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Research Article

Numerical simulation of heat transfer characteristics of taper helical and spiral tube heat exchanger

Amit SHARMA[1], C. S. RAJORIA[1,*], Dharmendra SINGH[1], J. P. BHAMU[1], Ravi KUMAR[1]

[1]*Department of Mechanical Engineering, Government Engineering College Bikaner, India*

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ABSTRACT

The paper investigates the heat transfer characteristics of the heat exchanger under two different configurations namely taper helical tube and spiral tube type heat exchanger. The working fluid is water which flows in tubes kept horizontally and the mass flow rate ranges from 0.04 kg/s to 0.16 kg/s. The effect of tube configuration parameters and Reynolds number on heat transfer coefficient and Nusselt number were studied. Regression equations have been developed for both the configurations under horizontal orientation using various process parameters and regression tools available in Minitab Software. A comparison has been made between the theoretical results and the results obtained from simulation gives an error of 15% which is well within the permissible limit. The result indicates that the highest value of Nusselt number is greater by 21% than taper helical tube heat exchanger which is attributed to better heat transfer characteristics of spiral tube heat exchanger and hence the former is thermally superior over the latter. In this study, temperature and velocity contour have been developed for both the configurations which clearly indicates that, when heat transfer is considered, spiral configuration should be preferred over taper helical configuration.

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INTRODUCTION

Heat transfer between different fluids is one common physical process that occurs daily. This physics is used by engineers for various applications using heat exchanger devices. Various type of Heat exchanger is utilized in various kind of application, as in processing food, automobile radiators, condenser, air pre-heater, cooling towers, etc. The motivation behind developing a heat exchanger is to provide an effective way to transfer the heat from one to another fluid, by the method of direct or indirect contact. Three principles are used to transfer heat via conduction, convection, and radiation. Transfer of heat by radiation is considered to be negligible concerning conduction and convection in the heat exchanger. The conductive heat transfer can also be optimized by consideration of the minimum thickness of extremely conductive material wall. But convection plays

*Corresponding author.
*E-mail address: shakes_2001@gmail.com

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an important role to influence the overall performance characteristics of a heat exchanger. The heat transfer in pipe takes place from one moving stream to another due to forced convection. The cooler fluid gains heat from the heated fluid as it flows alongside or across it. Fig. 1 shows the classification of heat exchangers which can be used according to the need and application. Several studies have been carried out by researchers on laminar and turbulent flows with different system layout and designs. An experimental and numerical study was performed by Lui *et al.* [1] to observe the characteristics of friction factor and heat transfer of laminar flow (480<Re<1750) in a circular tube equipped with central slant rods under unchanging heat flux condition. The effect of three variables of central rods such as the pitch, slant angle, and length in the direction of the radius was discussed numerically. Several longitudinal vortex structures were observed due to the effect of the slant rod. Their investigation results revealed that friction factor and Nusselt number increased with a reduction in rod pitch but both have decreased with slant angle and length of the rod in the direction of radius. A similar numerical investigation on turbulent flow (5000<Re<20,000) under the uniform wall of heat flux has been conducted by Oni *et al.* [2] which used circular tube fitted with various twisted tapes in it for turbulent flow. The results from the renormalized group (RNG) k-ϵ model depicted that triangular cut twisted tape has a higher impact on the execution of the tube due to the additional mixing phenomenon. Nusselt number, friction factor and thermal performance factor were approximate 2.18, 3.15, and 1.43 times than plain tube respectively. Some useful numerical investigations have been carried out on corrugated geometries Mohammed *et al.* [3] have numerically investigated heat and fluid flow in the transversely corrugated circular tube having turbulent flow (5000<Re<60,000) under constant heat flux, and the water is used as a working fluid. The result indicates that the Nusselt number raises because of the roughness height and width while the Reynolds number was inflated with the reductions of roughness pitch. The friction factor additionally expanded with diminishing pitch and width or increasing the height while it is found independent of Reynolds number.

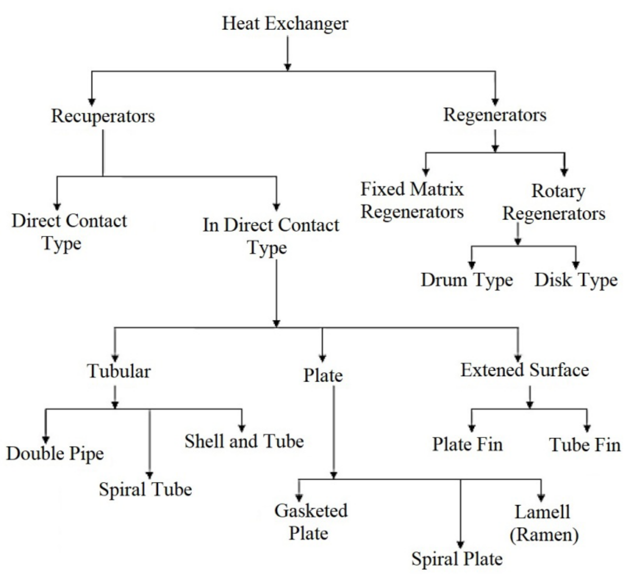

Figure 1. Types of the heat exchanger.

