Experimental and Numerical Investigation of a Multi Louvered Plate Fin-and-Tube Heat Exchanger using Computational Fluid Dynamics

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ABSTRACT

Heat exchanger is a thrust area, which required constant improvement in the performance. Recently multi-louvered plate fin heat exchangers, a kind of compact heat exchanger, have been the subject of vast research. Multi-louvered plate heat exchanger has unique advantage, in that it is easy to control fluid temperature beside the general peculiarity of compact heat exchanger; so a large number multi-louvered plate fin heat exchanger, especially plate fin heat exchanger, are used in chemical industry. A numerical investigation of the air side performance for fin-and-tube heat exchanger having circular tube and oval tube configuration is presented in this study. The geometrical louvered fin parameters of louver angle, louver pitch, louver length are examined. The fixed louver length (6.25mm) and louver pitch (2mm) only the louver angle is varying. However, an optimal value of heat transfer performances is observed due to the difference in the development of thermal boundary layer on the upper and lower louver surface. The heat transfer characteristic for oval tube is slightly higher than the circular tube. In the present study, successively increased or decreased louver angle patterns are proposed and numerical analysis on heat transfer and pressure drop are carried out. Louvered fins are commonly used in many compact heat exchangers to increase the surface area. For 25 heat exchangers with different louver angles (1^0–29^0), fin pitches (1.0, 1.2, 1.4,1.6,1.8,2 mm) and flow depths (16, 20, 24 mm), a series of tests were conducted for the air-side Reynolds numbers of 100-1200, at a constant tube-side water flow rate of 0.043 kg/s. The inlet temperatures of the air and water for heat exchangers were 30 ^{0} c and 90 ^{0} c, respectively. The air-side heat transfer performance data were analyzed using effectiveness-NTU method for cross-flow heat exchanger.

Keywords: *Fin-and-tube heat exchanger; Heat transfer Co-efficient; Pressure drop.*

I. Introduction

Fin-and-tube heat exchangers are commonly used heat exchanger type in air conditioning mostly because of its flexible manufacturing process. The capacity and characteristics of the heat exchanger can be scaled up or down per the requirements of the application. Fin-and-tube heat exchanger is constructed from fin plates that are penetrated with tubes that are connected to each other with a header from start and finish of the circuit. Heat is transferred between the fluid that circulates inside the tubes and the air that flows between the fins. Simulations are done in computational means with using only open source software. Junqi Dong et al (2007) have done the work on heat transfer and pressure drop correlations for the multi-louvered fin compact heat exchangers. It was observed that the heat transfer coefficients increase with fin length decreasing at the same frontal air velocity while the pressure drop decreases. Michael J. Lawson et al (2007) have done the work on heat transfer augmentation along the tube wall of a louvered fin heat exchanger using practical delta winglets. It was observed that louvered fin geometry to improve the heat transfer on the tube wall surface, and also increasing the pressure drop. Paul A. Sanders et al (2006) have done the work on the increasing the tube wall heat transfer by using the delta winglets placed on the louvers near the tube wall. It was observed that an increase in heat transfer with

increasing number of winglets. They showed results for winglets for winglets staggered and alternating direction, which produced higher heat transfer augmentations that on the in-line winglets in the same direction. Lyman.A.C et al (2002) have done the work on the study of heat transfer along louvers embedded in an array of louvered fins, which is representative of what occurs in a compact heat exchanger. It was observed that an increase in heat transfer coefficients as the flow transitioned from being duct directed to being louver directed. Ching-TSun Hsieh et al (2006) have done the work on analysis for louver fin heat exchanger with variable angle. It was observed that in higher heat transfer performance for large louver angle. A large louver angle would also contribute to the increase in the pressure drop. Man-Hoe Kim et al (2000) performed air-side thermal hydraulic performance of multi louvered fin in aluminium heat exchanger. Heat transfer and pressure drop characteristics of multi louvered fin using 45 different louvered fins for experimentally. The pressure drop increases with louver angle and flow depth and decreases with increasing fin pitch. Dohoy Jung et al (2006) has numerically investigated the fluid flow and heat transfer in multi louver plate fin heat exchanger. A numerical modeling methodology using finite difference method with staggered grid system based on the thermal resistance concept has been developed from the compact heat exchanger with louvered fin and flat tube. Malapure.V.P et al (2006) has numerical simulation of compact louvered fin heat exchanger is performed for the heat transfer and pressure drop characteristics. 15 different configurations are used. The simulation results are compared to the experimental data. Thomas perrotin et al (2003). The heat transfer performance predictions by CFD. The air-side using fin and tube heat exchanger, they present the result of 2D and 3D CFD model of compact multi louvered plate fin heat exchanger for the determination of heat transfer and pressure drop characteristics. This project investigates the effect of pin pitch, thickness, fin length and louver fin angle and heat transfer and pressure drop inside the plate fin heat exchanger.

