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Performance Optimization of the Helical Heat Exchanger With Turbulator

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In this paper, optimization on a two-tube helical heat exchanger with a fin is represented. The spiral pipes heat exchanger which is made of the cooper is adopted for investigation. The effects of three types of fins with the proposed geometric shapes on the overall heat transfer coefficient and pressure loss are investigated. The fins are located on the inner surface of the outer pipe. The obtained numerical results are compared with the experimental results, and a good agreement is observed between the results. The studies show that the total heat transfer coefficient has increased by 170% compared to an exchanger with no fin. Therefore, the best fin has been selected based on the benefit-cost-ratio (BCR) factor. Finally, using the new represented optimization algorithm, the height of the represented triangular fin is optimized to represent the best values for overall heat transfer coefficient and pressure loss of the helical heat exchanger. In addition, the results indicate that reducing the density and height of the triangular fin increases heat transfer and reduces pressure loss.

Keywords: helical heat exchanger, fins effects, heat transfer, pressure loss, multi-disciplinary design optimization method

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INTRODUCTION

The process of transferring the heat (Ghalandari et al., 2019; Zhao et al., 2021a, Zhao et al., 2021b) between two fluids at different temperatures separated by a solid wall is common in many engineering applications (Che et al., 2021; Mahariq and Erciyas, 2017; Mahariq et al., 2020; Panahi and Zamzamian, 2017; Prabhanjan et al., 2002; Shafee et al., 2020; Zhao et al., 2021c). Heat exchangers are devices that allow heat to be transferred from one fluid to another without mixing the two fluids (Assad et al., 2021; Che et al., 2021; Chu et al., 2020; She and Fan, 2018; Zhou et al., 2020). Significant issues such as savings in materials, space, energy, and the global economy have led to the development of more efficient heat exchanger equipment and reductions in costs (Ghalandari et al., 2020; Irandoost Shahrestani et al., 2020; Shadloo et al., 2016; Zhao et al., 2020a). On the other hand, the main hydraulic-thermal goals are to reduce the required heat exchanger dimensions according to the intended heat capacity and increase the performance capacity with the lower temperature difference (Aryanfar, 2020; Maleki et al., 2020; Pishkariahmadabad et al., 2021). Therefore, the engineers proposed several types of heat exchangers with different dimensions and above capacities depending on their application (El Haj Assad et al., 2021; Kumar et al., 2006; Ramezanizadeh et al., 2019; Zhao et al., 2020c; Zhou et al., 2020).

Among heat exchangers, the two-tube heat exchanger in which hot and cold fluids move in two concentric tubes, either in the same direction or on the opposite side, is the simplest heat exchanger (Han et al., 2005; Rennie and Raghavan, 2005; Panahi and Zamzamian, 2017; Zhao et al., 2020b). The Two-pipe exchangers can be applied with different series and parallel arrangements to meet the pressure loss and the average temperature difference (Chen et al., 2021; Prabhanjan et al., 2002). These types of heat exchanger could be suitable for one or both hot and cold fluids at high pressure. The main disadvantage of these exchangers is their low heat transfer per unit area (Dean, 1927, Dean, 1928; Verma and Ram, 1993). Another type of heat exchanger which is called shell and tube exchangers has a bunch of tubes with circular cross-sections placed inside a large cylindrical shell whose axis is parallel to the axis of the shell (Ghorbani Mianroudi et al., 2008). The features of this type of exchanger are the possibility of surface cleaning of the pipes after the formation of sediment and the relative flexibility between the pipes and the shell due to thermal strain. One of the problems in employing this type of exchanger is the presence of death zones, which results in corrosion problems. It should be noted that these heat exchangers are less efficient than the other heat exchangers (El Maakoul et al., 2016; Irandoost Shahrestani et al., 2021).

To overcome some of the problems in using the above heat exchangers, a new design was needed to limit dead areas and make the converter geometry something easier to describe. In fact, one proposal was combining these two types of heat exchangers and finally providing a helical two-tube exchanger for cold thermal energy storage application (Bhanvase et al., 2018; Naik and Vinod, 2018; Song et al., 2021a; Song et al., 2021b; Song et al., 2021c). Numerous studies in the field of heat transfer of different types of heat exchangers indicated that in the heat exchangers that use helical pipes, they were more preferable than direct pipes in terms of heat

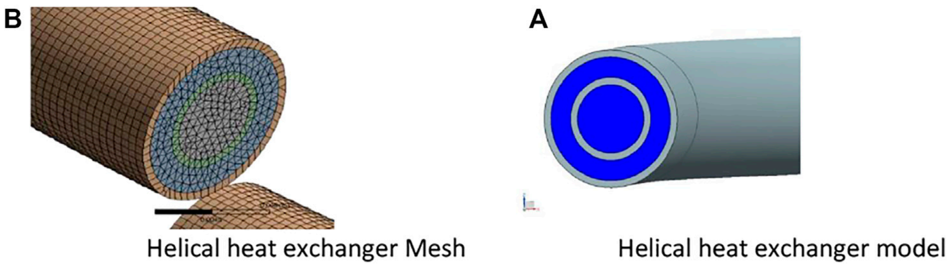

FIGURE 1 | The represented helical heat exchanger **(A)** CFD Mesh **(B)** Heat exchanger model.

