

Thermo-Hydraulic Performance Analysis of Plain Rectangular Plate Fin Heat Exchangers in Laminar Region -A Numerical Study

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ABSTRACT: Recent advancement in the chemical process plants, power industries, cryogenic applications, aerospace applications and electronics technology requires efficient and economic design of compact heat exchangers. This is always a challenge to design a compact heat exchanger because of its complex structure. Plate Fin Heat Exchanger (PFHE) is very common compact heat exchanger used in industries. But many of the cases they are not well designed from the point of view of heat transfer characteristics and pumping power requirement. In this research work an extensive numerical study of the rectangular plate fin heat exchanger is done to evaluate heat transfer characteristic j factor and heat exchanger core pressure drop characteristic Fanning friction factor f as function of heat exchanger geometry and Reynold's number. Colburn j factor and friction factor are the primary parameters required to find the effectiveness and pumping power requirement of the PFHE.

KEYWORDS: Plate Fin Heat Exchanger, Numerical study, Colburn j factor, Fanning friction factor f

<https://doi.org/10.29294/IJASE.6.S1.2019.25-29>

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1. INTRODUCTION

The compactness of heat exchanger is the index of progress in the present-day scenario of industrial growth [1] especially with increasing the need for developing the cryogenics field. Usually, plate fin heat exchangers are suitable for a wide range of industries [2-5] like chemical process, power industries, aerospace, automobile and electronics. Plate fin units are normally arranged for counter flow heat exchanger but cross flow arrangements are also very common. Plate fin heat exchanger has thin corrugated fins. Surface density (large surface area per unit volume) of compact heat exchangers is very high and it could be as high as $1800 \text{ m}^2/\text{m}^3$, the plate fin heat exchanger is suitable for a close approach temperature as low as $2 - 1^\circ\text{C}$ and two or more streams can be used by changing the section.

Plate fin heat exchanger is significant now a days and most widely used due to high heat transfer rate. Norris and Spofford [6] reported the first experimental report they draw out the effect of heat transfer coefficient on the basis of length, thickness and pitch of fins and also reduced the friction factor and Colburn j factor. As the practical demand of plate fin heat exchanger has increased experimental studies, have been made by London and Shah [7] in 1967 they been brought to a conclusion that small offset spacing fin thickness and a large number of fins per inch gives better heat transfer. In 1975 Wieting [8] setup a relationship between the variables from earlier experimental heat transfer and fluid flow friction data for a plate-fin heat exchanger of offset fin and by using this relationship (untested offset fin geometries can be predicted realistically and accurately within the parameter range of the

correlation). So that one can predict virtually and correctly within the parametric range of newest offset plate-fin heat exchanger having no previous tested data.

Experimental validation of numerical simulation also provides a comparison of experimental result in computationally obtained results from the effects of fin thickness and freestream turbulence. In 1977, a set of experiments was performed by Sparrow [9] to observe the heat transfer for a mass flow rate with varying Reynolds number. The thickness ratio and the spacing ratio are the other factors establish that the Nusselt number varies while changing the plate thickness and also come upon after searching that it is not necessarily equal spacing and length gives optimal results.

Though many works had been done on study of PFHE but the results are not consistent and rather highly conflicting in few cases. So, more research in this area is required. A numerical simulation using Finite Volume Method is done to analyse the performance of the plain rectangular PFHE in a Reynolds number range of 200-800 for different height and space ratio with fin thickness. A multiple regression analysis has been done on the simulation data and correlations are suggested for f and j as function of Re and other non-dimensional geometrical terms.

2. NUMERICAL ANALYSIS

2.1 Physical description of the model

The PFHE of rectangular cross section as shown in Fig 1 is of our interest of study for different ranges as mentioned below with water as cold fluid to be heated in the heat exchanger.

