

Numerical Investigation of Finned-tube Heat Exchanger with Circular, Elliptical & Rectangular Tubes

MD. HASIBUL HASAN¹

Department of Mechanical Engineering
Khulna University of Engineering & Technology
Khulna, BANGLADESH

DIPAYAN MONDAL²

Department of Mechanical Engineering
Khulna University of Engineering & Technology
Khulna, BANGLADESH

Abstract: - A three dimensional numerical study has been conducted on finned-tube heat exchanger with multiple rows of tubes using ANSYS (Fluent). The objective of this study is to numerically investigate finned tube heat exchanger with different type of tubes such as circular, elliptical and rectangular tubes. As circular tubes has much pressure drop so elliptical and rectangular tubes has been introduced in order to reduce pressure drop. As well as heat transfer has also been examined. The finite volume based CFD code ANSYS Fluent 16.2 is used to calculate the flow and temperature fields and by applying SIMPLEC algorithm. At low velocity of air and water, nothing significant occurred for the combination of tubes. At high velocity in maximum tube combination there was heat transfer (HT) enhancement and pressure drop reduction when compared with circular tubes only in case of air. When the combinations of circular, elliptical and rectangular tubes has been compared with circular tube heat exchanger (CTHX) heat transfer reduces as well as pressure drop (PD) also reduces for air. In case of water vapor HT and PD behaves the same. When those combinations has been compared with elliptical tube HX, for air in some cases heat transfer remains same and on other case it increases. For pressure drop in case of air, in some cases it reduces and on other cases it reduces. For elliptical tube HX for the fluid water vapor HT and PD both remains same or reduces. This work has not been with conducted any numerical simulation on rectangular Heat exchanger reason behind it there isn't an y existence of this kind of heat exchanger. However, it could be numerically conducted to examine the results between those combination and rectangular heat exchanger.

Keywords: Conventional and modified heat exchanger; Darcy friction factor; Fin and tube Heat exchanger; Heat Transfer; Numerical simulation.

Received: October 11, 2019. Revised: March 9, 2020. Accepted: April 5, 2020. Published: April 30, 2020.

NOMENCLATURE

B	Baseline or Conventional heat exchanger	N	Number of tube rows
C_p	Specific heat capacity in J/Kg K	Nu	Nusselt number
D	Diameter of the circular tube in m	Pr	Prandtl number
f	Darcy Friction factor	Q	Heat transfer rate in W
H	Fin Height in m	Re	Reynolds number
h	heat transfer co-efficient in W/m ² k	T	Temperature in K
k	Thermal conductivity in W/mk	U	x-component velocity at inlet in m/s
L	Length of fin in m	Δp	Pressure drop in N/m ²
m	Mass flow rate of air in Kg/s	ΔT_{lm}	Logarithmic mean temperature difference
M	Modified Heat Exchanger		

1 Introduction

Plate fin heat exchangers consists of a number of corrugated metal plates in mutual contact, each plate having four ap paratus serving as inlet and outlet ports, and seals designed to direct the fluids in alternate flow passage. The plates are clamped

together in a frame that includes connections for the fluids. Since each plate is generally provided peripheral gasket to provide sealing arrangement [1]. The present analysis is focused on plate fin and three different type of tubes such as circular, elliptical & rectangular tube. Air is passed in between plate fin and tubes for combination of

tube. In this present analysis, fluid streams remain separate and the heat transfer takes place continuously through a dividing impervious wall. In cross flow exchanger, two fluid are normal to each other and both fluids are unmixed. For unmixed-unmixed case, fluid temperature variations are idealized as two-dimensional only for the inlet and outlet section [2]. Thermal effectiveness for the cross flow heat exchanger falls in between parallel flow and counter flow arrangement. In many cases one gas is compressed to increase pressure so the density is high while the other at lower pressure and low density. Compared to liquid-liquid exchangers, the size of gas-gas exchanger is much larger, because the convective heat transfer co-efficient on the gas side is low compared to the liquid side [3]. Circular tube is used in conventional heat exchanger. But now a days, elliptical tube is introduced. Elliptical tube is used due to its good aerodynamic shape. Elliptical tube has less cross sectional area than circular one. Influence of aerodynamic shape of an elliptical tube on the thermo-hydraulic performance is found quite significant by reducing the ellipticity ratio. Elliptical tubes also helps to reduce drag force [4]. Rectangular tube in heat exchanger is a new concept. This type of tube is used as a duct in air conditioning system. It is used to increase the heat transfer area which additionally enhance heat transfer. In this present analysis, fluid streams remain separate and the heat transfer takes place continuously through a dividing impervious wall.