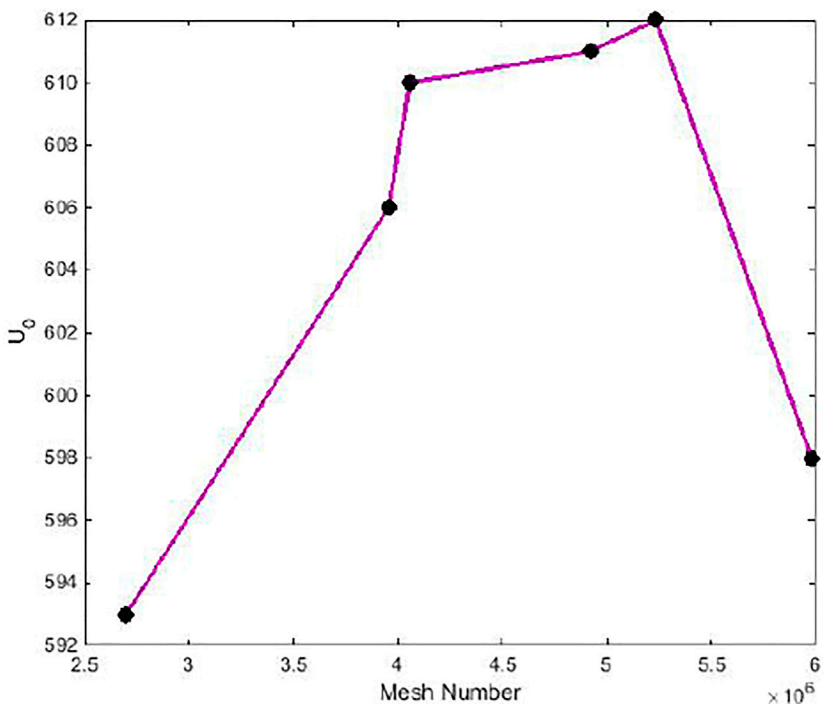

FIGURE 2 | Mesh convergences investigation.

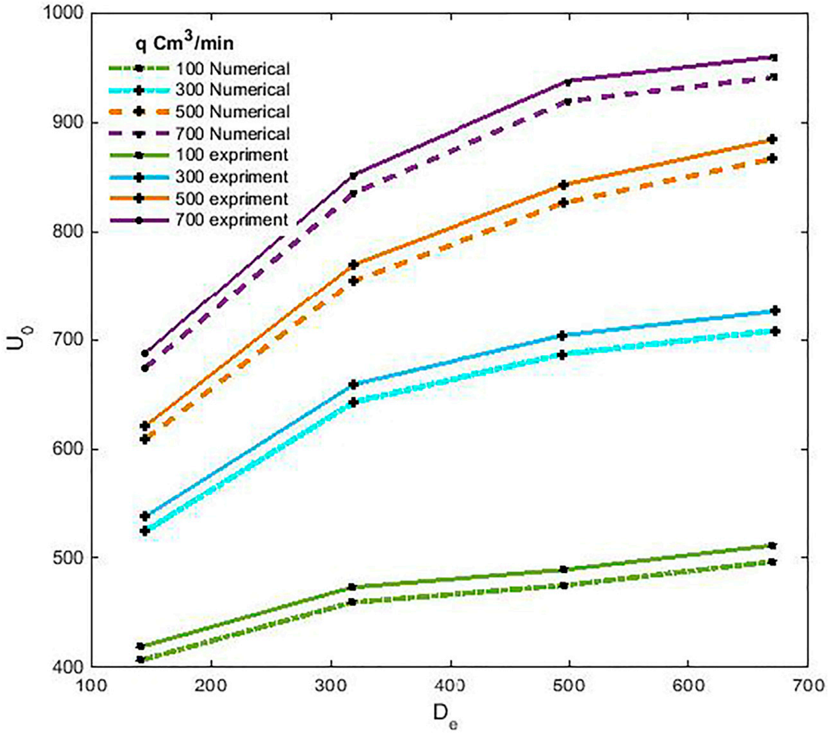

FIGURE 3 | The comparison between the numerical and experimental results for heat transfer coefficient.

