Numerical Investigation on the Performance of the Helical and Conical Coil Heat Exchanger Configurations in the Dynamic Mode of Heat Extraction

Hitesh Khurana,[1, a)] Rudrodip Majumdar,[2, b)] and Sandip K Saha[1, c)]

[1)] *Department of Mechanical Engineering, Indian Institute of Technology Bombay, Mumbai- 400076, INDIA.*
[2)] *EEP, School of Natural Sciences and Engineering, National Institute of Advanced Studies, Bengaluru-560012, INDIA*

[a)] *Electronic mail: khurana.hitesh25@gmail.com*
[b)] *Electronic mail: rudrodip@nias.res.in*
[c)] *Corresponding author: sandip.saha@iitb.ac.in*

Abstract. Owing to the high heat transfer performance, compact design and large surface area per unit volume, the curved and helical circular tubes have many applications such as heating, refrigeration, air conditioning, steam generation, heat extraction etc. One of the ways to improve the performance of water heating systems is to enhance the performance of immersed heat exchanger. In solar heating systems, immersed heat exchangers of curved shape are used to charge and discharge sensible heat water storage tanks. Effect of arrangement and position of the tubes, geometry of coil, configuration, shape, flow rate, type of working fluid, Reynolds number associated with flow, and inlet temperature to coil, on the performance of storage systems have been studied in the literature. Rate of discharge, heat transfer coefficient, effectiveness of heat exchanger, heat transfer rate, discharging efficiency and increment in temperature of outlet fluid are the standard performance parameters to evaluate the heat transfer and fluid flow phenomenon in the curved tubes. Heat transfer characteristics are investigated and analyzed for one helical and two conical coil (conical and inverted conical) configurations using a three-dimensional unsteady numerical model. The numerical model is validated against reported experimental result and a good agreement is found. For the same length of the coil, the inverted conical configuration presents more heat transfer surface area to the incoming hot fluid entering the thermal energy storage tank, as compared to conical coil and helical coil configurations, leading to higher extraction of thermal energy. Based on the performance parameters, inverted conical coil experiences enhanced heat transfer, high overall heat transfer coefficient and better effectiveness of heat exchanger as compared to helical and conical coil configurations.

INTRODUCTION

Heat transfer in curved and helical circular tubes draw ample interest due to relatively high heat transfer coefficients and large surface area per unit volume associated with them. Helical coil tubes are used frequently in heating, refrigeration, HVAC applications, steam generation, condenser designs in power plants, nuclear industry, process plants, heat recovery systems, food industry etc. [1, 2, 3]. The main characteristics of coiled pipes are the compactness, high heat transfer performance, ease of maintenance, improved thermal efficiency, high operating pressures and extreme temperature gradients [4]. As the Newtonian fluid flows through a curved pipe, a secondary flow exists in addition to the main flow caused by longitudinal pressure gradients, which further give rise to centrifugal forces, resulting in the higher rate of momentum and heat transfer rates. Curved shape of the tube in helical coil of heat exchanger facilitates passive heat transfer enhancement [5]. The critical Reynolds number (Re_{cr}) in case of coiled tubes can be determined as follows [6].

$$Re_{cr} = 2300[1+8.6(\frac{r}{R_c})^{0.45}] \tag{1}$$

Where r is the tube radius and R_c is the coil radius. For curvature ratios $\frac{r}{R_c}$ less than $\frac{1}{860}$, Re_{cr} becomes equal to that of a straight tube. Modern designs of domestic hot water (DHW) system involve cold water flowing through a heat exchanger coil immersed in the hot water of the thermal energy storage (TES) tank for effective extraction of stored heat content [7]. Immersed heat exchangers have been used recently to discharge indirect integral collector storage (ICS) systems [8]. Optimization of the hot water tank geometry [9] and enhancement in the immersed heat exchanger performance [10] are some of the prominent ways to improve the performance of water heating systems. Published literature demonstrates that the placement of baffles around the heat exchanger coil fixed inside the storage tank under natural convection heat transfer mode improves the rate of discharge [11] and increases the effectiveness of heat exchanger [12]. Feeding of working fluid to the heat exchanger from the bottom enhances heat transfer between the hot and cold working fluids [13]. Literature suggests that the immersed heat exchanger should be coiled upwards inside the TES tank and should be located in upper region of the tank to achieve enhanced rate of heat extraction and improved discharging efficiency [7, 10, 14]. Circular truncated cone structure of storage tank with helical coil heat exchanger provides high discharging efficiency [15]. Further, surface modification of the tube of heat exchanger leads to high effectiveness of heat exchanger [16]. Several Nusselt number (Nu) correlations have been developed by performing experiments on helical coil heat exchangers with different geometrical parameters such as curvature ratios ($C.R$), number of turns (N), pitch to diameter ratios (p/D), pitch (p), considering varying flow rates of the working fluid (water, oil or steam) [17, 18, 19, 20]. The correlations formulated through the experiments and mathematical modelling brought forth that heat transfer coefficient is dependent on the tube diameter [1]. For the helical coils, although the heat transfer rate increases with the number of turns, the heat transfer coefficient is inversely proportional to number of turns. This behaviour is attributed to the rise of the plumes from the lowest to highest turns successively, thereby reducing the effectiveness of upper turns. Consequently, the average heat transfer coefficient decreases with the increase in number of turns [19]. The total height of the coil is considered as the characteristic length, suited best to the correlations [21].