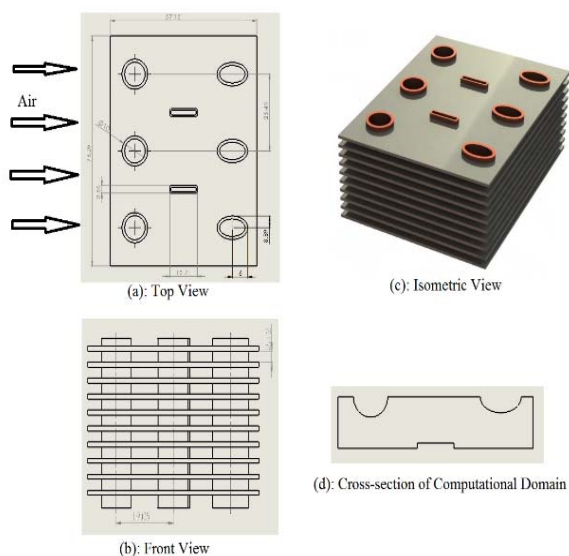


Fig 1: Schematic configuration of a heat exchanger for N3M1

2 Literature Review

Based on the intersection angle between the velocity and temperature gradient, Guo et al. [5] have proposed an approach to identify the region where the heat transfer should be enhanced in parabolic type flows. Later Based on field synergy principle Li et al. [6] have studied the fluid flow and heat transfer characteristics of strip fin w.r.t its position (upstream, downstream) and observed that strip fin in downstream arrangement performs better than others. A significant contribution on heat transfer enhancement and the study on flow characteristics of flow past various tube arrangements has been reported Jayavel and Tiwari et al. [7-9]. It is noticed that for the same heat transfer area using different diameter of tube in the successive row yields better performance over the same diameter tube. A multiple correlations of friction factor and Nusselt number for large tube diameter and large number of rows have been proposed by Wu and Tao et al [10], it is noticed that for the same heat transfer area using different diameter of tube in the successive row yields better performance over the same diameter tube. A multiple correlations of friction factor and Nusselt number for large tube diameter and large number of rows have been proposed by Xie et al [11]. The geometric effects on the performance of plate finned-tube heat exchanger with different fin pitch, tube pitch, fin thickness, and tube diameter have been investigated by Lu et al [12]. From their work it is noticed that, with a fixed inlet frontal velocity there is enhancement in heat transfer by decreasing tube diameter. The influence of tube shape on the thermo-hydraulic performance is found quite significant from the work of Lotfi et al. [13], it is identified that the heat transfer performance is enhanced by reducing the tube ellipticity ratio.

3 Theoretical Aspects

3.1 Flow across tube banks

Because many heat-exchanger arrangements involve multiple rows of tubes, the heat transfer characteristics for tube banks are of important practical interest. The heat-transfer characteristics of staggered and in-line tube banks were studied, and on the basis of a correlation of the results of various investigators. The original data were for gases with $Pr \sim 0.7$. To extend the use to liquids, the present writer has modified the constants by the same $1.11Pr^{1/3}$ factor employed. The Reynolds number is based on the maximum velocity occurring in the tube bank; that is, the velocity

through the minimum-flow area. This area will depend on the geometric tube arrangement.

Heat transfer in tube banks can be computed from the general correlation [14]:

$$Nu_D = C_1 C_2 Re^n Pr^m \left(\frac{Pr}{Pr_s} \right)^{0.25} \quad (1)$$

3.2 Physical Model

In this present simulation, a three dimensional numerical study of overall performance of the heat exchanger has been performed with combination of circular, elliptical and rectangular tube. Plate-fin with the staggered tube arrangement is taken to study with water-vapor and airside heat transfer and pressure drop characteristics. The tube geometries considered are circular, elliptical and combination of circular, elliptical and rectangular tube. Too closely represent the real time heat exchanger, multiple tube rows (number of tube rows, $N = 3, 6$) are considered. The effect of number of tubes has also been studied. Increased number of tube leads to decrease in heat transfer and when the number exceeds 6, the corresponding increase in heat transfer is negligible [4], therefore present work concerns up to six rows of tube.

The circular and elliptical tubes are designed in such a way that perimeter of the tubes are same, which allows ease of manufacturing and ensures the same heat transfer area for circular, elliptical and rectangular tube. Eccentricity of the elliptical tube is taken as which is closer to Rocha et al [15]. The range of the inlet velocity is taken between 0.5-2.5m/s with 0.5 intervals assuming the flow to be laminar. As the region of interest of the present work is focused on to identify the tube geometric effect, the fin surface is assumed to be a constant wall temperature.

3.3 Geo-metric Details

Perimeter of all tube e.g. circular, elliptical and rectangular is taken as equal. For an ellipse having semi axes of lengths a and b , the perimeter (P) is approximately [16].

$$P = 4 \times (a + b) \times \left(\frac{\pi}{4} \right)^{\left(\frac{4ab}{(a+b)^2} \right)} \quad (2)$$



Where, $a = 6$ mm and $b = 3.89$ mm by trial and error method taking eccentricity $e = b/a = 0.6$ for rectangular tube perimeter is $= 2 \times (a + b)$ $a = 10.71$ mm and $b = 2.5$ mm

Perimeter of the circular tube $= 2\pi r$; where radius of the circular tube $= 5$ mm

Table 1: Geometrical data of the simulation.