transfer (Majid Etghani and Amir Hosseini Baboli, 2017; Prabhanjan et al., 2002). In fact, in helical two-tube heat exchangers, the curvature of the tube causes centrifugal forces to affect the flow inside the tube (Webster and Humphrey, 1997). As a result, the rate of heat transfer increases (Sheeba et al., 2019). Many studies have been conducted on heat transfer and flow characteristics in curved tubes. Indeed, it is worth noting that in this type of configuration, the entire area of the coil is exposed to the moving fluid, and the dead areas that may exist in the shell and tube types of the heat exchanger are removed, and then the flow inside the outer tube can experience secondary flow as well. The first attempt to mathematically describe the flow in a helical tube was conducted by Dean (Dean, 1927; Dean, 1928). In fact, he studied the motion of a stable incompressible fluid flow along with a coil with a circular cross-section. Han et al. (2005) experimentally investigated the heat transfer of condensation R-134a in a circular tube. Their results showed that saturation temperatures have a significant effect on their condensing heat transfer coefficient. Another experimental and numerical study on helical heat exchangers has been performed by Jayakumar et al. (Mahajani et al., 2008; Xiong et al., 2021). Their study showed that the boundary conditions of constant heat flux and constant wall temperature for a real exchanger have bad effects on efficiency. In their investigation, which was based on experimental results, a relationship had been proposed to calculate the heat transfer coefficient of helical inner tubes. In another study, Jayakumar et al. (2010) numerically studied (Chu et al., 2021) single-phase currents within helical tubes. The results of their study showed changes in the local Nusselt number (Nu) along the length and circumference of the wall of a helical tube. The heat transfer characteristics of a helical two-tube exchanger have been numerically and experimentally studied by Kumar et al. (2006). They showed that the total heat transfer coefficient of the exchanger increases with increasing De number of the inner tube at a constant flow rate in the space between the two tubes, and vice versa. Heat transfer studies of helical coils immersed in a water bath have been investigated by Parabhanjan et al. (2004).

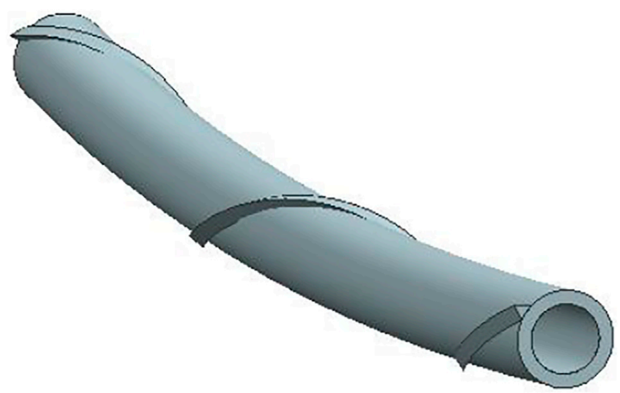

FIGURE 4 | The heat exchanger with triangular Turbolator in inner pipe.

In this paper, a method based on the computation fluid dynamic is employed for predicting temperatures outside of the pipe and evaluating the heat transferring parameter in the helical heat

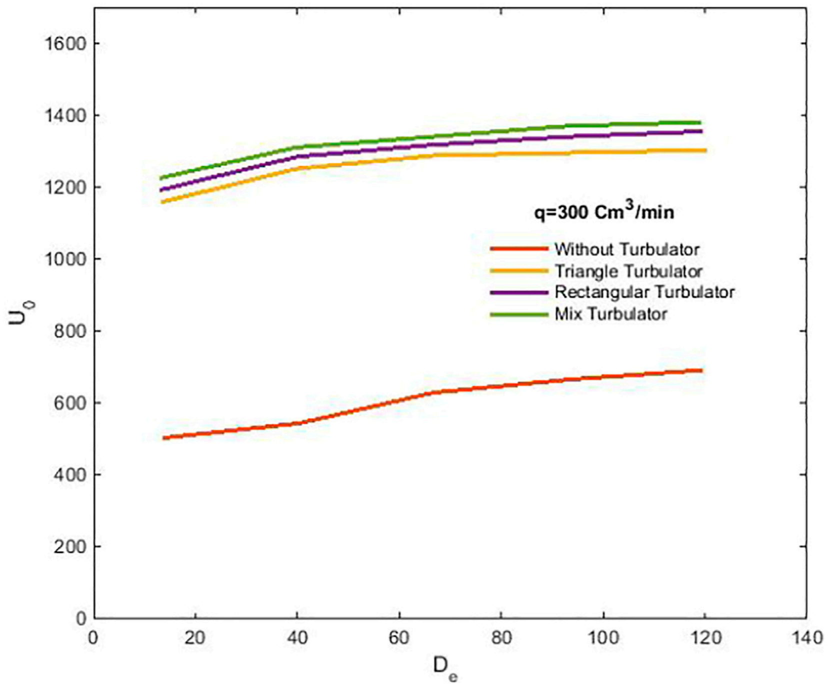

FIGURE 5 | The heat transfer coefficient of the helical heat exchanger with Turbulator.