Figure 1. Different sizes of fin-and-tube heat exchangers are designed for different applications

II. METHODOLOGY

- The various fin types are modeled for the standard specifications using the modeling software (pro-E).
- Each fin configurations are analyzed for heat transfer and pressure drop for the specified boundary conditions using CFD analysis CFX.
- The numerically obtained heat transfer and pressure drop characteristics of the fin arrangement will be validated experimentally.

III. PLATE FIN HEAT EXCHANGER

Numerical analysis is preferred because of their ability to exactly simulate the models with certain assumptions. It involves mathematical representation of physical process undergone by the system as a set of differential equations. The differential equations are solved as a set of simultaneous algebraic equations. They possess various advantages,

- Possible to view all the parameters of the system simultaneously.
- Speed of computing is high, consumes less time and cheaper.
- Versatile tool to solve complex problems.

There are various tools available for simulating and analyzing the models. The tool used for modeling the plate fin arrangement is Pro-E {Wildfire version 2.0}. The tool used for the analysis of plate fin models is CFX {Version 10.0 & 11.0}. They both are versatile and compatible tools for the heat transfer and flow analysis.

A. *Specifications of plate fin model*

The plate fin arrangements are modeled as per the standard specifications

PLATES

- Material : Aluminium
- Length : 101.6mm
- Width : 38.1mm
- Height : 0.3mm

FIN

- Material : Aluminium
- Louver Length : 6.25mm
- Louver Pitch : 2.4mm
- Louver Height : 1.4mm
- Louver Angle : 26°
- Fin Pitch : 2mm

TUBE

- Material : Copper
- Outer Diameter : 9.5mm
- Inner Diameter : 6mm
- Wall Thickness : 3.5mm

B. *Governing equations*

The governing equations are one dimensional unsteady energy equation and momentum equation.

X-momentum equation

$$\rho\frac{\partial u}{\partial t}+\rho u\frac{\partial u}{\partial x}+\rho v\frac{\partial u}{\partial y}=\mu(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2})-\frac{\partial p}{\partial x}$$

Y-momentum equation

$$\rho\frac{\partial v}{\partial t}+\rho u\frac{\partial v}{\partial x}+\rho v\frac{\partial v}{\partial y}=\mu(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2})-\frac{\partial p}{\partial y}$$

C. *Boundary conditions*

Plate temperature : 90°C
Air inlet temperature : 30°C
Air velocity : $3ms^{-1}$
Air density : $1.19kgm^{-3}$
Fin material : Aluminium {$k=204Wm^{-1}K^{-1}$}

D. *Fin configurations considered for analysis*

1. Plain Fin
2. Louvered Fin

IV. DESIGN CALCULATIONS

The parameters and non-dimensional numbers necessary for the analysis are calculated. The heat transfer coefficient signifies the amount of heat transferred from the fin to the air. This shows the heat transfer ability of the fin configurations. Reynolds number signifies whether the flow inside the plate fin model is laminar or turbulent. Heat flux is the amount of heat liberated per unit area of the plate and the fin in the plate fin arrangement.