The thermal performance of three starts spirally corrugated tube for water has been investigated by Kareem *et al.* [4] at low Reynolds number (100<Re<1300) with constant wall temperature. Effects of new spiral corrugation characteristics on overall thermal performance were studied and it was found that spiral corrugation has improved the heat transfer. The best thermal performance occurred in a narrow range of Reynolds numbers. Nanofluid is becoming a promising fluid in various heat transfer applications and has drawn the attention of the researchers to investigate its effect on various designs and geometries of heat exchangers. Saeedinia *et al.* [5] experimentally investigated the pressure drop properties in a circular tube inserted with different coils for laminar flow under uniform heat flux and the working fluid used was nanofluid Cuo/Base oil. The experimental outcomes indicated that the wire coil inserted in the tube increases the heat transfer coefficient. Inserting Nanoparticle with increased wire diameter had increased the heat transfer. The empirical correlations were generated for foreseeing the Nusselt number and friction factor of the Nanofluid flow within the coiled wire fitted tube. Another investigation made by Sundar *et al.* [6] analyzed forced convection heat transfer coefficient and friction factor properties of Fe_3O_3 Nanofluid for turbulent stream inside the circular tube with constant wall temperature. They studied different volume concentrations of Nanofluid and demonstrated that heat transfer was enhanced with volume concentration nanofluid. Also, the Nusselt number and friction factor have increased with an increase in volume concentration of Nanofluid. Wu *et al.* [7] have experimentally investigated the pressure drop and convective heat transfer characteristics of H_2O and alumina/ H_2O Nanofluid in a double helical pipe type heat exchanger for both laminar flow and turbulent flow inside the helical coil using different weight concentration of Nanofluid. Hot water and Nanofluid were allowed to flow in closed circuit and cold water was constrained flow in the open circuit. It has been observed that there is an increase in Nusselt number and reduction in friction factor concerning an increase in Reynolds number. An overall investigation of pressure drop-off and convective heat transfer behavior of Nanofluid (CuO) has been done by Akbaridoust *et al.* [8] for steady-state and laminar flow (200<Re<1000) in the helically coiled tube at a uniform wall temperature. They have used coils with different curvature and torsion ratio for in the study. It was found that heat transfer intensified with a greater curvature ratio. Heat transfer and pressure drop increase with a higher value of particle volume fraction in Nanofluid.

Akhavan-Behabdi *et al.* [9] used oil and Nanofluid with different weight concentration as working fluid and carried out experimental studies on convective heat transfer with laminar flow inside the vertically helical coiled tube with uniform wall temperature condition Results indicated that heat transfer rate increased with increase in Nanofluid mass concentration and Reynolds number. Decreasing the coil to tube diameter ratio and increasing the coil pitch to the tube diameter ratio resulted in augmenting the heat transfer rate for base fluid as well as Nanofluid. Nusselt numbers of Nanofluid in helical coils were higher than base fluid. In a similar investigation Naphon [10] experimentally investigated on heat transfer and flow characteristics of Nanofluid in the horizontal spirally coiled tube for turbulent flow (2000<Re<12000) under constant wall temperature condition and TiO_2 Nanofluid in de-ionized water as base fluid. The results showed that heat transfer enhances on decreasing curvature ratio and increasing Nanofluid concentration. The Friction factor tends to raise with decreasing curvature ratio. Two correlations of Nusselt number and friction in the case of a horizontal spiral coiled tube were proposed. Dogan *et al.* [11] have experimentally investigated the effect of inlet temperature of cold fluid on the thermal performance of compact heat exchanger. They performed experiment on two mini channel flat tube with different number of multi-louvered fin rows at constant temperature. They have inferred that two fin rows are more effective in terms of overall thermal conductance. An extensive focus has been laid on the different orientations of tubes used in heat exchangers the most popular among them are spiral, conical, helical, etc. Purandare *et al.* [12] has performed an experimental study of the thermal analysis of a conical coil heat exchanger with hot and cold water. The results of the experiment with stream rate 10–100 lph (500<Re<5000) and 30–90 lph separately indicated that the Nusselt number and friction factor are directly related to stream rate, tube diameter, curvature ratio, and cone angle. Nusselt number improved with an increase in tube side stream rate and reduced with an increase in shell side stream rate and increment in cone angle or tube diameter. The numerical investigation of flow structures developed around various layouts of impingement plate with or without tube bank under isothermal condition was carried out by Maurya *et al.* [13]. Jamshidi *et al.* [14] experimentally studied the enhancement rates of heat transfer in shell and coil tube heat exchangers. Hot water flows in the helical tube and cold-water flow in the shell side. Experimental apparatus and Taguchi method are utilized to study the effect of fluid flow and geometrical variables such as coil diameter, coil pitch, and mass flow on heat transfer. Results revealed that increment variables in shell and tube can improve the heat transfer rate. Fakiri *et al.* [15] have done numerical analysis of turbulent forced convection flow and heat transfer characteristics in rectangular pipe with baffles attached inside the pipe under constant heat flux condition. They observed performance with variations of baffle distance and baffle thickness. Naphon *et al.* [16] have studied the effect of curvature ratio on heat transfer and flow development in the horizontal spirally coiled tube under constant wall temperature. They used water as base fluid and used k-ϵ standard model and finite volume method for simulation. It was found that the centrifugal force has a substantial effect on the enhancement of heat transfer and pressure drop. The heat transfer rate is directly proportional to the mass flow rate of water. Centrifugal force increased on increasing curvatures ratio. Nusselt number was higher for the lower curvature ratio. Pressure drop per unit length found from a higher curvature ratio was lower. Jalaluddin and Miyara [17] have discussed thermal performance and pressure drop of the spiral tube ground source heat pump (GHE). They used water as the base fluid for laminar flow (Re=1900) and turbulent flow (Re=7600) with constant temperature wall conditions. The heat exchange rate per meter borehole depth of the spiral tube GHE with different pitches and their pressure drop are studied. It was found that the spiral pipe provides a better heat exchange rate than a straight pipe. A pressure drop of water flow increases due to an increase in the length of pipes per meter borehole depth, therefore pumping power is also increased.

Jundika C. Kurnia *et al.* [18] have numerically investigated flow behavior in a helical tube with twisted tape inserts for a range of Reynolds number (100–2000) under constant wall temperature condition. They reported optimal heat transfer with tape insert on straight and helical coil tube and numerical outcomes indicated four times enhancement in heat transfer in a helical tube as compared to the straight tube heat exchanger. The heat transfer characteristics of helical coil tube was investigated by Nashine *et al.* [19] They have numerically studied the variations of inlet velocity or pressure and Dean number for both turbulent and laminar flow under constant wall heat flux. It was reported that heat transfer phenomena enhanced with increase in Dean number. Yunmin Liang *et al.* [20] analyzed heat transfer and flow characteristics of laminar flow in a plain tube fitted with center tapered wavy tape insert under constant heat flux condition and compared with conventional wavy tape insert. The results depicted that center tapered wavy tape inserts have a significant effect on the reduction of flow resistance and improvement of overall performance when the amplitude of laminar flow was high. Mahmoud Abdel Maged [21] has studied thermal performance characteristics of a triple spiral tube in the turbulent region Reynolds number (6400–57400) and also compared with a double spiral tube heat exchanger using alumina-water nanofluid as base fluid and observed 20.8% increment in heat transfer due to use of nanofluid. The effects of screw pitches, rotation angles and nanoparticle mass fractions on flow and heat transfer performances were conducted by

Xinfeng Zhai *et al.* [22]. It was found that Nusselt number enhanced with decreasing screw pitch and increasing nanoparticle mass fraction. It was also found that maximum heat transfer enhancement occurs at rotation angle $\beta = 45^0$ and minimum at $\beta = 90^0$.

