RESEARCH ARTICLE OPEN ACCESS

CFD Analysis of Plate Fin Tube Heat Exchanger for Various Fin Inclinations

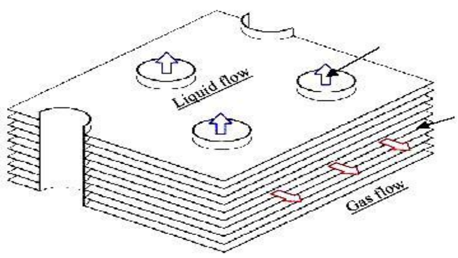
Subodh Bahirat, P. V. Joshi
Mechanical engineering department S.S.C.E.T Bhilai Chhattisgarh India
Mechanical Engineering Department S.S.C.E.T Bhilai Chhattisgarh India

Abstract
ANSYS Fluent software is used for three dimensional CFD simulations to investigate heat transfer and fluid flow characteristics of six different fin angles with plain fin tube heat exchangers. The numerical simulation of the fin tube heat exchanger was performed by using a three dimensional numerical computation technique. Geometry of model is created and meshed by using ANSYS Workbench software. To solve the equation for the fluid flow and heat transfer analysis ANSYS FLUENT was used in the fin-tube heat exchanger. The fluid flow and heat transfer are simulated and result compared for both laminar and turbulent flow models k-epsilon and SST k-omega, with steady state solvers to calculate heat transfer, flow velocity and temperature fields of variable inclined fin angles (□ $= 0^0, 10^0, 20^0, 30^0, 40^0, 50^0$). Model is validate by comparing the simulated value of velocity, temperature and colburn factor with experimental and numerical results investigated by WANG [1] and GHORI KIRAR [10]. Reasonable agreement is found between the simulations and other results, and the ANSYS Fluent software is sufficient for simulating the flow fields in tube fin heat exchanger.

Keywords— Plate fin Heat Exchanger, Computational Fluid Dynamics(CFD), ANSYS Work Bench, FLUENT.

I. Introduction

Heat exchanger is frequently used product in the profession as a result of offer heat transfer involving two fluids which can be in diverse temperatures in addition to divided by way of a reliable retaining wall. The plane fin-tube heat exchanger have been used in winter executive applications, including power stations, element facilities, meal sectors, heat-chilling methods, aircrafts, automotive areas, Nuclear executive, and many others. There are numerous fin types which used plate, louver, convex louver, wavy in addition to tube geometries which used spherical, elliptical, and many others. A plate fins continue to be the fin pattern in the fin-tube heat exchanger applications to its durability, ease, flexibility in addition to solidity. Any plate fin tube heat exchanger is actually a kind of heat exchanger which employs to transfer heat between two fluids. It's an essential gain more than a conventional heat exchanger because fluids experience the larger expanse for the reason that fluids spread out above the surfaces. This kind of heat exchanger allows for heat transfer from low to heat; in addition to drastically improve the rate in the heat alter. The plate heat exchanger (PHE) has been developed through Dr Richard Seligman inside 1923 in addition to revolutionized methods of oblique heating and cooling connected with fluids.



Turbulence

Turbulent flows are filled with swirling and spiralling motions. This is especially true if the object itself is spinning like a planet or star where the Coriolis Effect causes winds and currents to curve and wiggle around. Turbulence consists of fluctuations in the flow field in time and space. It is a complex process, mainly because it is three dimensional, unsteady and consists of many scales.

Most widely used Turbulence Models
- Standard k-ε Model
- Zero Equation Model
- RSM- (Reynolds Stress Model)
- RNG - (Re-normalized Group Model)
- NKE - (New k-ε Model due to Shih)
- Standard k-ω Model

Computational Fluid Dynamics (CFD)

CFD limitations tend to be set up round the statistical algorithms that could be tackle liquid complications. In order to offer easy accessibility on

their solving electrical power many business CFD plans include things like superior user interfaces suggestions problem guidelines in order to verify the outcomes.
Hence all codes contain three main elements:
1. Pre-processing.
2. Solver
3. Post-processing.

II. Literature Survey

Study of heat transfer and friction characteristics of typical louver fin-and-tube heat exchangers were done by Wang [1]. A fresh air straightener was useful to retain movement moving in the particular x-direction, the 8-thermocouple fine mesh was place into the actual inlet and a16-thermocouple fine mesh inside wall socket spots which established by ASHRAE suggestions. Just about all equipment related to information exchange thermocouples, stress transducer, venting way of measuring section, and movement meter were checked out intended for accuracy and reliability previous to operating the actual findings [2]. Mineral water while using the inlet occurred from 60ºC, circulation velocities were tested inside cover anything from 0. 3 m/s to 6m/s. a couple of m/s. Energy scales were checked during the test intended for the hot- and cold-side and claimed to get in two [3]. This questions for that main measurements (mass movement rate intended for fresh air and mineral water, anxiety fall, and heat of the mineral water and air) were tiny and as a consequence these measurements is usually presumed to get appropriate [4]. The exact performance associated with fin-and-tube heat exchangers tend to be connected to geometric details. Early on test outcomes reached through Prosperous [5]. Lu C. N. Highlighted the end results associated with geometric details which include tube pitch, tube width and tube dimension in depth. The best possible benefit intended for Q/ΔP seemed to be observed through statistical simulation [6]. Tang et al. examined the actual air-side heat transfer and friction attributes associated with 5 varieties of fins. In addition to, to be able to enhance performance, different types of techniques are utilized within finned tube heat exchangers [7]. He et al. employed winglet type of vortex generators to bolster air-side heat transfer performance. A statistical analyze was completed intended for boosting fresh air part heat transfer performance. Improved upon heat transfer rate was observed by employing winglet type of vortex power generator [8]. A different procedure was carried out through Tao et al. exactly who employed triangular wavy fins to produce the actual performance much better. It is usually noticed in which vortex generators and Samsung waves 8500 fins tend to be generally manufactured to boost the heat transfer for that air-side. These hydrophobic prosperities are important intended for chemistry apps [9]. Fluent software is used by Ghori and Kirar to investigate colburn factor and friction factor in a two-row simple tube in conjunction with Fin heat exchanger. Heat transfer and pressure drop is also found [10].