Calculation of Reynolds number

Re = ρvd/μ

$= 1.165*3*(6.9*10^{-3}) / (18.63*10^{-6})$

Re=1294.444

Re<2300 (Laminar)

For **Re**=1294.444 and Multi Louver plate fin arrangement with specified dimensions,

Colburn j factor, **j = 0.0038**

Fanning friction factor, **f = 0.019**

Calculation of heat transfer coefficient

Heat transfer coefficient can be found from Colburn j factor,

$$j = Nu / (Re*Pr^{1/3})$$

$0.0038 = (h * 25.4*10^{-3}) / (0.02675 * 1294.444 * 0.888)$

$$h = 4.6\ Wm^{-2}K^{-1}$$

Table1. The plate fin arrangements are modeled as per the standard specifications

Multi-louvered fin		Tube	
Material	Aluminium	Material	Copper
Louver Length	6.25mm	Outer Diameter	9.5mm
Louver Pitch	2.4mm	Inner Diameter	6mm
Louver Height	1.4mm	Wall Thickness	3.5mm
Louver Angle	26°	No. of row in tube	2
Fin Pitch	2mm	Longitudinal Pitch	19.05mm
Fin thickness	0.3mm	Transverse pitch	25.4mm

Fin pitch = 2mm
Plate spacing = 2mm
Fin length flow direction = 38mm
Flow passage hydraulic diameter = 6.5mm
Fin metal thickness for Aluminium = 0.3mm

V. MODELING AND ANALYSIS

The various plate fin configurations are modeled using Pro-E {Wildfire version 2.0} software to the given specifications. Then the models are meshed using tetrahedral mesh with a mesh size of 0.5. The boundary conditions are the velocity inlet and the pressure outlet. The continuum types are specified for plate, fin and fluid. The continuum specified plate and fin is solid whereas for air is fluid. The meshed models are exported to CFX {Version 11.0}.

5.1. Air flow analysis

The computation domain of the tube-fin heat exchanger as rendered in the figure 2 is a symmetrical section. The symmetrical section of air flow region of 101.6 mm length and height is 38.1 mm and 0.3 mm width is modeled in PRO engineer as shown in figure 3. Grid tetrahedral meshes are applied in ANSYS Workbench. The section

represents the air flow region over three fins of the heat exchanger. The inlet velocity and temperature of air is applied to the section and symmetricity is applied as required. The pressure drop and temperature drop obtained for the symmetric airflow region has been extrapolated to calculate the drop for the whole system. The type of analysis conducted in the airflow region is laminar flow thermal unsteady state analysis.

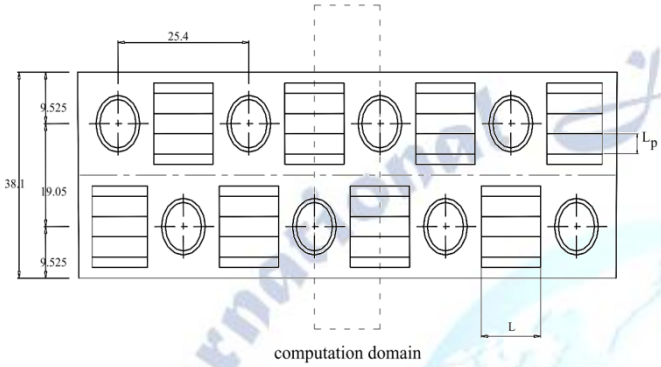

Figure 2. Computational domain of louvered fin heat exchanger

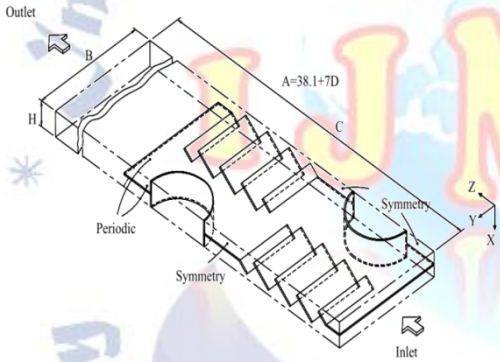

Figure 3.Symmetrical section of airflow region of the heat exchanger

5.2. Hot water analysis

The hot water flow analysis is a turbulent flow thermal unsteady state analysis with a second order of upwind differential scheme. The analysis of hot water flow in a tube fin heat exchanger involves further complexity.