Transverse Pitch (P_t)	24.7 mm
Longitudinal Pitch (P_l)	19.05 mm
Circular Tube Diameter (D)	10 mm
Perimeter of circular, elliptical & rectangular tube	31.42 mm
Fin Height (H)	2.115 mm
Number of Rows	3,6

Table 2: Combination of tube in various arrangement.

Designation	Schematic Representation	Category
N3B1		Baseline-1
N3B2		Baseline-2
N3M1		Modified-1
N3M2		Modified-2
N3M3		Modified-3
N3M4		Modified-4
N3M5		Modified-5
N3M6		Modified-6
N6B1		Baseline-1
N6B2		Baseline-2
N6M1		Modified-1
N6M2		Modified-2
N6M3		Modified-3
N6M4		Modified-4
N6M5		Modified-5
N6M6		Modified-6

3.4 Mesh Generation and Solution Methodology

The computational domain is discretized into a finite number of control volume. In fin region, inflation mesh control is done with four edges of circular and elliptical in both sides. First layer thickness inflation option is selected. After that body sizing is done in fin region. In upstream and

downstream region. Several edge sizing is done in various direction.

The finite volume based CFD code ANSYS Fluent 16.2 is used to calculate the flow and temperature fields and by applying SIMPLEC algorithm. Under relaxation factor for pressure correction is taken as 1 for faster convergence [16]. To obtain improved accuracy of the solution, second order spatial discretization of the pressure is employed. As the grids are structured hexahedral and are aligned with flow direction, QUICK scheme is used for discretizing higher order convective terms in momentum equation. The residual is 10^{-6} for continuity and momentum, whereas for energy equation, it is taken as 10^{-8} .

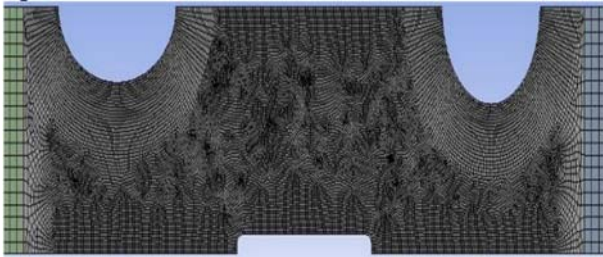


Fig 2: Schematic representation of grid.

3.5 Parametric Definition

The definitions of non-dimensional parameters such as Reynolds number (Re), Nusselt number (Nu) and Darcy friction factor (f) are defined as follows [9],

$$LMTD = \frac{(T_w - T_{in}) - (T_w - T_{out})}{\ln \frac{(T_w - T_{in})}{(T_w - T_{out})}} \quad (3)$$

Heat Transfer rate

$$Q = mC_p (T_{out} - T_{in}) \quad (4)$$

$$h_m = \frac{Q}{LMTD \times A_s} \quad (5)$$

$$Re = \frac{\rho U_{in} H}{\mu} \quad (6)$$

$$Nu = \frac{hH}{k} \quad (7)$$

$$f = \frac{\Delta P}{0.5 \rho U_{in}^2} \frac{H}{L} \quad (8)$$

4 Numerical Results and Discussion

4.1 Mesh Sensitivity

For checking mesh dependency baseline N3B1 is taken for consideration. Grid 1 = 264935 nodes, Grid 2 = 361911 nodes, Grid 3 = 498982 nodes. Mesh dependency is checked for heat transfer coefficient at different inlet velocity. From Fig 4.1, it is observed that at grid 1 and grid 3 gives almost same heat transfer coefficient. For further

calculation, grid 1 is taken as best option considering time for solution to converge.

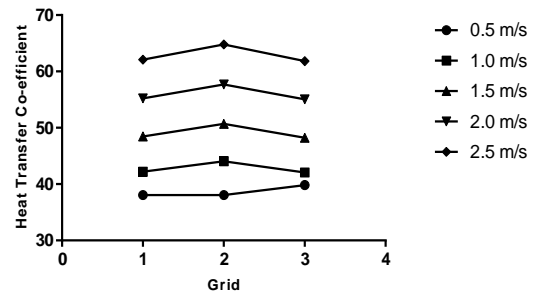


Fig 3: Grid independence results (N=3)

4.2 Validation of Numerical Results

For the fin and tube heat exchangers with plain fin configuration, the air side performance characteristics have been examined experimentally for various samples (varying geometrical parameters) [17]. The present results are validated with the experimental work of Wang et al. [17] and a close agreement has been observed as shown in Fig 4

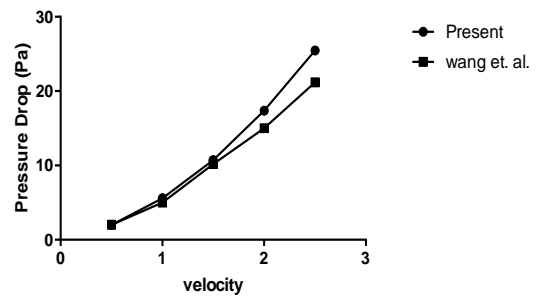


Fig 4: Validation results for N=2

4.3 Heat Transfer of Air

The connection of all tubes may pose manufacturing difficulty, but with advanced manufacturing technology, this connection is quite possible if all tubes are maintained at the same perimeter [16].