Research objectives:
1. Modeling of plate fin tube heat exchanger in ANSYS Workbench.
2. Considering laminar and turbulent fluid flow.
3. Apply laminar, k-epsilon and k-omega turbulent fluid flow model.
4. Applying two different velocity magnitudes which are 0.3m/s and 6.2m/s.
5. Obtain temperature distribution over the surface of fin.
6. Obtain Prandtl number and Stanton number, which will help to calculate colburn factor.
7. Obtain result for velocity, Nusselt number, heat transfer coefficient and heat transfer rate.

III. Problem description and Solution Method

Looking at this literature, many experts have found the influence of geometrical parameters with heat exchange as well as force decline to the numerous b kinds as well as tube geometries on the fin-tube heat exchangers have been researched over the new many years. Even so, these successful aspects of likely b angles with heat exchange as well as force decline around within a heat exchanger haven't recently been analyzed numerically. Possessing additional basic composition, adjustable likely fin point of view devices are generally simpler to fabricate as opposed to flue gasoline stream point of view devices for example louvered fins. In reality, making of louvered fin geometries changing flue gasoline stream angles used by heat exchange advancement inside heat exchangers will be more difficult as well as expensive as opposed to fin point of view technique. These types of adjustable fin point of view technique with all the exact same volume of fins seeing that immediately flue gasoline technique provide additional heat exchange advancement. In case of cutting down pertaining to immediately flue gasoline course additional b figures will probably have to provide you with the exact same heat exchange advancement. With this regard, adjustable likely fin angles utilized in ordinary platter tube heat exchangers were being considered to investigate. As varying via other work, the main aim of this examine would be to analyze pertaining to diverse likely fin angles by using a 3D statistical calculation approach, aid from this FLUENT, a new CFD personal computer signal. The result of these likely fin point of view conditions about the heat exchange as well as force decline were being researched for an airplane platter fin tube heat exchanger on this examine. The sort of stream taking place within water within a station is very important parameter which includes considering inside platter fin tube heat exchanger. The particular dimensionless Reynolds number is definitely a significant parameter inside equations which describe no matter if stream ailments bring about laminar or maybe turbulent stream.

Three flow models are considered to analyze the heat transfer rate of various inclined fin angles. Out of these three models one is laminar flow model and the other two are turbulent flow model. The three different models are:

1. Laminar flow model
2. K-epsilon flow model
3. K-omega flow model

Problem description: The plate fin tube heat exchanger shown in figure 1 is to be analyzed for velocity profile, temperature profile, Nusselt number, colburn factor and heat transfer rate. For the purpose of analysis a single plate is modeled in ansys workbench software as shown in figure 3. Various inlet angles are considered to analyses the temperature effect on colburn factor, Nusselt number, heat transfer coefficient and rate of heat transfer. Three CFD models namely laminar flow model, k-epsilon turbulent flow model and k-omega turbulent flow model are applied for laminar and turbulent flow. The results for velocity, temperature, colburn factor, Nusselt number and heat transfer rate obtained from three models with various inclinations and two different velocities are presented and compared.

Procedure:

The ANSYS WORKBENCH software is used to create and mesh the computational model. Diagram of studied model is shown in figure3. The diagram consists of inclination of fluid inlet, fin geometry, air flow area between two fins and surface of rows of tubes. A schematic of the model with dimension is shown in figure2, with the geometrical dimensions listed in table 1.

Table 1: Geometrical dimension for fin and tube

Fin thickness	T	0.130mm
Fin pitch	Fp	2.240mm
Fin collar outside diameter	Dc	10.23mm
Transverse pitch	P_t	25.40mm
Longitudinal pitch	P_l	22.00mm
Tube wall thickness	Δ	0.336mm

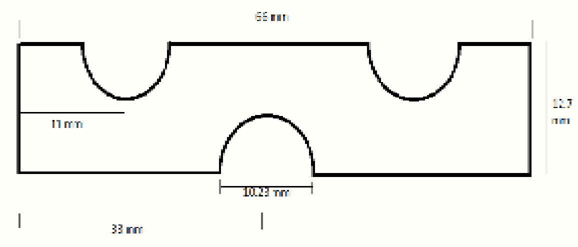


Figure2: Dimension of fin with three tubes in staggered arrangement.

Model Geometry: The very first step is to make a flow model in any computational heat transfer problem. In geometry section part of fin tube heat exchanger is drawn. Five different geometries of fin inclination and one straight fin geometry are modeled. During the fluid simulation setup it is necessary to define boundary conditions to apply specific physics. It is important to define where the air enters the geometry or where it exists. We can define the location of boundaries by defining name sections, such as inlet, outlet, fin, tube, symmetry, wall etc. as shown in figure 2.