Most of the research groups have worked on a passive method for enhancement of heat transfer and the present work also investigates the effect of coil geometry and secondary flow characteristics. However, the proposed coil geometry that is taper helical and spiral tube type heat exchanger has not been exclusively compared based on fluid flow and heat transfer. An experimental set up is fabricated for a simple straight tube having water as base fluid. The experimental results of the plain tube are validated with CFD simulation while using the Nusselt number and heat transfer coefficient as a responsive parameter.

Experimental Setup and Heat Exchanger Design

An experimental analysis has been carried out for a simple straight tube (SST) and its overall performance has been investigated. The simulation was carried out for 3 tube configurations namely SST, taper helical, and spiral tube. The schematic diagram of the experimental setup used for SST is shown in Fig. 2(a-b). All three proposed tubes were made of copper and the length of SST is 650 mm, diameter 15 mm, and thickness 1 mm. Helical pipe and spiral pipe tubes are developed using bending a 15 mm diameter horizontal copper tube into a helical and spiral coil of 5 turns. The exploratory arrangement has been shown in Fig. 3 (a-b), assembled with measuring devices such as a thermocouple, a stream meter with Arduino chip, a U-tube manometer for measuring pressure difference, and a heater for water heating. Water is used as a working fluid for this test with a controlling valve for changing the mass flow rate of hot water. Heating coils are made of excessive conducting material and cross-segment of the pipe is polished from the inside, to avoid pressure variation.

Error Analysis

Error analysis is performed to authenticate the results obtained to calculate the heat transfer performances with the help of Eq. 1 [22].

$$\frac{U_{Nu}}{Nu} = \sqrt{\left(\frac{U_h}{h}\right)^2 + \left(\frac{U_k}{k}\right)^2} \tag{1}$$

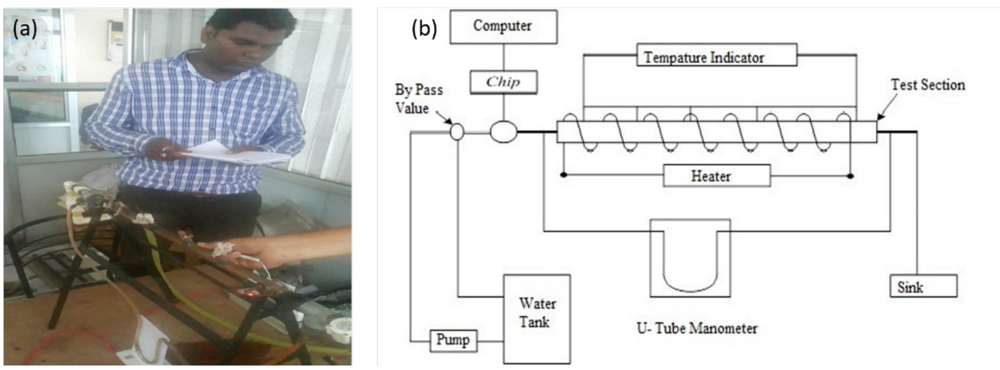

Figure 2(a-b). Experimental setup of test rig for simple straight tube.

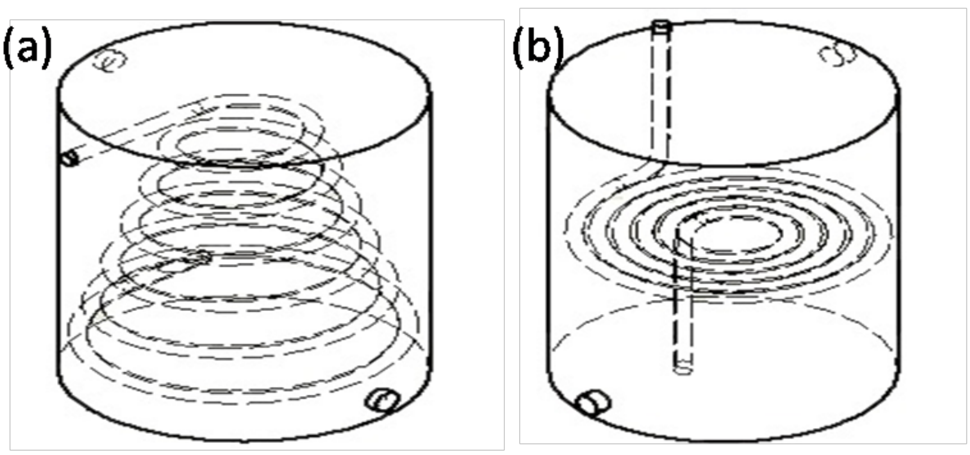

Figure 3. (a) Helical heat exchanger tube design, (b) Spiral heat exchanger tube design.

Table 1. An error of physical quantities

Physical quantities	% Error	Physical quantities	% Error
Mass stream rate	±2.9	Resistance	±0.4
Viscosity	±3.9	Heat flux	±2.3
Reynolds number	±4.7	Temperature	±3.9
Voltage	±0.054		

It can be found that the error of the Nusselt number is approximately ±5.27% and the error of other physical quantities are shown in Table 1.

Numerical Approach

The problem under investigation in the present work was assumed to be three-dimensional and the physical properties of the fluid were assumed to be constant. The governing equation of fluid flow, mass conservation, conservation of momentum and conservation of energy that describe the turbulent heat transfer flow characteristics of the spiral and taper helical tube type heat exchangers are expressed in Eq. 2–9 as follows [23].

