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Optimization of Micro-Fin Tubes in Dual-Tube Heat Exchangers Using Genetic Algorithm

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Abstract

In this study, laminar flow is considered, and the heat transfer oil is investigated numerically in horizontal, smooth and micro-fin tubes at different fin heights and spiral angles with boundary conditions of constant wall temperature. The variables considered in the micro-fin tubes are the number of fins that varies from 40 to 60, the micro-fin's height that varies is from 0.2 to 0.5 mm, and the micro-pin spiral angle that varies between 5 and 47 degrees. This study was conducted for Reynolds number ranging from 500 to 100, inner tube diameter of 8.62 and outer diameter of 9.52 mm. The numerical resolution was done using ANSYS fluent v.15 commercial software.
The numerical method was validated by comparing the results of simulating laminar flow of oil through a smooth and micro-fin tube with experimental results and previous observations. The simulation results are then used in the genetic algorithm for optimizing the micro-fin geometry. The results showed that increasing Reynolds number and micro-fin height increases the optimum surface. However, a different trend is observed for the desired spiral angle.

Keywords: Heat exchangers, Heat transfer coefficient, Micro-fin, Genetic algorithm, Pressure drop.

1. Introduction

Today, heat transfer plays an important role in advanced technologies such as mechatronics, microelectronics, power generation, air conditioning and petrochemical, and oil and gas industries. One of the reasons for the low heat transfer rate is the low heat transfer from conventional liquids such as water, ethylene glycol, oil, and so on. The poor thermal conductivity of ordinary liquids limits the amount of heat transfer. The use of micro-fin spiral tubes, which significantly increases the heat transfer surface without affecting the overall size of the heat transfer device, is considered as a common method to achieve the goal of increasing the efficiency of the heat transfer of fluids.

Commercial applications of increased interior tubes has significantly increased in recent years to achieve desired heat transfer rates.

Nomenclature

Cp	specific heat. J/kg.K	v	velocity, m/s
d	tube diameter.m	U	Uniform inlet velocity, m/s
e	fin height.m		
g	gravitational acceleration, m/s2	Greek symbols	
h	heat transfer coefficient,W/m2.K	α	helix angle
k	thermal conductivity, W/m.K	β	apex angle
L	tube length, m	μ	dynamic viscosity, N.s/m2
ṁ	mass flow rate, kg/s	θ	dimensional coordinates
Nu	Nusselt number	ρ	density, kg/m3
p	pressure, Pa, and fin pitch, m	Subscripts	
Re	Reynolds number	f	fluid
t	fin width, m	i	inlet
r, z	dimensional coordinates, m	o	outlet
T	temperature, K	w	wall

A number of recent experimental studies have been conducted on single-phase fluid flow through micro-fin tubes. Kapitti et al. [1] studied the characteristics of heat transfer and pressure drop for testing the flow of water in micro-fin tubes. Their observation was that the heat transfer coefficient for the micro-fin tubes in the turbulent flow regime increased up to 190%. Nevertheless, they concluded that the heat transfer coefficient has increased only by about 20% in the laminar-flow regime in comparison with smooth tube. In another experimental study, Han and Lee [2] investigated the characteristics of single-phase heat transfer and the characteristics of fluid flow in micro-fin tubes. They introduced an efficiency index to evaluate the increase in heat transfer performance. They observed that tubes with larger relative roughness and smaller spiral angle show greater heat transfer. Lee et al. [3] investigated the heat transfer mechanism in galvanized tubes through flow imaging. They concluded that the flow has parabolic patterns in the laminar regime; however, these patterns were broken due to accidental separation from vortices in turbulent regime. Testing the flow of water inside a plexiglass tube was performed by a twisted tape. They observed that fluid flow was dominated by a rotary pattern for silicon for an angle of less than 30 degrees, and by a crossed pattern for a spiral angle of 70 degrees.