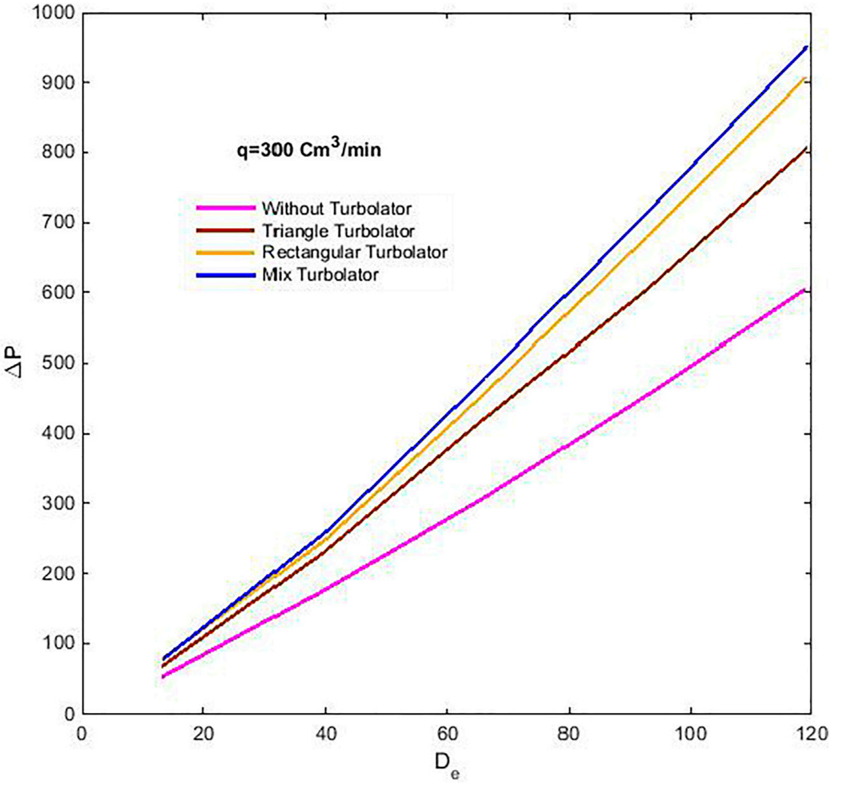

FIGURE 6 | The pressure loss comparison of the heat exchanger with and without Turbulator.

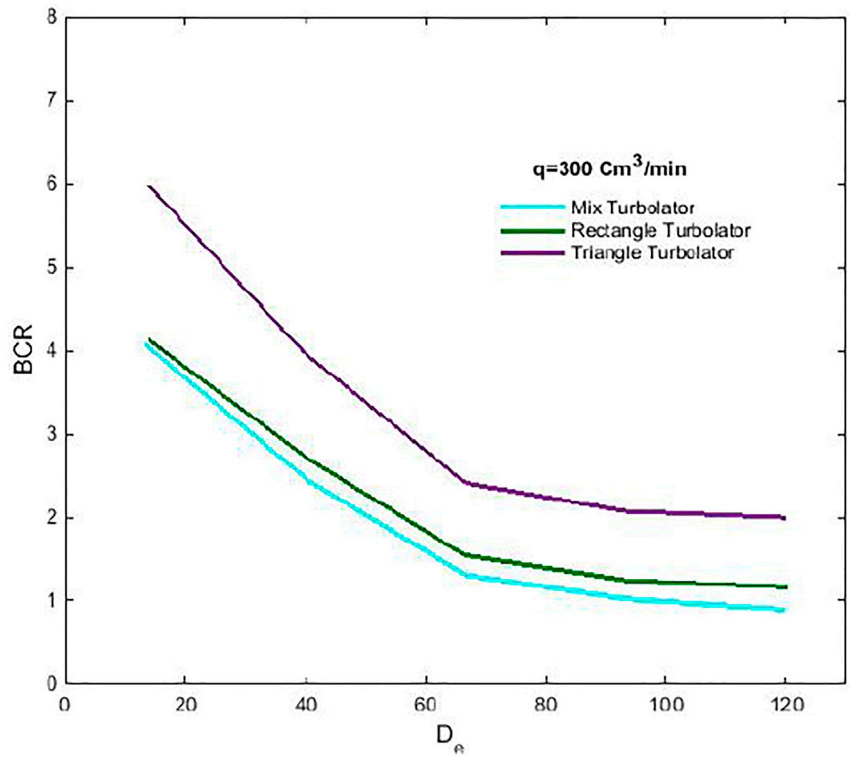

FIGURE 7 | Comparison of BCR factor for the represented Turbolator.

exchanger, firstly. As a new contribution in this area, the rectangular, triangular, and compound shapes of the fin are employed on the inner side of the outer pipe surfaces of the exchanger to study the heat transfer characteristics. Then, based on the represented novel optimization algorithm (Song et al., 2015), the best shape of the selected fin is proposed.

METHODOLOGY

To examine the precision of the represented numerical approach, a heat exchanger with helical case studies is adopted. The first step in heat transfer study of a represented heat exchanger is tubes modeling of the exchanger using computation fluid mechanic technique. The existing experimental results for the proposed case study which was extracted for a simple helical exchanger can be considered as the numerical results validation. The exchanger has one turn and consists of two copper horizontal pipes that are placed in the same center. The outer surface diameter of the outer pipe and its thicknesses are 0.0159 and 0.008 m, respectively. The diameter of the inner pipe is 0.0095 m which has 0.008 m thickness. Also, the pitch dimension and the curvature of the heat exchanger (R) are 0.159 and 0.2359 m, respectively. The validation procedure is conducted for 100, 300, 500, and 700 $(Cm^3/_{min})$ water mass flow rate in the inner tube and 300, 500, 700, and 900 $(Cm^3/_{min})$ mass flow rate for the fluid between the two tubes. The fluids flow in parallel with each other in the layer regimes. The inlet temperature of water between two pipes and the inlet temperature and inner pipe is adjusted 60 and 20.1°C, respectively. The boundary condition in the outer surface of the outer pipe is also considered adiabatic.