Published literature also demonstrates the comparisons made on the basis of curvature ratio, fluid flow rate, rise in temperature of target fluid, pitch circle diameter and heat transfer coefficient [22, 23], coil geometry [24], friction factor associated with coil [5], positioning of coil inside storage tank [25], inlet temperature to coil [26], cone angle of conical coils [27], fluid type [28], Reynolds number and pitch and tube diameter [4]. Coronel and Sandeep [22] performed experiments on both helical and straight heat exchanger under turbulent conditions to determine the convective heat transfer coefficient. The results indicated that the overall heat transfer coefficient (U) is higher for helical heat exchanger and it increases with curvature ratio and flow rate. Jayakumar et al. [23] performed CFD simulations to study the variation of local Nu along the length and circumference of pipe for vertically oriented helical coils by varying coil parameters. Authors found that the Nu on the outer side of the coil is higher than that at any other location at that cross-section. The effect of coil curvature on flow and average Nu were found to decrease with increasing pitch circle diameter (PCD). Prabhanjan et al. [24] experimentally compared the helical coiled heat exchanger with straight tube heat exchanger and the results showed that the average heat transfer coefficient of helical coil is greater than that of the straight tube heat exchanger for every flow rate studied. Conte et al. [25] numerically investigated the forced laminar fluid flow over helical and conical pipes with circular cross-section by considering exterior flow arrangements for three different Re. Authors concluded that the higher heat transfer is associated with the conical coil as compared to the helical coil for the same surface area. Shinde [26] experimentally showed that the conical coil heat exchanger performs better in terms of the Nu on the inner tube, heat exchanger effectiveness and heat transfer rate as compared to the straight tube counterpart. However, the inner tube Nu was found to increase with increasing coil side flow rate, whereas the heat exchanger effectiveness decreased. Purandare et al. [27] experimentally investigated the thermal performance of fifteen conical coil heat exchangers with different cone angles and tube sizes for varying flow rates of hot and cold fluids and concluded that Nu increases, and effectiveness of heat exchanger decreases with the increase in Re based on the inner diameter. Wongwises and Naphon [28] experimentally investigated the heat transfer characteristics of spiral coil heat exchanger under sensible cooling conditions, where the heat exchanger effectiveness was found to be inversely proportional to the mass flow rate of air and directly proportional to the mass flow rate of water. Ghorbani et al. [29] studied the mixed convection heat transfer in coil-in-shell heat exchanger for different Re, curvature ratios and coil pitch under laminar and turbulent flow conditions by performing experiments. Authors found that the heat transfer coefficient decreased rapidly as the coil surface area was increased and was not affected by the tube diameter. Mirgolbabaei et al. [4] numerically simulated the mixed convection heat transfer from vertical helical coiled tubes in cylindrical shell at various flow and geometrical conditions. Their results demonstrated that shell side heat transfer coefficient increases with tube diameter, and it initially increases with increasing pitch of coil and thereafter decreases.

Forthgoing discussion shows that the rise in outlet temperature of cold working fluid is affected by coil geometry, flow rate, configuration of heat exchanger, and placement of baffles and surface modifications of coil. Although several experimental and numerical studies have been conducted to analyze the performance of helical coil heat exchanger, the conical coil configurations have not been extensively explored thermodynamically and hydro-dynamically. For a coiled pipe with one fluid moving through it and a second fluid flowing over the outer surface, an increase in thermal performance may be obtained through a suitable geometrical configuration. Generation of secondary flows is the reason of high thermal performance of coiled pipes because it produces additional transport of the fluid over the cross-section of circular pipe which in turn affects the heat transfer. Very few studies have compared straight, spiral, conical and helical coil tube heat exchangers based on the performance parameters, i.e., heat transfer coefficient, amount of heat transfer and the associated Nu, efficiency and effectiveness of heat exchangers, experimentally, as well as numerically. In the present study, to find the optimal heat transfer intensification, one helical coil and two conical

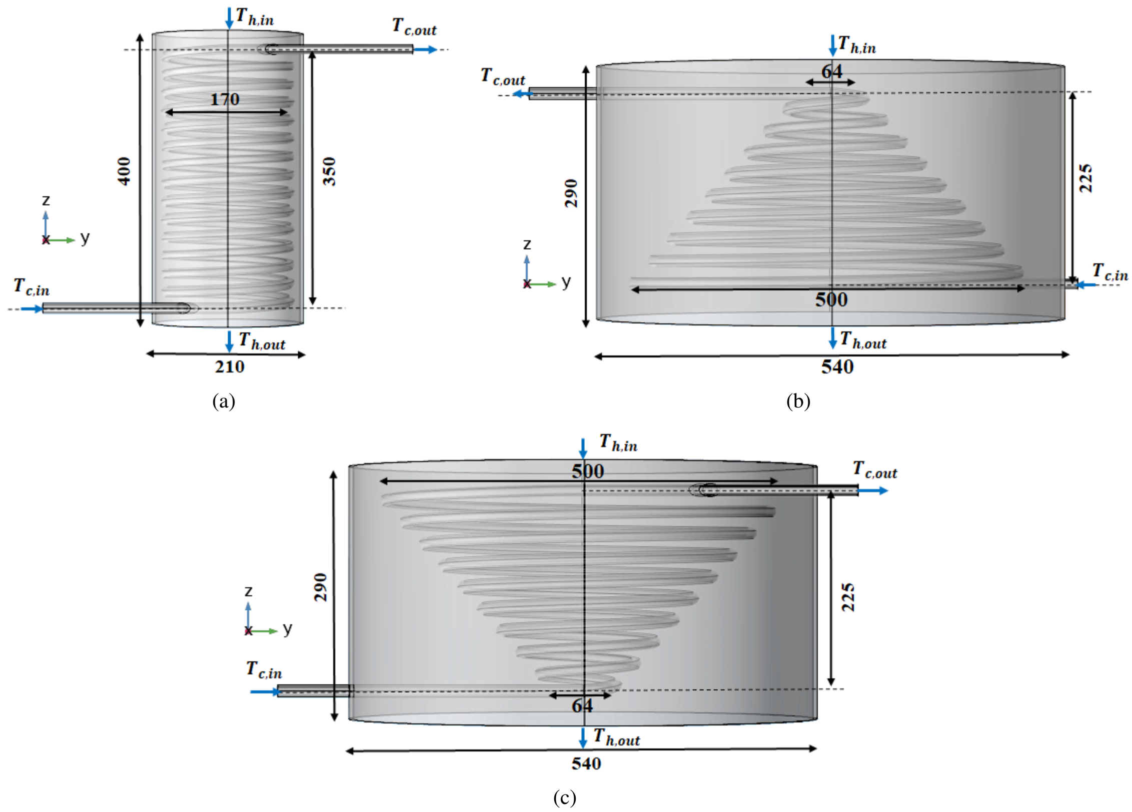


FIGURE 1. Three-dimensional numerical domain of three cylinders with a) helical, b) conical and c) inverted conical coil configuration (all dimensions are in mm and are not to scale).

