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Numerical study of heat transfer in a finned double pipe heat exchanger

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Abstract. In the present study the performance of a concentric tube heat exchanger is analyzed with passive heat transfer technique. The performance of the heat transfer process in a given heat exchanger is determined for three different longitudinal fin profiles, rectangular, triangular and parabolic. Numerical analysis was carried out in a parallel flow double pipe heat exchanger for the above profiles for varied mass flow conditions both in the inner and outer tube. Base width and height of the fin were kept constant for all the three types. Simulated results indicated an enhancement in the heat transfer rate for a finned tube compared to the unfinned one. Among the different configurations, fin with rectangular profile showed marginal improvement over triangular and concave parabolic profiles in terms of heat transfer characteristics. For a constant value of $mc_c = 0.02kg/s$ and varying mc_h, rectangular finned tubes showed an average improvement of 6.1% over the triangular and 9.2% over parabolic finned tube. Similarly For a constant value of $mc_h = 0.02kg/s$ and varying mc_c, it showed an improvement by 2 and 5% over the triangular tube and parabolic finned tube respectively Fins with concave parabolic profiles exhibited minimum pressure drop and has reduced by 38% and 65% compared to the triangular and rectangular finned tube.

Keywords: double pipe heat exchanger, triangular fins, parabolic fins, fin effectiveness, pressure drop

1 Introduction

Heat transfer enhancement in a heat exchanger is getting industrial importance because it gives the opportunity to reduce the heat transfer area for the heat exchanger. Increase in the heat exchanger performance can help to make energy, material and cost saving related to a heat exchange process. Double pipe heat exchangers are the simplest devices in which heat is transferred from the hot fluid to the cold fluid through a separating cylindrical wall. They are primarily adapted to high temperature and high pressure applications due to their small diameters. They are fairly cheap, but the amount of space they occupy is relatively high compared to the other types. Hence for the given design and length of the heat exchanger heat transfer enhancement in a double pipe heat exchanger is possibly achieved by several methods. These techniques are divided into active and passive techniques. Active methods involve some external input for the enhancement of heat transfer like induced vibrations, injection and suction of fluids and jet impingement etc. Another method is the passive method without the stimulation by external power such as surface coating, surface roughness and extended surfaces. Chen et al.[4] used dimples as the heat transfer modification on the inner tube. Bhuiya et al.[3], Eiamsan et al.[5] and Liao et al.[12] used circular tube equipped with perforated twisted tape inserts with different configurations to enhance the heat transfer through the tube. In order to intensify the heat transfer from the heat exchanger surface to fluid, it is possible to increase convection coefficient (by growing the fluid velocity), widen temperature difference between surface and fluid or increase the surface area across which convection occurs. Extended surfaces, in the form of longitudinal or radial fins are common applications where the need to enhance the heat transfer between a surface and an adjacent fluid exists.Several researchers used extended surfaces for the enhancement in the heat transfer[2, 9, 11, 15].

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Masliyah et al.[13] studied heat transfer characteristics for a laminar forced convection fully developed flow in an internally triangular finned circular tube with axially uniform heat flux using a finite element method. For a given fin geometry, the Nusselt number based on inside tube diameter was higher than that for a smooth tube. Also, it was found that for maximum heat transfer there exists an optimum fin number for a given fin configuration. Agarwal et al.[1] studied Laminar flow and heat transfer magnitudes in a finned tube annulus. Pressure drop and heat transfer characteristics of the fins are obtained in the periodically fully developed region by varying geometric and flow parameters. Geometric parameters are annulus radius ratio (0.3 to 0.5), fin height/annular gap (0.33 to 0.67) and fin spacing/annular gap (2 to 5). Flow parameters are Reynolds number (100 to 1000) and Prandtl number (1 to 5). Comparisons are made with a plain tube annulus having the same length, heat transfer surface area, volume flow rate, and Reynolds number. They observed that at Prandtl numbers less than 2, the use of fins may not be justified because the increase in pressure drop is more pronounced than the increase in heat transfer. At a Reynolds number of 1000 and A Prandtl number of 5, the heat transfer increases by a factor of 3.1, while the pressure drop increases by a factor of 2.3.

mc_h	:	Mass flow rate of the hot fluid(kg/s).
mc_c	:	Mass flow rate of the cold fluid (kg/s).
ε	:	Fin effectiveness.
$h_{annulus}$	:	Effective heat transfer coefficient in the annulus side.
h_{normal}	:	Heat transfer coefficient in the annulus side without the fins.

