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Performance characteristics of shell and helically finned-tube heat exchanger

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Abstract: Experimental and numerical studies were conducted for different fin shapes (solid fins, serrated fins-8 wings, serrated fins-11 wings, and serrated fins-14 wings) and spacing of shell and helically coiled heat exchanger. Investigations were made on the heat exchanger's fin spacing. The FLUENT 19 numerical code was used to illustrate the thermofluid behaviors through the shell and finned tubes heat exchanger. Using an inlet hot water temperature of 80 ^oC for the coil side and inlet cold water temperature of 10 ^oC for the shell side, mass flow rates along the sides of the coil and shell were 0.05-0.2 kg/s. The impact of fin shape on temperature distribution was clear. The use of solid fin results raised Nussult number than serrated fins-8wings, serrated fins-11wings, and serrated fins-14wings with an increase of around 6.3%, 9.5%, and 12.6%, respectively, and in the case without fins about 17.8% for the same value of Dean number=6000.

Keywords: Shell and a coiled helical tube, Fin, Heat exchanger, Fin space, Fin shapes.

NOMENCLATURE

Symbol	Description
A	Area (m^2)
c	Specific heat ($kJ.kg^{-1}.K^{-1}$)
D	Coil diameter (mm)
d	Tube diameter (mm)
h	Heat transfer coefficient ($W.m^{-2}.K^{-1}$)
H	Height (mm)
k	Thermal conductivity ($W.m^{-1}.K^{-1}$)
$\dot{m}$	Mass flow rate ($kg.s^{-1}$)
Nu	Nusselt number (-)
$\dot{Q}$	Heat transfer rate (kW)
Re	Reynolds number (-)
T	Temperature (^oC)
U	Overall heat transfer coefficient ($W.m^{-2}.K^{-1}$)
v	Average velocity ($m.s^{-1}$)
S	fin spacing
Greek Symbols	
ε	Effectiveness (-)
μ	Fluid viscosity ($kg.m^{-1}.s^{-1}$)
β	fin spacing ratio

Abbreviations	Description
CFD	Computational fluid dynamics
LMTD	Log- mean temperature difference
NTU	Number of transfer units
Subscripts	
av	average
c,i	cold-inlet
h,i	hot-inlet
ht	helical tube
hy	hydraulic
i	inlet/inner
o	outlet/outer
sh	shell
t	Tube
to	Tube outer
ti	Tube inner
W	Water

1. Introduction

Providing a suitable living environment, including housing, job opportunities, and services (health, education, etc.), is one of the most important requirements for building sustainable human settlements. It is a way of life for social mobility in an integrated urban environment. Thus, the search for a better and safer life was the main motive for many groups of Egyptian people to migrate from rural areas to the cities. Growth was concentrated in the central cities, especially Cairo, as it is the administrative and economic capital, and industrial activities are concentrated there. This led to increased rates of rural migration, and consequently, slums were created in Greater Cairo over the agricultural areas around the Nile Basin. [1]

Most industries designed and thermally evaluated heat exchangers in order to save money, materials, and energy while achieving maximum heat transfer. The main problem in heat exchanger design was to get maximal heat transmission in the least amount of area [1]. Senapati et al. [2] identified the ideal fin spacing for maximal heat transmission in turbulent flow and the conjugate heat transfer properties were explored. A vertical cylinder of diameter (d) 25 mm with fins of uniform thickness (t) 1 mm having a constant inter-fin spacing (S) is taken for the analysis. The transport of heat during laminar flow from the heated isothermal tube surface to the fins increases, whereas, in turbulent flow, heat transfer decreases after reaching a maximum value. For the turbulent domain, the optimal fin spacing to tube diameter ratio (S/d) for maximum heat transfer lies in S/d = 0.28 and 0.31, corresponding to 7.0 mm to 7.7 mm spacing, respectively. Elias et al. [3] studied the effects of different particle forms (cylindrical, bricks, blades, and platelets) on the overall heat transfer coefficient, heat transfer rate, and entropy generation were investigated using segmental baffles and shell and tube heat exchangers with changing baffle angles. The predicted results compared to other baffle angles and the segmental stutter that the heat transfer rates were higher for the cylinder-like form with a 20° stutter angle. Puttewar and Andhwere [4] investigated the heat exchangers with helical coils and shells by studying their thermal analysis in a counter-flow configuration. Several parameters, consisting of the rate of flow of cold and hot water, the temperature, the effectiveness, and the overall heat transfer coefficient were taken into consideration when doing thermal tests [5]. Wang et al. [6] investigated how the shape of the fins and the shell's input mass flow rate affected energy loss. Heat transfer, fan work, number of transfer units (NTU), fin height and number, and shell-side flow velocity all increased with increasing exergy loss. By adding annular fins to the outside of the helical coil, the increased rate of heat transfer in heat exchangers with shells and helical coils has been quantitatively investigated [5]. A. Alimoradi et al. [7] studied the impact of height or number of fins on the shell side Reynolds indices (Re_{sh}=7,500, 15,000 and 30,000). The numerical model has been validated using two different techniques. For fin pattern, the ideal conditions were attained having the fastest rate of heat transmission and requiring the least amount of material. It was suggested that rather than higher height fins with fewer fins, consume the fin-related materials in a way that incorporates more fins with a higher number and lower height. Miansari et al. [8] tried to optimize a using a coiled tube for a heat exchanger and a helically grooved shell. The data demonstrate a thermal performance difference of up to 20% between grooved and un-grooved shell and helically coiled tube heat exchangers.