Nafone and Cerimon [4] presented a study of heat transfer and the characteristics of two-tube horizontal pressure drop with and without spiral fins. They observed that the insertion of spiral fin has had a significant effect on the increase in heat transfer. However, the friction factor of the tube with the spiral fin increases at the same time. Afroz and Mira [5] measured the pressure drop in a single-phase turbulent flow. They investigated the flow of fluid inside the micro-fin spiral tubes for different fin sizes to create a general correlation for the friction factor and the single-phase fluid flow for such tubes. Their results showed that larger angle and fin height caused pressure drop in inside the angularly-shaped micro-fin tubes and, furthermore, the pressure drop in the spiral tube was significantly higher than that in the smooth tubes, which works depending on the geometric parameters of the sheet and the velocity of the mass fluid. Zendyuk et al. [6] investigated the heat transfer coefficient and friction factor for eight different micro-fin tubes and also for testing a smooth tube. Their results showed that a micro-fin tube with spiral angles of 48, fin-height to pipe-diameter ratio of 0.0244 and number of fins of 48 can be appropriate due to the above factors and the mean factors f considered for all Reynolds numbers. Seydik et al. [7] linked the characteristics of heat transfer and integrated pressure drop to the flow of fluid inside a two-tube heat exchanger with their experimental micro-fin tubes for heat transfer and pressure drop with Dittus-Boelter and Blasius-type relations, respectively. They concluded that the micro-fins had a significant effect on the amount of heat transfer and pressure drop in such tubes. They considered a wide range of fluid flow regimes from laminar to completely turbulent one. For a constant pumping force, their results showed that the finned tube with a twisted tape creates the maximum heat transfer of 400% in the laminar range and 140% in the turbulent range compared to the heat transfer of the smooth tube. A similar comparison was made for a spiral tube with twisted tape and resulted in a maximum increase of 600% in the laminar range and 140% in the turbulent range compared with the appropriate values smooth tube.

Selen et al. [8] investigated the characteristics of the single-phase pressure drop of smooth and micro-fin tubes. They used a horizontal flow in a two-tube heat exchanger as a test section. Their measurements showed that the friction factor and pressure drop were generally higher for the micro-fin tubes than the corresponding values for the smooth tube, which means that the micro-fin tubes cause more flow disturbances as a result of rotation and circulation caused by the fins, causing more pressure drop. Recently, Ikhvan Baghladi et al. [9] tested the heat transfer of copper oxide nano-fluid using smooth and finned horizontal tubes with constant wall temperature. Based on these experiments, they concluded that Nusselt's number of the base fluid in the micro-fin tube was 56% higher than that of the base fluid in the flow through the smooth tube. Derakhshan et al. [10] investigated the heat transfer characteristics of a mixture of nanowire MWCNT oil, which flows through smooth and micro-fin tubes for boundary conditions of equal wall heat flux. Their results showed that the use of nano-fluids instead of pure fluids was a more effective way to increase the heat transfer coefficient than the use of a micro-fin tube.

2. Shape Definition and Geometric Model

Figure 1a shows a photograph of the studied micro-fin tube. The geometric parameters described in the micro-fin tubes are: the fin height, the fin step, the helix angle, and the inner diameter of the tube. The cross sectional shape also strongly affects the friction factor and the heat transfer coefficient in the micro-fin tubes. The fins of the micro-fin tube have thickness, whose shape is determined by the base width, the fin angle and the fin height. Some parameters are shown in Figure 1b. The geometric parameters of the present micro-fin tube are given in table No.1.