Soliman et al.[17] studied steady, laminar, forced convection heat transfer in the thermal entrance region of internally finned tubes for the case of fully developed hydrodynamics. Results were presented for 16 geometries including the local Nusselt number and developing length corresponding to each boundary condition. These results indicate that internal finning influences the thermal development in a complicated way, which makes it inappropriate to extend the smooth tube results to internally finned tubes on a hydraulic diameter basis. Totala et al.[19] conducted experiments in a double pipe heat exchanger by providing threads in the inner pipe. They observed that Nusselt number, heat transfer coefficient were increased for the threaded pipe. But the pumping power required also increased compared to the plain tube. Khannan et al.[10] studied the heat transfer through a double pipe heat exchanger with annular fins. Three different configurations annular ring, spiral rod and rectangular projection were considered on the outside surface of the outer tube. Experiments were done with varied mass flow rates. It was observed that heat transfer rate was increased for a finned tube. Fin with annular ring showed better performance than other methods. Nagarani et al.[16] used circular and elliptical annular fins as a heat enhancement devise in a double pipe heat exchanger. It was observed that heat transfer rate was higher in elliptical fins than circular fins. Heat transfer coefficient depends on the fin spacing, flow condition and fluid properties. Fin efficiency was higher for elliptical fins. Mir et al.[14] studied Numerical simulation of the steady, laminar, forced convection heat transfer in the finned annulus for the case of fully developed incompressible flow corresponding to thermal boundary condition of uniform heat input per unit axial length with peripherally uniform temperature at any cross section. Various heat transfer and fluid flow characteristics were investigated for a range of values of the ratio of radii of inner and outer pipes, fin height and number of fins. The results calculated are in good comparison with the correlation results with considerable gain in the computational time. Syed et al.[18] numerically simulated the laminar convection flow in the fully developed region of finned double pipe subjected to constant heat flux boundary conditions. They found a significant enhancement in heat transfer rate and also nusselt number. For small number of fins fin geometries like fin height, ratio of radii, and half fin angle were found to be less influential. Iqbal et al.[8] investigated optimal configuration of finned annulus with parabolic, triangular and trapezoidal fins using finite element methods and genetic algorithms. They concluded that no single fin shape is best in all situations and for all criteria. Zhang et al.[20] studied heat transfer enhancement for shell side of a double-pipe heat exchanger with helical fins and pin fins, the three-dimensional velocity components for the shell side with and without pin fins were measured experimentally by using laser Doppler anemometer (LDA) under cylindrical coordinate system and the fluid flow characteristics. The results showed that, for the shell side only with helical fins at large pitch, there was a pair of vortex near the upper and lower edge of the rectangular cross section the weakest secondary flow occurred at the center. By pin fins being installed, the three-dimensional velocity components in the helical channel were strongly changed. Patel et al.[6] simulated the industrial experimental results of a double pipe heat exchanger using ANSYS14-CFX. Results indicated heat transfer, pressure drop, pumping power increased with mass flow rate whereas friction factor decreased.

Optimization aims at improving the performance with lowering the fin mass by means of changing the shape of the fin. The fin shape modification influences not only the mass of the heat exchanger, but also affects the flow direction that causes the temperature changes on the fin contact surfaces. For heat exchangers, built with many fins and designed for real industry, it is important to pay attention to and calculate the heat transfer considering the fluid flow and flow paths. The resistance of the body results in a pressure drop. The fin and tube surface orientation also modifies the resistance of the body that results in a pressure drop. Hence a comparison study of heat transfer characteristics using different configurations of fins is very essential. Literature survey reveals that most of the analysis was done by considering constant heat flux or constant wall temperature boundary condition. Literature regarding the numerical study of enhancement in heat transfer characteristics using different configurations of internal longitudinal fins for a double pipe heat exchanger with conjugate heat transfer is still scare. Hence the present work aimed at comparison of heat transfer characteristics using different fin profiles for a double pipe heat exchanger under various operating conditions to evolve with the best possible configuration. Three different configurations namely rectangular, triangular and concave parabolic were selected. Base width,height and number of the fins were kept identical for the sake of comparison. Numerical simulation was done using commercial CFD package[7]. Heat transfer characteristics like temperature variation, heat transfer rate, and heat transfer coefficient and fin effectiveness for the above said models were compared and are presented. The paper has been organized into 5 sections. Section 1 deals with introduction, Section 2 describes the experimental set up and experimentation procedure. Section 3 describes the numerical scheme followed for the double pipe heat exchanger for various fined configurations. Section 4 discusses the results and in Section 5 conclusions were drawn.

2 Experimental set up and experimentation

Fig .1 shows the experimental set up of the concentric tube double pipe heat exchanger. It consists of inner tube made of copper where in hot water flows from a geyser attached to it. Cold water flows in the annulus which can be admitted at any one of the ends enabling the heat exchanger to run as a parallel or as a counter flow exchanger. This can be done by operating the valves provided. Specifications of the heat exchanger are mentioned in Tab. 1. Temperatures of the fluid can be measured using thermocouples with digital display. Flow rates of hot and cold water can be measured by rotometers connected to the pipes. Outer tube is provided with insulation to minimize the heat losses to the surroundings. The inlet temperature of the hot fluid was maintained at $332K$ and the cold fluid at $303K$. Experiments were conducted for parallel flow arrangement at various mass flow rates of hot water (mc_h) ranging from $0.02kg/s$ to $0.1kg/s$ with an increment of $0.02kg/s$ at each time keeping the mass flow rate of cold water (mc_c) through the annulus constant at $0.02kg/s$. Range of the Reynolds's number in the present study was 3000 to 20000. Outlet temperatures of the hot water and cold water were noted each time. Similarly the experiments were repeated by changing the cold water flow rates keeping the hot water flow rate constant at $0.02kg/s$. The results obtained were compared with results obtained from CFD model.