coil (conical and inverted conical coil) configurations are considered, and their thermal performances are compared based on the standard performance parameters. Inverted conical coil has larger length exposed to hot incoming fluid from the inlet of tank which leads to more heat extraction time for the cold working fluid at a fixed flow rate, leading to good thermodynamic quality of the extracted energy.

NUMERICAL DOMAIN

Geometrical domain

A numerical model for the cylindrical tank, as well as the coil configuration is developed to analyse the performance of different coil configurations under four coil side flow rates of working fluid while the flow rate at cylinder side is kept constant. Two cylindrical tanks, one for the helical coil and other for the two conical coil configurations are used in the model. Height and inner diameter of cylindrical tank used to accommodate the helical coil configuration are 400 and 210 mm respectively. Cylindrical tank of height and inner diameter 290 and 540 mm respectively are used for both conical and inverted conical coil configurations. The hot working fluid is fed from the top of the cylinder and the cold working fluid is circulated through the coil. To compare the thermal performance of three heat exchanger configurations, coil length (L), pitch (p), inner and outer diameters of tube (d_i and d_o) are kept same for helical, conical and inverted conical coil configurations. Number of turns (N), coil height (H_{coil}), tank diameter (D_{tank}) and tank height (H_{tank}) are taken different to keep the coil volume same for every configuration. Pictorial representation of three-dimensional domains for the aforesaid tank arrangements and coil configurations is shown in the figure 1. For the same length of the coil, the heat transfer surface area exposed to the flow in cylindrical tank is more in case of conical coil heat exchangers as compared to helical coil heat exchanger, leading to high extraction of thermal energy

TABLE I. Common geometrical properties of helical and conical coils

Parameters	Values
Length of coil (L, mm)	7540
Axial pitch (p, mm)	25
Inner diameter of pipe (d_i, mm)	10
Outer diameter of pipe (d_o, mm)	14
Surface area of coil (A, m^2)	0.329

TABLE II. Different geometrical properties of helical and conical coils

	Helical coil	Conical coil	Inverted conical coil
Number of turns (N)	14	9	9
Height of coil (H_{coil}, mm)	350	225	225
Radial pitch (P_r, mm)	-	25	25
Top diameter of coil (d_T, mm)	170	64	500
Bottom diameter of coil (d_B, mm)	170	500	64
Volume of tank (V, m^3)	0.0138	0.0664	0.0664

[25]. For the three types of configurations, the same length of circular pipe, axial pitch pipe inner and outer diameters are considered. The number of turns (N) is 14 for the helical coil, and 9 for the conical coils, respectively. The geometrical properties for helical and two conical coils are specified in tables I and II, respectively.In this study, the water is considered as the working fluid for both the coil (as the cold working fluid) and the tank (as the hot working fluid). The thermo-physical properties of the water are listed in table III. The operating conditions are specified in table IV.

Numerical Formulation

The characteristics of heat transfer and the behaviour of fluid flow in the three-dimensional domain of the cylinder and coil system are evaluated by solving a set of standard mass, momentum, and energy equations. In the z-momentum equation, Boussinesq approximation, which considers the variation of density of working fluid with temperature only, is applied to include the buoyancy effect to the system. The assumptions made to develop the numerical model are listed below, (i) Newtonian and incompressible fluid, (ii) constant hot and cold working fluid inlet temperature, (iii) isotropic and constant thermo-physical properties of the working fluid for the range of temperature considered in the study (iv) no internal heat source , and (v) radiation heat transfer is neglected. The critical Reynolds number based on equation (1) are 7827 and 6701 for the helical and the conical coil configurations, respectively. The maximum Reynolds number evaluated based on the inlet velocity and the inner diameter of coil is 4326. Laminar flow regime is considered inside the coil for the configurations considered in the study as the maximum Reynolds number is less than the critical Reynolds number. The governing equations are expressed as:
a) Conservation of mass equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{2}$$

b) Conservation of momentum equations:

$$\rho(\frac{\partial u}{\partial t}+\frac{u\partial u}{\partial x}+\frac{v\partial u}{\partial y}+\frac{w\partial u}{\partial z})=-\frac{\partial P}{\partial x}+\mu(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}) \tag{3}$$

TABLE III. Thermo-physical properties of water

Specific heat $(C_p, J/kg.K)$	**Density** $(\rho, kg/m^3)$	**Thermal conductivity** $(K, W/m.K)$	**Dynamic viscosity** $(\mu, Pa.s)$
4200	998	0.59	0.00098