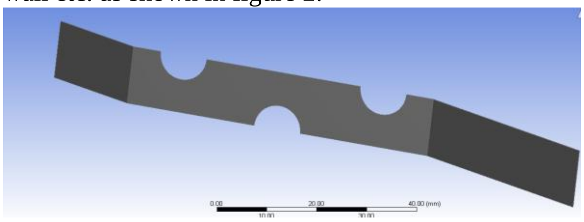


Figure3: Fin model with inlet and outlet in ANSYS WORKBENCH

Mesh: Meshing is the second step in which the domain is divided into number of small elements. The mesh generation process in the meshing section is fully automatic. The background element size, type of mesh to generate and mesh refinement is the available computing resources by which we can get best possible fluid solution. Body and face meshing is done for domain and 808580 nodes and 3288942 elements are created. Mesh size is shown in figure4.

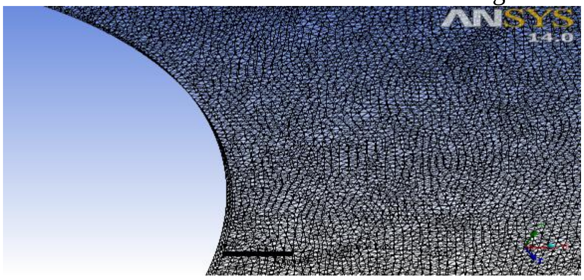


Figure4: Meshing of two fin segment.

Setup: The computation domain contains the boundary conditions. These boundary conditions are described in this section. 3D version of fluent is open for simulation of computation domain. Than model solver has to be define in which segregated solver, implicit formulation, space steady and time steady are considered. Model solver is defined in which three different models are selected one by one. These three models are LAMINAR FLOW MODEL, K-EPSILON TURBULENCE FLOW MODEL and K-OMEGA TURBULENCE FLOW MODEL. The next step is to select the material and its property. Fin and tube material is selected as aluminium and the air is passing between two fins. Default property of solid and fluid are considered from fluent software.

The solid material selected was ALUMINIUM with properties.

Dynamic Viscosity, μ = 1.7894 x 10-5
Density, = 2719 kg/m^3
Thermal Conductivity, K=202.4 W/mK
Specific heat, Cp= 871 J/kg- K
The fluid material selected was AIR with properties.
Dynamic Viscosity, μ = 1.7894 x 10-5
Density, = 1.225 kg/m^3
Thermal Conductivity, K=0.0242 W/mK
Specific heat, Cp= 1.225 J/kg- K

After selecting the fluid properties operating and boundary conditions are to be selected. At the inlet velocity inlet function is selected and the velocity magnitude of air is taken as 0.3m/s and 6.2m/s for different cases. At the same time temperature of air is selected as 278K. For outlet conditions pressure outlet is selected and the operating pressure is about 101.325KPa. Fin 1 and Fin 2 are considered as wall. Space between fin1 and fin 2 is considered as symmetry. Tubes are also considered as wall at 333K temperature.

Solution: In solution process some solution controls under relaxation factors are to be considered. The Pressure= 0.3, Density= 1, Body Force= 1, Momentum= 0.7, Turbulent Kinetic Energy= 0.8, Energy= 1 are taken as default from fluent software. Than the solution is initializing with standard initialization method with following initial values.
Compute from –all zones
Reference frame – relative to cell zone
Initial values
Gauge pressure = 0.3281 Pa
X Velocity = 0.301 m/s
Turbulent kinetic energy= 1 m^2/s^2
Temperature=321.77 k
After initialize the solution, calculation has to be run to calculate the equations and iterate the calculation for 100 times for better results.

Results: After run the calculation fluent software calculate the different properties of interest and result is shown in the form of contour. Filled contour of velocity, pressure, temperature and kinetic energy are saved in the form of image. Prandtl number, Nusselt number and Stanton number are also observed for calculate the heat transfer rate. Result can also be saved in form of Graphs and animations.

IV. Result and Discussion

Validation:

The ansys fluent software provides the velocity contour, temperature distribution over fin surface, Nusselt number, colburn factor and rate of heat transfer for laminar flow model, k-epsilon and k-omega turbulent flow model with various fin inclination. For validation purpose the colburn factor for zero degree inclination of the entire three flow model is compared with the Wang [1] and Ghori [55]. A good agreement is found between experimental, numerical and simulation result. Reynolds number is calculated with the help of fluid velocity and with the help of Prandtl number and Stanton number colburn factor (j) is calculated. Relation between colburn factor and Reynolds number is presented in below graph.

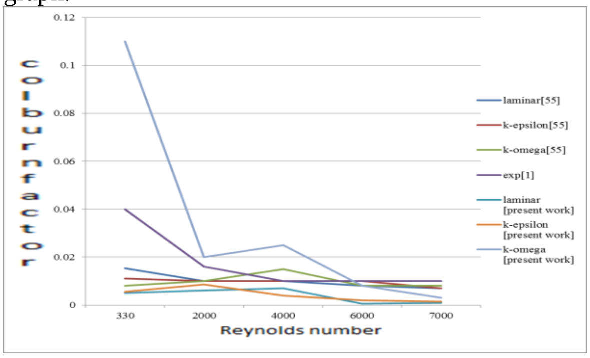


Figure 5: Shows validation graph for zero degree inclination

Velocity Observation:

The air enters at the inlet on the left and exits at the out let on the right hand side. Laminar, k-epsilon and k-omega turbulence models are presented with different inclination of fin. With the help of these model contours we can easily understand the flow characteristics of air in heat exchanger between two fins. The two different magnitude of velocity i.e. 0.3 m/s and 6.2 m/s are taken in consideration.