Table 1. Specifications of the double pipe heat exchanger

S.No	Specifications	Dimension (mm)
1	Inner diameter	9.5
2	Thickness of the inner tube	1.5
3	Inner diameter of the outer tube	28.5
4	Thickness of the outer tube	2
5	Length of the heat exchanger	1500

Fig. 1. Photograph of the experimental test rig

3 Numerical scheme

3.1 Double pipe heat exchanger without fins

The geometric model of the double pipe heat exchanger was constructed using work bench in ANSYS 14 environment. The complete domain consists of 157254 Elements. Grid independence test was performed to check the validity of the quality of the mesh on the solution as shown in the Fig. 2. Further refinement did not change the result by more than 0.9% which was taken as the appropriate mesh quality for computation. Inlet temperature and mass flow rate of the cold and hot fluid was specified. At the outlet, a pressure outlet boundary is enforced. Pressure velocity coupling was resolved using SIMPLEC algorithm with a skewness correction factor of 1. For pressure linear discretization was used. For momentum turbulent kinetic energy and turbulent dissipation rate with power law scheme was used. For the energy equation second order upwind was used. Conservation equations were solved for the control volume to yield the velocity and temperature fields for the water flow in the heat exchanger. Convergence was affected when residuals fell below 10^{-3} in the computational domain. A sample convergence plot is shown in the Fig. 3 Simulations were done by changing the mass flow rates of hot fluid as well as cold fluid as discussed in the previous section.A comparison of numerically simulated and experimental outlet temperatures of the cold water is as shown in Tab. 2. It reveals that the average error for constant mc_c and varying mc_h is 0.157% and for constant mc_h varying mc_c it is 0.183% which is within the acceptable limits. Hence the use of CFD modeling for studying the heat transfer characteristics in a double pipe heat exchanger can be employed with confidence.

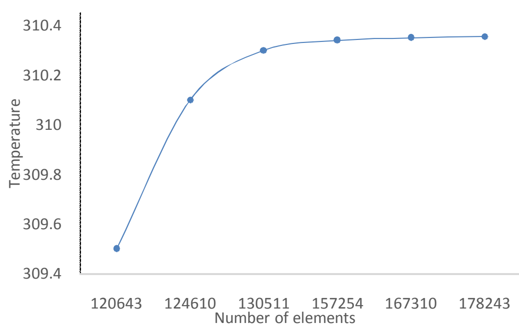

Fig. 2. Variation of cold water outlet temperature with number of elements

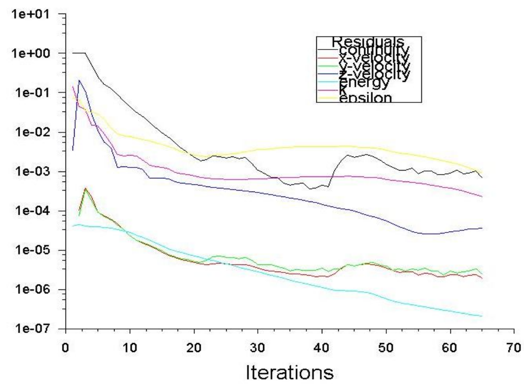

Fig. 3. Sample of Residual convergence plot for the double pipe heat exchanger

Table 2. Comparison of experimental and CFD simulated results for unfinned the double pipe heat exchanger

S.No	Experimental condition	Mass flow rate (kg/s)	Experimental measur_ ed cold water outlet temperature (K)	CFD simulated cold water out_ let temperature (K)	Relative error (%)
1	mc_c constant at 0.02 kg/s and mc_h varying	0.02	309.8	310.34	0.175
2		0.04	311.05	311.5	0.161
3		0.06	311.4	311.97	0.183
4		0.08	312.6	312.22	0.123
5		0.1	312.2	312.45	0.144
6	mc_h constant at 0.02 kg/s and mc_c varying	0.02	309.8	310.34	0.175
7		0.04	307.5	308	0.163
8		0.06	306.2	306.92	0.236
9		0.08	305.6	306.28	0.222
10		0.1	305.4	305.83	0.140