Governing Equation of Fluid Flow

$$\frac{\partial}{\partial t}(\rho\phi) + div(\rho\nabla\phi - \Gamma_\phi grad\phi) = S \qquad (2)$$
$$Transient\ +\ Advection\ -\ Diffusion\ =\ Source$$

Governing Equation of Mass Conservation

$$\frac{\partial\rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \qquad (3)$$

In compact vector form notation

$$\frac{\partial\rho}{\partial t} + div(\rho u) = 0 \qquad (4)$$

Governing Equation of Conservation of Momentum

$$\rho\frac{Du}{Dt} = \frac{\partial(-p+\tau_{xx})}{\partial x} + \frac{\partial\tau_{yx}}{\partial y} + \frac{\partial\tau_{zx}}{\partial z} + S_{Mx} \qquad (5)$$

The expression in Eq. 5 shows the "Navier-Stokes" equation in the x-direction. The expressions for y and z-direction are as follows.

$$\rho\frac{Dv}{Dt} = \frac{\partial\tau_{xy}}{\partial x} + \frac{\partial(-p+\tau_{yy})}{\partial y} + \frac{\partial\tau_{zy}}{\partial z} + S_{My} \qquad (6)$$

$$\rho\frac{Dw}{Dt} = \frac{\partial\tau_{xz}}{\partial x} + \frac{\partial\tau_{yz}}{\partial y} + \frac{\partial(-p+\tau_{zz})}{\partial z} + S_{Mz} \qquad (7)$$

Governing Equation of Conservation of Energy

The fluid energy (E) is directly equal to the summation of internal energy (i), kinetic energy 1/2 ($u^2 + v^2 + w^2$), and potential energy. So, the expression for energy equation is

$$\begin{aligned}\rho\frac{DE}{Dt} = -div(pu) + \Bigg[&\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{xy})}{\partial y} + \frac{\partial(u\tau_{xz})}{\partial z} \\ &+ \frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} \\ &+ \frac{\partial(w\tau_{xz})}{\partial x} + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z}\Bigg] \\ +div\left(K\, grad T\right) + S_E\end{aligned} \qquad (8)$$

$$Here\ \ E = i + \frac{1}{2}(u^2 + v^2 + w^2) \qquad (9)$$

According to Dittus-Boelter [24], co-relation equation for Nusselt number is mentioned as Eq. 10

$$Nu = 0.02Re^{0.8}Pr^{0.3} \qquad (10)$$

According to Gnielinski [25], the Co-relation equation for Nusselt number is mentioned as Eq. 11

$$Nu = \frac{\left(f/8\right)\left(Re-1000\right)Pr}{1+12.7\left(f/8\right)^{0.5}\left(Pr^{0.67}-1\right)} \qquad (11)$$

Generation of Correlation Equation Using Matlab

MATLAB/ Simulink software is used to generate a correlation equation that is used to define the relationship between Nusselt, Reynolds, and Prandtl number (Eq. 12–13). Fig. 4 shows the results of the correlation equation for both taper helical and spiral tube configurations in the curve fitting tool of MATLAB. The unknown variables "a" and "b" were derived with confidence bounds of 95%. The correlations for both tube configurations were fitted by numerical results, which can be utilized for evaluating the heat transfer coefficient in designing of tubes.

For taper helical tube;

$$Nu = Re^{1.042}Pr^{-2.773} \qquad (12)$$

For spiral tube;

$$Nu = Re^{1.152}Pr^{-3.185} \qquad (13)$$

RESULTS AND DISCUSSION

The performance of all the three tube configurations was investigated. The experimental results of heat transfer

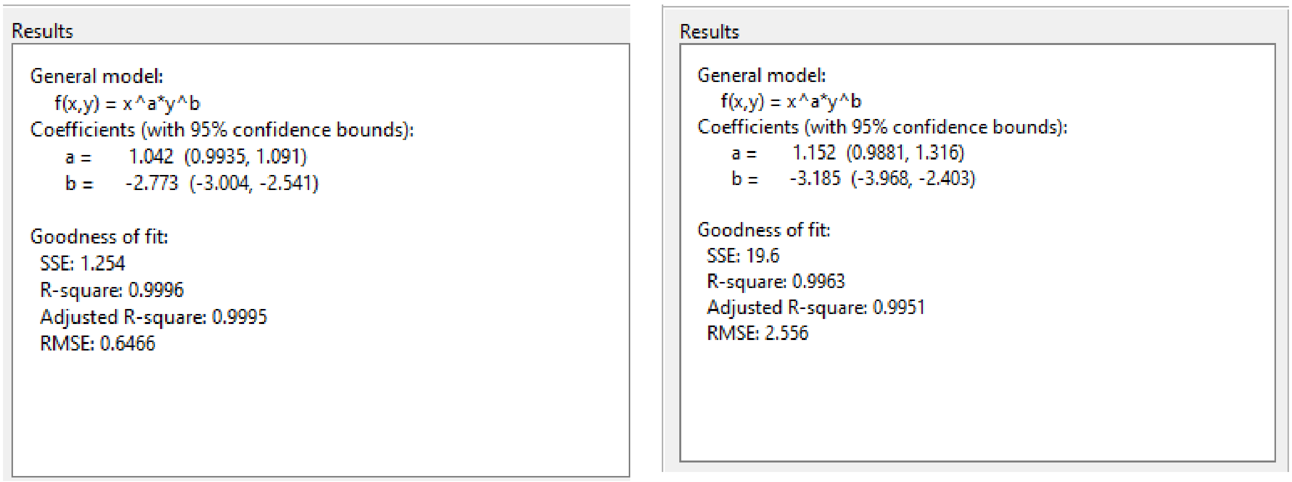

Figure 4. Values of unknown variables in correlation.

on a simple horizontal tube were simulated using ANSYS FLUENT 14.5 software to validate the outcomes of CFD analysis. The simulation results of the remaining two tubes (taper helical tube and spiral tube) were validated based on a simple tube. For simulation work, the same dimensions were considered for both taper helical and spiral tubes and the co-relation equations were also generated for both tubes for further analysis. The grid independence test and validation test were performed for the simple horizontal tube, afterwards, the comparative analysis of simulated results for both taper helical and spiral tube configurations were carried out.

Grid Independency Test

The grid independency test was carried out to examine the variation in outlet temperature concerning the alteration inside the grid size. The modification in the grid size was assured by considering the fluent solver and its outcomes were observed as the output temperature.