The result has been presented on Normalized Nu number and friction factor. The results are normalized (X_i/X_o), where i stands for modified cases and o stands for baseline cases which are conventional method such as heat exchanger with circular and elliptical tube rows. The hydraulic diameter of circular and elliptical tubes are different due to differences in cross section under constant perimeter, hence, Reynolds number based on fin height is used for representing the results. From Fig 5, it shows that N3M1 performs better than N3M2 when compared to N3B1. Though heat transfer gradually decreases, but at lower inlet velocities both perform better than conventional grouped circular tubes. Circular tube has a higher

heat transfer rate than elliptical and rectangular tubes. Heat transfer for elliptical tube almost half circular tubes. At lower inlet velocity heat transfer of N3M1 is greater than N3B1 due to no circulation occurs at this velocity. Mostly heat transfer by diffusion in this low velocity. And in this case for forced convection to takes place. In contrast, at high Re number N3M4 performs quite better than N3M3. N3M5 and N3M6 acts almost in similar fashion when compared with N3B1. N3M5 and N3M6 also acts almost similar fashion when compared with N3B2. N3M6 is a better option than N3M5. From Fig 6, the curve is upward when compared with N6B1. But N6M1 is better than N6M2 at high Re number. But for N6M2 almost all the tube gives higher circulation. For this reason high heat transfer for N6M1 than N6M2. In both cases, as velocity increases heat transfer increases, but heat transfer rate of N6B1 is much higher than N6M3. N6M4 also acts like N6M3 when compared with N6B1. N6M3 when compared with N6B2 it was seen the heat transfer for both of the cases are almost similar. N6M5 and N6M6 acts similar when compared with N6B1. But at high inlet velocity, heat transfer for both cases increases.

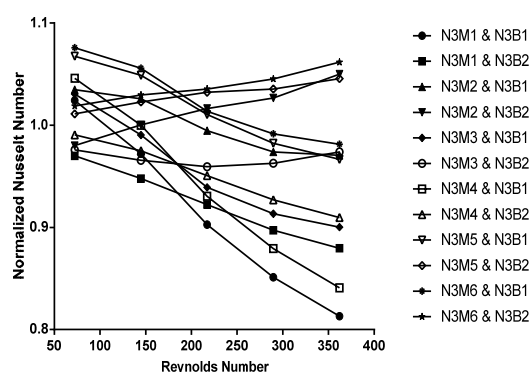


Fig 5: Nusselt Number of air of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=3

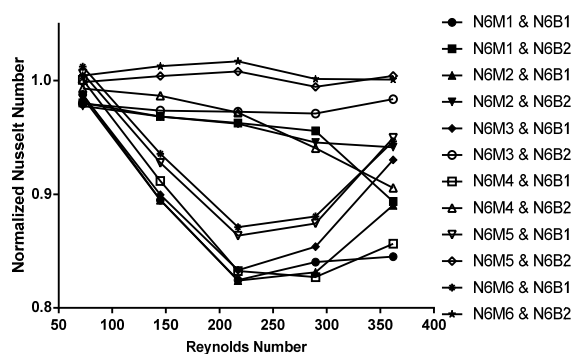


Fig 6: Nusselt Number of air of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=6

4.4 Heat Transfer rate for water-vapor

Water vapor is also taken in steads of air with all those combinations. Due to having different properties of water vapor there is difference in heat transfer and Nusselt number.

From Fig 7, N3M2 is a better option than N3M2 for water-vapor. But in reverse action is shown for N3M2 when compared with N3B2. For water-vapor N3M3 and N3M4 both have a low heat transfer rate from low inlet velocity to the high inlet velocity when compared with N3B1. Similar things happen for N3M3 and N3M4 when compared with N3B2. N3M5 & N3M6 combinations are better than the previous two named as N3M3 and N3M4. For water-vapor N3M5 and N3M6 both have a low heat transfer rate from low inlet velocity to the high inlet velocity when compared with N3B1. Similar things happen for N3M5 and N3M6 when compared with N3B2.

From Fig 8, N6M1 and N6M2 are compared with N6B1, at that time heat transfer is so much lower in high inlet velocity. But N6M1 and N6M2 are compared with N6B2 which is better than compared with N6B1.