3.2 Rectangular finned double pipe heat exchanger

A double pipe heat exchanger with rectangular longitudinal fins was modeled in the Ansys work bench environment. Base width of the fin was $2mm$ ($18\ degrees$) kept constant throughout the study. Simulation was done using different fin heights $3mm$, $6mm$ and $7mm$. 12 fins were placed circumferentially around the thickness of the inner tube which remained constant throughout the study. Simulation was carried out for various mass flow rates of cold and hot water. For $mc_h = 0.1kg/s$ and $mc_c = 0.02kg/s$ temperature distribution at the outlet for both unfinned and finned (height $= 6mm$) heat exchanger is shown in Fig .4 which clearly indicates the increase in the average temperature in the annulus for a finned tube. Annulus fluid temperature along the length of heat exchanger was plotted for different fin heights and is as shown in Fig .5. It can be observed that the temperature rise for the finned heat exchanger is more than that of the unfinned type. As the fin height increases the average water temperature increased. But on comparing the temperature rise for different fin heights, as the height increases, rise in average temperature was found to be lower. Also the curve shows that the temperature rise along the length of exchanger is gradual and not steep. Fig .6 represent the temperature plot along the vertical length at the center of the fin for $3mm$, $6mm$ and $7mm$ height fins at a midsection of the heat exchanger. Fin tip temperature for $3mm$, $6mm$ and $7mm$ height fins are 326.84, 325 and $323.95K$ respectively. It indicates that there was a decrease of temperature by $1.84K$ when the height was increased from $3mm$ to $6mm$ and a decrease of $1.05K$ when the height was increased from $6mm$ to $7mm$ which indicates the reduced fin performance on increasing the fin height. Simultaneously increasing the fin height increases the weight of the finned assembly. Hence an optimum fin height of $6mm$ was chosen for further comparison between different profiles for fins.

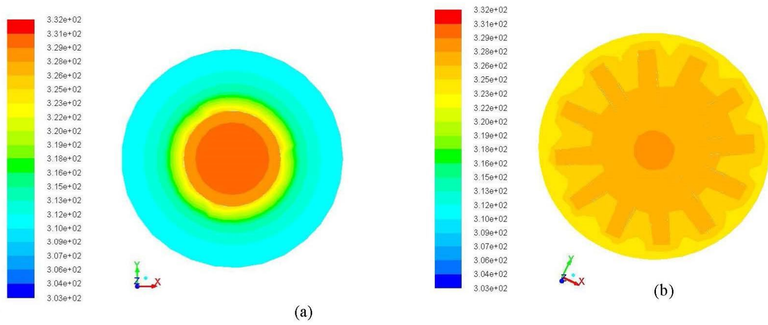

Fig. 4. Temperature distribution at the outlet surface for (a) unfinned (b) rectangular finned double pipe heat exchanger ($6mm$) for $mc_h = 0.1$ and $mc_c = 0.02kg/s$

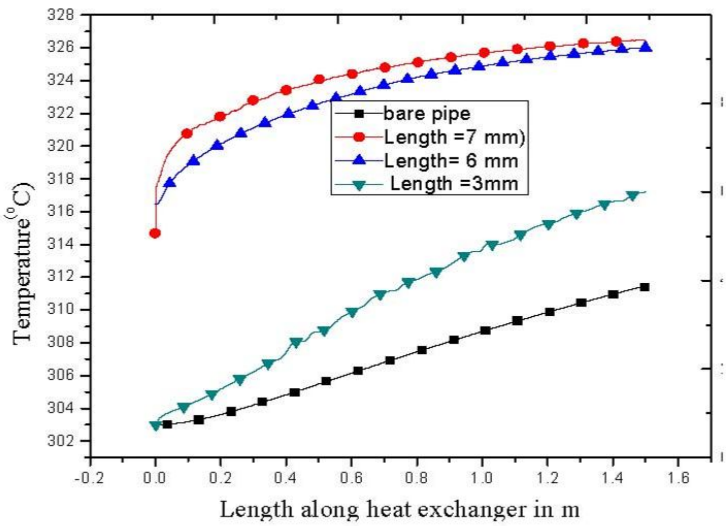

Fig. 5. Variation of cold water temperature along the length of exchanger for different heights of rectangular fins for $mc_h = 0.1$ and $mc_c = 0.02kg/s$

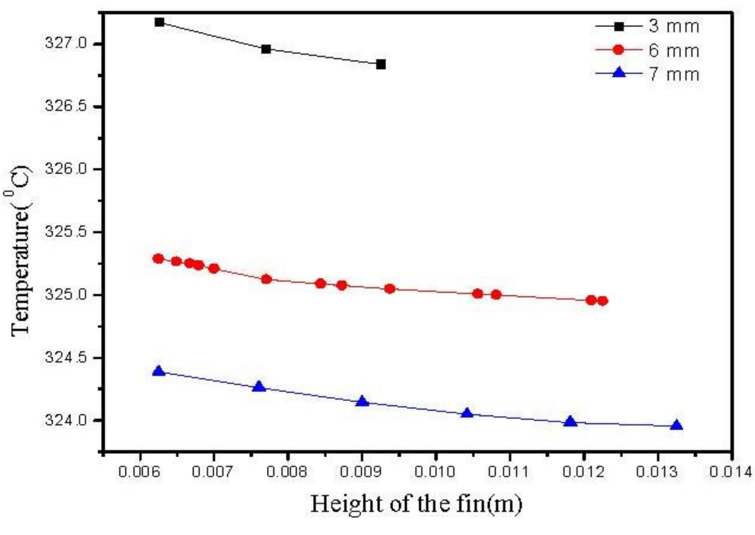

Fig. 6. Variation of temperature along the height of fin at the mid-section of the heat exchanger $mc_h = 0.1$ and $mc_c = 0.02kg/s$