The above literature review reveals that no work is available in heat exchangers with different fin shapes and distances. In the present work, the following fin shapes; solid fins, serrated fins-with 8 wings, serrated fins-with 11 wings, and serrated fins-with 14 wings are examined. The fins are located with spacing of 11 cm, 8 cm, and 5 cm.

2. DETAILS OF THE EXPERIMENT

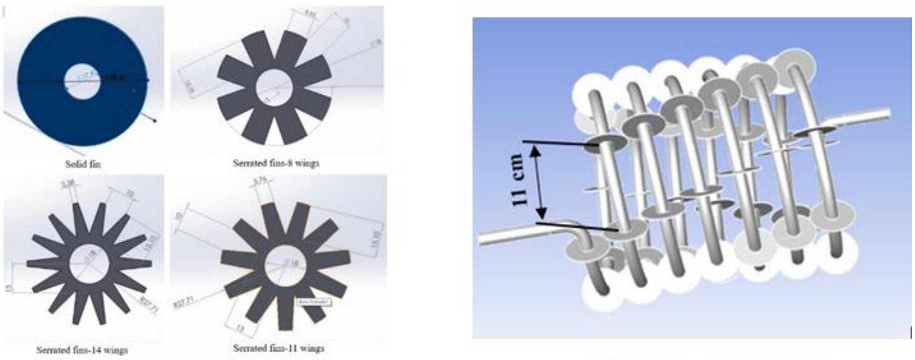
2.1 EXPERIMENTAL SETUP

A shell and helical coiled tubes heat exchanger is designed as an experimental apparatus in Fig. 1. The copper helical coiled tube of 12.88 mm with a coil diameter of 250 mm, 50 mm coil pitch, 6.4 turns, and 1.24 mm thickness. The shell's inner diameter was 350 mm, while 12.88 mm was the tube's inner diameter.

Figure 1 depicts a photo of a test rig with its instrumentations.

Four different copper fin shapes were placed on the external surface of circular coiled tubes namely solid fin, serrated fins-with 8 wings, serrated fins-with 11wings, and serrated fins-with 14 wings; their shapes and dimensions are shown in Fig. 2.a. Fins were fixed at a 90-degree angle with different spacing of 11 cm, 8 cm, and 5 cm. A closed loop of cooled water was used to feed the shell side. The tank for the cooled water cycle had a capacity of 0.285 m^3. A helically coiled tube evaporator attached to a 2.25 horsepower compressor's condensing unit. By altering the temperature controller, the water was managed. A tank with dimensions 0.70x0.7x0.70 m^3 was used for the hot water cycle. Four immersion heaters with a power of 12 kW were used. Water was circulated using centrifugal pumps 0.75 HP. Table 1. displays the study variable limitations.

FIGURE 1: Experimental apparatus photo with instrumentations.