The boundary conditions applied in round tube,

- Inlet velocity of hot water = 1m/s
- Outflow condition at the outflow of hot water tube.
- In the Wall convective coefficient of air = 4.6W/m^{2}K^1 at a temperature of 303K.
- Tube flow is simulated and solved to obtain total surface heat transfer flux.
- Arrangement will be validated experimentally.

5.3. Plate Fin Models

The plate fin models are a small representation of the PFHE. A single passage alone is taken into consideration. The various fin configurations are plain fin, multi-louvered fin as shown in figure.4, 5, 6 and 7.respectively. A simple fin model is analyzed.

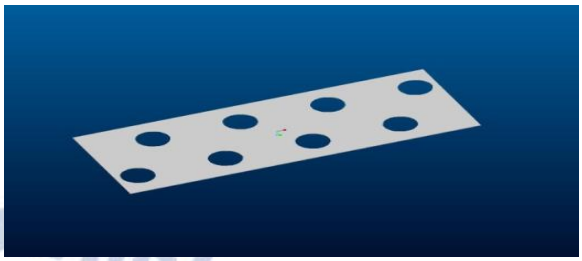

Figure.4. simple plain fin

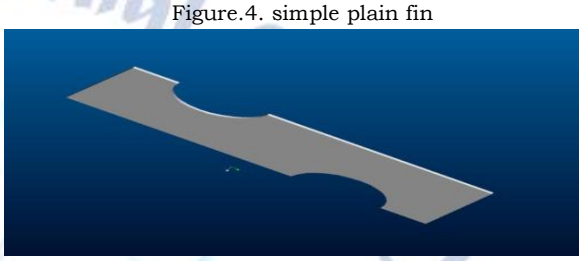

Figure.5.cut section airflow region of plain fin

5.4. Multi-Louvered Plate Fin

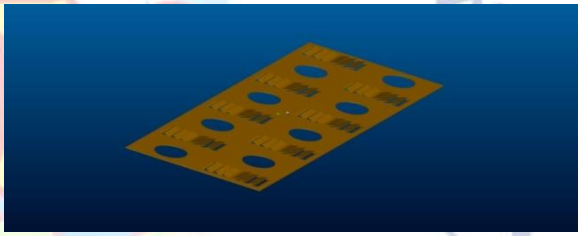

Figure.6.schematic of multi-louvered fin heat exchanger

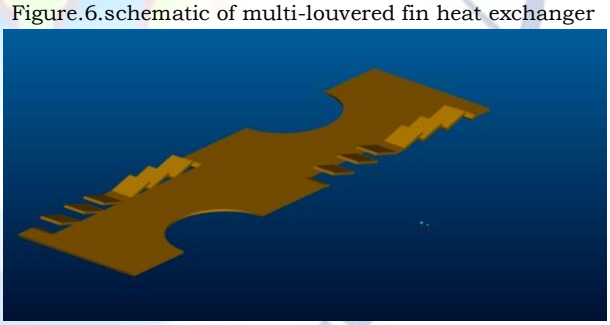

Figure.7. symmetrical section of airflow region of the louvered fin

5.5. Steady state analysis

Steady state analysis is carried out to find the temperature, pressure and velocity variations inside the plate fin model. Steady state analysis is performed for simple fin, multi-louvered fin.

5.6. Fundamentals of Heat Transfer

Heat transfer can be divided into three different modes: conduction, convection and radiation, where only first two are considered in this paper due to the low temperature levels in fin-and tube heat exchangers. The law that governs heat conduction was first proposed by J.B. Fourier in

1822 (Mills, 1999). It states that the heat will flow through the medium in the direction of decreasing temperature gradient. Therefore, the one-dimensional heat flux can be defined as:

$$q = -k \frac{dT}{dx}$$

where k is called the thermal conductivity of the material. The minus sign comes from the definition that when the gradient is negative the heat flux is positive in the direction of positive x-axis. Convective heat transfer is a combination of conduction and advection. Advection can be described as the measure of mass that is transferred with the flow field. Convection is then an integration of conduction and advection in the flow medium. The amount of heat that is transferred from a surface to the flowing medium can be described with a heat transfer coefficient that is defined as:

$$h = \frac{q}{A\Delta T}$$

which is the amount of heat transferred ‘ q ’ per the surface area ‘ A ’ and the temperature difference ΔT