The grid dimension range lies between 4–5 millimeters. The tetrahedral mesh was preferred with the RNG k-ε turbulence model and it was observed that the outlet temperature as noted from the analysis is aligning around the experimental results pointed in Fig. 5. The results from the grid independency test are shown in Table 2 with the optimized mesh size mentioned as highlighted until no major change in the result was found.

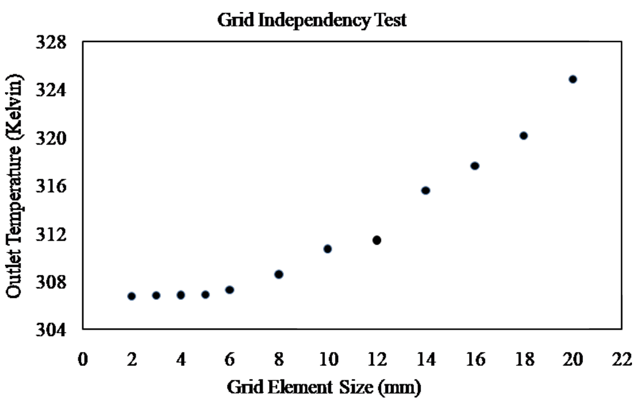

Figure 5. Grid independency test for horizontal tube.

Prescribed Boundary Conditions

The boundary conditions followed to develop the proposed model are as follows [15]

i. **Inlet:** At the inlet, constant mass stream rate and corresponding range Reynolds numbers of 3000–14000 at a constant temperature.
ii. **Outlet:** At the outlet, the prescribed pressure is atmospheric and at zero temperature gradient through the stream.
iii. **Wall:** At the wall, constant wall temperature condition with no-slip is applied set.

Table 2. Grid independency test results

Element Size	Straight Tube	Helical Tube	Spiral Tube
20	324.83	354.08	367.63
18	320.16	344.52	357.85
16	317.64	332.71	346.33
14	315.59	326.75	335.05
12	311.39	323.59	330.6
10	310.73	320.41	329.68
8	308.6	318.55	326.41
6	307.3	315.26	320.85
5	306.9	314.85	319.85
4	**306.89**	**314.35**	**319.35**
3	**306.85**	**314.30**	**319.30**
2	**306.84**	**314.28**	**319.27**

Table 3. Heat transfer database for proposed taper helical tube

Mass stream rate (kg/s)	Velocity (m/s)	Hot Junction (Temp in °C)		Cold Junction (Temp in °C)		Nu_{CFD}
		Inlet	Outlet	Inlet	Outlet	
0.04	0.23	50	34.34	27	40.12	22.07
0.07	0.40	50	34.13	27	40.76	39.55
0.10	0.57	50	34.27	27	40.95	55.62
0.13	0.74	50	34.05	27	41.02	74.13
0.16	0.91	50	33.96	27	41.12	92.17

Table 4. Heat parameter for proposed taper helical tube

Q	T_{wall}	T_{bulk}	H	Re	Nu_{CFD}	Nu_{D-B}
2619.60	32.11	42.17	1471.50	3432.80	22.07	38.79
4645.78	32.11	42.06	2637.19	5970.09	39.55	60.39
6578.28	32.11	42.13	3708.11	8507.37	55.62	80.18
8671.37	32.11	42.02	4942.19	11044.67	74.13	98.80
10732.68	32.11	41.98	6144.91	13581.95	92.17	116.57

Table 5. Heat transfer database for proposed spiral tube

Mass stream rate (kg/s)	Velocity (m/s)	Hot Junction (Temp in °C)		Cold Junction (Temp in °C)		Nu_{CFD}
		Outlet	Inlet	Outlet	Inlet	
0.04	0.23	50	33.1	27	33.55	27.43
0.07	0.40	50	33.92	27	35.2	43.59
0.10	0.57	50	33.22	27	35.73	67.63
0.13	0.74	50	32.84	27	36.87	91.94
0.16	0.91	50	32.5	27	37.66	117.78

Comparative Study of Proposed Designs for Different Mass Stream Rates

A comparison of results for both taper helical and spiral tubes is presented at different mass stream rates for the same physical conditions. The heat transfer database for a taper helical tube and the spiral tube is presented in Tables 3 and 5 respectively. The tables present temperature at the hot and cold junction as inlet and outlet temperature respectively at different mass flow rates and the velocity of fluid flow.

Heat parameters for helical tube and spiral tube are presented in Tables 4 and 6 respectively. The Reynolds number (Re) was considered in the range of 3000 to 14000 for the study of both tubes. The CFD results were correlated with Dittus-Boelter's (D-B) correlation to observe the values of Nusselt number

Variation in heat transfer coefficient and Nusselt Number at different mass stream rates are presented in Table 7 and the corresponding plots are shown in Fig. 6 and 7. The heat transfer coefficient of both the tubes increases with the mass stream rate and at the mass stream rate of 0.16 kg/s, there is a significant improvement of 21 % in the Nusselt number of the spiral tube as compared to taper helical tube. Therefore, it can be inferred that a spiral tube is better in terms of performance as compared to taper helical and indicative of better heat transfer characteristics of the spiral tube owing to its improved secondary heat flow.

Variation in heat transfer coefficient and Nusselt Number with Reynolds number is shown in Fig. 8 and 9, and a similar trend has been observed which indicates that as the flow approaches towards more turbulence, better heat transfer is achieved.