1) For velocity inlet 0.3 m/s :

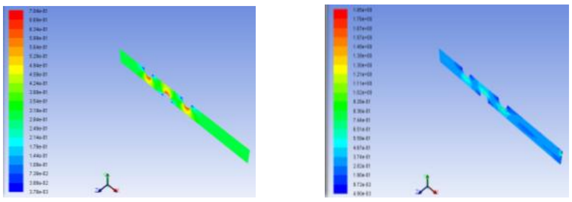
In zero degree inclined laminar velocity contour the velocity is greater near the tube surface. This is because of the free flow area of air decreases, which showed that the velocity going around the tube is faster than that going around the other area. The flow is forced to speed up, as the tube is act as a type of pipe contraction in the air flow channel. The pressure of the air decreases due to the velocity of air is increases. Figure 6(a) shows that the velocity near the tube surface is about 0.71 m/s, whereas the initial velocity is about 0.3m/s. In figure 6(d) ten degree inclined laminar flow velocity contour the velocity is decreases due to inclination of fin inlet and then increases about 0.62 m/s near the tube surface. Figure 6(g), 6(j), 6(m) and 6(p) showing the laminar velocity contour at twenty, thirty, forty and fifty degree inclination of fin and it is observed that the velocity is about 0.61, 0.60, 0.61 and 0.63 m/s. Velocity near the tube surface is decreasing with increase of inclination of fin up to thirty degree. But in case of forty and fifty degree the velocity is increasing.

In zero degree inclination k-epsilon and k-omega model it is observed that the velocity contour are quiet similar. It is observed that the velocity is low near the first tube, & it continues increases near the second and the third tube. For ten degree inclination the velocity is greater than the zero degree inclination k-epsilon and k-omega turbulence model. At twenty degree inclination velocity is ten times greater than the ten degree inclination k-epsilon and turbulence model. Similarly for thirty degree inclination the velocity is about 20 m/s near the tube surface. Figure 6(q) and 6(r) shows the velocity contour at fifty degree fin inclination of k-epsilon & k-omega turbulence model respectively. It is clearly observed in the figure the velocity is high near the second tube and it is increasing near the third tube. The maximum velocity is found at thirty degree inclination near the third tube.

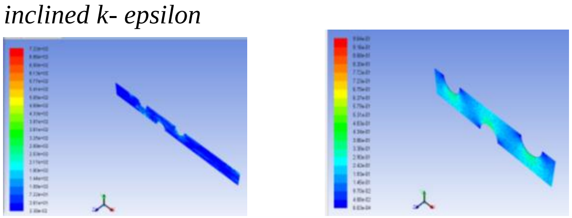
2) For velocity inlet 6.2 m/s:

In zero degree inclined laminar velocity contour the velocity is greater near the tube surface. This is because of the free flow area of air decreases, which showed that the velocity going around the tube is faster than that going around the other area. The flow is forced to speed up, as the tube is act as a type of pipe contraction in the air flow channel. The pressure of the air decreases due to the velocity of air is increases. Figure 7(a) shows that the velocity near the tube surface is about 9.56 m/s, whereas the initial velocity is about 6.6 m/s. In figure 7(d) ten degree inclined laminar flow velocity contour the velocity is decreases due to inclination of fin inlet and then increases about 10 m/s near the tube surface. Figure 7(g), 7(j), 7(m) and 7(p) showing the laminar velocity contour at twenty, thirty, forty and fifty degree inclination of fin and it is observed that the velocity is about 11.8, 14.0, 10.1 and 9.87 m/s. Velocity near the tube surface is decreasing with increase of inclination of fin up to thirty degree. But in case of forty and fifty degree the velocity is increasing.

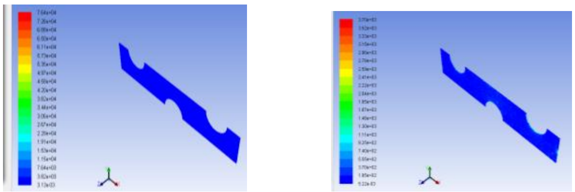
In zero degree inclination k-epsilon and k-omega model it is observed that the velocity contour are similar. It is seen that the maximum velocity is 17.9 and 22.5 m/s in k-epsilon and k-omega turbulence model respectively is much higher than the laminar flow model. For ten degree inclination the velocity is greater than the zero degree inclination k-epsilon and k-omega turbulence model. At twenty degree inclination velocity is ten times greater than the ten degree inclination k-epsilon and turbulence model. Similarly for thirty degree inclination the velocity is about 20 m/s near the tube surface. Figure 7(q) and 7(r) shows the velocity contour at fifty degree fin inclination of k-epsilon & k-omega turbulence model respectively. It is clearly observed in the figure the velocity is high near the second tube and it is increasing near the third tube. The maximum velocity is found at thirty degree inclination near the third tube.