TABLE IV. Fluid flow and temperature conditions for helical and conical coils

Parameters	Values
Inlet temperature to tank $(T_{hi}, ^{o}C)$	70
Inlet temperature to coil $(T_{ci}, ^{o}C)$	35
Inlet flow rate to tank $(Q_{in,tank}, L/min)$	1
Inlet flow rate to coil $(Q_{in,coil}, L/min)$	0.5-2

$$\rho(\frac{\partial v}{\partial t}+\frac{u\partial v}{\partial x}+\frac{v\partial v}{\partial y}+\frac{w\partial v}{\partial z})=-\frac{\partial P}{\partial y}+\mu(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}) \quad (4)$$

$$\rho(\frac{\partial w}{\partial t}+\frac{u\partial w}{\partial x}+\frac{v\partial w}{\partial y}+\frac{w\partial w}{\partial z})=-\frac{\partial P}{\partial z}+\mu(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2})+\rho g_z \quad (5)$$

c) Conservation of energy equation:

$$\rho C_p(\frac{\partial T}{\partial t}+\frac{u\partial T}{\partial x}+\frac{v\partial T}{\partial y}+\frac{w\partial T}{\partial z})=K(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2})+\mu\phi \quad (6)$$

where ρ, μ, g, C_p and K are the density, dynamic viscosity, acceleration due to gravity, specific heat capacity and thermal conductivity of the working fluid.
The boundary conditions of the cylinder and coil system considered are:
(i) No slip condition at the wall of tank and coil, $u|_s = 0$
(ii) At inlet of tank, $z = H_{tank}$, $T = T_{hi}$, $u = u_{hi}$
(iii) At inlet of coil, $T = T_{ci}$, $u = u_{ci}$
(iv) At outlet of tank and coil, $P = 0$
(v) Tank surface is adiabatic,

$$(\frac{\partial T}{\partial n})_s = 0 \quad (7)$$

(vi) Coupled wall condition at heat exchanger wall,

$$K_w(\frac{\partial T}{\partial n}) = h_h(T_w - T_h) \quad (8)$$

The initial conditions considered are:
(i) Initial temperature of system, at t = 0, $t = T_{ini}$
(ii) Initial velocity of fluid in tank and coil, at t = 0, $u = u_{ini} = 0$
where, s stands for the tank surface and n is the coordinate measured normal to the surface. h_i, K_w, h_h, T_w and T_h represents the hot inlet, thermal conductivity of heat exchanger wall, heat transfer coefficient of the hot fluid near the wall of heat exchanger, wall temperature of coil and the temperature of hot water respectively. A commercial software, *COMSOL Multiphysics* $5.4a$ which is based on finite element method is used to solve the the governing equations $(2,3,4,5,6)$ together with the initial and boundary conditions. For the velocity and pressure terms, the first order element discretization is done. Linear discretization is used by the solver for temperature variable. Nested dissection multithread algorithm is utilised in the *PARDISO* solver for simulations. The criteria of convergence for the energy and momentum equations are set as 10^{-6} and 10^{-3}, respectively.

Performance parameters

a) Heat transfer
The heat transfer rate q $(\frac{J}{s})$ is expressed as:

$$q = \frac{(q_h + q_c)}{2} \quad (9)$$

where, q_h is the heat transferred by hot working fluid; $q_h = m_h C_{(p,h)}(T_{in} - T_{out})_h$ and q_c is the heat transferred by cold working fluid, $q_c = m_c C_{(p,c)}(T_{out} - T_{in})_c$. The parameters h, c, in, out and m represents the hot fluid, cold fluid, inlet, outlet condition and mass flow rate respectively.

b) Overall heat transfer coefficient
The overall heat transfer coefficient U_o, $(\frac{W}{m^2K})$ is expressed as:

$$U_o = \frac{q}{\Delta T_m A_o} \tag{10}$$

where, $A_o = \pi d_o L$, is the outer surface area of coil. ΔT_m is the log mean temperature difference based on the inlet temperature difference ΔT_i and outlet temperature difference ΔT_o, as expressed in equation 11.

$$\Delta T_m = \frac{\Delta T_i - \Delta T_o}{ln(\frac{\Delta T_i}{\Delta T_o})} \tag{11}$$

c) Heat exchanger effectiveness
The effectiveness of heat exchanger, ε is expressed in equation 12.

$$\varepsilon = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ho}} \tag{12}$$

where, T_{co} and T_{ci} are the steady state outlet and inlet temperature of cold working fluid respectively, and T_{hi} is the steady state inlet temperature of hot working fluid.

MODEL VALIDATION

Grid size independence study

In order to ensure the computational economy of the simulations performed, the grid size independence study of the numerical model is executed. Three grid sizes namely, coarse, normal, and fine are selected for the grid independence study. The temporal variations of the coil outlet temperature for the three chosen grid sizes are shown in figure 2(a). It is noticed that when the grid size is changed from normal (grid elements 133577) to fine (grid elements 172023), the maximum difference in the temperature of the cold working fluid at the outlet of the conical coil configuration is only 0.45%. Therefore, the normal grid size with 133577 elements is selected for further simulations.