Table 6. Heat parameter for the proposed spiral tube

Q	T_{wall}	T_{bulk}	H	Re	Nu_{CFD}	Nu_{D-B}
2827.03	33	41.55	1829.30	3432.80	27.73	38.79
4707.25	33	41.96	2906.55	5970.09	43.59	60.39
7017.39	33	41.61	4509.14	8507.37	67.63	80.18
9329.20	33	41.42	6129.91	11044.67	91.94	98.80
11709.6	33	41.25	7852.534	13581.95	117.78	116.57

Table 7. Heat parameters comparison of taper helical and spiral tube

Mass Stream rate (kg/s)	Nusselt Number (Nu)		Heat Transfer co-efficient (h) (W/m² – K)	
	Taper Helical	Spiral	Taper Helical	Spiral
0.04	22.07	27.43	1471.50	1829.30
0.07	39.55	43.59	2637.19	2906.55
0.10	55.62	67.63	3708.11	4509.141
0.13	74.13	91.94	4942.19	6129.91
0.16	92.17	117.78	6144.918	7852.53

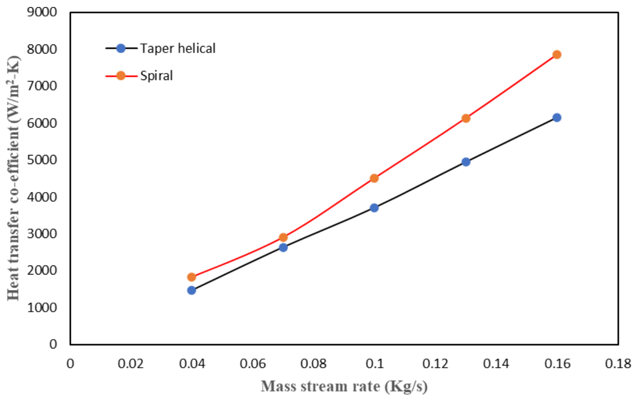

Figure 6. Variation of the heat transfer coefficient at different mass stream rates.

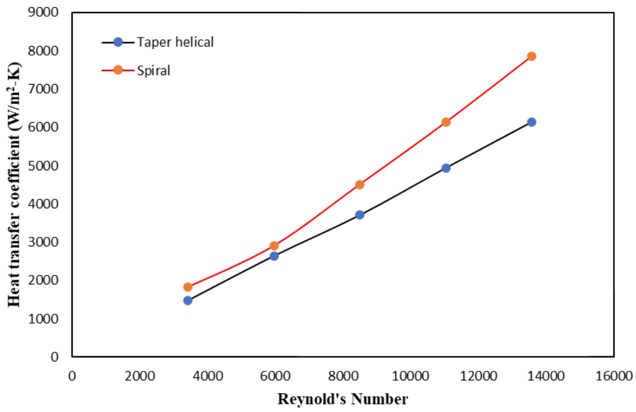

Figure 8. Variation of the heat transfer coefficient with Reynolds number.

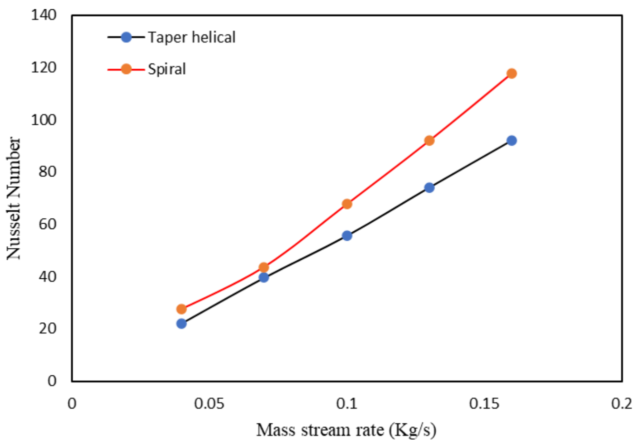

Figure 7. Variation of the Nusselt number at different mass stream rates.

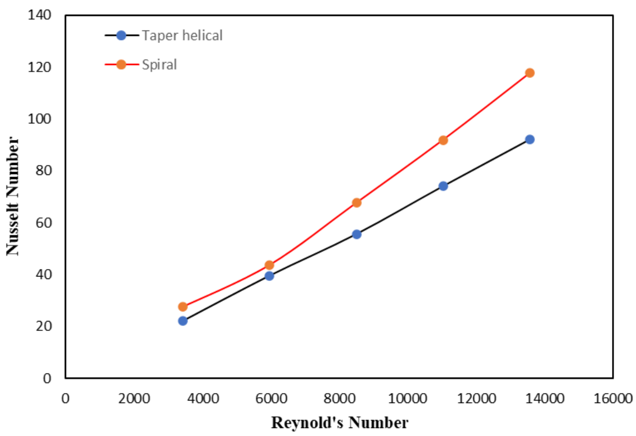

Figure 9. Variation of Nusselt number with Reynolds number.

CFD Generated Results

The CFD simulation tool (ANSYS FLUENT) has been used to find out heat transfer in curved tubes. The temperature contour represents the visual outcomes at different mass stream rates for both taper helical and spiral tubes.

Temperature Contour for Spiral Tube

The temperature and velocity contours are represented in Fig. 10–11 for the spiral tube at various mass flow rates.

Temperature Contour for Taper Helical Tube

The temperature and velocity contours are represented in Fig. 12–13 for taper helical coiled tube at various mass stream rates. The wall temperature contours are represented in Fig. 14 for spiral and taper helical tube at various mass flow rates.

The Nusselt number increases proportionally with the Reynolds number. This can be attributed to the changes in mass streaming concerning the rate of flow presented in Table 3 & 5 that leads to concentrated secondary flow in the tube because of the outward moving (centrifugal) action of fluid particles inside the twisted tube due to curvature. The intensity of the concentration flow effect is higher in the upper section as compared to the lower section of the tube. Therefore, the rate of heat transfer in the upper section is higher than the lower section of the tube as presented in Fig. 10–14 at various mass stream rate values.

Regression Analysis on Heat Exchanger Design

The algebraic expression of the regression lines is also known as the regression equation. It is used to express the values of the identified variable from the given values of independent variables. Essentially, we use the regression equation to express the values of an identified variable (Eq. 14–15). This variable is referred to as the **outcome variable.**