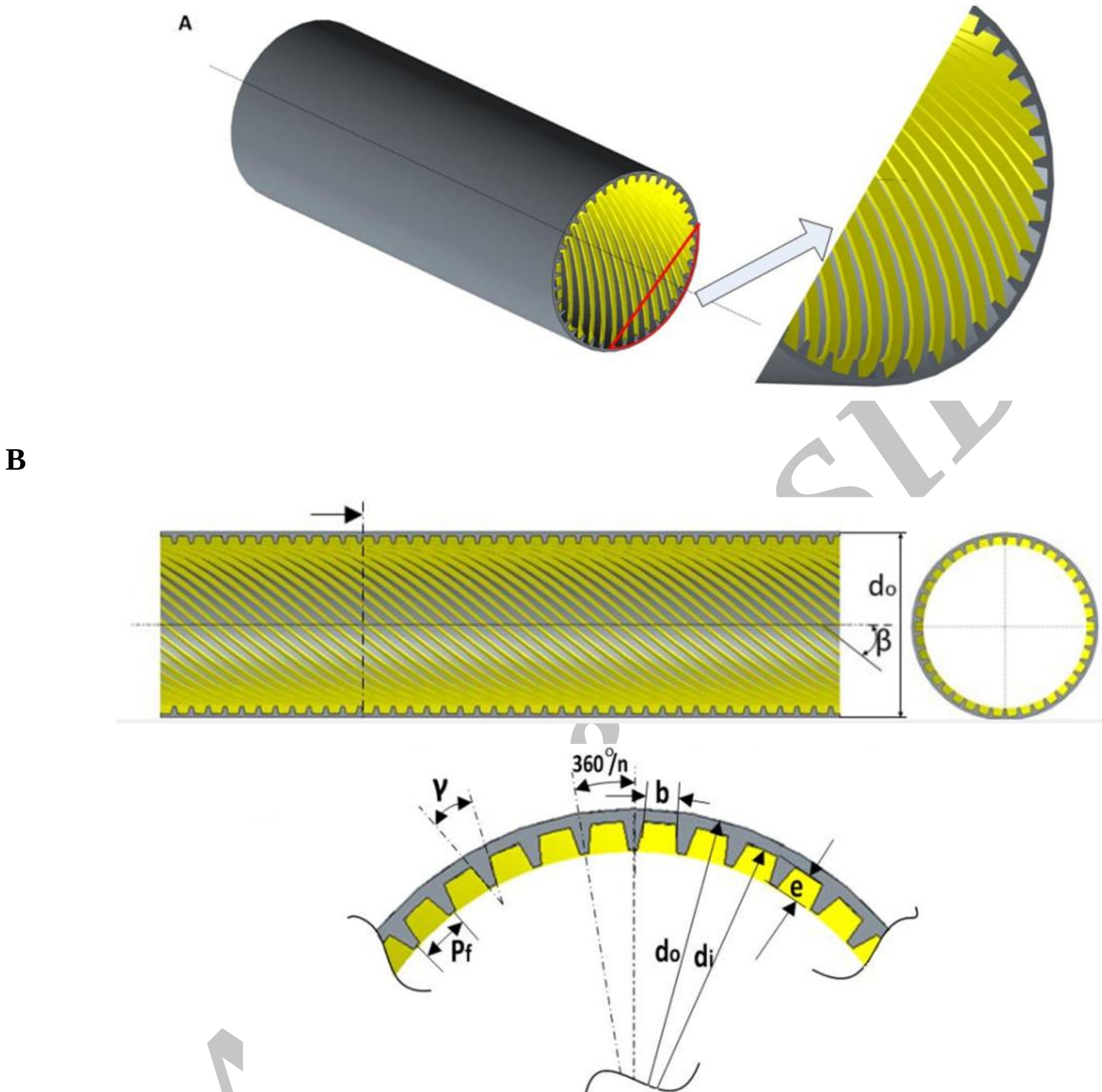


Figure 1- Micro-Finn tube tested: (a micro-fin tube image and (b) finite geometry parameters.

Table 1- geometric parameters of the present micro-fin tube

inner diameter	8.62mm
Outer diameter	9.52mm
Length	500mm
Fin thickness	0.2mm
Fin height	0.2-0.45mm
Fin pitch	0.40mm
Helix angle	5-45
Helical pitch.($\pi d/tan\ \alpha$)	27.07-309.37mm
Apex angle	40

In addition, a smooth tube was also tested. The outer diameter of the tube is 9.52 mm, the wall thickness of the tube is 0.45 mm and the tube length is 500 mm. The roughness of the smooth tube is less than 0.02 mm. Both smooth and micro-fin tubes are made of copper. The influx boundary conditions of the wall are applied on the wall of the tube. The liquid enters the tube and passes at a uniform velocity and constant temperature of 15 °C. Changing the uniform inlet velocity will change Reynolds number. The wall of the tube is kept in an insulated compartment that gives us a constant temperature of 98 °C at the outlet of the tube. The pressure is set to zero at the outlet of the tube. These boundary conditions have been applied equally to both smooth and finned tubes.

3. Numerical Implementation

The governing equations for laminar non-isothermal flow of the heat transfer oil through smooth and micro-fin tubes for fixed wall boundary condition are applied numerically using ANSYS fluent v.15 commercial software, which is an element-based finite volume method for solving the governing equations. The computations are performed using parallel processing on a computer that has 2.8 GHz i5 processors and can reach 4 GB of RAM. The CPU time is about 9 hours. In order to save computation time, the boundary layer mesh was applied near the walls, and we set it to 4. The considered geometry was originally designed in the Catia r21 software, and then designed and modeled in the geometry section of ANSYS software due to the lengthy import time.

4. Numerical Pattern

The governing equations for laminar non-isothermal flow of the heat transfer oil through smooth and micro-fin tubes for constant wall temperature boundary conditions are applied numerically using ANSYS CFX v.15 commercial software which is an element-based finite volume method based on limiting the governing equations [26] for solving the governing equations. A "high resolution" design (second order) is used to interpret the transformer conditions in transport equations. To obtain convergent solutions, a time scale equation of 0.2 is used for momentum equations, while an automated time scale is used for the energy equation.