The governing equation of the fluid mechanic and heat transfer in a Cartesian coordinate system for constant density and viscosity of the fluid which is water can be represented as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\nabla^2 u+\rho f_x \tag{2}$$

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

$$\rho\left(\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\nabla^2 w+\rho f_z \tag{4}$$

$$\rho C_v\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k\nabla^2 T+\mu\varphi \tag{5}$$

Here, u, v, and f_n are the relative displacements and forces in x, y, and z directions, ∇ is gradient operator, and φ is displacement field, respectively. Also, ρ is density, p is pressure, T is temperature, and μ is dynamic viscosity. Calculation of the overall heat transfer coefficient (U_0) can be performed using the following equation:

$$U_0=\frac{q}{A_0 LMTD} \tag{6}$$

where A_0, q, and $LMTD$ are surface area, heat transfer rate, and mean logarithmic temperature differences, respectively.

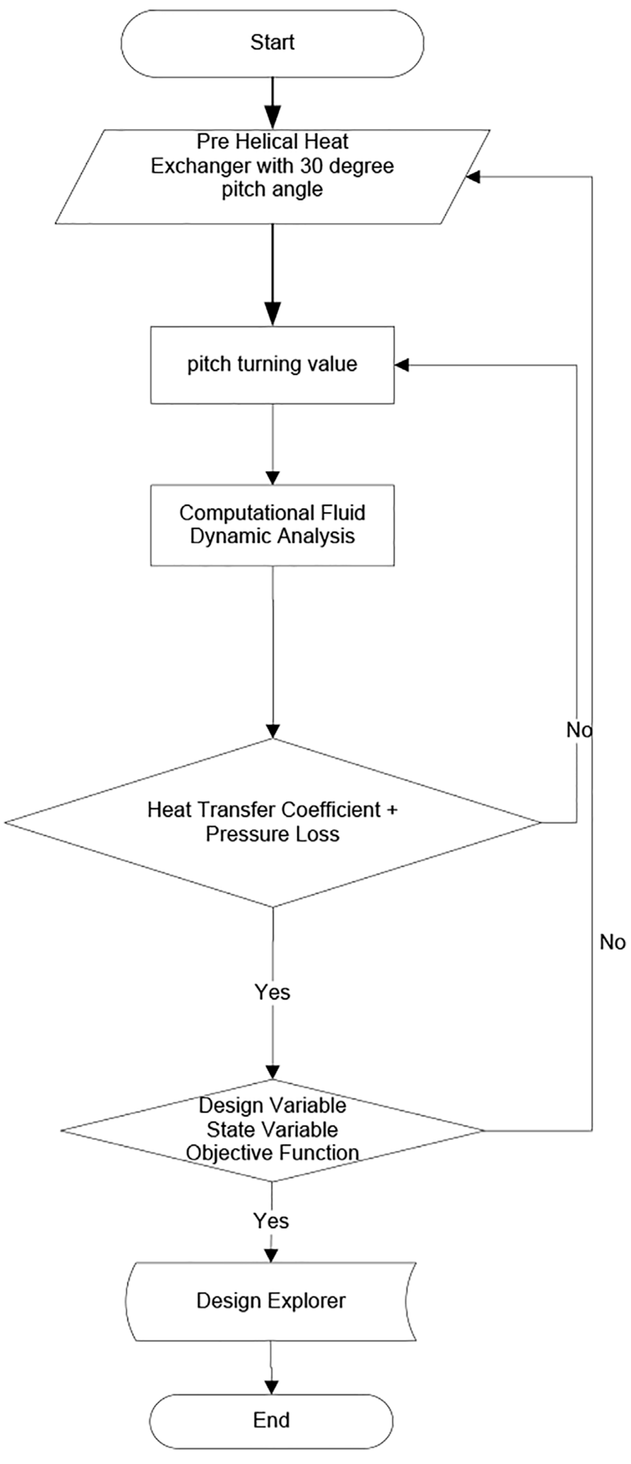

FIGURE 8 | Optimization procedure algorithm.

The log means temperature difference ($LMTD$) can be obtained as:

$$LMTD = \frac{(\Delta T_2 - \Delta T_1)}{\ln\left(\Delta T_2/\Delta T_1\right)} \tag{7}$$

where ΔT_2 and ΔT_1 are the temperature differences between the fluids in the inlet and the outlet. The inner Dean number employed for study of the fluid behavior in curved tubes can be represented as the following:

$$De = \frac{\rho V d}{\mu}\left(\frac{D}{2R}\right)^{0.5} \tag{8}$$

Here V is the fluid velocity and d is the tube diameter, R is radius of curvature, and D is diameter of annulus. The modified Dean number for between the two tubes is calculated using the following equation:

$$De^* = \frac{\rho V}{\mu}\left(\frac{D_0^2 - D_i^2}{D_0^2 + D_i^2}\right)\left(\frac{D_0 - D_i}{R}\right)^{0.5} \tag{9}$$

where D_i and D_o are inner and outer diameter of annulus. Taking advantage of the CFD in-house code, the fluid dynamic analysis is performed for the helical heat exchanger. **Figure 1** shows the helical heat exchanger model and CFD mesh. Specific fluid characteristics of the helical heat exchanger are calculated by definition of the wall boundary conditions and fluid domain analysis. CFD calculations are conducted based on the K–ε model, and the efficiency value is controlled within ± 0.5% of the main working interval. A mesh study is conducted to prove the adequacy of convergence values for the CFD calculation, and a 5236 grid number is selected for further analysis (**Figure 2**).