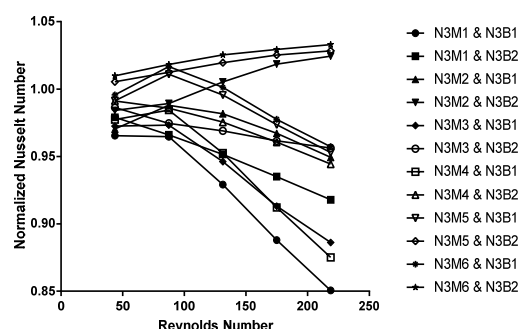


Fig 7: Nusselt Number of water-vapor of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=3

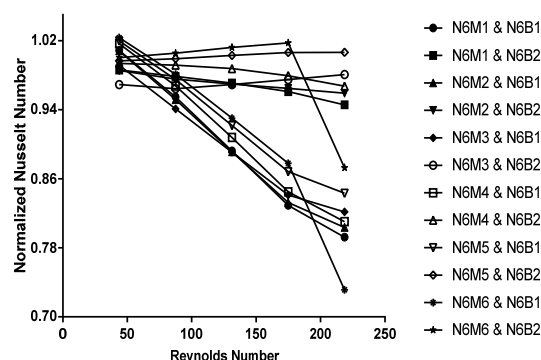


Fig 8: Nusselt Number of water-vapor of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=6

4.5 Friction factor for air

Pressure drops are represented in terms of friction factor and inlet air velocity is taken as in terms of Re number.

From Fig 9, it is shown that friction for N3M2 increases having high friction factor. In contrast friction factor for N3M1 decreases when compared with N3B2. N3M1 combination is better when compared with N3B1. N3M1 is better when compared with N3B2 because it has lower friction co-efficient. It is shown that N3M3 is the best combination because it has a much lower friction factor. But for the N3M3 pressure drop is higher when compared with N3B2. N3M4 also have lower pressure drop when compared with N3B1 than N3B2. Pressure drop decreases as velocity increases. It is shown that for N3M5 and N3M6 pressure drop reduces as velocity increases when compared with N3B1. Reverse action occurs when compared with N3B2.

From Fig 10, it is shown that N6M1 is the best combination because it has a much lower friction factor. But for the N6M1 pressure drop is higher when compared with N6B2. N6M2 also have lower pressure drop when compared with N6B1 than N6B2. N6M3 and N6M4 for both cases pressure drop reduces as velocity increases when compared with N6B1. But slight in pressure drop for N6M3 when velocity crosses 2.0 m/s. For N6M5 pressure drop is higher when compared with N6B2. N6M6 also have lower pressure drop when compared with N6B1 than N6B2. Pressure drop decreases as velocity increases. Reverse when N6M5 and N6M6 are compared with N6B2.

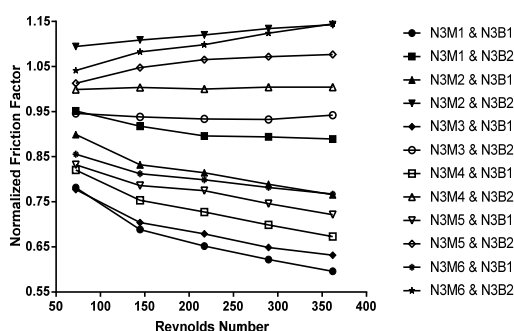


Fig 9: Normalized Friction Factor of air of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=3

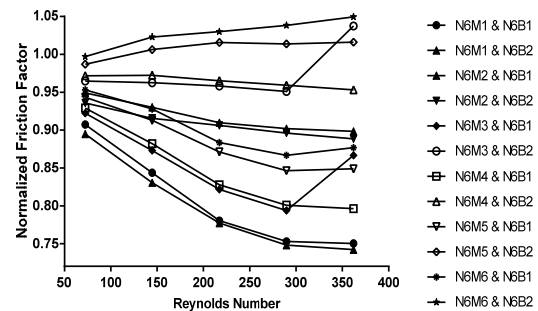


Fig 10: Normalized Friction Factor of air of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=6

4.6 Friction factor for water-vapor

Friction factor for water-vapor is estimated and plotted in terms of friction factor and Re number for number of tubes 3 and 6 for various combinations. From Fig 11, it is almost similar to air. N3M1 is better when compared with N3B2 because it has lower friction co-efficient. In contrast friction factor for N3M1 decreases when compared with N3B2. N3M1 combination is better when compared with N3B1. It is observed that at low inlet velocity for N3M3 and N3M4 value is respectively 0.85 and 0.83 but when velocity increases pressure drop reduces and respectively the values are 0.73 and 0.68 when both are compared with N3B1. N3M3 and N3M4 are compared with N3B2 friction factor values are 0.95 and 0.98. For N3M3 pressure drops values almost equal to N3B2. N3M5 and N3M6 pressure drop reduces as velocity increases when compared with N3B1. Friction factor for N3M5 and N3M6 at lower velocity, friction factor velocity are respectively 0.86 and 0.87. But for higher velocity those values decrease to 0.80 and 0.78. Reverse action occurs when compared with N3B2.

