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Numerical Analysis of Fin Side Turbulent Flow for Round and Flat Tube Heat Exchangers

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Abstract: *Numerical three dimensional simulation of turbulent flow in round and flat tube fin heat exchangers having two rows of staggered arrangement has been carried out to investigate fluid flow and heat transfer characteristics using ANSYS Fluent 14® software. HYPERMESH10® Software has been used for the creation of models as well for meshing. The cases have been simulated for different fin side Reynolds number in turbulent regime to observe the effect of various parameters like fin pitch, tube pitch and fin temperature on Colburn j factor and Friction factor f for both round and flat tubes. Fin side flow has been simulated using various steady flow models in the software for same velocity range. As simulation using k-ε model resulted in close agreement with that of experimental in turbulent regime, it is considered for further analysis. The performance of round tubes is compared with that of flat tubes with same flow area and geometrical parameters. For both round and flat tube domains with all the geometrical configurations simulated in this work Colburn j factor varied inversely with the inlet air velocity. The heat transfer is more with the higher fin spacing for both round and flat tubes following the above said trend. On the other hand, the pressure drop across the tubes is more with the lesser fin spacing in contrast to the heat transfer. Due to lesser turbulent intensity in flat tubes, they exhibit slightly lesser Colburn j factor and considerably lesser pressure drop compared to round tubes. Although flat tubes exhibit slightly lesser Colburn j factor, due to larger exposed tube area increase in the air temperature in the fin side is comparable with that of round tubes. Higher fin temperatures result with lesser Colburn j factor and higher pressure drop across the tubes although the fin temperature affects the pressure drop to lesser extent.*

Keywords: *Colburn j factor, Extended Surface, Fins, Flat tube, Friction factor, Heat Exchanger, Numerical Analysis, Round Tube.*

I. Introduction

Extensive work has been carried out by many researchers on various configurations of heat exchangers both numerically as well as experimentally. Even though, literature relevant to the field is available in abundance, due to parameters to be observed are too many with varied configurations still the scope exists to research furthermore. In this work performance assessment of heat exchangers by varying the parameters associated has been undertaken for two rows by numerical simulation. This involves building a model of plain fin and tube heat exchangers using HYPERMESH® software for modeling and creation of suitable mesh, selection of solvers and numerical solution methods by using ANSYS FLUENT® software. The presented work is focused on fin side flow and heat transfer characteristics particularly on turbulent flow regime. Colburn J factor and Friction factor f are considered to best suggest the performance characteristics. In this work the effect of parameters such as fin pitch, tube pitch and fin temperature on performance of heat exchangers is studied. Flow in two tube axi-symmetric model is simulated for a range of inlet velocities using various steady flow models available with the used solver software. The simulated results are compared with the experimental results with same flow and geometrical configurations partly as validation of the numerical approach followed and to select most suited model for turbulent regime. Many researchers have worked on flat tubes with same perimeter as of round tubes with different tube arrangements in their comparative study. In the presented work, the performance of plain fin flat tube heat exchanger is compared with that of round tube by keeping the flow area and the pitches as same in respective cases.

The work also includes the study of available literature in related areas as per which many experimental works has been carried out on plate fin and tube heat exchangers. Wang et.al conducted experimental works on plate fin heat exchangers with different geometrical parameters including number of tube rows, fin spacing and fin thickness to study their effect on heat transfer and friction characteristics [1]. Further, Wang along with K Y Chi gave an improved experimental data on plane fin and tube heat exchangers. Study stated that the heat transfer coefficients are strongly dependent on number of tube rows in case of laminar flow with decrease in fin pitch and increase in tube diameter leads to increase of pressure drops [2]. In parallel, Jang et.al reported that heat transfer practically remains independent of number of tube rows for more than four rows and showed higher Colburn and friction factor for staggered tube arrangement [3]. Yonghan Kim and Yongchan Kim conducted an experiment and found that for the one row heat exchanger fin pitches had negligible effect on heat transfer and increasing fin pitches by increasing the number of tube rows lead to the increase in heat transfer. For the staggered tube alignment with more than 4 tube rows the heat transfer coefficient is independent of number of tube rows [4]. Recently, Gurjeet Singh and Gulshan Sachdeva, conducted CFD simulations for both round and flat tube heat exchangers with same perimeter of the tubes and concluded that in turbulent region the friction factor for round tube was 40 to 45% more than the flat tubes and same Colburn j factor was achieved [5]. Experimental results from Wang et.al have been taken to validate numerically simulated results and to select most suitable model to solve the cases of fin side turbulent flow [1][2].