The comparison result between the numerical calculation and what is represented by Rennie and Raghavan (2005) for the two helical heat exchangers is illustrated in the following **Figure 3**.

As can be calculated, the maximum differences between the numerical and experimental are nearly about 7% and prove the accuracy of the modeling.

FLUID CHARACTERISTICS V.S HELICAL HEAT EXCHANGER WITH FINS

In this section, the effect of the fins (Turbulator) configurations (**Figure 4**) and dimensions on heat transfer and pressure loss of the above helical heat exchanger is investigated. Firstly, three types of fins are selected. The fins have triangular, rectangular, and compound (mix fin) shapes which are located on the top of the inner tube in a spiral form with an above-represented pitch angle. The current for the exchanger is parallel, and for all three different geometries of the fin, the mass flow is the same as those used for a simple two-pipe exchanger (without fin). Indeed, the mass flows are selected in such a way that value of the minimum difference exists between the experimental and numerical results. The comparison proves that the mass flow 300 ($Cm^3/_{min}$) has about a 3% difference between the numerical and experimental results.

The fins are added in order that the effective surfaces have equal levels in terms of heat transfer performances. This means that the levels of heat transfer are considered equal for three types of the Turbolator. The dimensions of Torbulator are 0.001, 0.001 × 0.002 for equilateral triangle and rectangular fins, respectively. The extracted pressure loss and heat transfer coefficient of the helical pipe with the above fins are represented in the following figures (**Figures 5**, **6**).

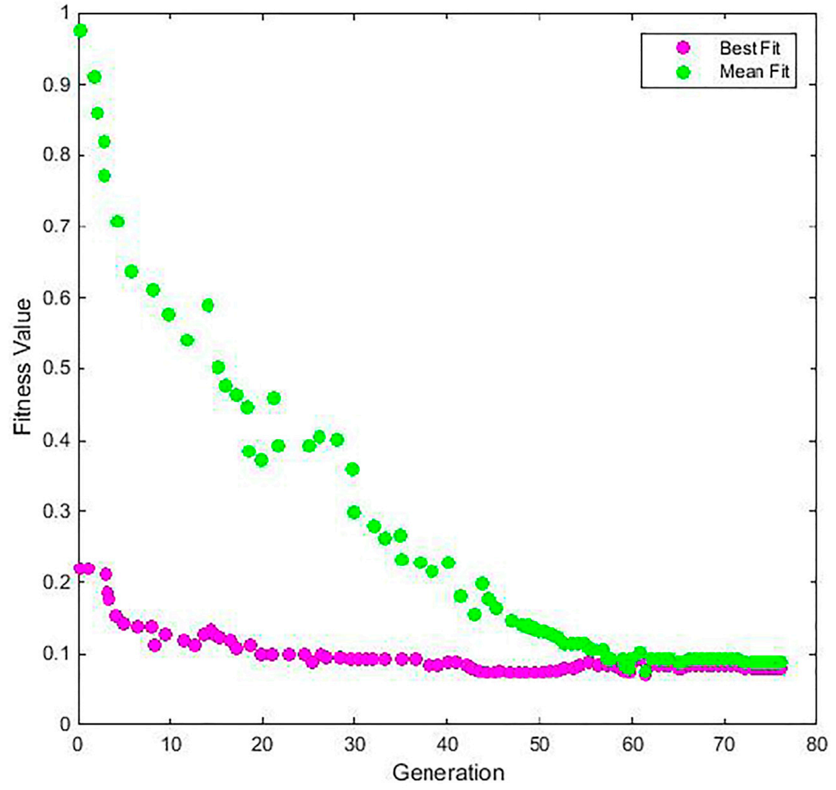

FIGURE 9 | Fitness value vs. generation numbers.