From Fig 12, it is shown that for N3M5 and N3M6 pressure drop reduces as velocity increases when compared with N3B1. N6M2 is one of the best combination because it has much lower friction factor having friction factor 0.925. For N6M2 and N6B2 values reduce to 0.92 while it started from 0.96, N6M1 also have lower pressure drop when compared with N6B1 than N6B2. Friction factor for N6M3 & N6B1 reduces while friction factor for N6M3 & N6B2 increases. It is also monitored that N6M3 & N6B2 and N6M4 & N6B2 pressure drop increase in a similar fashion.

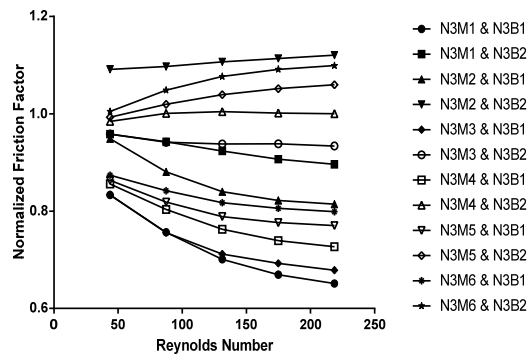


Fig 11: Normalized Friction Factor of water-vapor of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=3

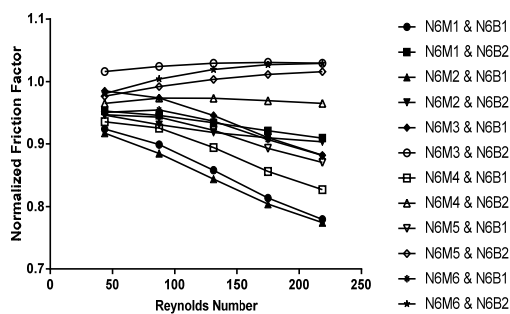
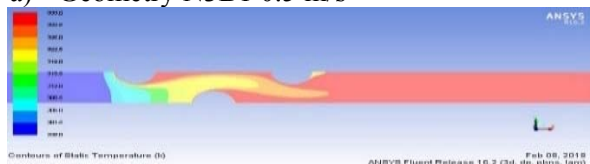


Fig 12: Normalized Friction Factor of water-vapor of Modified Heat Exchanger (HX) with corresponding Reynolds Number of Conventional HX for N=6

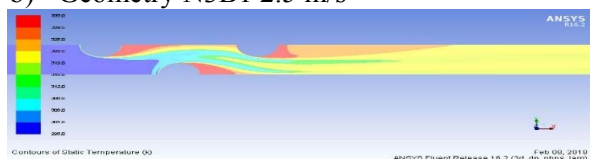
4.7 Temperature contour of air for various combination

Temperature contour means the distribution of temperature for air. In the front portion of fin, heat transfer is large because high temp difference but at rear portion heat transfer reduced because difference of tube surface temp and air is lower. Only one conventional and modified temp contour is shown here [18].

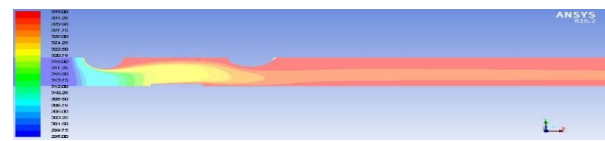
a) Geometry N3B1 0.5 m/s



b) Geometry N3B1 2.5 m/s



c) Geometry N3M1 0.5 m/s



d) Geometry N3M1 2.5 m/s

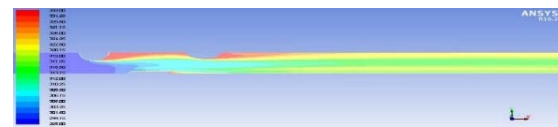


Fig 13: Temperature contour of air a) N3B1 at 0.5 m/s b) N3B1 at 2.5m/s c) N3M1 at 0.5m/s d) N3M1 at 2.5 m/s

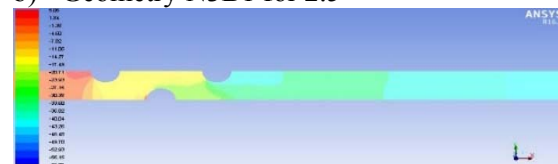
4.8 Pressure Contour for Air

Pressure contour of air three rows of tube as static pressure distribution. As the velocity increases pressure drop increases. Pressure contour shown for three rows of tube one conventional and another for modified heat exchanger [18].

a) Geometry N3B1 for 0.5 m/s



b) Geometry N3B1 for 2.5



c) Geometry of N3M3 at 0.5 m/s



d) Geometry of N3M3 at 2.5 m/s



Fig 14: Pressure contour for air a) N3B1 at 0.5 m/s b) N3B1 at 2.5 m/s c) N3M3 at 0.5 m/s d) N3M3 at 2.5 m/s

4.9 Temperature contour of water-vapor

Temperature distribution is shown for water-vapor for three rows of tube. There is little difference in outlet temp of water-vapor than air [18].