Conclusive equation of regression for Helical Tube

$$Nu = -0.00460 + 0.000014\,Re + 0.014971\,h \qquad (14)$$

Conclusive equation of regression for Spiral Tube

$$Nu = 0.4836 - 0.000317\,Re + 0.015488\,h \qquad (15)$$

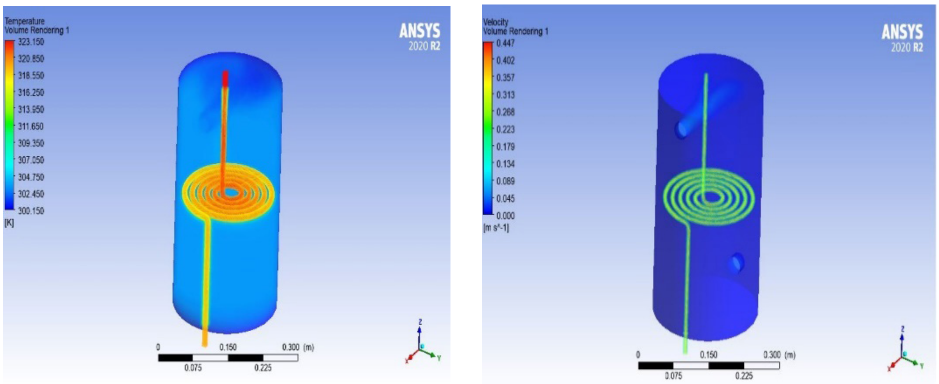

Figure 10. Temperature and velocity fluctuation at a mass stream rate of 0.04 kg/s (Spiral Tube).

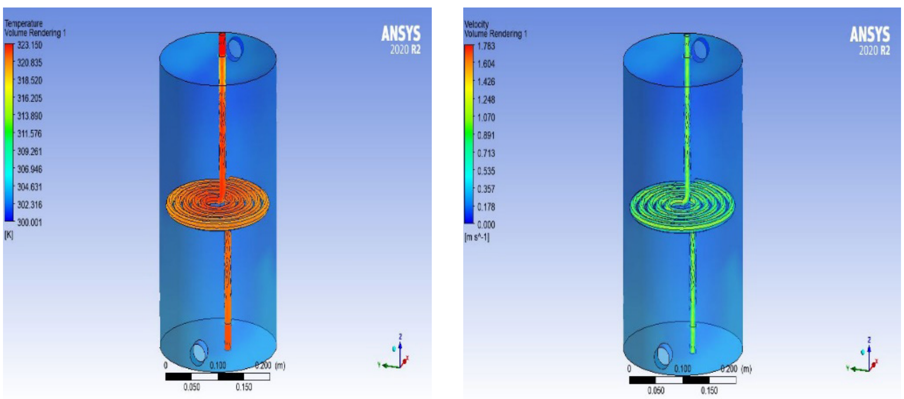

Figure 11. Temperature and velocity fluctuation at a mass stream rate of 0.16 kg/s (Spiral Tube).

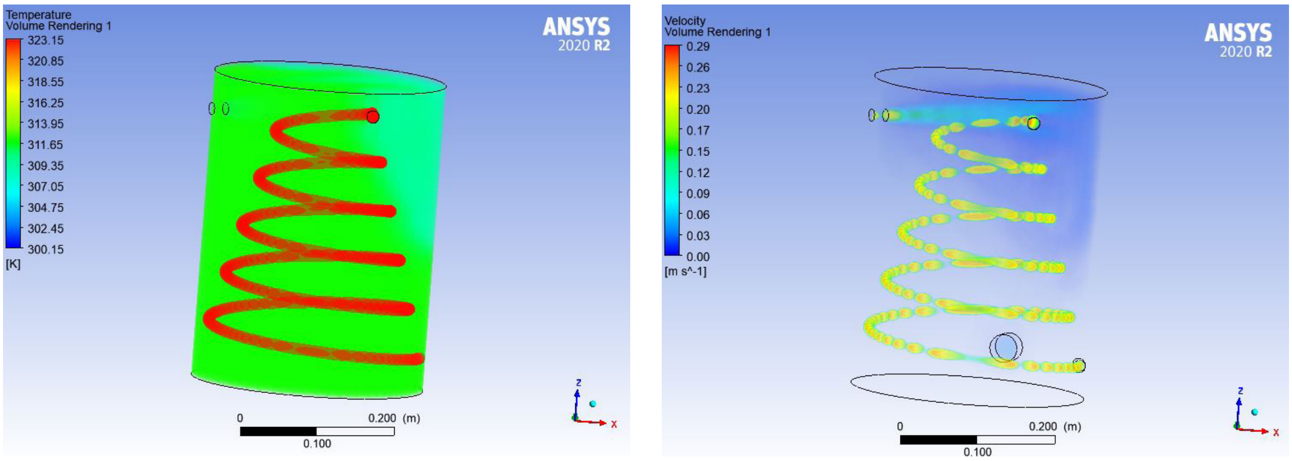

Figure 12. Temperature and velocity fluctuation at a mass stream rate of 0.04 kg/s (Helical Tube).

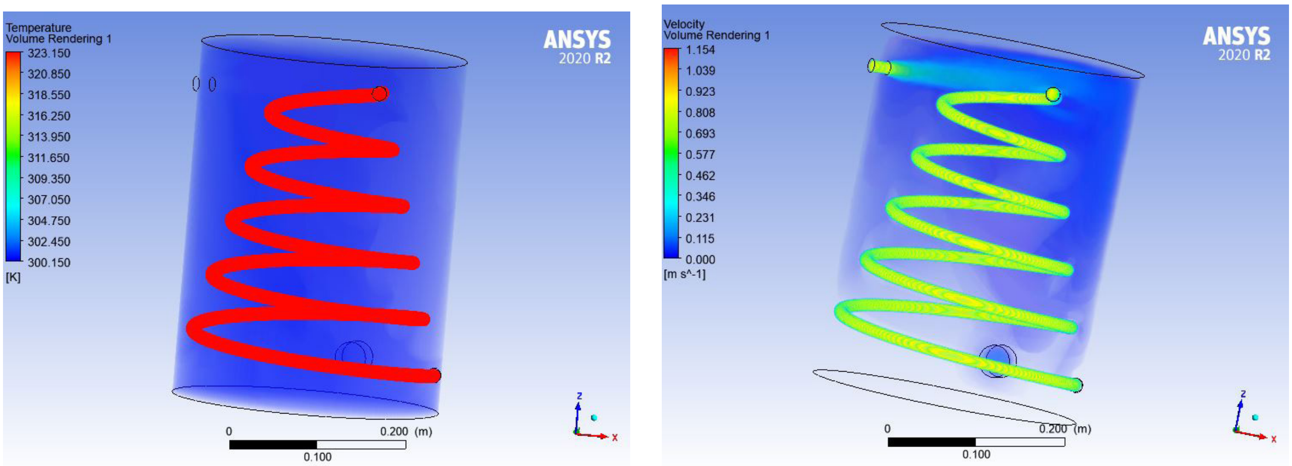

Figure 13. Temperature and velocity fluctuation at a mass stream rate of 0.16 kg/s (Helical Tube).