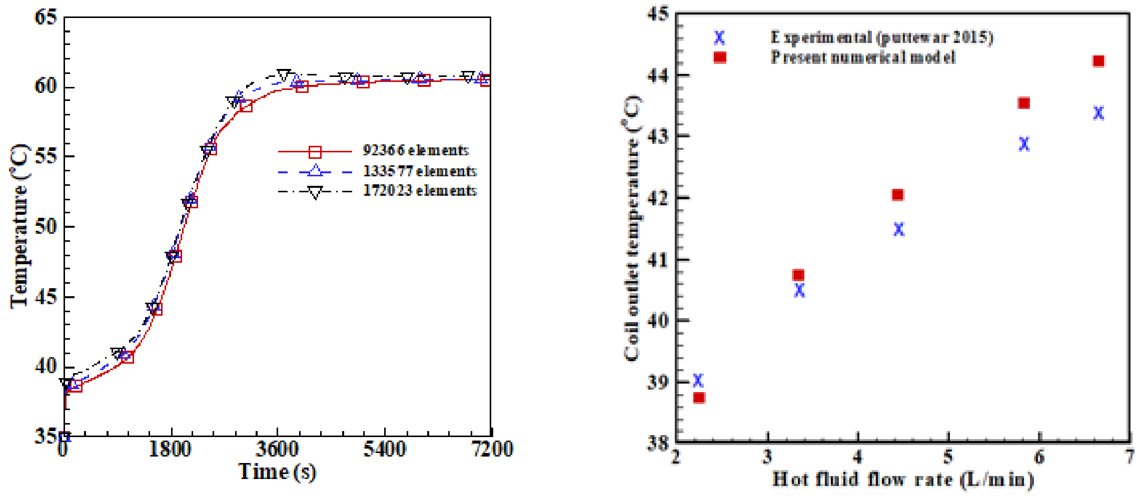
Numerical model validation

In order to ensure accuracy of the calculation and results, the working process of shell and helical coil heat exchanger is imbibed from Puttewar [30] and the results are presented in figure 2(b). The fluid flow and temperature conditions along with initial and boundary conditions are kept similar to that of experimental conditions specified by Puttewar [30]. Figure 2(b) shows the variation of coil outlet temperature with the coil side flow rate at a fixed shell side flow rate. The numerical results are found to be in good agreement with the experimental results. The numerical model can capture the coil side working fluid's temperature with the maximum error of 1.96%.

RESULTS AND DISCUSSION

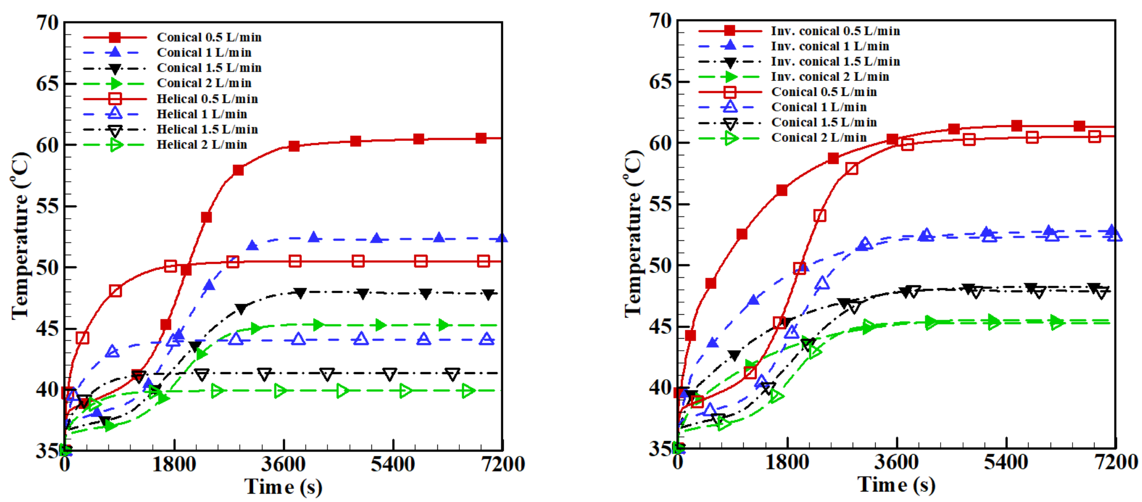
Comparison of helical and conical coil

a) Coil outlet temperature variation with time
Figure 3(a) shows the temporal variation of the outlet temperature of cold working fluid for helical and conical coil



(a) (b)

FIGURE 2. a) Temporal variation of coil outlet temperature with different grid elements for conical coil configuration at 0.5 L/min flow rate and b) Variation of coil outlet temperature with flow rate of hot water, at a constant flow rate of cold water (1.11 L/min).



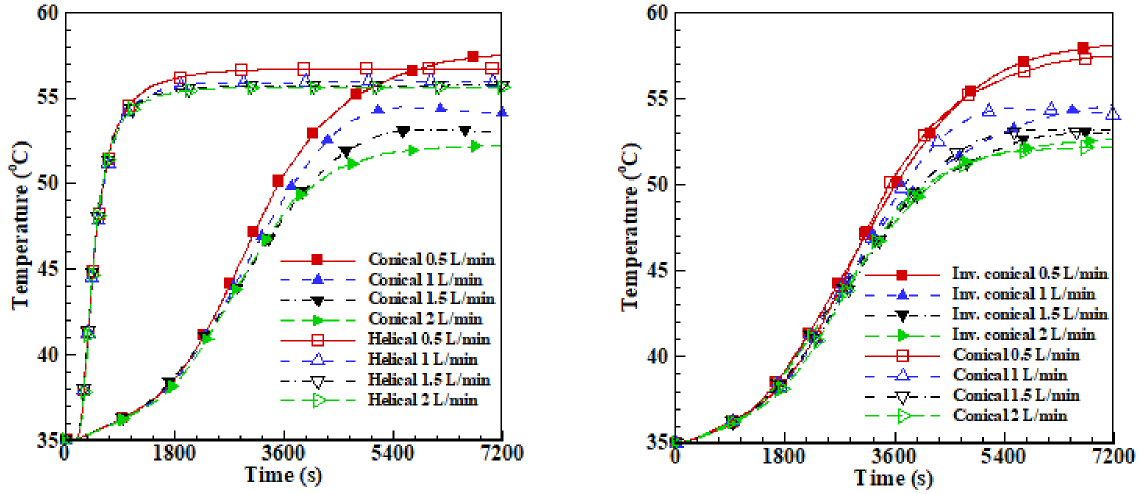
(a) (b)

FIGURE 3. Temporal variation of coil outlet temperature for a) helical and conical coils and b) conical and inverted conical coils, at different flow rates at the coil side

configurations at four different flow rates. The outlet temperature of the cold working fluid increases with time for both coil configurations, upon gaining heat from the surrounding hot working fluid in the tank. The outlet temperature of cold working fluid becomes steady once the thermal equilibrium is attained. As the cold working fluid flow rate is increased from 0.5 to 2 L/min, the steady temperature attained by the cold working fluid decreases for both the coil configurations. At a particular flow rate, the steady state outlet temperature of cold working fluid is higher for conical coil, as it gains more heat energy owing to the larger surface area exposed to the surrounding hot fluid in the tank. The higher temperature of working fluid at the outlet of coil depicts higher thermodynamic quality of energy.