5.7. Turbulent Flow

Turbulent flow is used in practical heat exchanger applications to enhance heat transfer. Turbulent flow still always comes with a cost, which is the increased pressure drop in the fluid medium. In this paper, the flow is modeled with turbulence models due to the lack of super computers and aim to achieve an efficient way to study heat exchangers in the future, forces us to use computationally cheaper approaches.

5.8. Turbulence Characteristics

Irregularity

This characteristic emphasizes the nature of turbulence to be irregular, random or even chaotic, meaning that prediction of turbulent behavior is, if not possible, a very challenging task. Despite of its irregular nature, it can be studied with deterministic approaches and Navier-Stokes equations still describe the full nature of the flow.

Diffusivity

Turbulent flow is more diffusive than laminar one. Increased diffusivity can be said to be the most important feature of turbulent flow since it increases the transfer of momentum, heat and mass. This can be used in practical applications for example to delay the separation of the flow from the body to increase the possible angle of attack for airfoils and to increase the heat exchange in heat exchangers. It is also a source of resistance of the flow in ducts and pipes.

Large Reynolds number

Turbulent flow always occurs in flow situations with high Reynolds numbers. This can be said to be over ReD >= 2300 for the flow inside a circular tube or Rex >= 500000 for the flow over a plate. When the Reynolds number increases the instabilities in the flow increase with it.

Three-Dimensional

Turbulence is always a three-dimensional phenomenon. This is mainly because the main feature that enhances turbulence is vortex stretching. It is the main mechanism of transforming turbulent energy to smaller scales.

Dissipation

In a turbulent flow, the big eddies are created by the mean flow and their size can be as large as the length scale of the mean flow. This turbulent energy is then transferred to smaller, medium sized eddies and again to smaller and smaller eddies, until the smallest eddies are transferred into thermal energy. This process is called the cascade process, and it explains why turbulent flow is dissipative in its nature.

Continuum

The length a scale of the turbulent eddies can be considered much bigger than the molecular length scales.

VI. Results and Discussion

The heat transfer and pressure drop characteristics of the two fin shapes are compared with j- and f-factor and then the flow field is analyzed more carefully with streamline, glyph and contour plots. Calculation times with 60 cores ranged from around 5 hours to up to 100 hours depending on the cell count and inlet velocity conditions. If we now compare the efficiency parameters of the two studied fin shapes and tube configurations, some very interesting characteristics can be found. If we look at figure 8, where the j-factor is plotted without any fin efficiencies, the plain fin shape seems to transfer heat more efficiently than the x-slit fin.

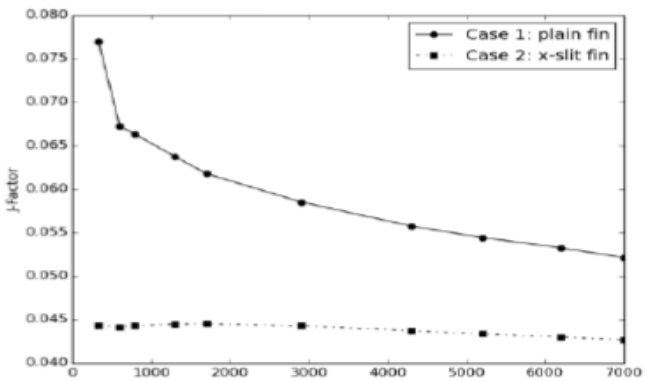

Figure 8. Comparison of the j-factor between plain and x-slit fin

This is due to the denser disposition and the size of the tubes. It must be remembered that even though the temperature at the outlet rises higher with the x-slit fin shape, compared to the plain fin geometry, a much higher potential mass thermal capacity flows through it due to the bigger frontal face area. If other parameters would have been kept constant and only the fin shape would have been changed, it would be sure that x-slit fin shape would transfer more heat than the plain fin geometry. If we now look at the normalized pressure drop in figure 9 we see that x-slit fin has a much higher fanning friction factor with all the flow velocities studied in this paper.