(a) Fin shapes (b) Distance between fins

FIGURE 2: Different fin shapes and distances

TABLE 1. The study variable ranges**.**

Variables	Limitations
Chill water inlet temperature	10 ˚c
Temp. of the hot water supply	80 ˚c
Shell and coil side fluid flow rate inlet	0.05-0.2 kg/s
Heat exchanger inclination angle	90˚
Coil diameter ratio	0.056
Tube geometry shape	Circular
Fin spacing, cm (see Fig. 2.b)	11,8,5 cm
Fin shapes	Solid fin, serrated fins-8 wings, serrated fins-11 wings, and serrated fins-14 wings

The suggested coiled tube diameter to the coil diameter ratio was 0.056 mm. To reduce heat loss, the heat exchangers was insulated. Two sensors at each point of the intake and exit via the heat exchanger were fixed and related to a data acquisition unit entirely through computer. Additionally, another 24 thermocouples were distributed by 195 mm and linked by a temperature controller. Moreover, it employed a heat exchanger with a pressure drop. The inclination angle of heat exchangers was set constant at 90^o.

2.2 MEASURING TECHNIQUES

Atemperature of 80˚C and 10˚C were used to provide the hot and chilled water, respectively. Eight K-type thermocouples were used to measure temperatures at the test rig's entrance and exit. Using a set of 24 types, K thermocouples was employed to measure surface temperatures of helical coil. Before sensors were fixed, all thermocouple sensors were calibrated against a reference thermometer. The rates of flow of both cold and hot water were measured via two rotameters. Using a pressure transmitter, the pressure drop across the heat exchanger was determined. Eight sensors were connected to a data logger, and the remaining 24 temperature sensors were linked by a selection switch to a digital monitor.

2.3 UNCERTAINTY OF THE RESULTS

The error analysis approach was estimated by Kline and McClintock to determine the degree of uncertainty in the computed outcomes. In Holman [9], the uncertainty estimating approach was presented. The estimate method was based on an uncertainty value that was defined by several initial experimental measurements. Table 2. illustrates the instrumentations accuracy and data result uncertainty.

TABLE 2. Uncertainty of calculated parameters

Instruments	Uncertainty
Reynolds number	±2.9
Friction factor	±3.3
Nusselt number	±2.4
Effectiveness	±2.9
Amount of heat transfer units	±2.7
Heat transfer rate per unit pumping power	±3.6

3. DATA REDUCTION

The basic formulae below can be used to compute the rate of heat transfer from hot fluid moving through the helical coiled tube and cool fluid moving through the shell:

$$\dot{Q}_{sh} = \dot{m}_{sh} c_{sh} (T_{sh_o} - T_{sh_i}) \quad (1)$$

$$\dot{Q}_{ht} = \dot{m}_{ht} c_{ht} (T_{ht_i} - T_{ht_o}) \quad (2)$$

In real-world experiments, Under the same conditions, there is no difference greater than 1.5% due to measurement mistakes and the effectiveness of thermal insulation. As a result, the computations can use the arithmetic average of the heat gained and lost by the heat exchanger is known as the SHCTHE heat load, as given below [11]:

$$\dot{Q}_{av} = \left(\frac{\dot{Q}_{ht} + \dot{Q}_{sh}}{2}\right) \tag{3}$$

The typical helical side heat transfer coefficient, (h_{ht}) is computed as:

$$h_{ht} = \frac{\dot{Q}_{av}}{A_{ht_i(T_{ht_m} - T_{ht_s})}} \tag{4}$$

The following equations illustrate how the Reynolds number, Dean number, and Nusselt number of a helical coiled tube's fluid side are represented:

$$Re_{ht} = \frac{4.\dot{m}_{ht}}{\pi.d_{ht_{hy}}.\mu_{ht}} \tag{5}$$

$$De_{ht} = Re_{ht}\left(\frac{d_{ht_{hy}}}{D_c}\right)^{0.5} \tag{6}$$

$$Nu_{ht} = \frac{h_{ht}.d_{ht_{hy}}}{k_{ht}} \tag{7}$$

The following estimate is made for the overall heat transfer coefficient:

$$U_o = \frac{\dot{Q}_{av}}{\Delta T_{LMTD}.A_{ht_o}} \tag{8}$$

The effectiveness is obtained as follows:

$$\varepsilon = \frac{\dot{Q}_{av}}{\dot{Q}_{max}} = \frac{\dot{Q}_{av}}{(\dot{m}c)_{min}(T_{ht_i} - T_{sh_i})} \tag{9}$$

As defined by the unit of measure of heat transfer (NTU):

$$NTU = \frac{U.A}{c_{min.}} \tag{10}$$

4. CFD SIMULATION MODELING

The behaviors of helically coiled tubes and shell heat exchangers as a result of heat transmission and pressure drop were investigated experimentally and numerically. A non-linear partial differential equation was used to determine the single-phase fluid behaviors of heat exchangers with a shell and helical coil. The fluid flow properties were described the mass, a momentum, and energy equations as [10]:

Conservation of Mass equation:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{11}$$

Momentum equation:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{12}$$

Conservation of Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_j}\right] \tag{13}$$

The governing equations for heat exchangers with shell and helical coils were solved using exchangers. With the help of the drawing tool and the ANSYS mesh programme, heat exchanger with a shell and helical coil was created. The equation for momentum, turbulence, and energy was discretized. via the second-order upwind approach. Furthermore, the 10-6 continuity, velocity, and energy convergence criteria were applied.

The boundary criteria were as follows: the tube side with the helical coils was 80 ˚C warmer than the shell side, which was 10 ˚C warmer. On the shell's side and the helix tube's side, the mass flow rate was 0.05 to 0.2 kg/s. The pressure was set to zero. Adaptive grid refinement was also incorporated in the preliminary estimations. To modify the model solution, 18 layers were developed with nodes of 4248658 and cells of 14508866. To guarantee The reliability of the computational solution, the independence of various grid arrangements (A:E) was examined. Six distinct cellular mesh grids were constructed as seen Figure 3.

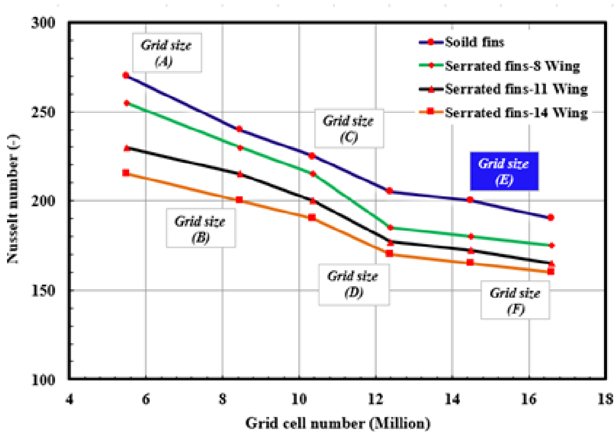


FIGURE 3: Nusselt number grid dependency for six different grid systems.

The differences in the Nusselt number in (grid size-D) $12.39\text{x}10^6$, (grid size-E) $14.5\text{x}10^6$, and (grid size-F) $16.6\text{x}10^6$ are less than 4.2%, but solving the grid systems with mesh F requires more than 28% more time due to the complexity of the model meshing approach. As a result, the mesh grid system (E) saves time while obtaining the required convergence with solution precision.

5. RESULTS AND DISCUSSION

The following subsections look at different fin geometries and fin spacing.

5.1 VALIDATION OF THE PRESENT RESULTS

To fulfil the research goal, the precision of the test apparatus and numerical modelling dependent on the method of choice must be ensured. The constructed numerical model is evaluated by comparing it to experimental data.

Current experimental data and numerical outcomes in Figure 4 shows a graphic for the shell and coiling helically finned-tube heat exchanger when the Nu number varies with the Dean number. The figure demonstrates that, the Nu number has maximum variances of ±3.7% comparing experimental data and numerical findings for sold fins. Strong confidence is given by this validation in the ability to generalize the application of the numerical model change in line with research limitations and test settings across simulations.