In order to save computation time, it is possible to create a fine mesh near the micro-fin tube wall, and in order to minimize mesh generation error, a three-dimensional spiral domain containing one of the fins of the micro-fin tube is selected as the computational range. This domain was created geometrically using Catia V5 software and subsequently entered into CFS ANSYS. Alternating courses and symmetry boundary conditions on the two lateral surfaces of the computational range were considered for the microorganism and smooth tubes respectively. Figure 2 shows the geometry of the computational domain for the micro-fin tubes. After solving the governing equations and obtaining the flow and temperature field, the friction factor is calculated from the following equation.

$$f = \frac{\Delta\rho . d_h}{2 L_p U^2} \quad (1)$$

Where dh is the hydrodynamic diameter of the pipe, it corresponds to formula 2:

$$d_h = \frac{d}{1 + \frac{de}{p}} \tag{2}$$

Steam pressure temperature, which is required to calculate the heat transfer coefficient 3:

$$t_b = \frac{1}{m} \iint_A puTrdA \tag{3}$$

The current heat transfer coefficient and Nusselt then calculate their number using the following formula:

$$h = \frac{mc_p}{\pi dl} \mathrm{In}\left(\frac{t_{w-t_{b.i}}}{t_{w-t_{b.o}}}\right) \qquad Nu = \frac{hd_n}{k} \tag{4}$$

Where Tb, I and Tb, o are, respectively, the inlet and outlet fluid temperatures. Additionally, the Reynolds number is estimated based on the integrated inlet fluid velocity, thermo-physical properties at the inlet temperature of the fluid hydraulic diameter for determining the optimal conditions related to pressure drop and heat transfer for the micro-fin tubes. The performance index used is as defined below:

$$\eta = \frac{h_{microfin}/h_{smooth}}{f_{microfin}/f_{smooth}} \tag{5}$$

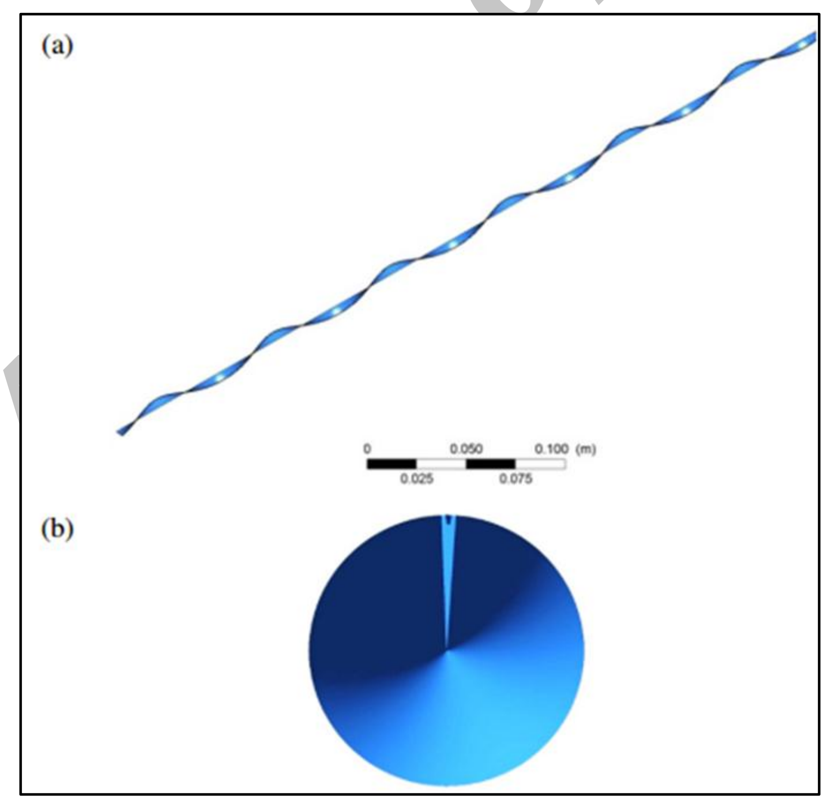


Figure 2 - the geometry of the computational field, which includes a micro-fin and a single fin. (a) A three-dimensional view (b) A sectional view

4.1. Discretization (mesh)

The domain is divided into a finite set of control volumes or discrete cells, which are referred to as mesh discrete domain. The meshes have different types and geometric shapes, and have their own application and performance. In this paper, according to almost complex geometry, we used triangular mesh to provide a high-quality solution and less cell count. To determine a suitable mesh for numerical simulations, a mesh-independence study is conducted for the laminar flow and the heat transfer fluid through the smooth and micro-fin tubes for Re=1000 and Re=500.