II. COMPUTATIONAL FLUID DYNAMICS

Computational fluid dynamics or CFD is the analysis of systems involving fluid flow heat transfer associated phenomena such as chemical reactions by means of computer based simulation.

There are three distinct streams of numerical solution techniques which are used by the Solver;

- Finite difference methods
- Finite volume methods
- Spectral methods

Numerical methods that form the basis of the solver perform the following steps.

- Approximation of the unknown flow variables by means of simple functions.
- Discritisation by substitution of the approximations into the governing flow equations and subsequent mathematical manipulations
- Solution of the algebraic equations.

The numerical algorithm consists of the following steps

- Formal integration of the governing equations of fluid flow over all the control volumes of the solution domain.
- Discretisation involves the substitution of a variety of finite difference type approximations for the terms in the integrated equation representing flow processes such as convection, diffusion and sources. This converts the integral equations into a system of algebraic equations.
- Solution of the algebraic equations by an iterative method.

Three mathematical concepts are useful in determining the success: Convergence, consistency and stability

- Convergence is the property of a numerical method to produce a solution which approaches the exact solution as the grid spacing, control volume size or element size is reduced to zero.
- Consistent numerical schemes produce systems of algebraic equations which can be demonstrated to be equivalent to the original governing equation as the grid spacing tends to zero.
- Stability is associated with damping of errors as the numerical method proceeds. If a technique is not stable even round off errors in the initial data can cause wild oscillations or divergence [6].

Governing equations used are:

Continuity equation: $\frac{\partial(\rho u_i)}{\partial x_i} = 0$

Momentum equation: $\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}(\mu \frac{\partial u_j}{\partial x_i}) - \frac{\partial p}{\partial x_j}$

Energy equation: $\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}(\frac{k}{C_p} \frac{\partial u_j}{\partial x_i})$

III. COMPUTATIONAL DOMAIN AND BOUNDARY CONDITIONS

The commercial software HYPERMESH is used to create and mesh the computational models. The dimensions of the basic domain are taken from the experimental works [1]. The computational models for round and flat tube with plain fins domains considered for simulation are shown in figure 1 and 2.

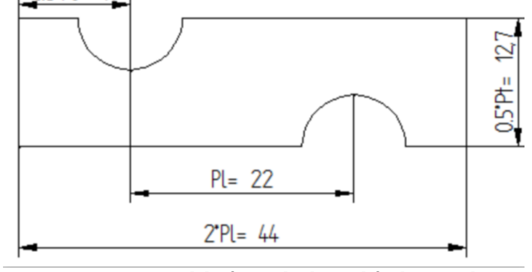


Figure1: Geometric model of round tube and fin heat exchangers

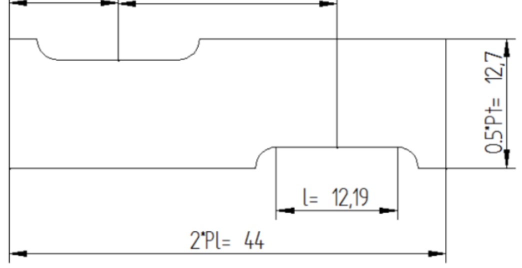


Figure2: Geometric model of flat tube and fin heat exchangers

The geometric details of round tube and fin configuration is listed in table 1 and flat tube and fin heat exchangers is shown in table 2

Geometrical Parameters	Symbol	Dimensions
Fin thickness	T	0.130 mm
Fin pitch	F_P	2.240 mm
Fin collar outside dia	D_C	10.23 mm
Transverse pitch	P_t	25.40 mm
Longitudinal pitch	P_l	22.00 mm
Tube wall thickness	Δ	0.336 mm

Table 1: geometric details of round tube and fin configuration

Geometrical Parameters	Symbol	Dimensions
Fin thickness	T	0.130 mm
Fin pitch	F_P	2.240 mm
Tube outside dia	d	4.14 mm
Transverse pitch	P_t	25.40 mm
Longitudinal pitch	P_l	22.00 mm
Tube wall thickness	Δ	0.336 mm

Table 2: geometric details of flat tube and fin heat exchangers

As it is difficult to conduct CFD simulations for entire heat exchanger with multiple rows and columns, a symmetrical model about both the axes of one channel of air between two fins with the air flowing by two tubes is considered as a geometrical model. Computational models for round and flat tube heat exchangers are shown in figure 3 and 4 respectively. Dimensions of the flat tube have been computed with l/d ratio of 4 maintaining same tube side flow area as of round tube. The gap between the two fins is considered as flow area for air and the model consists of structured hexahedral mesh throughout and the areas around the tubes are densely meshed. Grid independence test has been conducted and results found to be not much sensitive with the further refinement after the number of elements 61750 for round tube domain and 73500 elements for flat tube domain.