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Received: 01.05.2019

Accepted: 28.06.2019

Published on: 20.07.2019

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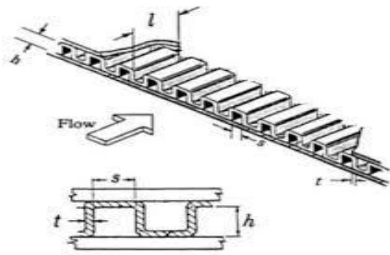


Fig 1: Geometry of Plain Rectangular Plate Fin Heat Exchanger

Table 1: Working Ranges of Model Parameters

Parameters	Working range
Reynolds number	$200 \leq Re \leq 800$
Height of the fin	$2mm \leq h \leq 8mm$
Fin spacing	$2mm \leq s \leq 6mm$
Thickness of the fin	$0.0254mm$
Length of the fin	$100mm$

2.2 Governing equations

The governing equations to describe the fluid flow and heattransfer phenomena are continuity, momentum and energy equation. The mentioned equations are solved to get the velocity, pressure and temperature field. We have considered 3-D, laminar and steady flow model, so the governing equations are as following,

Continuity equation:

$$\nabla \cdot (\rho \mathbf{V}) = 0 \quad (1)$$

Momentum equation:

$$\nabla \cdot (\rho \mathbf{V} \mathbf{V}) = -\nabla P + \nabla \cdot (\mu \nabla \mathbf{V}) \quad (2)$$

Energy equation:

$$\nabla \cdot (\rho c_p \mathbf{V} T) = \nabla \cdot (k \nabla T) \quad (3)$$

2.3 Boundary conditions

Atinlet, outlet and wall boundary conditions are as following.

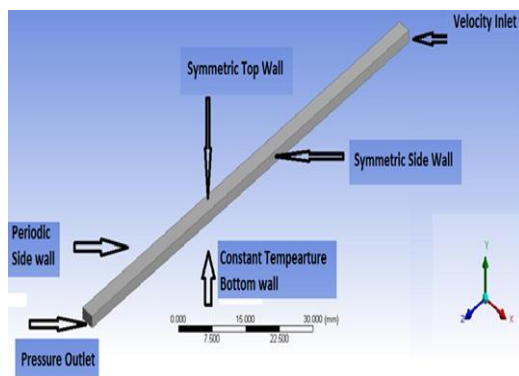


Fig 2: Geometry of the Model and Boundary Conditions

Top, side wall, symmetric top, symmetric side wall and bottom wall are assumed no slip boundary condition. Sidewall is kept at periodic boundary condition and bottom wall is kept at constant temperature boundary condition. Inlet set as velocity

inlet boundary condition corresponding to the Re range from 200 to 800 to assure the flow is laminar.

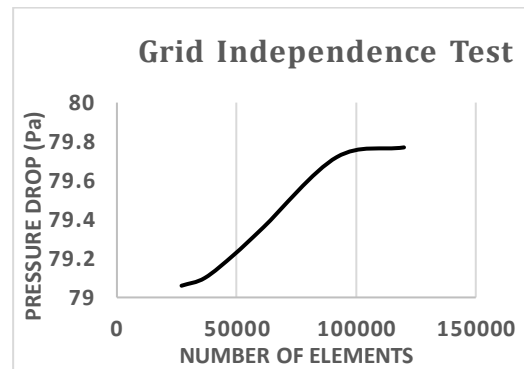
2.4 Meshing and model setup

The mesh of the computational domain was generated using a triangular patch conforming method. This mesh contains hexahedral cells having rectangular faces at the boundaries. The generated mesh consisted of 91875 elements and 96731 nodes.

A 3-D, incompressible and pressure-based solver was chosen for the computational domain. A 2nd order upwind interpolation formula was used for discretization of momentum and energy equation. The conventional SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm was used to solve the pressure velocity coupled equations, where several iterations were performed to ensure convergence. Convergence of the numerical solution was assured by monitoring the scaled residuals to a constant level of 10⁻⁶ for each variable.

2.5 Grid-independency

The solution obtained in this work is mesh independent as we have studied the mesh independency test. Result is as following



2.6 Experimental validation of the model

Validation is done to check the authenticity of the CFD tool. We have validated the model by comparing the results of the experiment done by Kays and London (1984), Table 10-3, FPI 6.2.