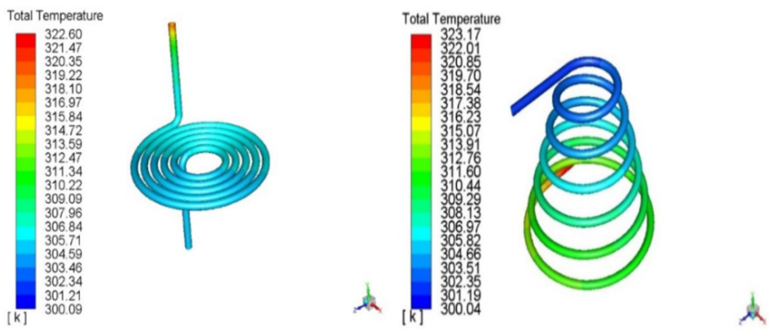

Figure 14. Temperature contour for the spiral and taper helical tube.

Experimental Validation

In the present investigation, a total of eight experiments were performed for different mass stream rates of cold fluid and hot fluid with three different designs of heat exchangers i.e. simple horizontal tube, taper helical, and spiral tube heat exchanger. All experiments were carried out in the month of November 2019 at Bikaner. The validation is done by using a heat transfer correlation with experimental and CFD results. Table 8 presents experimental data and simulation results of hot fluid outlet temperature and cold fluid outlet temperature. The error percentage is less than 15% which is well within the limit. Fig. 15 shows the

Table 8. Validation of experimental and CFD results with correlation equations

$\dot{m}$ (kg/s)	Re	Experimental Readings			Simulation Readings			Nu (D-B)	Gnielinski Correlation
		T_{in} (K)	T_{out} (K)	Nu (Exp)	T_{in} (K)	T_{out} (K)	Nu (CFD)		
0.06	3694.80	303	308.40	38.24	303	308.91	53.78	36.38	30.80
0.08	4926.40	303	307.90	52.58	303	308.82	63.07	45.80	42.26
0.10	6158.00	303	306.70	63.26	303	307.23	73.52	54.75	53.06
0.12	7509.39	303	306.30	70.11	303	306.61	83.82	64.16	64.40
0.14	8760.95	303	305.40	79.6	303	305.89	90.99	72.59	74.54
0.16	10012.52	303	305.35	89.12	303	305.56	106.31	80.77	84.41
0.18	11264.08	303	304.60	99.8	303	304.85	113.12	88.75	94.04
0.20	12515.64	303	304.29	105.17	303	304.49	114.47	96.56	103.49

results obtained in Table 8 and the simple horizontal tube is used to validate both experimental and CFD results. It is evident from the figure that the CFD results are sufficiently close to the experimental results while both CFD and experimental results are close to the results obtained from the two correlations which justify the use of the CFD tool for the analysis of the thermal capability of both the heat exchangers.

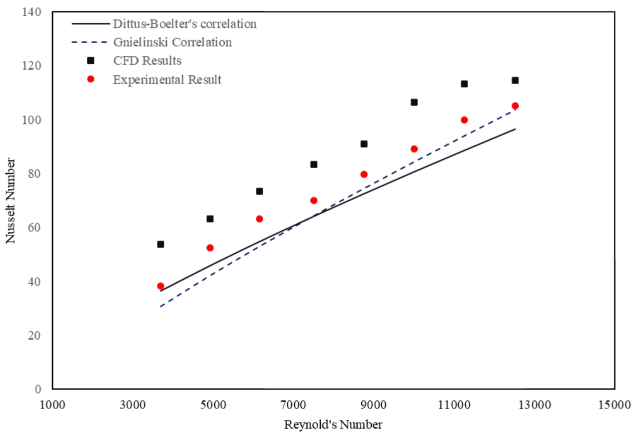

Figure 15. Validation of experimental and simulation results with correlation equations.

CONCLUSION

Simulation results were ensured and validated, that both simulation and experimental data are similar for the same tube type (simple horizontal). The regression equation was generated for both the tubes. The consequent results are drawn from the present investigation as:

1. Experimental and CFD outcomes variant for the simple horizontal tube has shown accurate agreement with each other, and the error is less than 15%.
2. The contours shown in the investigation reveals that the heat transfer area of the spiral tube is more than the taper helical tube. Due to more contact surfaces, the heat transfer is better in a spiral tube.
3. Based on outcomes acquired from the numerical solution, a mathematical relationship has been built up in Minitab software for determining Nusselt number.

. Conclusive equation of regression for Taper Helical Tube

$$Nu = -0.00460 + 0.000014\, Re + 0.014971\, h$$

Conclusive equation of regression for Spiral Tube

$$Nu = 0.4836 - 0.000317\, Re + 0.015488\, h$$

4. The spiral tube is preferred over the taper helical tube due to the improved heat transfer coefficient over the spiral tube.
5. The correlation obtained for the calculation of Nusselt number from MATLAB are:
   For taper helical tube;

$$Nu = Re^{1.042}\, Pr^{-2.773}$$

   For spiral tube;

$$Nu = Re^{1.152} Pr^{-3.185}$$

NOMENCLATURE

U	Uncertainty
Re	Reynolds Number
ρ	Fluid density (kg/m^3)
T	Temperature (K)
Q	Heat (J)
ε	Turbulent Dissipation Rate (J/kg. s)
k	Turbulent Kinetic Energy
Pr	Prandtl Number
h	Heat Transfer Coefficient (W/m^2 – K)
D	Diameter (m)

F	Force (N)
M	Mass (kg)
Nu	Nusselt Number
T_{bulk}	Bulk Temperature
T_{wall}	Wall Temperature (K)
E	Kinetic Energy
S	Body Force
F	Friction Factor
V	Velocity of Fluid (m/s)
K	Thermal Conductivity (W/m – K)
L	Length (m)
A	Acceleartion (m²/s)

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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