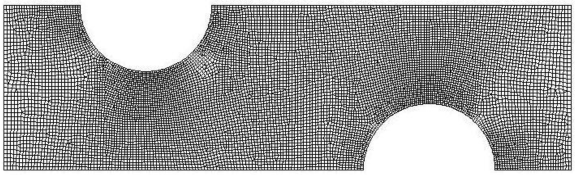


Figure 3: Meshed model for round tube domain

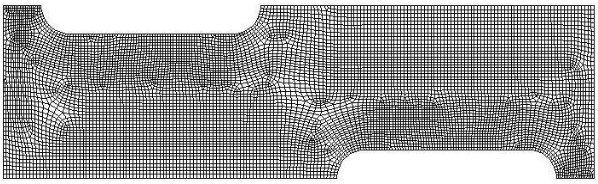


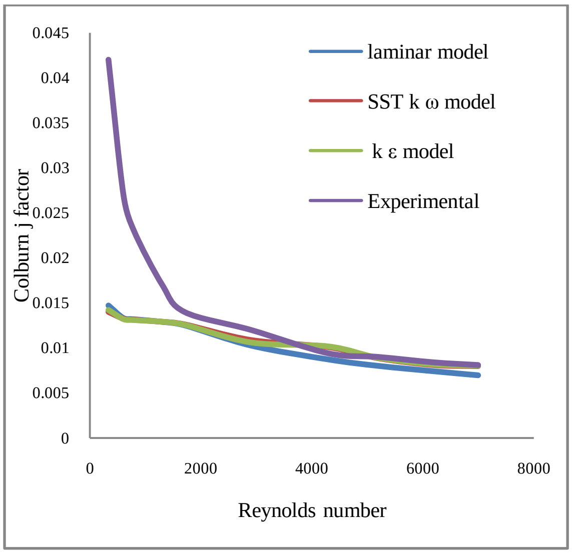
Figure 4: Meshed model for flat tube domain

The fluid is assumed to be incompressible with constant properties and the flow is turbulent and in steady state. All numerical simulations are carried out using a finite-volume method. The boundaries of the computational domain consist of inlet and outlet, symmetry planes and solid walls. Boundary conditions for the domain are applied as tabulated in table 3. A steady state unidirectional uniform velocity at inlet plane and uniform wall temperature of 55˚C are applied to simplify the computations. A constant temperature of 5˚C is set at the flow inlet to meet the experimental conditions. At the outlet, stream wise gradient (Neumann boundary conditions) for all the variables are set to zero. No-slip boundary condition is used at the fins and the tube surfaces.

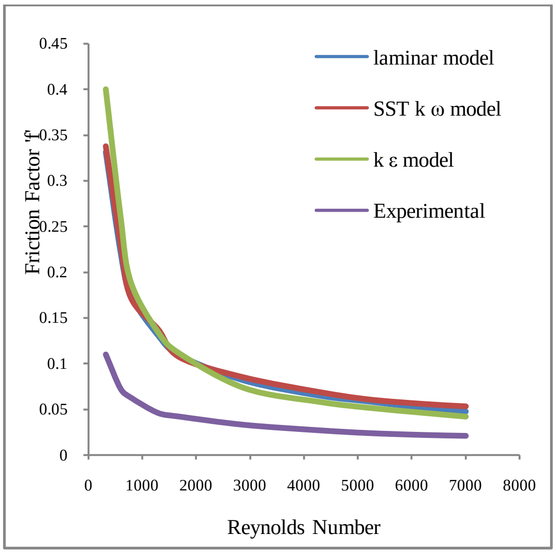
Tube surfaces	Dirichlet boundary condition Air velocity u = v = w =0	T= T_w=60^0C=333K
Fins	Dirichlet boundary condition Air velocity u=v=w=0	T= T_{fw}=60^0C=333K
Inlet	Dirichlet boundary condition Uniform velocity ‘u’	u=u_{in}, u_{in} ranging from 3.7 m/s to 6.2 m/s. T=5^0C=278K
Outlet	Neumann boundary conditions that is zero gradients of pressure temperatures and velocities	-
Side planes	Symmetry Conditions	$\frac{\partial u}{\partial y}$=0, v =0, $\frac{\partial w}{\partial y}$=0, $\frac{\partial T}{\partial y}$=0