Table 1: Validation of the model with experimental results

Re=800	Our Simulation Result	Experimental Result of Kays and London	Error
Friction factor(f)	0.024	0.021	12.5%
Colburn j factor	0.0049	0.0058	18.6%

3. RESULTS AND DISCUSSION

This is clear from the solution of the numerical analysis that with the increase of Re in the laminar region from 200 to 800 friction factor decreases. The

relationship is nonlinear and f also depends on other non-dimensional geometric parameters as shown in Fig 4(a),4(b) and 4(c).It is observed (Fig 4(c)) that with the decrease of (t/s) friction factor is increasing. It is also observed that for constant Re and (t/s) friction factor increases with increasing (h/s) . Another set of graphs shown in Fig 5(a), 5(b) and 5(c) reveals that Colburn j factor decreases with Re in a nonlinear relation and also depends on the fin geometrical parameters. It is observed in Fig 5(b) that

j factor increases with increasing (h/s) which is because of the fact that fin lengths are increased improving heat transfer.It is observed in Fig 5(c) with decreasing fin thickness with respect to fin spacing j factor.It is clear that with increasing fin frequency heat transfer parameter j factor increases at the cost of increasing friction factor.So, it is required to optimize the fin frequency and size of the fin in a good heat exchanger design.

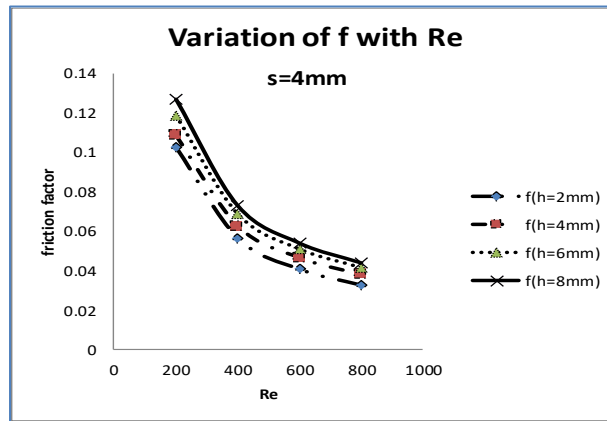


Fig 4a: Variation of friction factor with Re

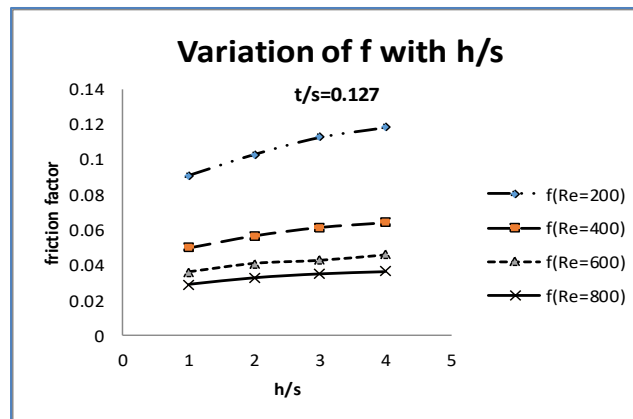


Fig 4b: Variation of friction factor with (h/s)

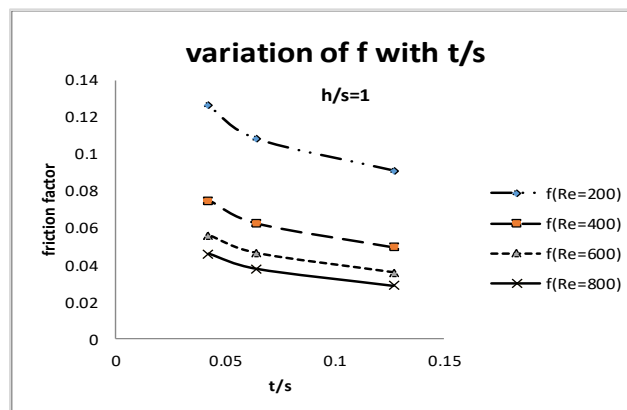


Fig 4c: Variation of friction factor with (t/s)