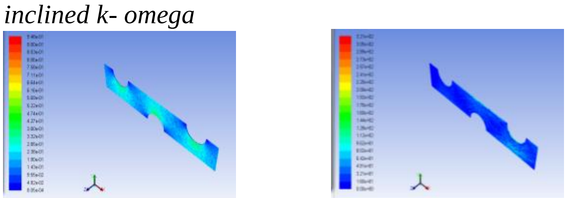
(a)zero degree inclined laminar (b) zero degree inclined k- epsilon



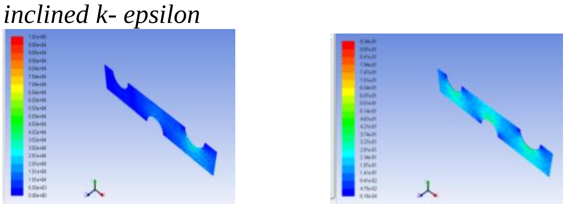
(c)zero degree inclined k- omega (d) ten degree inclined laminar



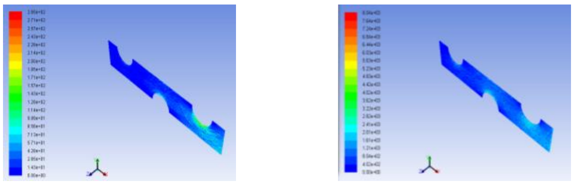
(e) ten degree inclined k- epsilon (f) ten degree inclined k- omega



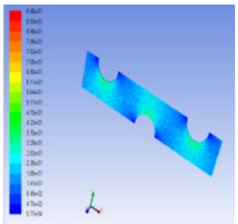
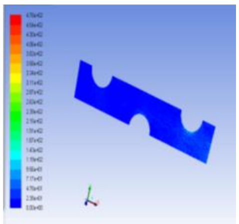
(g) twenty degree inclined laminar (h) twenty degree inclined k- epsilon



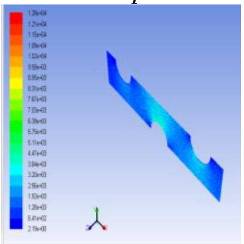
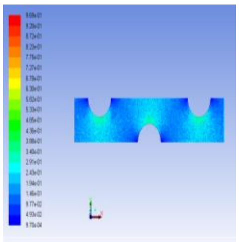
(i) twenty degree inclined k- omega (j) thirty degree inclined laminar



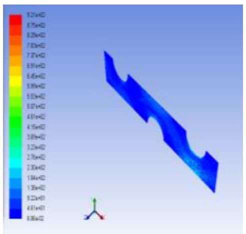
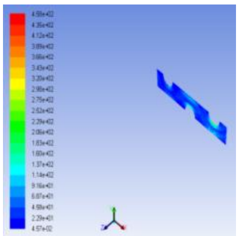
(k)thirty degree inclined k-epsilon (l) thirty degree inclined k- omega

(m) forty degree inclined laminar (n) forty degree inclined k- epsilon

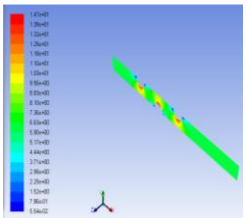
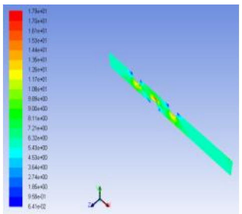



(o) forty degree inclined k- omega (p) fifty degree inclined laminar

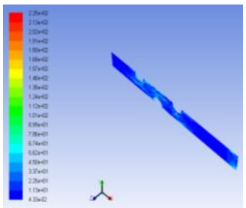
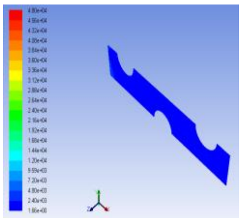



(q) fifty degree inclined k- epsilon (r) fifty degree inclined k- omega

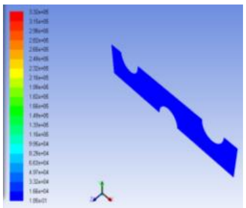
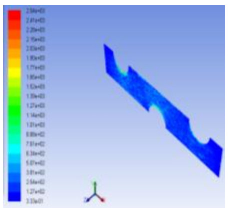
Figure 6: Velocity contours at various inclination of fin for 0.3 m/s velocity inlet

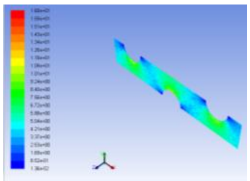
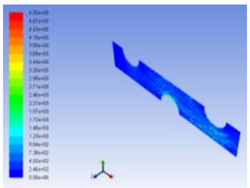
(a)zero degree inclined laminar (b) zero degree inclined k- epsilon

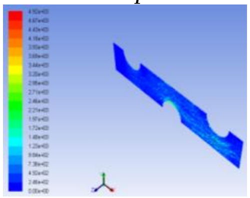
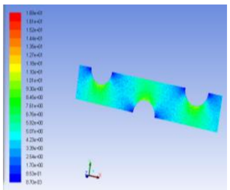
(c) zero degree inclined k- omega (d) ten degree inclined laminar

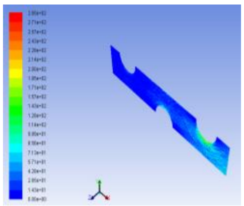
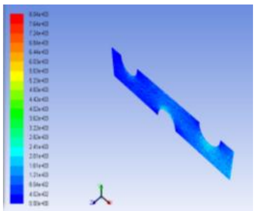
(e) ten degree inclined k- epsilon (f) ten degree inclined k- omega