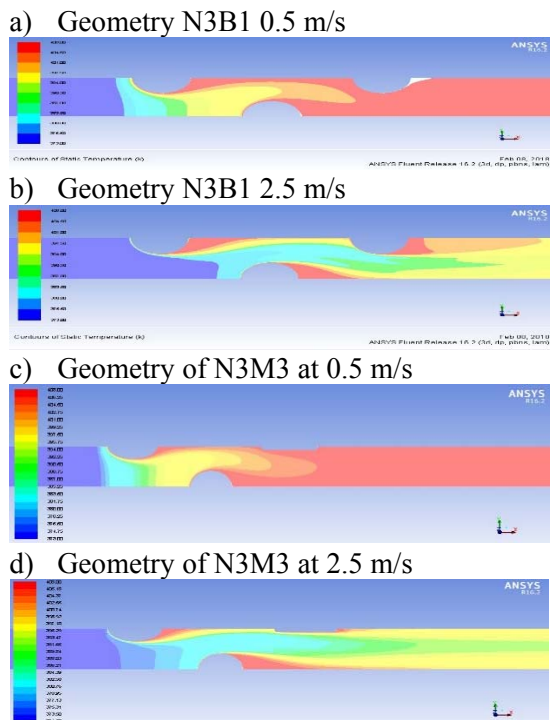


Fig 15: Temperature contour of water-vapor a) N3B1 at 0.5 m/s b) N3B1 at 2.5m/s c) N3M3 at 0.5 m/s d) N3M3 at 2.5 m/s

4.10 Pressure contour of water-vapor

Static pressure distribution are shown for water-vapor are presented for one baseline case another for modified case [18].

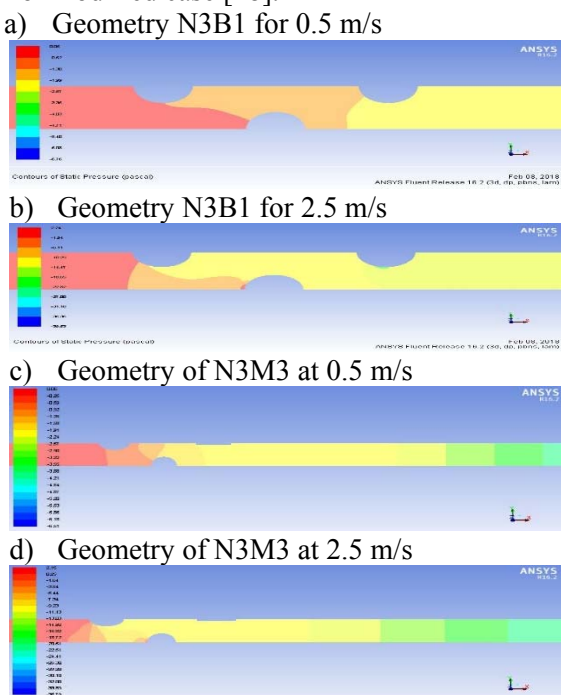


Fig 16: Pressure contour for water-vapor a) N6B1 at 0.5 m/s b) N6B1 at 2.5 m/s c) N3M3 at 0.5 m/s d) N3M3 at 2.5 m/s

4.11 Discussion

In this research work, several combinations of three rows of tube with circular, elliptical and rectangular tubes have been investigated. These tubes have also been used for six rows of tube. First of all, fluid is taken at the air. Then air is replaced with water-vapor. Present simulation has investigated with an established paper and compared with that paper in terms of pressure drop vs. velocity. It has been inspected from the graph of normalized Nu number and Re number that heat transfer for several geometries gives a high heat transfer rate than conventional grouped circular and elliptical tubes. For example for air geometry N3M6 performs better than conventional grouped elliptical tubes. It has been observed that as velocity increases pressure drop also increases. In contrary, friction factor decreases as velocity has increased being friction factor and velocity anti-proportional. For example, friction factor for geometry N3B1 at velocity 0.5 m/s was 0.95 and for velocity 2.5 m/s it was 0.37. For verification, the result has been compared with the heat transfer correlation for tube banks. 1.41 % of error occurred with the actual co-relation result. The results show that, at low inlet velocity for air, modified heat exchanger 4 and 6 have 7.54% and 4.59% increase in heat transfer co-efficient from the conventional circular tube heat exchanger when $N=3$. But heat transfer co-efficient is decreases 2.42% for modified heat exchanger 3 from grouped elliptical tube when $N=3$. When the number of tubes are six, for all the modified heat exchangers heat transfer co-efficient decrease almost 2-3% from grouped circular and elliptical tube heat exchanger. In case of water-vapor, modified heat exchanger 1 and 2 have 3.48% decrease in heat transfer from conventional circular tube heat exchanger. Also modified heat exchanger 1 and 2 have 2.11% decrease in heat transfer from conventional elliptical tube heat exchanger. On the contrary, at high inlet velocity, modified heat exchanger 2 have 10.45% higher from grouped elliptical tube heat exchanger, for modified heat exchanger 6 gives 5.80% higher heat transfer from grouped elliptical tube heat exchanger when $N=3$. This has been due to flow separation point of the fluid over the tube surface. As long as flow adhered to the tube surface, fluid was heated up. Vortex generator could have been used to delay flow separation.

