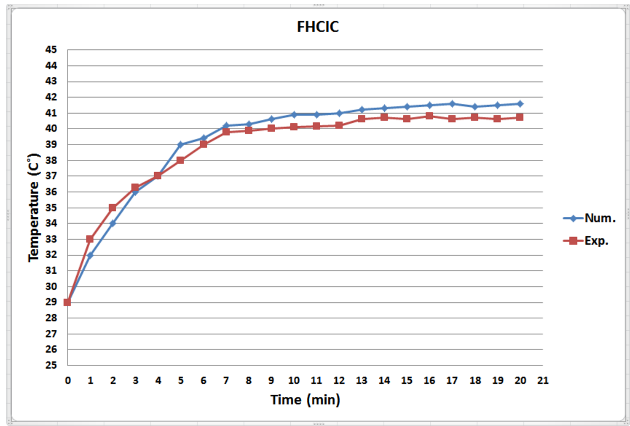


Figure 23. Comparison between experimental and numerical results for finned helical coil with internal core

5. Conclusions

From the results, Using finned helical coil heat exchanger with internal hollow core has direct consequences on the output performance, Increasing the heat transfer rates and coefficients, outlet temperature and water velocity inside the shell side and reducing the reaching time to the steady state condition and the difference of reaching time directly proportional to the size of the internal hollow core. With increasing of the water flow rates in the shell side lead to increasing of the mass fraction transferred from the vapor phase to liquid phase (dryness factor of refrigerant). Found that when using a heat exchanger with internal hollow core reduces the time needed to reach the steady state outlet temperature and directly proportional to the size of the core by 20% and slightly increasing the water outlet temperature by 5%.

NOMENCLATURE

Cp	Specific heat at constant pressure
g	Acceleration of gravity
P	Pressure
r	Radius
v	Velocity
x	Mass fraction factor
ρ	Density
μ	Viscosity
θ	Curvature angle

Subscripts

tp	Two Phase
a	Saturated vapor
f	Saturated liquid
θ	Tangential direction

Conflict of interest

There are not conflicts to declare.

6. References

1. A. Ameen, S. A. Mollik, G. A. Quadir, and K. N. Seetharamu, "Investigation into the Phase Change of Refrigerant in a Wire–and–Tube Consender of Refrigerator," *J. Teknol.*, vol. 43, no. 1, pp. 37–48, 2005.
2. Q. S. Mahdi, S. A. Fattah, and O. M. Jasim, "Experimental and Numerical Investigation to Enhance the Performance of Helical Coiled Tube Heat Exchanger by Using Turbulators," *J. Eng. Dev.*, vol. 18, no. 6, pp. 1–16, 2014.
3. M. Zareh, H. Shokouhmand, M. R. Salimpour, and M. Taeibi, "Numerical simulation and experimental analysis of refrigerants flow through adiabatic helical capillary tube," *Int. J. Refrig.*, vol. 38, no. 1, pp. 299–309, 2014.
4. H. Saffari and R. Moosavi, "Numerical study of the influence of geometrical characteristics of a vertical helical coil on a bubbly flow," *J. Appl. Mech. Tech. Phys.*, vol. 55, no. 6, pp. 957–969, 2014.
5. M. H. Alhamdo, M. A. Theeb, and A. S. Golam, "Finned double-tube PCM system as a waste heat storage," *IOP Conf. Ser. Mater. Sci. Eng.*, vol. 95, no. 1, 2015.
6. S. A. Nada, E. I. Eid, G. B. Abd El Aziz, and H. A. Hassan, "Performance Enhancement of Shell and Helical Coil Water Coolers Using Different Geometric and Fins Conditions," *Heat Transf. - Asian Res.*, vol. 45, no. 7, pp. 631–647, 2016.
7. F. Bagherzadeh, M. Saffar-Avval, M. Seyfi, and A. Abbassi, "Numerical investigation of nanofluid heat transfer in helically coiled tubes using the four-equation model," *Adv. Powder Technol.*, vol. 28, no. 1, pp. 256–265, 2017.
8. A. A. Hussain and A. S. Golam, "Experimental and Numerical Study of the Heat Transfer Coefficients of Double Pipe Helically Coiled Heat Exchanger," *J. Eng. Sustain. Dev.*, vol. 20, no. 06, 2016.
9. E. M. Abed and A. Abdulkadhim, "Modelling of phase change for Two-Phase Refrigerant Flow inside Capillary Tube under Adiabatic Conditions," *Int. J. Curr. Eng. Technol.*, vol. 6, no. 5, 2016.
10. V. M. Hameed and A. R. Al-Khafaj, "Experimental and numerical investigation of temperature distribution through shell and helical coil tube heat exchanger using Lab VIEW as a data acquisition program," *Int. J. ENERGY Environ.*, vol. 8, no. 3, pp. 253–270, 2017.
11. A. Alimoradi, M. Olfati, and M. Maghareh, "Numerical investigation of heat transfer intensification in shell and helically coiled finned tube heat exchangers and design optimization," *Chem. Eng. Process. Process Intensif.*, vol. 121, pp. 125–143, 2017.
12. T. H. Ariwibowo, F. H. Solihah, and B. Harjanto, "A numerical study of condensation heat transfer for R-134a in annular flow regime inside horizontal tube," in *AIP Conference Proceedings*, 2017, vol. 1788, no. January.
13. M. Sepehr, S. S. Hashemi, M. Rahjoo, V. Farhangmehr, and A. Alimoradi, "Prediction of heat transfer, pressure drop and entropy generation in shell and helically coiled finned tube heat exchangers," *Chem. Eng. Res. Des.*, vol. 134, pp. 277–291, 2018.
14. J. Wang, S. S. Hashemi, S. Alahgholi, M. Mehri, M. Safarzadeh, and A. Alimoradi, "Analysis of Exergy and energy in shell and helically coiled finned tube heat exchangers

and design optimization," *Int. J. Refrig.*, vol. 94, pp. 11–23, 2018.
15. R. López *et al.*, "Thermo-hydraulic analysis of the condenser (helically-coiled type) of an air-water heat pump, using CFD numerical simulation," *J. Phys. Conf. Ser.*, vol. 1221, no. 1, 2019.
16. M. Sabah Kassim and S. Fahad Lahij, "Numerical and Experimental Study the Effect of SiO2 Nanoparticles on the Performance of Double Pipe Heat Exchanger," *J. Eng. Sustain. Dev.*, vol. 23, no. 05, pp. 148–171, 2019.
17. *Ansys Fluent Theory Guide*, vol. 1. ANSYS, Inc., 2013.
18. J. Tu, G.-H. Yeoh, and C. Liu, *Computational Fluid Dynamics*, vol. 53, no. 9. Elsevier Ltd., 2018.