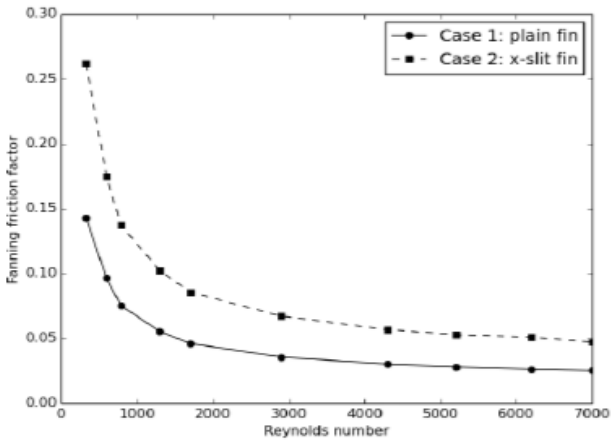

Figure 9. Comparison of the fanning friction factor between plain and x-slit fin

6.1 Flow Characteristics

If we look closer at the flow structures between the fins and before the first tube, we can say that it looks to be combined from two fundamental fluid study cases: channel flow and an impingement flow. The features of both of these can be seen on figure 10.

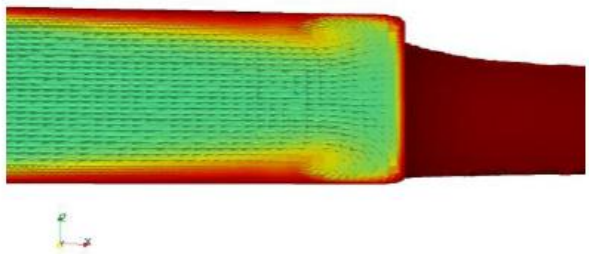

Figure 10. Impingement flow on the first tube

On the left side of the picture the velocity profile can be seen to have its curvy shape as the velocity reaches its maximum value on the middle of the channel and gradually goes to zero on the wall. On the right side of the picture before the round shape of the tube, an impingement flow and its two vortices can be seen on both upper and lower parts of the channel. Recognizing these kinds of basic flow features can be highly useful in the process of validating the computational domain but also more importantly when thinking about enhancing the heat transfer and utilizing all the research done on these subjects.

VII. Conclusion

In this paper a simulation method for convective heat transfer on the air side of a fin-and-tube heat exchanger was developed. All the programs used were open source and distributed with the public license. The method developed is fully capable of simulating arbitrary fin shapes and comparing the hydrothermal characteristics of different fin shapes. Two different fin shapes: plain and x-slit fin with differing tube configurations were studied. Plain fin case was used as a model validation case and two different types of meshes were created for this fin shape: a structured curvilinear mesh and an unstructured mesh. This unstructured mesh creation method was developed because of the increased complexity on the fin shape and its influence to the difficulty level in the meshing procedure with structured handmade meshes. A lot of problems were encountered with the meshing procedure for the unstructured mesh, as snappy Hex Mesh is prone to crashing for many several reasons. These reasons are accumulated over the different meshing phases in the meshing procedure of the snappy Hex Mesh. Then plain fin geometry with smaller tube size was compared to x-slit fin geometry with a bigger tube size and disposition of the tubes. It was noticed that the heat transfer efficiency parameter j-factor, with ten different flow velocities, was higher for the plain fin than for the

x-slit fin even though x-slit fin opposes a lot more turbulent structures in the flow field. Then the situation was completely turned around when the normalized pressure drop called fanning friction factor parameter was compared with ten different inlet velocities. It was found to be higher for the x-slit fin than for the plain fin. This is due to the non-disturbed flow field in the plain fin, even though the tubes are smaller and closer to each other. It is clear that the efficiency parameters are an important tool in the process of comparing different fin shapes and that comparing only one output variable like temperature should always be avoided.

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