b) Tank outlet temperature variation with time

Figure 4(a) shows the temporal variation of the temperature of hot working fluid at the tank outlet for four different flow rates of cold working fluid passing through the helical and conical coil configurations. The outlet temperature of hot working fluid increases with time for both coil configurations as inlet of tank is being fed continuously with



(a) (b)

FIGURE 4. Temporal variation of tank outlet temperature for a) helical and conical coils and b) conical and inverted conical coils, at different flow rates at the coil side

hot working fluid at 70^oC. As the cold working fluid flow rate is increased from 0.5 to 2 L/min, the steady state temperature of hot working fluid at the outlet of tank decreases for both helical and conical coil configurations. At a particular flow rate, the average outlet temperature of hot working fluid is lower for the conical coil, as compared to helical coil configuration. This is attributed to the higher amount of thermal energy captured by the cold working fluid through a larger heat transfer surface area, while flowing through the conical coil configuration.

c) Heat transfer variation with time

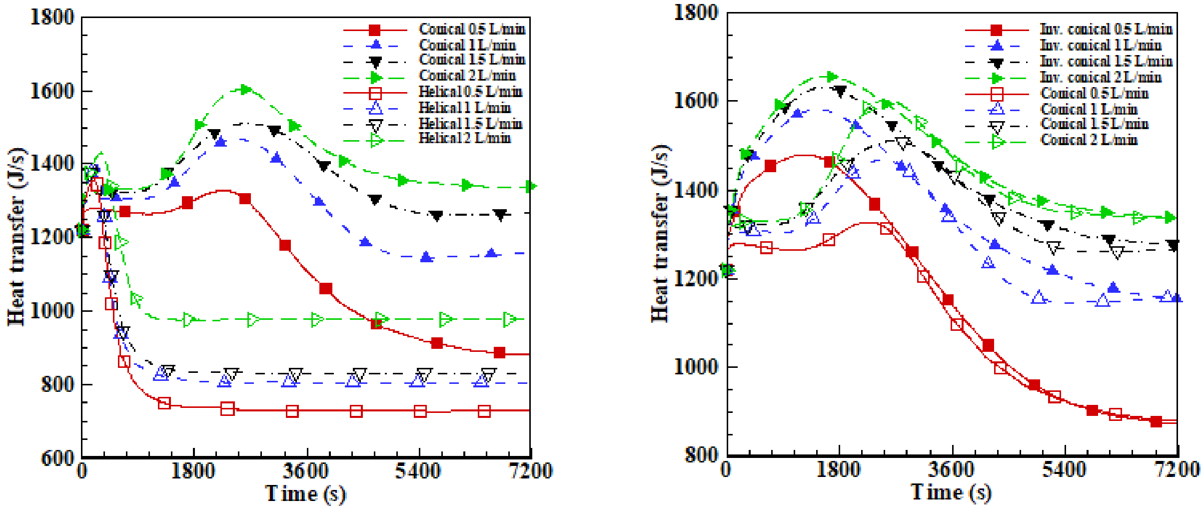
Figure 5(a) shows the temporal variation of heat transfer from hot to cold working fluid for four different flow rates of cold working fluid passing through helical and conical coil configurations. During the early stages of the heat transfer process, the temperature of the cold working fluid at outlet increases slowly from the initial value of 35^oC, indicating a smaller extent of heat transfer. As the time elapses, the sudden rise in temperature is observed indicating higher extent of heat transfer. As the working fluid temperature approaches steady state, the extent of heat transfer starts to diminish. Subsequently, a peak can be clearly observed in the temporal evolution of the heat transferred into the cold working fluid. As the cold working fluid flow rate increases from 0.5 to 2 L/min, the extent of heat transfer also increases for both coil configurations. During the process, for a particular flow rate, the maximum heat transferred from the hot to cold working fluid is higher for the conical coil configuration as compared to helical coil configuration because of the improved heat transfer coefficient, which can be owed to the continuously changing curvature ratio of the conical coil.

Comparison of conical and inverted conical coil

a) Coil outlet temperature variation with time

Figure 3(b) shows the temporal variation of outlet temperature of cold working fluid at four different flow rates of cold working fluid through the conical and inverted conical coil configurations. The outlet temperature of cold working fluid increases with time for both the configurations as it gains heat from the surrounding hot working fluid in the tank. Steady state is achieved, when the hot and cold working fluids reach thermal equilibrium. The steady state outlet temperature of cold working fluid decreases for both the configurations with increasing cold fluid flow rate. The inverted conical coil has higher surface area exposed to the hot working fluid at the inlet as compared to the conical coil configuration. Consequently, for a fixed flow rate of cold working fluid, the average outlet temperature of cold working fluid is higher for the inverted conical coil configuration as compared to conical coil counterpart.

b) Tank outlet temperature variation with time



(a) (b)

FIGURE 5. Temporal variation of heat transfer from hot to cold working fluid at different flow rates of cold working fluid for a) helical and conical coil and b) conical and inverted conical coil configurations