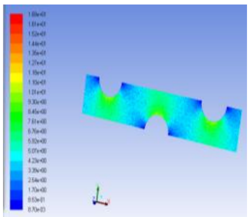
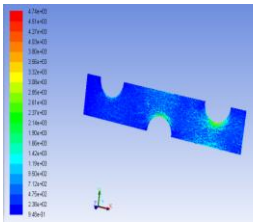
(g) twenty degree inclined laminar (h) twenty degree inclined k- epsilon

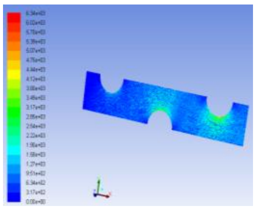
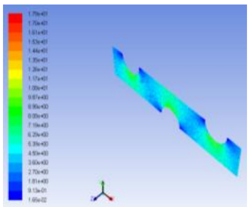
(i) twenty degree inclined k- omega (j) thirty degree inclined laminar

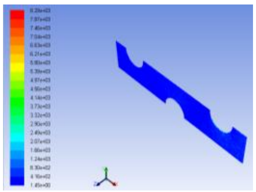
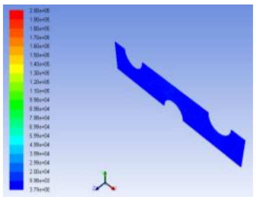
(k) thirty degree inclined k- epsilon (l) thirty degree inclined k- omega

(m) forty degree inclined laminar (n) forty degree inclined k- epsilon

(o) forty degree inclined k- omega (p) fifty degree inclined laminar

(q) fifty degree inclined k- epsilon (r) fifty degree inclined k- omega

Figure 7: Velocity contours at various inclination of fin for 6.2 m/s velocity inlet

Heat Transfer Rate: At different fin angle with laminar and turbulent flow model heat transfer coefficient and heat transfer rate is calculated in table2 and table3. In table2 the Nusselt number is ranging from 101 to 425 for 0.3m/s velocity magnitude, heat transfer coefficient and rate of heat transfer is compared for laminar and turbulent flow model of zero to fifty degree fin inclinations. For zero degree inclination maximum heat transfer rate 37.96W between two fin segments is found in k omega flow model. In laminar flow model heat transfer rate is 9.58W and for k-epsilon model is 29.04W, is less than k-omega model. For ten degree inclination heat transfer rate for three different models are 9.7W, 30.8W and 39W respectively. For twenty degree inclination it is about 10.W, 30.9W and 39.85W respectively. For thirty degree inclination heat transfer rate is about 10.53W, 31.79W and 40.23W. Maximum heat transfer rate is found in thirty degree inclination k-omega model. For two fin segment rate of heat transfer is about 40.3W. For forty and fifty degree inclination heat transfer rate is again decreases

In table3 Nusselt number is ranging from 376 to 924 for velocity magnitude of 6.2m/s. for zero degree inclination Nusselt number for three different models are 376, 586 and 668, for these Nusselt number heat transfer rate is 35.6W, 55.6W and 63.3W. For ten degree inclination increase heat transfer rate is about 39W, 57.4W and 64.7W. For twenty degree fin inclination it is 60.6W, 60.2W and 65.6W. For thirty degree inclination Nusselt number is decreasing and heat transfer rate is about 42.9W, 60.7W and 67.3W. For forty degree inclination more heat transfer rate is found in laminar flow model rather than k-epsilon and k-omega model. Fifty degree laminar flow model is having maximum heat transfer rate 87.69W.

***Table 2 Heat transfer coefficient and rate of heat transfer*.**

INLET VELOCITY= 0.3 m/s					
S.NO.	**INLET ANGLE**	**MODEL**	**NUSSELT NUMBER Nu=(h*D)/k**	**HEAT TRANSFER COEFFICIENT h=Nu*k/D**	**HEAT TRANSFER RATE Q=h*A*t**
1	0 DEGREE INCLINED	LAMINAR	101	238.865	9.58517472
2		K-EPSILON	306	723.69	29.040232
3		K-OMEGA	400	946	37.961088
4	10 DEGREE INCLINED	LAMINAR	103	243.595	9.77498016
5		K-EPSILON	317	749.705	30.084162
6		K-OMEGA	411	972.015	39.0050179
7	20 DEGREE INCLINED	LAMINAR	109	257.785	10.34439648
8		K-EPSILON	326	770.99	30.93828672
9		K-OMEGA	420	993.3	39.8591424
10	30 DEGREE INCLINED	LAMINAR	111	262.515	10.53420192
11		K-EPSILON	335	792.275	31.7924112
12		K-OMEGA	425	1005.125	40.333656
13	40 DEGREE INCLINED	LAMINAR	115	271.975	10.9138128
14		K-EPSILON	330	780.45	31.3178976
15		K-OMEGA	424	1002.76	40.238753
16	50 DEGREE INCLINED	LAMINAR	115	271.975	10.9138128
17		K-EPSILON	348	823.02	31.12809216
18		K-OMEGA	328	775.72	39.3846288

Table 3 Heat transfer coefficient and rate of heat transfer

INLET VELOCITY= 6.2 m/s					
S.NO.	**INLET ANGLE**	**MODEL**	**NUSSELT NUMBER Nu=(h*D)/k**	**HEAT TRANSFER COEFFICIENT h=Nu*k/D**	**HEAT TRANSFER RATE Q=h*A*t**