Table 3: Boundary Conditions

For the validation of the numerical approach followed and to select most suitable model for simulation, geometry of round tube domain is maintained same as referred experimental work. The simulation is carried out for the velocities ranging from Reynolds number 330 to 7000 in the mentioned domain is done with all the steady state flow models and compared with the experimental results. As the present work is focused on the turbulent regime, from the graphs 1(a) and 1(b) it is evident that the k-ε model computed most proximate results to the experimental work [1]. Further, same validated numerical approach and model selected is used to simulate rest of the cases in the work.



Graph 1(a): Reynolds Number v/s Colburn j factor



Graph 1(b): Reynolds Number v/s Friction factor f

IV. Results and Discussions

In the present work, round and flat tube domains are simulated for fin side turbulent flow using k-ε model for different fin pitches and fin temperatures. Figure 5(a), 5(b) and 5(c) are simulated contours of pressure, velocity and temperature for round tube domain with fin pitch of 2.24mm for inlet velocity 5.4m/s. Figure 6(a), 6(b) and 6(c) are simulated contours of pressure, velocity and temperature for flat tube domain with fin pitch of 2.24mm for inlet velocity 5.4m/s. As the study is more focused on Colburn j factor and friction factor, above said contours are not discussed in detail.

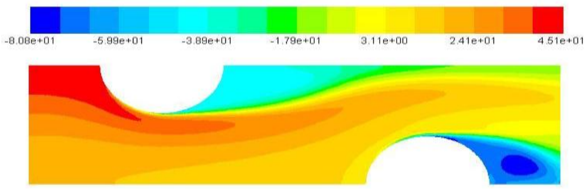


Fig 5(a): Pressure Contour for Round Tube

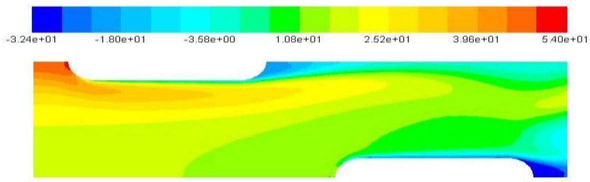


Fig 6(a): Pressure Contour for Flat Tube

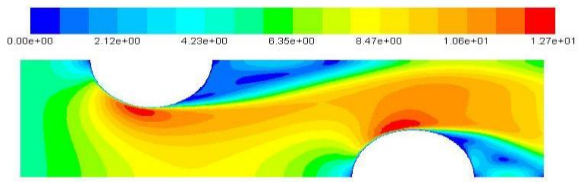


Fig 5(b): Velocity Contour for Round Tube

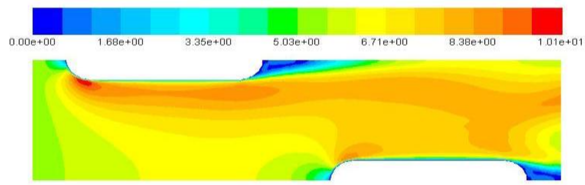


Fig 6(b): Velocity Contour for Flat Tube

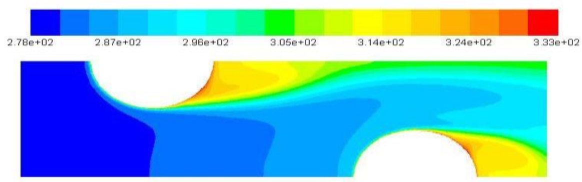