4 Results and discussions

Geometrical modeling of the heat exchanger was constructed by changing the fin configuration to triangular and concave parabolic profiles keeping the fin base width, height and number of fins constant as discussed in Section 3. Fig. 7 shows the different fin configurations undertaken in the present study. Simulation was done for different mass flow rates of hot water through the inner tube for a constant cold water flow rate of $0.02kg/s$ in the annulus. Fig .8 indicates the temperature plot of all fin profiles with $mc_h = 0.1kg/s$ and $mc_c = 0.02kg/s$. It depicts that the cold water outlet temperature for the finned tube is higher than the bare tube. Fins promote boundary layer separation of the fluids and disturb the whole bulk flow field inside circular tubes. Separation and restarting of the boundary layers increases the heat transfer rate. It can be further noted that the bulk of the fluid layer is being disturbed the rise of temperature is also uneven along the length of the tube as shown in Fig. 8. Temperature for triangular and parabolic profile s is slightly higher than the rectangular at the entry of the heat exchanger, and as fluid progresses along the length of the heat exchanger temperature for the rectangular profile increases sharply than the other two types and at the outlet it is marginally higher than for triangular and parabolic type. The fin shape modification influences not only the mass of the heat exchanger, but also affects the flow direction that causes the temperature changes on the fin contact surfaces.

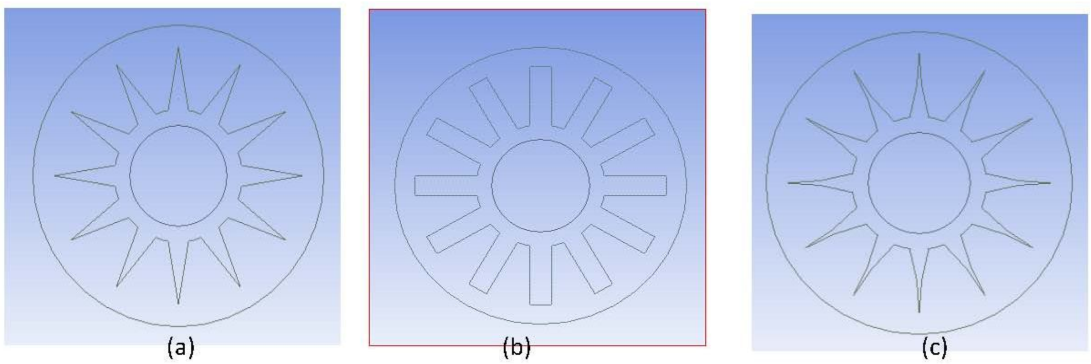

Fig. 7. Different types of finned configurations (a) triangular, (b) rectangular and (c) concave parabolic

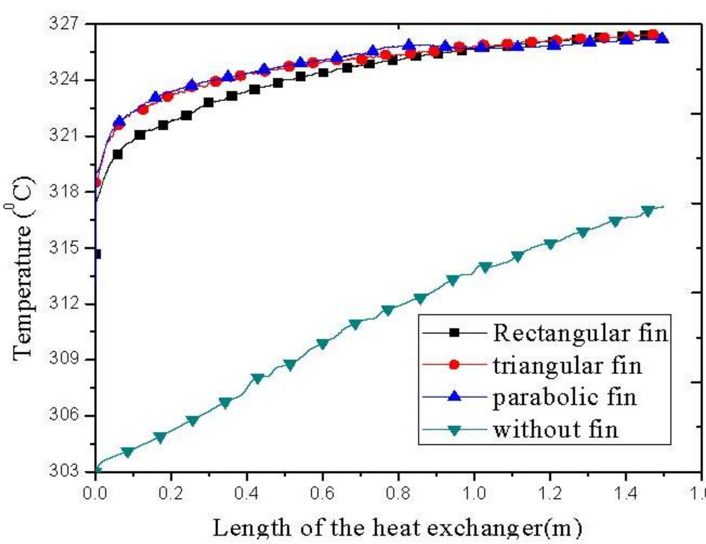

Fig. 8. Variation of temperature along the length of heat exchanger for different fin profiles at $mc_h = 0.1$ and $mc_c = 0.02kg/s$

Fig. 9 indicates the plot of cold water outlet temperature Vs mass flow rates of the hot water. The outlet temperature for the rectangular fin is 0.1% and 0.2% more that of triangular and parabolic shape for the highest mass flow rate of hot water. Fig. 10 shows temperature rise along the length for a particular value of $mc_h = 0.02kg/s$ and $mc_c = 0.1kg/s$. It can be revealed that rise in the temperature is higher for a finned tube than the unfinned one. In case of finned tube, the rise is not uniform and smooth as the entire flow field is being disturbed along the length. Rectangular finned tubes show a marginal improvement over the other types.