4.2. Numerical Method Validation

In order to evaluate the numerical method, the results of the present study for the extended flow through micro-fin tubes have been compared with the experimental results of Siddique and Alhazmy [13]. They performed the experiments in a two-tube heat exchanger, which included a cooling system and a hot water cycle system with a micro-fin tube. Two volume flow exchangers were used to measure the flow at the inlet of the experimental tube and the shell. The specifications of the micro-fin tubes used in their experiments and in the simulation carried out for investigating the numerical method are presented in Table (1). Figure 3 shows the experimental results of Siddique and Alhazmy for changing the mean Nusselt number for non-isothermal flow through a micro-finned tube according to Reynolds number.

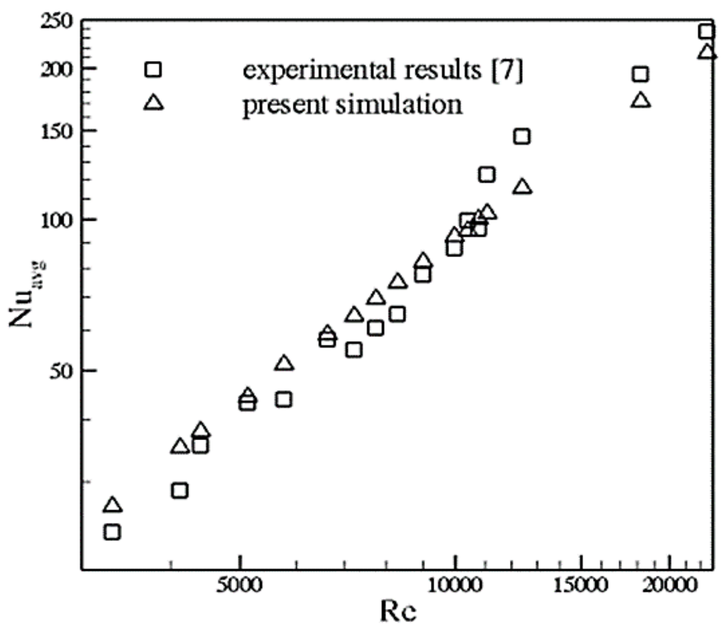


Figure 3- Comparison of the result of the variation of the average Nusselt number for the non-isothermic flow through the micro-fin tube according to the number of Reynolds number

5. Genetic Algorithm

The solution obtained to a problem solved by the genetic algorithm is constantly improved. The genetic algorithm begins with a set of solutions that are shown through the chromosomes. This set of solutions is called the initial population. In this algorithm, the solutions obtained from a population are used to generate the next population. In this process, it is hoped that the new population would be better than the previous population.

Selecting some solutions from all the solutions (parent) to create new solutions, or offspring, is done based on their fitness. It is natural that more appropriate solutions have a greater chance of re-production. This process continues until a predetermined condition (such as the number of populations or the improvement extent of the solution) is satisfied. The general schema of the genetic algorithm is as follows:

1. Generating random population of n chromosomes

2. Evaluating the fitness function f(x) of each chromosome x in the population

3. Creating a new population based on repeating the following steps:

4. Selecting two parent chromosomes from a population based on their fitness.

5. Considering a specified value for the possibility of applying a crossover operator and then performing combining operations on parents to create offspring. If there are no new combinations, the offspring will be the same as the parents.

6. Considering the possibility of mutation and then changing the offspring.

7. Replacing new offspring in the new population.

5.1. Steps to Implementing the Genetic Algorithm

After expressing the basic concepts, different stages in using the genetic algorithm are investigated. First, the variables to be determined are specified according to the problem form. Then, these variables are properly coded and displayed in the form of chromosome. Based on the objective function, a fitness function for the chromosomes is defined, and an arbitrary initial population is randomly selected. Subsequently, the fitness function value is calculated for each chromosome of the initial population.

6. Evaluation of the performance of spiral finned tubes

The purpose of the optimization method is to find the most optimal value of pressure drop, conduction heat transfer coefficient with respect to the changes made to the geometry, in order to provide the maximum heat transfer in the new and optimized geometry.

The variables of the micro-fin tube to be optimized include the number of fins varying from 40 to 60, fin height from 0.2 to 0.5 mm, and the micro-pin spiral angle between 5 and 47 degrees. This study is for Reynolds number of 500 to 1000 and for inner diameter of 8.62 mm.

To ensure a unique solution for optimal micro-fin tubes in each case, the genetic algorithm is repeated several times to ensure that similar results are obtained for optimal micro-fin tubes. The results of the genetic optimization method for micro-fin tubes of heat exchangers are presented below.

Initially, the parameters related to the pressure drop are entered to the algorithm and optimized. Figure 4 examines the optimal pressure drop with respect to different Reynolds numbers, which represents the mutual pressure drop index with respect to Reynolds numbers relative to the number of repetitions. These numbers will be for five different Reynolds numbers of 220, 340, 420, 580, and 700 for an inner diameter of 8.62 mm and an apex angle of 40 degrees.