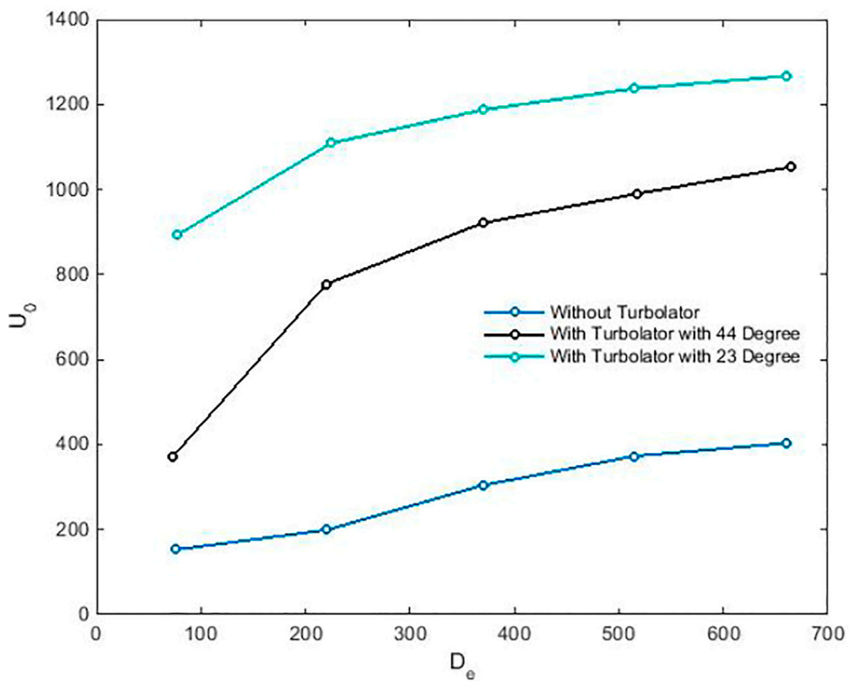

FIGURE 10 | The comparison between the heat exchanger without Turbolator and heat exchangers with optimized and specific turn of Turbolator.

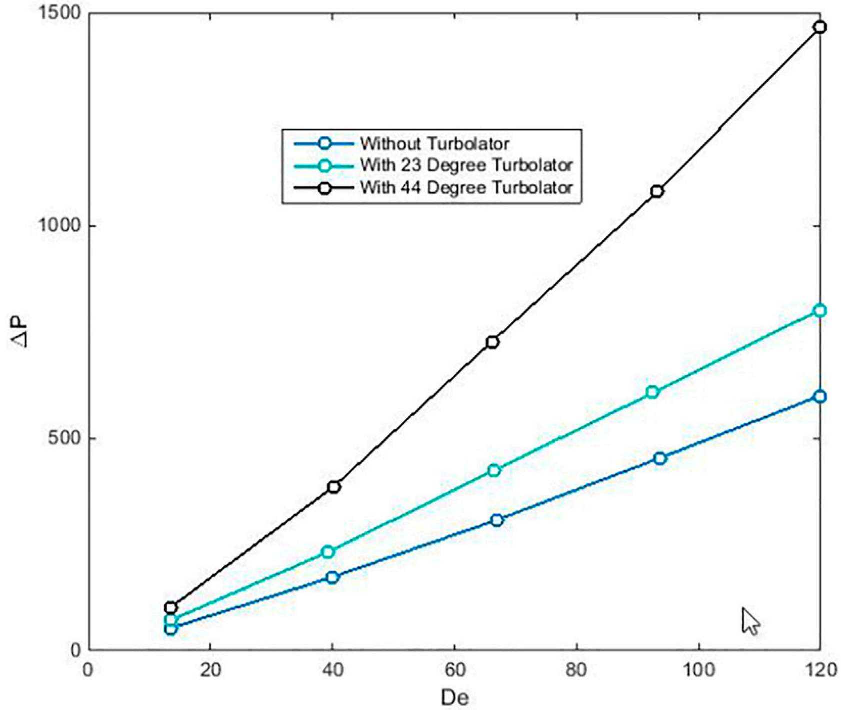

FIGURE 11 | The comparison between pressure loss value of the heat exchanger without Turbolator and heat exchangers with optimized and specific turn of Turbolator.

As shown in **Figure 5**, there is a remarkable increase in heat transfer coefficient for helical heat exchanger with fin (Turbolator) compared to the simple heat exchanger. Among the represented Turbolator, the rectangular and mix fins have more influence on heat transfer coefficient than the triangular Turbolator. The pressure loss between two pipe annulus areas for the helical heat exchanger is illustrated in **Figure 6**.

The study on pressure losses for different geometries shows a different behavior compared to heat transfer coefficient. As depicted in **Figure 6**, pressure loss has less value for equilateral triangle geometry than the rectangular and compound or mix geometries in a similar De number for represented mass flow with 300 $(Cm^3/_{min})$ value. Moreover, the investigation shows that the equilateral triangle geometry in De number nearly about 15 has minimum and nearly about 3% difference compared to the heat exchanger without fin. Evaluation of the best configuration has been conducted by the following formulation:

$$\mathrm{BCR} = \frac{\Delta u/_{u_s}}{\Delta p/_{p_s}} \tag{10}$$

where u and p are the heat transfer coefficient and pressure loss, respectively. Furthermore, the s index indicates the values of the heat exchanger without Turbolator. Indeed, the benefit-cost-ratio (BCR) is defined in terms of the total increasing amount of heat transfer coefficient overpressure loss of heat exchanger with Torbulator over the same heat exchanger without Torbulator. **Figure 7** represents the calculated BCR factor for three types of Turbulator in $q = 300\,(Cm^3/_{min})$.

Based on extracted results illustrated in **Figure 7**, the equilateral triangle has more values of BCR for a specific pressure loss and shows better performances than the others. Furthermore, the BCR performance calculation proves that the rectangular configuration is preferable to the mix configuration.