Fig 5(c): Temperature Contour for Round Tube

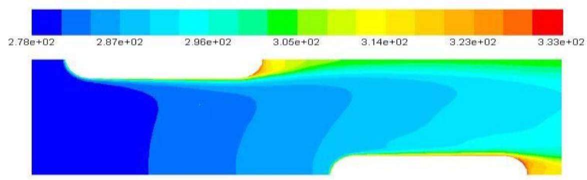
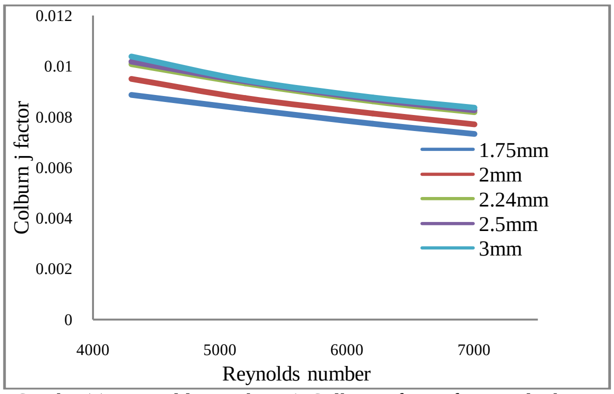
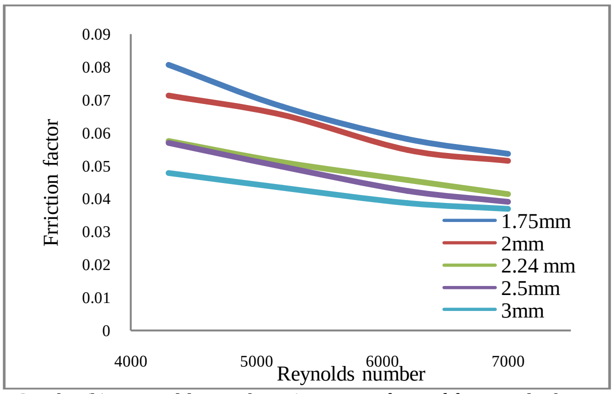


Fig 6(c): Temperature Contour for Flat Tube

The effect of the fin pitch on Colburn j factor and friction factor “f” for different Reynolds number in the turbulence regime is depicted in the graphs 2(a) and 2(b) respectively for the round tube domain. It is evident from the plot that heat transfer varies inversely with the Reynolds number. For the lower Reynolds numbers the air spends more time in the flow area and absorbs the more heat from the fins and tubes. It can also be observed that heat transfer is more with the higher fin spacing. On the other hand, the pressure drop across the tubes is more with the lesser fin spacing in contrast to the heat transfer.

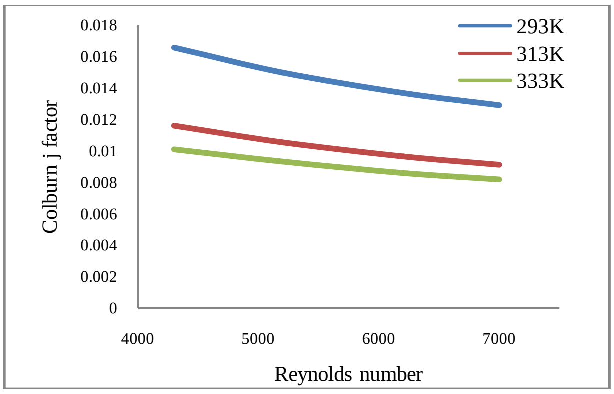


Graph 2(a): Reynolds Number v/s Colburn j factor for round tube

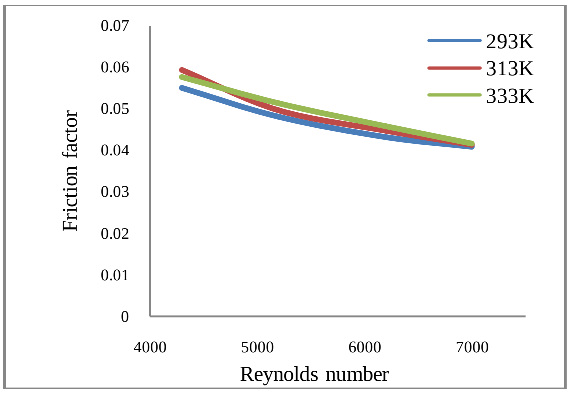


Graph 2(b): Reynolds Number v/s Friction factor f for round tube

The effect of the fin temperature on Colburn j factor and friction factor "f" for different Reynolds number in the turbulence regime is depicted in the graphs 3(a) and 3(b) respectively for the round tube domain. It is evident from the plot that heat transfer varies inversely with the fin temperature and also the variation in heat transfer was considerably larger with initial increase in fin temperature and varied much lesser with later increase in fin temperature. Lower fin temperatures resulted in higher Colburn j factor. On the other hand, it is evident that the fin temperature does not affect the pressure drop across the tubes to a higher extent. Also, it can be seen that increase in fin temperature results in slight decrease in friction factor.