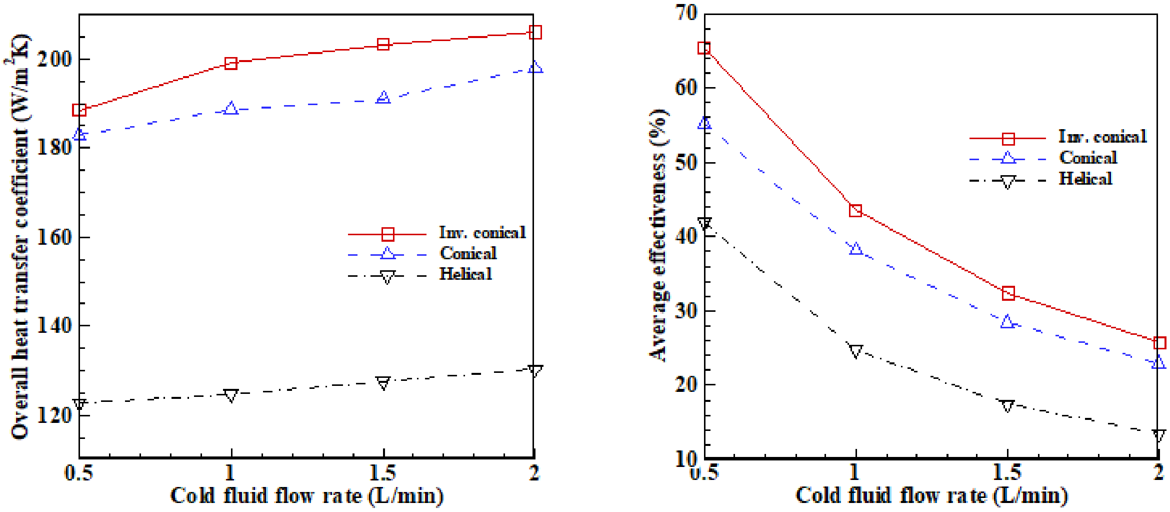
Figure 4(b) shows the temporal variation of hot working fluid temperature at the storage tank outlet for four different cold working fluid flow rates through the conical and inverted conical configurations. The temperature of hot working fluid at the tank outlet increases with time for both coil configurations. The steady state outlet temperature of hot working fluid decreases with increasing flow rate of cold working fluid for both coil configurations. For the whole duration of process, a major portion of the length of inverted conical coil is exposed to hot incoming fluid in the upper portion of tank which ensures effective transfer of heat content into the cold working fluid passing through the coil. On the contrary, in case of conical coil, the length exposed to hot incoming fluid in the upper portion of tank is considerably smaller. Therefore, at a particular flow rate of cold working fluid, the heat content transferred to the fluid in inverted conical coil is higher as compared to conical counterpart, leading to a higher average outlet temperature in inverted conical coil configuration.

c) Heat transfer variation with time
Figure 5(b) shows the temporal variation of heat transfer from hot to cold working fluid for different flow rates of cold working fluid through the conical and inverted conical coil configurations. Heat transferred from the hot side to the cold working fluid increases with the increase in coil side flow rate for both configurations. Evidently, at a particular flow rate of cold working fluid, the heat content captured by the inverted conical coil configuration is higher as compared to the conical configuration.

Comparison of helical, conical and inverted conical coil configurations

a) Comparison of overall heat transfer coefficient
Figure 6(a) shows the variation of overall heat transfer coefficient with the cold working fluid volume flow rate for helical, conical, and inverted conical coil configurations. As the flow rate is increased from 0.5 to 2 L/min, the overall heat transfer coefficient increases for each coil configurations demonstrating an enhancement in the extent of heat transfer as evident from figures 5(b) and 6(a). For a particular flow rate through the coil, the overall heat transfer coefficient is always higher for the inverted conical coil configuration as compared to conical and helical coil configuration indicating better heat exchange characteristics of inverted conical coil over other two configurations.

b) Comparison of average effectiveness of heat exchanger
Figure 6(b) shows the variation of heat exchanger average effectiveness for different cold fluid flow rates for the helical, conical, and inverted conical coil configurations. Effectiveness of heat exchanger is dependent on the coil side outlet temperature. Increasing the flow rate leads to decrease in the outlet temperature at the coil side, leading to



(a) (b)

FIGURE 6. Variation of a) overall heat transfer coefficient and b) average effectiveness of heat exchanger with cold fluid flow rate for helical, conical and inverted conical coils

decrease in average effectiveness of the heat exchanger. For a particular flow rate, heat exchanger average effectiveness is higher for inverted conical coil configuration as compared to conical and helical coil configurations.

CONCLUSION

The present study conducts a numerical investigation to compare the heat transfer characteristics among different geometrical configurations (in helical, conical and inverted conical coil) of heat exchanger for four different flow rates of the working fluid at the coil side. A numerical model is developed and is validated with the reported experimental results. In order to compare the thermal performance of three configurations of heat exchanger, coil length, pitch, as well as the inner and outer diameters of tube are considered to be the same. Number of turns, coil height, tank diameter, and tank height are taken different to keep the coil volume same for every configuration. For a particular flow rate of cold working fluid, the extent of heat transfer, overall heat transfer coefficient and effectiveness of heat exchanger are found to be the highest for inverted conical coil configuration and the lowest for helical coil configuration. For the same flow rates of hot and cold working fluids, inverted conical coil presents the largest heat transfer surface area to the hot fluid entering the storage tank, leading to a higher temperature of the working fluid at the coil outlet. Heat extraction time available for the cold working fluid is lesser in helical and conical coil configurations as compared to the the inverted conical coil configuration, and subsequently, the thermodynamic quality of the extracted energy is better in case of inverted conical heat exchanger configuration.