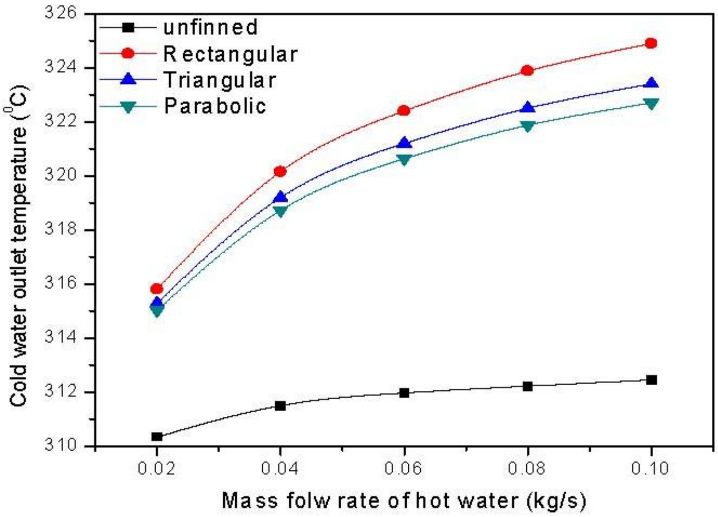

Fig. 9. Variation of cold water outlet temperature with mass flow rates of hot water

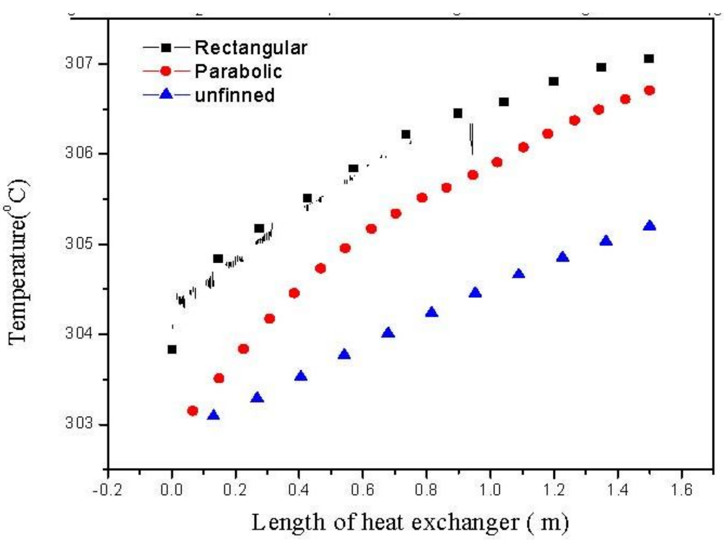

Fig. 10. Variation of temperature along the length of heat exchanger for different fin profiles at $mc_h = 0.1$ and $mc_c = 0.02kg/s$

Fig. 11 shows the variation of outlet temperature of cold water for varying mass flow rates of cold water with mass flow rate of hot water being fixed. It indicates as the mass flow rate increases the outlet temperature decreases for all profiles. This is because for a fixed value of mc_h heat energy dissipated from the inner tube remains constant and hence as mc_c is increased heat transfer rate is being increased due to the increased turbulence, but the retention time of the fluid in the exchanger will be reduced which decreases the cold water outlet temperature. Drop in outlet temperature or the rectangular profile is marginally lesser than the other two types showing its better performance.

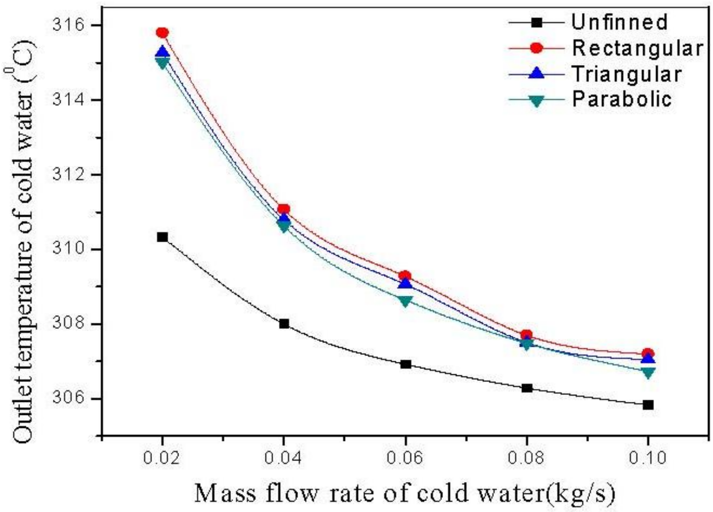

Fig. 11. Variation of cold water outlet temperature with varying mass flow rates of cold water