1	0 DEGREE INCLINED	LAMINAR	376	889.24	35.68342272
2		K-EPSILON	586	1385.89	55.61299392
3		K-OMEGA	668	1579.82	63.39501696
4	10 DEGREE INCLINED	LAMINAR	411	972.015	39.00501792
5		K-EPSILON	605	1430.825	57.4161456
6		K-OMEGA	682	1612.93	64.72365504
7	20 DEGREE INCLINED	LAMINAR	639	1511.235	60.64283808
8		K-EPSILON	635	1501.775	60.2632272
9		K-OMEGA	692	1636.58	65.67268224
10	30 DEGREE INCLINED	LAMINAR	452	1068.98	42.89602944
11		K-EPSILON	640	1513.6	60.7377408
12		K-OMEGA	710	1679.15	67.3809312
13	40 DEGREE INCLINED	LAMINAR	649	1534.885	61.59186528
14		K-EPSILON	640	1513.6	60.7377408
15		K-OMEGA	698	1650.77	66.24209856
16	50 DEGREE INCLINED	LAMINAR	924	2185.26	87.69011328
17		K-EPSILON	614	1452.11	58.27027008
18		K-OMEGA	694	1641.31	65.86248768

Plot of heat transfer rate versus Nusselt number: Figure 8 shows graph between Nusselt number and heat transfer rate for laminar flow model at velocity magnitude 0.3m/s. Figure 9 shows the value of heat transfer rate and Nusselt number at different inclination of laminar flow model. Value of Reynolds number is ranging from 101 to 115 and value of heat transfer is between 9.58W to 10.9W. Figure 10 describes the relation between Nusselt number and heat transfer rate for k-epsilon flow model. Figure 11 describes the value of Nusselt number and heat transfer at various inclinations for k-epsilon flow model. Value of Nusselt number is ranging from 306 to 335 for which heat transfer rate is between 29W to 31.7W. Figure 12 shows the relation between Nusselt number and heat transfer rate for k-omega flow model at velocity magnitude 0.3m/s. The dependence of heat transfer rate on Nusselt number can be observed in figure 14. Figure 13 shows the value of 37.9W for Nusselt number 400 and the maximum heat transfer rate 40.33 for Nusselt number 425.

Figure 14 to figure 19 shows the relation between Nusselt number and heat transfer rate. Different fin angles are also considered to observe the relation between heat transfer and Nusselt number. In figure 15 Nusselt number ranging between 376 to 924 and the value of heat transfer is between 35.6W to 87.7W. At various inclinations of fins minimum Nusselt number is 586 and maximum 640. Heat transfer for the same is ranging between 55.6W to 60.73W in figure 17. Figure 19 shows the maximum heat transfer 67.38W for k-omega model at Nusselt number 710.

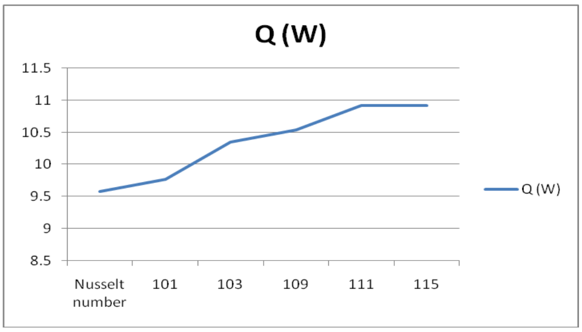


Figure 8: Relation between Nusselt number and heat transfer rate for laminar flow model and 0.3m/s velocity magnitude.

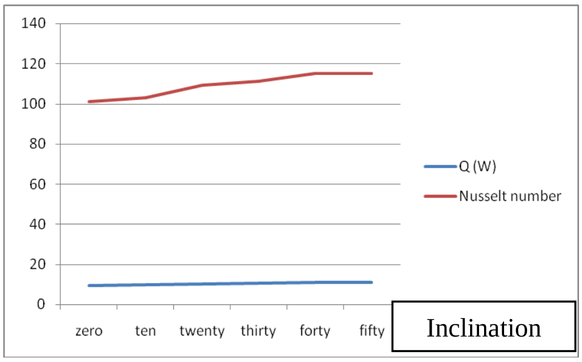


Figure 9: Relation between Nusselt number and heat transfer rate at various inclinations for laminar flow model and 0.3m/s velocity magnitude.

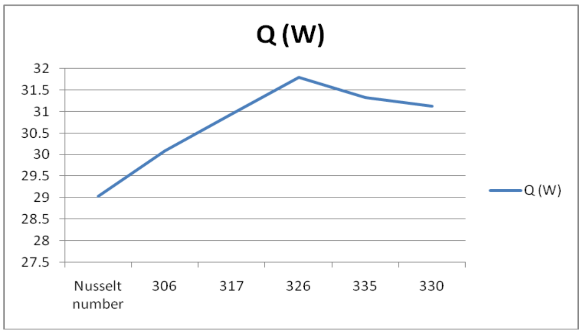


Figure 10: Relation between Nusselt number and heat transfer rate for k-epsilon flow model and 0.3m/s velocity magnitude.