REFERENCES

1. M. E. Ali, "Experimental investigation of natural convection from vertical helical coiled tubes," International Journal of Heat and Mass Transfer **37**, 665 – 671 (1994).
2. M. Abdalla, "A four-region, moving-boundary model of a once-through, helical-coil steam generator," Annals of Nuclear Energy **21**, 541 – 562 (1994).
3. B. Bai, L. Guo, Z. Feng, and X. Chen, "Turbulent heat transfer in a horizontal helically coiled tube," Heat Transfer—Asian Research: Co-sponsored by the Society of Chemical Engineers of Japan and the Heat Transfer Division of ASME **28**, 395–403 (1999).
4. H. Mirgolbabaei, H. Taherian, G. Domairry, and N. Ghorbani, "Numerical estimation of mixed convection heat transfer in vertical helically coiled tube heat exchangers," International Journal for Numerical Methods in Fluids **66**, 805–819 (2011), https://onlinelibrary.wiley.com/doi/pdf/10.1002/fld.2284.
5. S. Rajasekharan, V. Kubair, and N. Kuloor, "Heat transfer to non-newtonian fluids in coiled pipes in laminar flow," International Journal of Heat and Mass Transfer **13**, 1583 – 1594 (1970).

6. V. Kubair and N. Kuloor, “Flow of newtonian fluids in archimedian spiral tube coils-correlation of laminar transition+ turbulent flows,” Indian Journal of Technology **4**, 3 (1966).
7. R. Spur, D. Fiala, D. Nevrala, and D. Probert, “Performances of modern domestic hot-water stores,” Applied Energy **83**, 893 – 910 (2006).
8. W. Liu, J. H. Davidson, F. Kulacki, and S. C. Mantell, “Natural convection from a horizontal tube heat exchanger immersed in a tilted enclosure,” J. Sol. Energy Eng. **125**, 67–75 (2003).
9. H. Khurana, S. Tiwari, R. Majumdar, and S. K. Saha, “Comparative evaluation of circular truncated-cone and paraboloid shapes for thermal energy storage tank based on thermal stratification performance,” Journal of Energy Storage **34**, 102191 (2021).
10. S. Li, Y. Zhang, K. Zhang, X. Li, Y. Li, and X. Zhang, “Study on performance of storage tanks in solar water heater system in charge and discharge progress,” Energy Procedia **48**, 384 – 393 (2014), proceedings of the 2nd International Conference on Solar Heating and Cooling for Buildings and Industry (SHC 2013).
11. Y. Su and J. H. Davidson, “Discharge of thermal storage tanks via immersed baffled heat exchangers: numerical model of flow and temperature fields,” Journal of solar energy engineering **130** (2008).
12. J. F. Haltiwanger and J. H. Davidson, “Discharge of a thermal storage tank using an immersed heat exchanger with an annular baffle,” Solar Energy **83**, 193–201 (2009).
13. D. Austen and H. Soliman, “Laminar flow and heat transfer in helically coiled tubes with substantial pitch,” Experimental Thermal and Fluid Science **1**, 183–194 (1988).
14. R. Mote, S. Probert, and D. Nevrala, “Free-convective flows within a hot-water store, induced by a submerged, relatively cold heat exchanger,” Applied energy **39**, 207–234 (1991).
15. A. Li, F. Cao, W. Zhang, B. Shi, and H. Li, “Effects of different thermal storage tank structures on temperature stratification and thermal efficiency during charging,” Solar Energy **173**, 882–892 (2018).
16. R. Andrzejczyk and T. Muszyński, “Performance analyses of helical coil heat exchangers. the effect of external coil surface modification on heat exchanger effectiveness,” archives of thermodynamics **37** (2016).
17. A. Alimoradi, “Study of thermal effectiveness and its relation with ntu in shell and helically coiled tube heat exchangers,” Case Studies in Thermal Engineering **9**, 100–107 (2017).
18. D. G. Prabhanjan, T. J. Rennie, and G. V. Raghavan, “Natural convection heat transfer from helical coiled tubes,” International Journal of Thermal Sciences **43**, 359–365 (2004).
19. M. E. Ali, “Natural convection heat transfer from vertical helical coils in oil,” Heat Transfer Engineering **27**, 79–85 (2006).
20. G. Rogers and Y. Mayhew, “Heat transfer and pressure loss in helically coiled tubes with turbulent flow,” International Journal of Heat and Mass Transfer **7**, 1207–1216 (1964).
21. Y. Su and J. H. Davidson, “Transient natural convection heat transfer correlations for tube bundles immersed in a thermal storage,” (2007).
22. P. Coronel and K. Sandeep, “Heat transfer coefficient in helical heat exchangers under turbulent flow conditions,” International journal of food engineering **4** (2008).
23. J. Jayakumar, S. Mahajani, J. Mandal, K. N. Iyer, and P. Vijayan, “Cfd analysis of single-phase flows inside helically coiled tubes,” Computers & chemical engineering **34**, 430–446 (2010).
24. D. Prabhanjan, G. Raghavan, and T. Rennie, “Comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger,” International communications in heat and mass transfer **29**, 185–191 (2002).
25. I. Conté, X. Peng, and B. Wang, “Numerical investigation of forced fluid flow and heat transfer from conically coiled pipes,” Numerical Heat Transfer, Part A: Applications **53**, 945–965 (2008).
26. D. Shinde Digvijay and H. Dange, “Heat transfer analysis of a cone shaped helical coil heat exchanger,” International Journal of Innovations in Engineering and Technology (IJIET) **3**, 56–62 (2013).
27. P. S. Purandare, M. M. Lele, and R. K. Gupta, “Investigation on thermal analysis of conical coil heat exchanger,” International Journal of Heat and Mass Transfer **90**, 1188–1196 (2015).
28. S. Wongwises and P. Naphon, “Heat transfer characteristics and performance of a spirally coiled heat exchanger under sensible cooling conditions,” JSME International Journal Series B Fluids and Thermal Engineering **48**, 810–819 (2005).
29. N. Ghorbani, H. Taherian, M. Gorji, and H. Mirgolbabaei, “Experimental study of mixed convection heat transfer in vertical helically coiled tube heat exchangers,” Experimental Thermal and Fluid Science **34**, 900–905 (2010).
30. A. S. Puttewar and A. Andhare, “Design and thermal evaluation of shell and helical coil heat exchanger,” international journal of research in engineering and technology **4**, 2319–2326 (2015).