Tab. 3 indicates the heat transfer rate for different simulated conditions for the finned and unfinned tubes. It can be revealed that heat transfer rate increases with the increase in mass flow rate and finned tubes show higher value than the unfinned condition. For a constant value of $mc_c = 0.02kg/s$ and varying mc_h, rectangular finned tubes showed an average improvement of 6.1% over the triangular and 9.2% over parabolic finned tube. Similarly For a constant value of $mc_h = 0.02kg/s$ and varying mc_c, it showed an improvement by 2 and 5% over the triangular tube and parabolic finned tube respectively.

Table 3. Comparison of experimental and CFD simulated results for unfinned the double pipe heat exchanger

S.No	Condition	Mass flow rate (kg/s)	Unfinned tube (W)	rectangular finned tube (W)	triangular finned tube (W)	concave para_ bolic finned tube (W)
1	mc_c constant at 0.02 kg/s and mc_h varying	0.02	621	1069	1023	1000
2		0.04	718	1430	1348	1311
3		0.06	759	1620	1515	1463
4		0.08	784	1741	1623	1565
5		0.1	800	1828	1700	1619
6	mc_h constant at 0.02 kg/s and mc_c varying	0.02	621	1069	1023	1000
7		0.04	841	1372	1284	1270
8		0.06	992	1541	1487	1406
9		0.08	1102	1589	1507	1493
10		00.1	1187	1680	1664	1554

Fig. 12 shows the variation of annulus heat transfer coefficient for different varied mc_c keeping mc_h constant at $0.02kg/s$. It is observed that annulur heat transfer coefficient is found to be reduced for finned structure since effective heat transfer coefficient is related to the efficiency of the fin as $H_{annular} =$

$Fin_{efficiency} \times h_{normal}$. Further on comparing the values for different fin profiles it is observed that rectangular profiles the heat transfer coefficient is slightly higher than the others at all mass flow rate conditions.

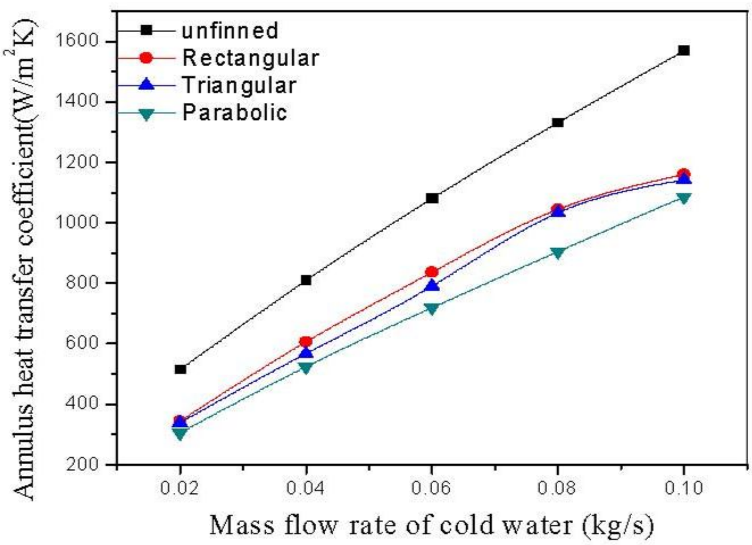

Fig. 12. Variation of annulus heat transfer coefficient with varying mass flow rates of cold water

Fig. 13 represents fin effectiveness for various types of fins under different conditions. Fin Effectiveness was calculated by the simple relation fin effectiveness, $\varepsilon = \frac{Q_{\text{withfin}}}{Q_{\text{withoutfin}}}$. For a constant value of mc_c of $0.02kg/s$, as mc_h was increased the effectiveness of the fins increased and rectangular fins showed highest effectiveness among the different types. Similarly for a constant mc_h and as mc_c are varied fin effectiveness decreases as seen in the graph.The rise in temperature for varying mc_c decreases for a finned tube compared with the unfinned one. Athigher mass flow rates the flow field is being disturbed severely when compared to the bare tube. Hence heat transfer capability of the fins decreases causing reduction in fin effectiveness. Hence in order to have a better performance by the fins for heating a liquid flowing in the annulus region, mass flow rate of liquid in the annulus should be minimum and mass flow rate of hot liquid flowing inside the inner tube should be maximum.