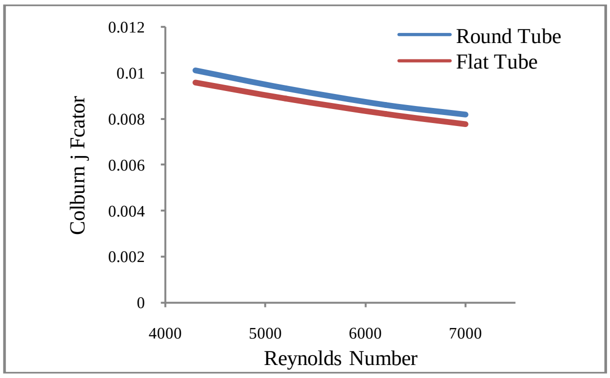


Graph 3(a): Reynolds Number v/s Colburn j factor for round tubes

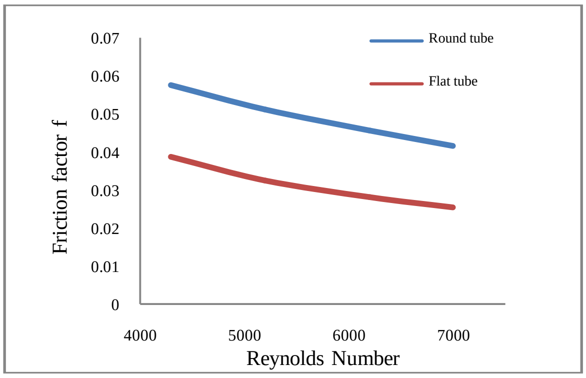


Graph 3(b): Reynolds Number v/s Friction factor f for round tubes

Simulation of fin side turbulent flow is also carried out on flat tube domains with same tube side flow area as that of round tube domain maintaining other geometrical configurations and range of inlet velocities as same. The graphs 4(a) and 4(b) shows the comparisons of variations of Colburn j and friction factors with Reynolds number for round and flat tubes. Similar to round tubes, the flat tubes showed higher heat transfer at lower inlet velocities and higher pressure drops at lower inlet velocities. From the comparisons, it is observed that round tubes exhibited slightly better Colburn j factor but the flat tubes exhibited larger reduction in pressure drop.

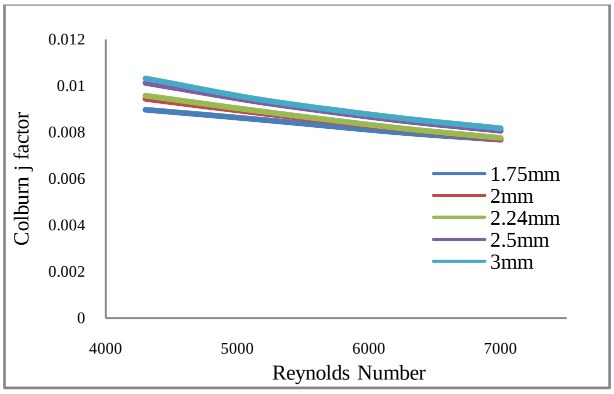


Graph 4(a): Reynolds Number v/s Colburn j factor

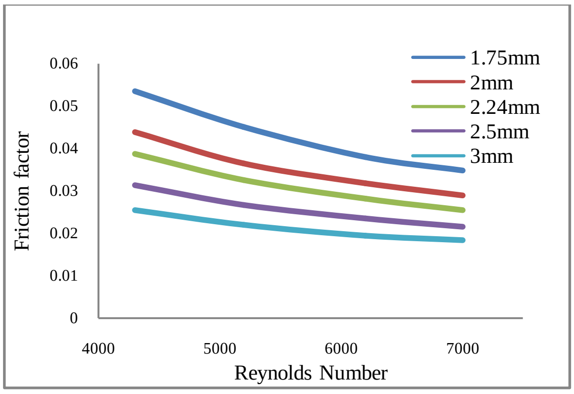


Graph 4(b): Reynolds Number v/s Friction factor f

Graphs 5(a) and 5(b) are plotted to show the variation of Colburn j factor and friction factor for flat tubes with Reynolds number for different fin spacing. Although the plot trends are similar to that of round tubes, flat tubes exhibited very slight variation in heat transfer with fins spacing as compared to that of round tubes. Also, they exhibited considerable reduction in pressure drop when compared to that of round tubes following a similar trend.