5 Conclusion

Air side and water-vapor heat transfer characteristics for different arrangements of circular, elliptical and rectangular tubes, has been numerically investigated in the laminar flow region for multiple rows. The result is presented as follows: The heat transfer enhancement is being quite good for air, the geometry N3M2 and N3M5 than the grouped elliptical tube heat exchanger in case of three rows of tube. On the other hand, for six rows tube N6M3 and N6M6 has performed better than the grouped circular tube. In case of water-vapor, the heat exchanger with geometry N3M1 and N3M2 has been performed quite better than both circular and elliptical tubes. Similar performance has been observed for geometry N6M5 and N6M6 when compared with conventional heat exchanger. The frictional resistance of air for geometry N3M1 and N3M3 when compared with grouped circular tube. For six rows of tube, N6M5 acts better than circular tube and N6M6 also performs better than elliptical tubes alone. The frictional resistance for water-vapor N3M1 N3M3 and N3M4 all has performed better than grouped circular and elliptical tubes. N6M1 and N6M2 has acted better than grouped circular tube heat exchanger.

References

- [1] M. N. Ozisik, 1985. "Heat Transfer a Basic Approach", International Edition, 1985, McGraw-Hill Book Company. ISBN 0-07-066460-9.
- [2] T. Kuppan, "Design of a heat exchanger handbook", Third Edition, Marcel Dekker Inc. 2010.
- [3] F.P. Incropera, D.P. Dewitt, "Fundamentals of Heat & Mass Transfer" Seventh Edition, 2013, ISBN 13 978-0470-50197-9
- [4] Xie G, Wang Q, Sunden B, "Parametric study and multiple correlations on air-side heat transfer and friction characteristics of fin-and-tube heat exchangers with a large number of large-diameter tube rows", Applied Thermal Engineering 2009; 29:1-16
- [5] Guo ZY, Li DY, Wang BX, "A novel concept for convective heat transfer enhancement", International Journal Heat Mass Transfer 1998; 41:2221-25.
- [6] Li J, Wang S, Chen J, Lei YG, "Numerical study on a slit fin-and-tube heat exchanger with longitudinal vortex generators", International Journal Heat Mass Transfer 2011; 54:1743-51
- [7] Jayavel S, Shaligram Tiwari, "Effect of tube spacing on heat transfer performance of staggered tube bundles in presence of vortex generators", Journal Enhance Heat Transfer 2010; 17:271-91.
- [8] Jayavel S, Shaligram Tiwari, "Effect of vortex generators and integral splitter plate on heat transfer and pressure drop for laminar flow past channel-confined tube banks", Heat Transfer Engineering 2010; 31:383-94.
- [9] Jayavel S, Shaligram Tiwari, "Finite volume algorithm to study the effect of tube separation in flow past channel confined tube banks", Engineering Applied Fluid Mechanics 2010; 4:39-57.
- [10] Wu JM, Tao WQ, "Impact of delta winglet vortex generators on the performance of a novel fin-tube surfaces with two rows of tubes in different diameters", Energy Conversion Manage 2011; 52:2895-901.
- [11] Xie G, Wang Q, Sunden B, "Parametric study and multiple correlations on air-side heat transfer and friction characteristics of fin-and-tube heat exchangers with large number of large-diameter tube rows", Applied Thermal Engineering 2009; 29:1-16.
- [12] Lu CW, Huang JM, Nien WC, Wang CC, "A numerical investigation of the geometric effects on the performance of plate finned-tube heat exchanger", Energy Conversion Manage 2011; 52:1638-43.
- [13] Lotfi B, Sunden B, Wang QW, "An investigation of the thermo-hydraulic performance of the smooth wavy fin-and-elliptical tube heat exchangers utilizing new type vortex generators". Applied Energy. 2016; 162: 1282-1302.
- [14] Zukauskas, A. "Heat transfer from tubes in cross flow, in Handbook of single phase convective heat transfer", Wiley International science, 1987.
- [15] Rocha LAO, Saboya FE M, Vargas J VC, "A comparative study of elliptical and circular sections in one- and two-row tubes and plate fin heat exchangers", International Journal Heat Fluid Flow 1997; 18:247-252.
- [16] R. Deepakkumar, S. Jayavel, "Air side performance of finned-tube heat exchanger with combination of circular and elliptical tubes" Applied Thermal Engineering, 2017, S1359-4311(16)31949-4
- [17] Wang CC, Chi KY, Chang CJ, "Heat transfer and friction characteristics of plain finned-tube heat exchangers, part II: Correlation", International Journal Heat Mass Transfer 2000; 43:2693-7
- [18] H. Hasan, "Numerical Investigation of Finned-tube Heat Exchanger with Circular, Elliptical & Rectangular Tubes", A Project of B.Sc. Engg. (ME) degree, Department of Mechanical Engineering, Khulna University of Engineering & Technology, Khulna-9203, Bangladesh, February 2018.