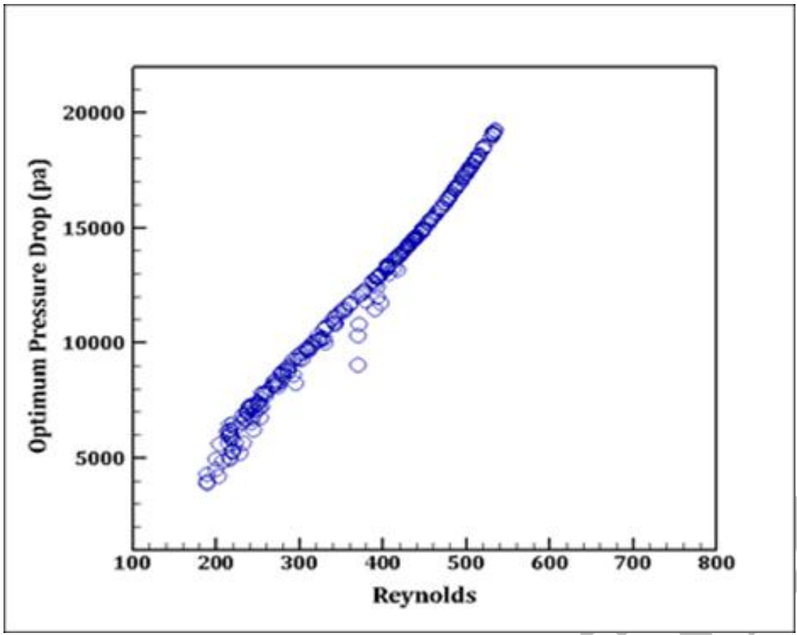
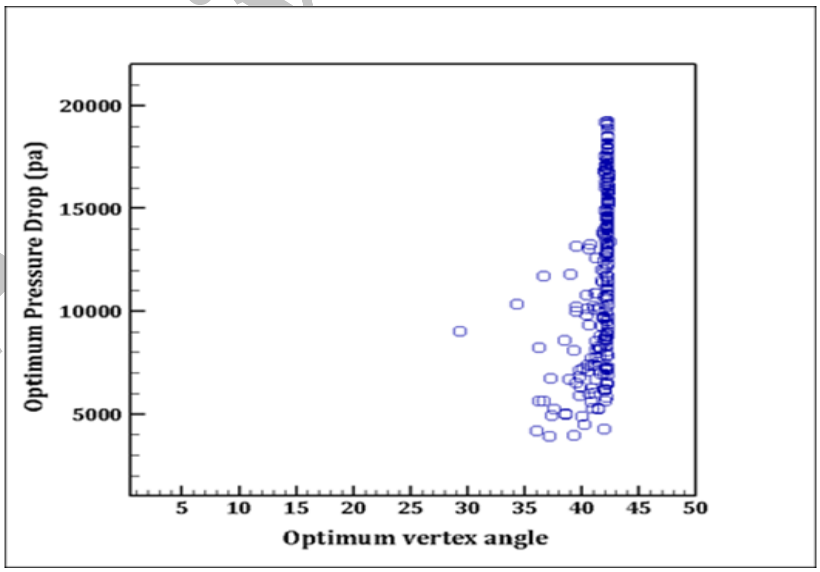


Figure 4- the pressure drop diagram on the number of Reynolds

The optimal values of the considered micro-fin variables in the different pressure values are presented in Figure 5. The results of Figure 5 are for an apex angle of 40 degrees and an inner diameter of 8.62 mm. In Figure 5-a, the variation of the desired spiral angle has been obtained with 6 different data and with considered micro-fin number of 48. As shown in Fig. 5-b, the height of the micro-fin and its relationship with the pressure drop are investigated with 6 repetitions using different data, and the optimal height value is ultimately determined.

a)



b)