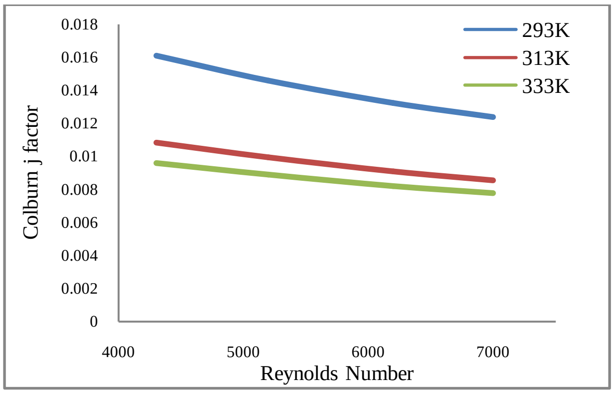


Graph 5(a): Reynolds Number v/s Colburn j factor for flat tubes

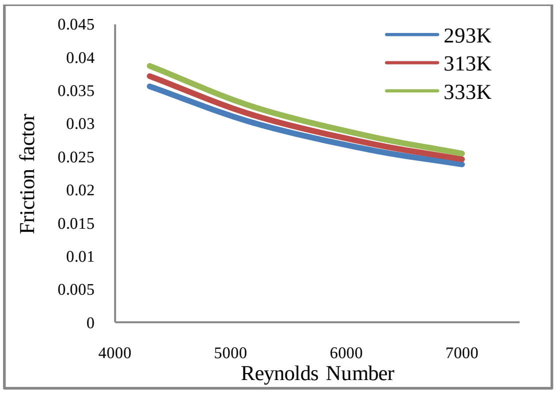


Graph 5(b): Reynolds Number v/s Friction factor f for flat tubes

The variations of Colburn j factor and friction factor for flat tube with Reynolds number at inlet for different fin temperatures are shown in graph 6(a) and 6(b), as per which variations followed a similar trend as that of round tube with almost same heat transfer and considerably better pressure drop.



Graph 6(a): Reynolds Number v/s Colburn j factor for flat tubes



Graph 6(b): Reynolds Number v/s Friction factor f for flat tubes

The increase in air temperature across two staggered tube rows is given in table 4. Although flat tubes exhibited slightly lesser Colburn j factor compared to that of round tubes, numbers tabulated herewith depict that the increase in the air temperature across two staggered rows is higher for flat tubes with lesser fin spacing and round tubes exhibited higher increase in air temperature with larger fin spacing.

Reynolds Number	Fin Pitch 1.75mm		Fin Pitch 2mm		Fin Pitch 2.24mm		Fin Pitch 2.5mm		Fin Pitch 3mm	
	Round Tube	Flat Tubes	Round Tube	Flat Tubes	Round Tube	Flat Tube	Round Tube	Flat Tube	Round Tube	Flat Tubes
4300	36.7	37.38	34.4	33.22	29.22	30.27	28.42	26.84	22.05	21.98
5200	33.4	34.5	31.73	29.97	26.14	26.86	25.23	23.65	19.59	19.35
6200	30.35	31.13	28.40	27.04	23.47	24.21	22.26	21.02	17.73	17.51
7000	28.11	28.88	26.13	24.99	21.44	22.35	20.35	19.31	16.72	16.08

Table 4: Increase in Air temperature across two staggered tube rows

V. Conclusion

For both round and flat tube domains with all the geometrical configurations simulated in this work Colburn j factor varied inversely with the inlet air velocity. The heat transfer is more with the higher fin spacing for both round and flat tubes following the above said trend. On the other hand, the pressure drop across the tubes is more with the lesser fin spacing in contrast to the heat transfer. Due to lesser turbulent intensity in flat tubes, they exhibit slightly lesser Colburn j factor and considerably lesser pressure drop compared to round tubes. Although flat tubes exhibit slightly lesser Colburn j factor, due to larger exposed tube area increase in the air temperature in the fin side is comparable with that of round tubes. Higher fin temperatures result with lesser Colburn j factor and higher pressure drop across the tubes although the fin temperature affects the pressure drop to lesser extent.

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