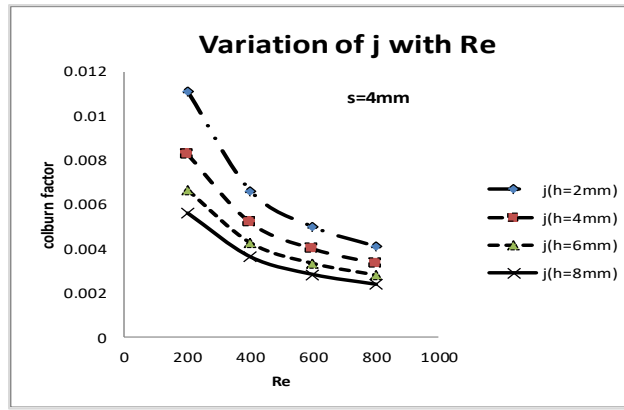


Fig 5a: Variation of Colburn j factor with Re

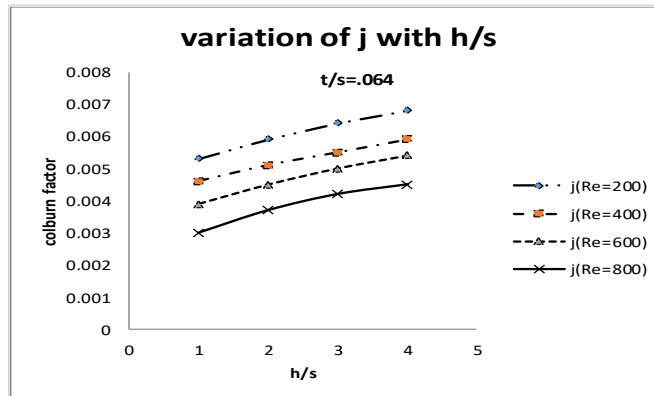


Fig 5b: Variation of Colburn j factor with (h/s)

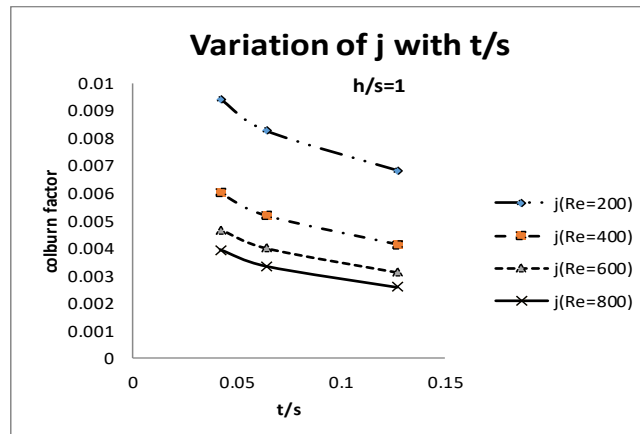


Fig 5c: Variation of Colburn j factor with (t/s)

Multiple regression analysis has been done on the simulation results and the correlation of 'f' and 'j' are suggested as function of Re and non-dimensional geometrical parameters (h/s) and (t/s). The correlations are valid in the range of $200 \leq Re \leq 800$, $1 \leq h/s \leq 4$ and $0.04 < t/s < 0.14$

$$f = 3.23Re^{-0.81} \left(\frac{h}{s}\right)^{0.16} \left(\frac{t}{s}\right)^{-0.34} \quad (4)$$

$$j = 0.105Re^{-0.68} \left(\frac{h}{s}\right)^{0.39} \left(\frac{t}{s}\right)^{-0.37} \quad (5)$$

4. CONCLUSION

It may be concluded that within a specified limit of fin geometry (i.e. $1 \leq h/s \leq 4$ and $0.04 < t/s < 0.14$) and laminar region of $200 \leq Re \leq 800$, f and j factor vary

in a non-linear way. It is observed that f and j both are decreasing with increase of Re and (t/s) but increases with h/s. A regression analysis has been done to find a correlation of f and j factor with Re and non-dimensional geometric parameters h/s and t/s. These correlations are very useful in finding out the heat transfer coefficient and pressure drop characteristic during heat exchanger design problems.

ACKNOWLEDGEMENT

We are thankful to college authority of MeghnadSaha Institute of Technology (M.S.I.T.) to carry out this research work in CFD Lab, Department of Mechanical Engineering.

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Selection and/or Peer-review under the responsibility of 2nd International Students' Conference on Innovations in Science and Technology (Spectrum - 2019), Kolkata

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