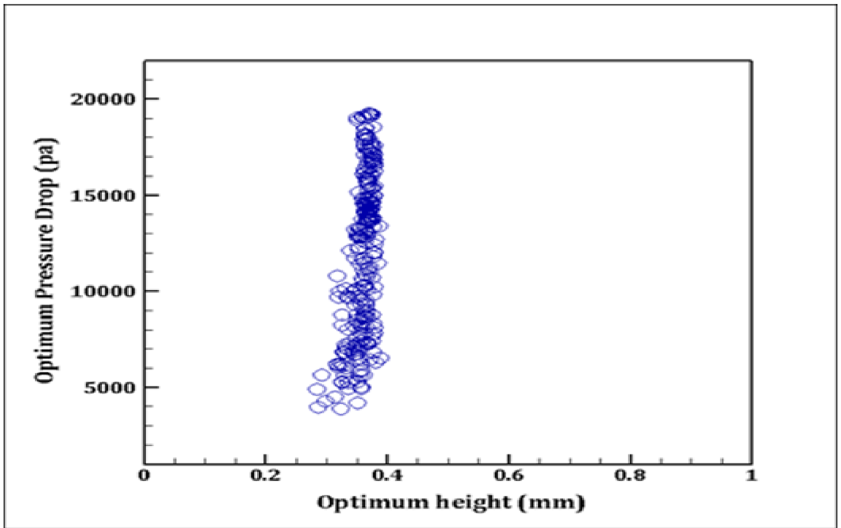


Figure 5-a) Optimal pressure drop relative to the spiral vertex angle, b) the optimal pressure drop compared to the micro-fin height

For fluid flow through micro-fin tubes, in addition to the main axial flow, a secondary spiral flow is generated due to the existence of micro-fins. The secondary spiral flow, which depends strongly on the spiral micro-fin angle, results in more fluid mixing between the fluid layers with different temperatures. This fluid flow pattern was observed by Bergles and Ravigururajan in their experimental study using the flow visualization technique [13]. Induction mixing increases heat transfer. This effect is enhanced by increasing the micro-fin spiral angle. However, the alternating secondary fluid flow increases the pressure drop through the micro-fin tube. Initially, i.e. for the low values considered for Reynolds number, the effect of increasing heat transfer is predominant, and using the micro-fin with relatively large helix angle, heat transfer is completely improved, refer to Figure 5-a.

In what follows, we investigate the optimal heat transfer coefficient with respect to different parameters and through a number of repetitions for achieving a comprehensive matching. Figure 6 shows the optimal value of the conduction heat transfer coefficient with respect to different heights for an inner diameter of 8.62 mm and an apex angle of 40 degrees. The desirable micro-fin height is considered by increasing the heat transfer for all fins while considering the inner tube. In addition, the higher micro-fin height increase rate, the higher the pressure drop across the tube, which in turn has a negative effect on the performance index. For low Reynolds numbers, that is when the optimal optic angle is large, the effect of the micro-fin height on the pressure drop is significant; hence, the micro-fin height has less effect on the increase of pressure drop in comparison with its effect on the increase of heat transfer. As a result, with increasing the optimal height, the Reynolds number increases. Figure 7 shows the height with respect to Reynolds number.

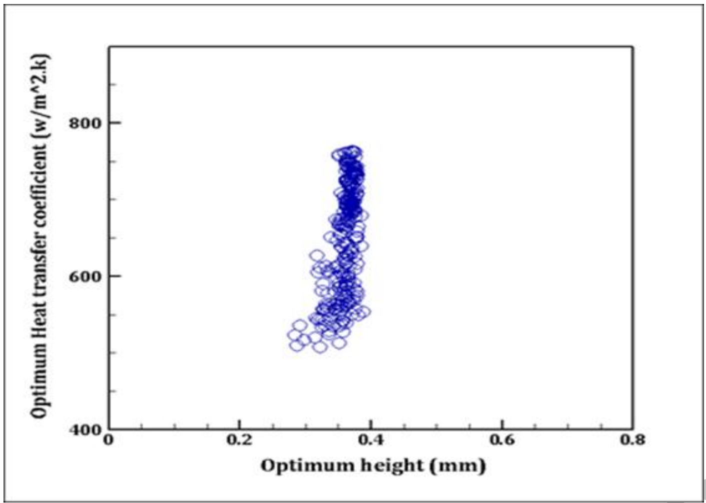


Figure 6- The amount of optimum heat transfer coefficient relative to the optimum micro-fiber height

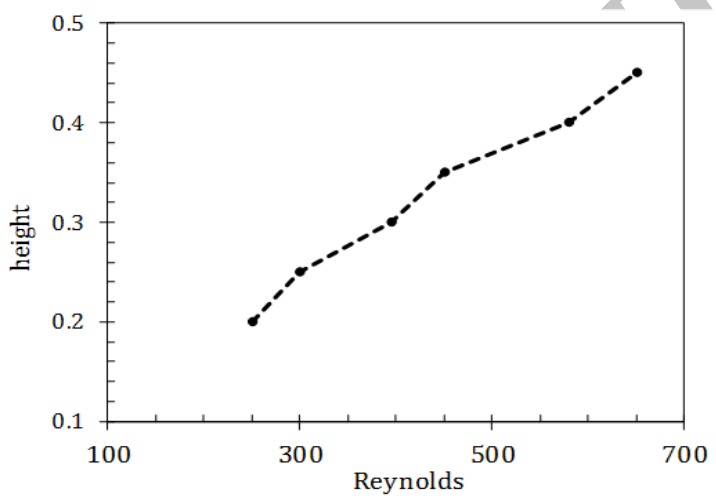


Figure 7- Altitude relative to Reynolds number

We also show the optimal variation of the heat transfer coefficient relative to the pressure drop (Fig. 8).