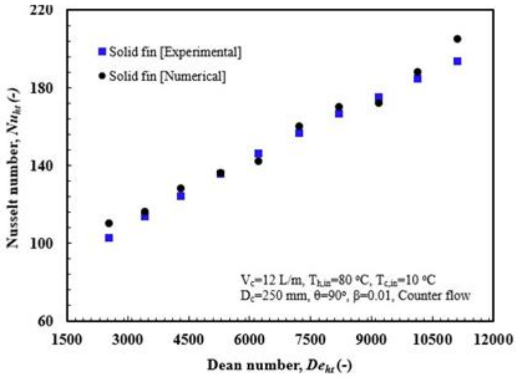


FIGURE 4: Validation between experimental and numerical results

5.2 RESULT OF ADDING FINS WITH VARYING SPACING

The fin spacing, also known as fin pitch, is a crucial design parameter with a major effect on a coiled heat exchanger's performance. Figure 5a illustrates the effect of different fin spacing S values (5 cm, 8 cm, and 11 cm) regarding heat transfer efficiency, while keeping the coil length constant at 500 cm. When S = 0, it represents the case of a tube without fins. Reducing the fin spacing results in a bigger area that can be used for heat transfer, leading to enhanced heat transfer rates due to more effective fluid to fin convectional heat transfer. This results in more efficient heat exchange between the fluids. However, as the fin spacing decreases, it also leads to higher frictional losses, resulting in increased pressure drop within the coiled heat exchanger. Upon analyzing Figure 5a, it becomes evident that there exists an optimum space S of 5 cm, The use of S=5 cm results in a higher Nu. number than S=8 cm , S=11 cm and S=0, with an increase of around 8.2%, 14.1%, and 18%, respectively, for the same vlue of Dean no.=6000.

The degree of effectiveness of a heat exchanger with coils is a crucial parameter that measures how efficiently heat is transferred between the two fluids in the system. It is described as the proportion between the actual and maximum heat transfer that can occur between the fluids. The relationship between the effectiveness, Dean number, and fin spacing in a coiled heat exchanger is interconnected. Smaller fin spacing generally improves effectiveness by expanding the heat transfer surface and improving convective heat transfer. Higher Dean numbers also improve effectiveness by promoting more complex flow patterns that enhance heat transfer. Figure 5b demonstrates that incorporating fins in the coil significantly improves the heat exchanger's effectiveness. Among the different fin spacing investigated. At using S =5 cm, effectiveness is higher than when using S =8 cm , S =11 cm , and S =0, with increases of approximately 6.5%, 13%, and 17.4%, respectively, for a similar value for Dean no.=6000.

The presence of fins on a coiled tube improves the tube's surface area and, as a result, its coefficient of heat transmission, as in Figure 6. And when changing the distances from 11cm to 8cm to 5cm, it will be noticed that in the case of the distance between the fins 5cm, it will be the best condition for temperature distribution and the highest temperature drop

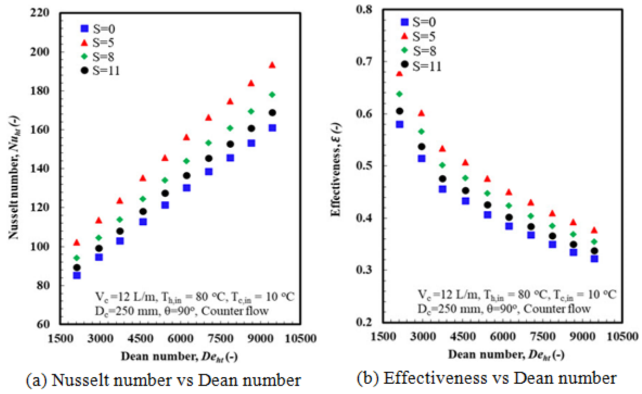


FIGURE 5: Performance of coiled heat transfer in relation to fins pitch.

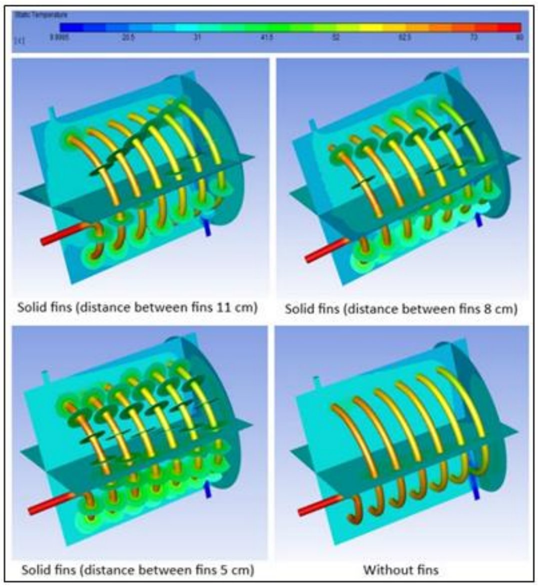


FIGURE 6: Effect of changing distance between solid fins on temperature contours.