OPTIMIZATION BASED ON FLUID PERFORMANCE CRITERIA

The represented multi-disciplinary design optimization (MDO) method is taken to propose the best performance of the mentioned helical heat exchanger for the equilateral triangular Turbolator. Genetic Algorithm (GA) and Artificial Neural Network (ANN) (Chen et al., 2020a; Chen et al., 2020b) are adopted to propose the best pitch turn of the above represented equilateral Turbolator. This design flow is constructed based on the parametrization of the pitch turning value, optimization algorithm, and a surrogate model on CFD results. Design of experiments (DOE) is also employed to create a sufficient database using the main mentioned heat exchanger along with Neural Network Algorithm. Based on the created database, the pressure loss and heat transfer coefficient criteria of the heat exchanger are evaluated. The pitch turning value is proposed using numerical calculation. The library is created based on the ANN results and converged to what is extracted in computational fluid dynamics. Finally, using the target goal satisfaction with results comparison procedure, the objective function is investigated. Indeed, the procedure is followed to minimize the pressure loss and increase the heat transfer coefficient using the following objective function:

$$OF = n_1 U_0 + n_2 \Delta P \tag{11}$$

where n_i, U_0, and ΔP are characterized as weighting coefficient, the heat transfer coefficient, and pressure loss, respectively. The variable n indicates the importance of the other of the multiplier. The calculations are conducted for equal impact factors for both heat transfer coefficient and pressure loss. Furthermore, the library of the DOE method is completed based on the meta-models theory, ANN analysis, and the CFD calculation of the helical heat exchanger with Turbolator and 3-degree pitch turning value. The investigation is followed by a feed forward-back propagation network with 15 hidden layers and one output neuron. The interval of the suitable pitch turning value is set to be in a 100% deviation of the original pitch turning value. Applying the ANN and the goal described in GA (**Eq. 11**), the optimization process is followed and the estimated values are compared with CFD calculations, which result in updating the ANN. The MDO optimization flow chart of the helical heat exchanger is proposed in **Figure 8**.

The Latin Hypercube type of 80 DOE experiments is applied and the validation procedure is conducted by 40% data and the remaining taken as network trainer. In the current training procedure of the ANN algorithm, the values of efficiency precision and deviation are 99.9 and 0.09%, respectively. The results of the three-dimensional CFD calculation and the values estimated with the network are compared in each cycle of genetic algorithm execution to evaluate the accuracy of the network. The network is re-trained if the three-dimensional CFD error is more than 0.3% and then results are added to the database. If the precision of the results is less than 0.3%, the CFD is ignored and just a neural network is employed for following the optimization procedure. Using 78 generations and 156 members in each generation, the process of optimization is conducted. As illustrated in **Figure 9**, fitness value has started at nearly 1 and reached to the value of less than 0.08. The estimated results are compared with the main composite wing characteristics to reach the best pitch turning the value of the pitch values. The result of the best value is obtained after seven iterations and is presented in Table 1.

The effect of the pitch angle and the best values of it based on the heat transfer coefficient and pressure loss for the represented helical heat exchanger is evaluated and depicted in **Figures 10**, **11**. As can be seen, the amount of the pitch turns for three modes prove that the 23-degree pitch turn shows the best performance for the helical heat exchanger.

From **Figure 10**, the heat exchanger with Turbolator has different values of heat transfer coefficient than the heat exchanger without Turbolator. Moreover, the comparison of the heat transfer coefficient between the heat exchangers with Turbolator and without Turbolator shows fewer difference values for a low De number. In other words, the evaluations for low De number reveal nearly 450% differences between the heat transfer coefficients of the heat exchanger with the presented optimized turn of Turbolator and the heat exchanger without the Turbolator.

However, the addition of the pitch turn of the Turbolator can increase the amount of heat transfer coefficient compared to the simple heat exchanger, and the study in **Figure 11** discloses that the 23° turn of Turbolator demonstrates the minimum value for pressure loss. The investigation in **Figure 11** proved that in low De number, the pressure loss has a minimum value in comparison to high De number values. The results also physically declared that increasing the pitch value can tend to blockage of the flow and decrease the percentage of the helical heat exchanger with Turbolator performances compared to the simple helical heat exchanger.

CONCLUSION

The optimization procedure of the helical heat exchanger was represented in this study. The effect of the mass flow rate on the helical heat exchanger performances were investigated and compared with what was extracted with experimental methodology. The result show good agreement and the maximum 7% differences between the numerical and experimental results were evaluated. Based on the minimum differences that occur between the mentioned numerical and experimental results, the flux value 300,300 $Cm^3/_{min}$ is selected. The effects of the different types of Turbolators involving equilateral triangle, rectangular, and mix shapes on the heat transfer coefficient and pressure loss were evaluated. The fluid characteristics prove that the equilateral triangle Turbolator which is located on the outer face of the inner pipe demonstrated the best results based on the definition of the new BCR represented formulation. Then, the presented MDO algorithm was applied to the helical heat exchanger geometry to propose the 23° pitch turning angle that is the best value with minimum pressure loss and maximum heat transfer coefficient factor.

DATA AVAILABILITY STATEMENT

The original contributions presented in the study are included in the article/Supplementary Material, and further inquiries can be directed to the corresponding authors.

AUTHOR CONTRIBUTIONS

WX and ZX collaborated in writing the manuscript and the representation of the Idea. IM and MS helped in the extraction of the result and the development of the optimization code. Also, MG and MA employed the optimization method on the model and feedback to the other authors.

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