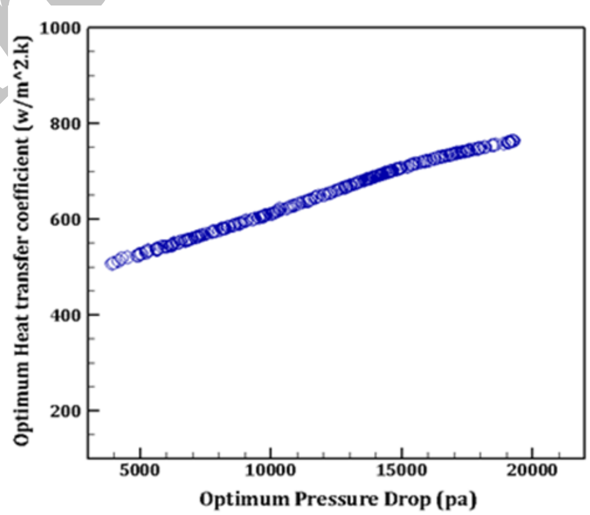


Figure 8- Optimal heat transfer coefficient on optimal pressure drop

Figure 9 shows the changes in the optimal heat transfer coefficient with respect to Reynolds number. As the results clearly show, the heat transfer coefficient is increasing and eventually shows that the applied changes, such as the spiral angle, the number of micro-fins and fin height are optimal.

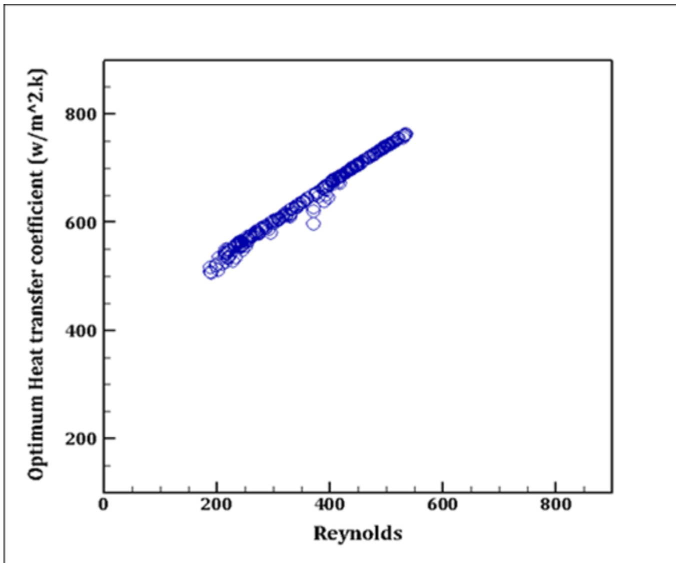


Figure 9- Optimal heat transfer coefficient relative to Reynolds

Based on the results of this research, the following recommended correlations for the desired micro-fin height, spiral angle, Reynolds number and optimal inner diameter are shown in the table in Figure (1).

7. Conclusion

The genetic optimization algorithm optimizes the heat exchangers with micro-fins, in order to maximize heat transfer, while minimizing the drop in pressure, in two-pipe heat exchangers with a smooth flow. The variables considered in this study are the number of micro-fins, the height of the fin, the vertex angle, and the spiral angle. In this study, for a standard diameter Diameter rutile and for different Reynolds number, the amount of optimal heat transfer coefficient is also introduced. Also, based on this study, proper height and optimal Reynolds amount are presented in the number of fins and different spiral angles, which include:

1. The optimal micro-fiber height also increases with the increase in the number of Reynolds, and as the Finn's height increases, the heat transfer coefficient also increases.

2. The optimum head angle is reduced by increasing the amount of Reynolds for the entire inner tube diameter.

3. The optimum height of the micro-fin is between 0.2 - 0.35 hp. A higher height reduces the pressure and does not have any effect on the heat transfer, the number of optimum fins between 50 and 60 is considered to be in the inner diameter of the pipe have taken.

4. We show the optimal heat transfer coefficient in relation to the Reynolds value, which has a parallel relationship to each other.

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