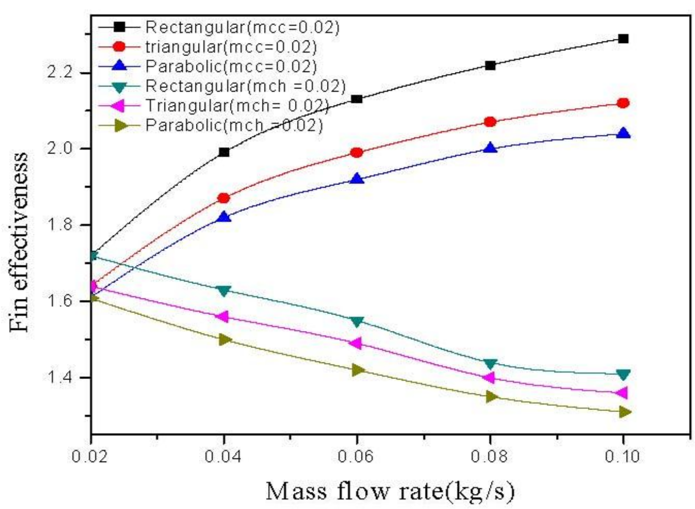

Fig. 13. Variation of fin effectiveness with varying mass flow rates of cold water and hot water

Fig. 14 represents the pressure drop variation for unfinned and the finned tube for various mass flow rates of cold fluid. It can be seen that pressure drop for unfinned tube is lesser than the finned tube. In case of a finned tube minimum pressure drop was observed in case of a fin with parabolic shape and highest for the fin with rectangular shape. The pressure drop for a concave parabolic shape has reduced by 38% and 65% compared to the triangular and rectangular finned tube.Higher pressure losses increase the pumping power. Since the pressure losses are more in case of rectangular finned tube the pumping power required will be more when compared to the triangular and parabolic finned tube. Marginal reduced performance of parabolic finned tubes is compensated by reduced weight and reduced pressure losses. Considering all these factors concave parabolic fins are preferred over the other types since required pumping power and weight of the finned structure will be reduced with sacrifice of a marginal benefit of heat transfer characteristics.

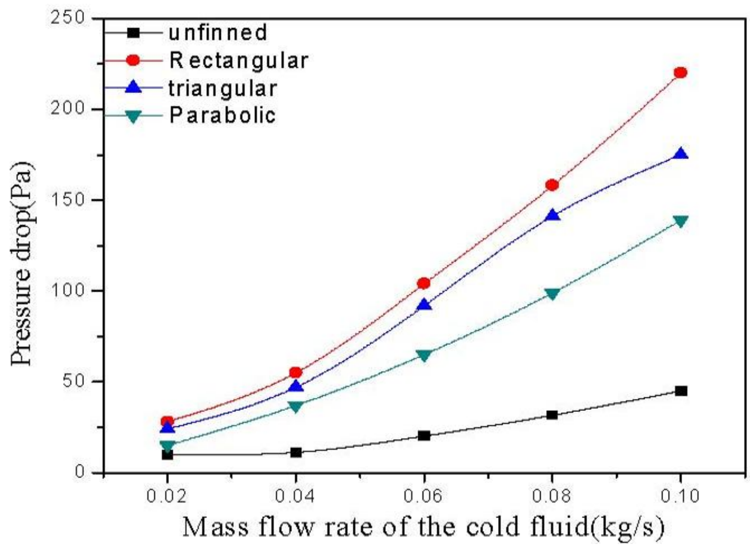

Fig. 14. Variation of pressure drop with varying mass flow rates of cold water

5 Conclusion

In the present study numerical simulation of finned double pipe heat exchanger is done with hot fluid flowing in the inner tube and cold fluid in the annulus. After validating the results for a bare heat exchanger with the experimental results Simulation was done using rectangular, triangular and concave parabolic finned configurations onthe outer body of inner tube. Results indicated finned configurations show an overall improvement in the thermal characteristics compared with unfinned one. For better Performance the mass flow rate of the cold fluid should be kept low where as that of the hot liquid should be high. Rectangular finned configuration show a marginal improvement over the other in terms of temperature rise, heat transfer rate and heat transfer coefficient. For a constant value of mc_c of $0.02kg/s$, as mc_h was increased, the effectiveness of the fins increased and rectangular fins showed highest effectiveness and is 21% and 11.5% higher than parabolic and triangular based fins.Parabolic fins showed minimum pressure drop for all mass flow rates. It has reduced by 38% and 65% compared to the triangular and rectangular finned tube. In addition to this even they provide the advantage of lesser material and hence reduced weight. Hence it can be concluded that parabolic finned configuration can be a better alternative compared to the triangular and rectangular because of reduced pressure drop and reduced weight of the finned assembly even though the thermal performance is being marginally reduced.

Recommendations for the future work: Numerical study of concentric double pipe heat exchanger can be extended by increasing the number of fins as well as by considering different fin profiles. Even the range of prandtl number can be varied by selecting different fluids flowing in the inner tube and in the annulus to study its influence on the heat transfer characteristics.

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