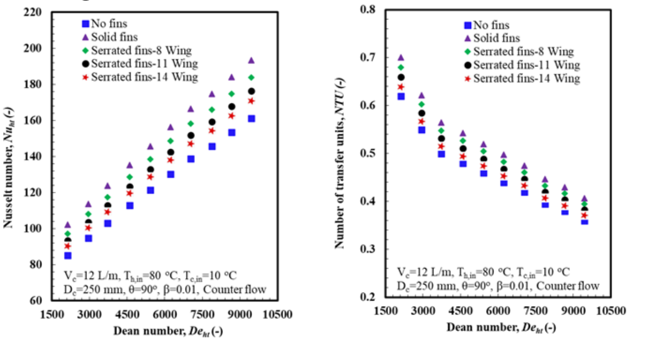
5.3 RESULT OF INCLUDING VARIOUS FIN SHAPES

Different fin designs can significantly influence the Nu. number, This represents a measure of how quickly heat is transmitted from a fluid to the surface. as Figure 7a shown soild fins exhibited superior performance compared to other design. The Nu. number depends on the temperature-transfer efficiency (h), which itself relies on the flow regime, fluid properties, and flow patterns. Conversely, certain fin designs may not provide sufficient surface area or promote favorable flow patterns, leading to lower heat transfer coefficients and reduced Nusselt numbers. Inadequate fin designs might result in inefficient heat transfer. For instance, optimized fin geometries can minimize the occurrence of Dean vortices, reducing pressure losses and improving overall heat exchanger efficiency. The use of solid fin results raised Nussult number than serrated fins-8wings, serrated fins-

11wings, and serrated fins-14wings with an increase of around 6.3%, 9.5%, and 12.6%, respectively, and in the case without fins about 17.8% for the same value of Dean no=6000.

Serrated fins with wings can provide significant improvement of heat transfer in heat exchangers with tubes and shells, especially as the flow transitions from laminar to turbulent regimes. The increased turbulence and fluid mixing generated by the combination of serrations contribute to higher NTU values and improved heat transfer performance. However, as with any fin design, the selection of serrated fins should consider factors such as pressure drop, manufacturing complexity, and the specific requirements of the heat exchanger application. At low Dean's numbers (laminar flow), due to the flow disruptions, serrated fins can still effectively transmit heat created by the serrations. This leads to a higher NTU value compared to conventional solid fins. As Dean's number rises and the flow becomes turbulent, the NTU value for serrated fins tends to increase further see Figure 7b. The additional turbulence generated by the wings results in more efficient heat transfer and higher NTU values compared to laminar flow conditions. The use of solid fin results raised Nussult number than serrated fins-8wings, serrated fins-11wings, and serrated fins-14wings, with an increase of around 4%, 7%, and 10%, respectively, and in the case without fins about 14% for the same value of Dean no.=6000.

It was observed in figure 8 that the solid fin performs greater compared to the other shapes. This improvement can be attributed to its larger surface area exposed to the flow, allowing for enhanced heat transfer.



(a) Nusselt number versus Dean Number (b) Number of transfer units versus Dean Number

FIGURE 7: Impact of different fin shapes on efficiency of heat transmission through coils.

6. Conclusions

Experimental and numerical investigation for the shell and helically coiled finned tube heat exchangers were carried out. The present study examines the effects of the operational and design aspects like fin geometry and fin spacing on the heat exchanger performance. The key conclusions are summarized as:

- Higher heat transfer coefficient is achieved by the solid fins compared with the other investigated fin geometries.
- A lower pressure drop was observed with a helically coiled tube without fins followed by serrated fins-8 wings, while the higher-pressure drop was observed with the solid fins.
- The impact of the fin shape on the temperature distribution was found, although the solid fin performed better than serrated fins-8 wings, serrated fins-11 wings, and serrated fins-14 wings with an increase of around 6.3%, 9.5%, and 12.6%, respectively, and 17.8% in case of without fins.

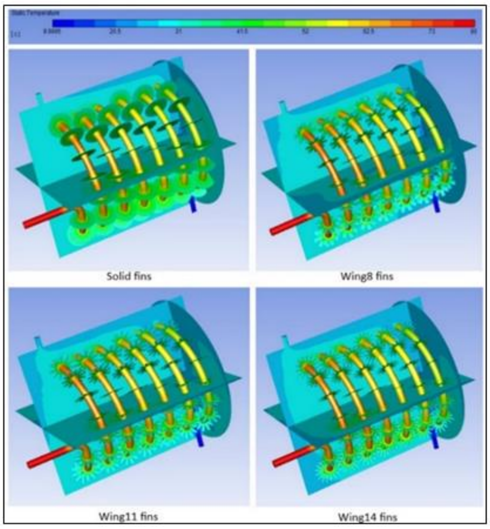


FIGURE 8: Different shapes of fins at 5 cm spacing.

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