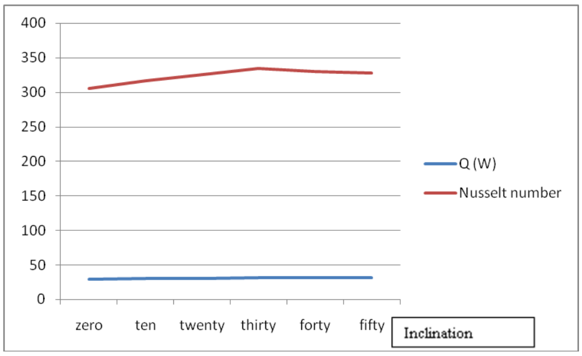


Figure 11: Relation between Nusselt number and heat transfer rate at various inclinations for k-epsilon flow model and 0.3m/s velocity magnitude.

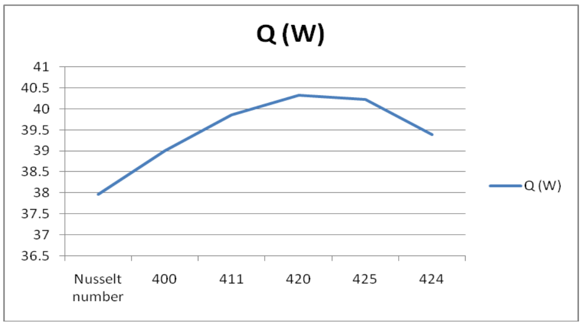


Figure 12: Relation between Nusselt number and heat transfer rate for k-omega flow model and 0.3m/s velocity magnitude.

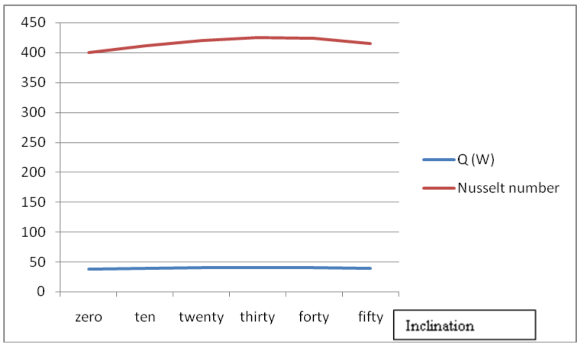


Figure 13: Relation between Nusselt number and heat transfer rate at various inclinations for k-omega flow model and 0.3m/s velocity magnitude.

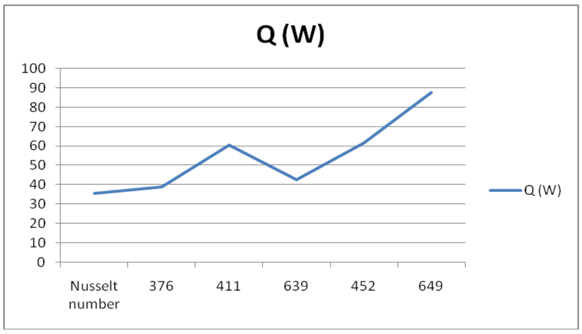


Figure 14: Relation between Nusselt number and heat transfer rate for laminar flow model and 6.2m/s velocity magnitude.

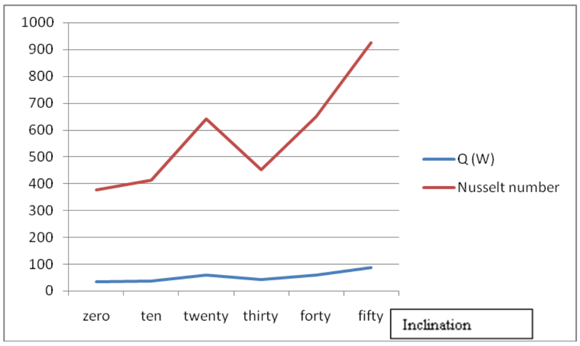


Figure 15: Relation between Nusselt number and heat transfer rate at various inclinations for laminar flow model and 6.2m/s velocity magnitude.

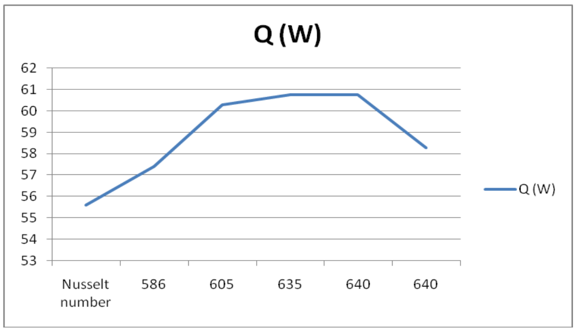


Figure 16: Relation between Nusselt number and heat transfer rate for k-epsilon flow model and 6.2m/s velocity magnitude.

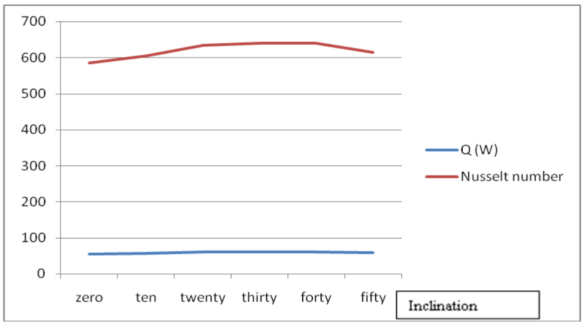


Figure 17: Relation between Nusselt number and heat transfer rate at various inclinations for k-epsilon flow model and 6.2m/s velocity magnitude.

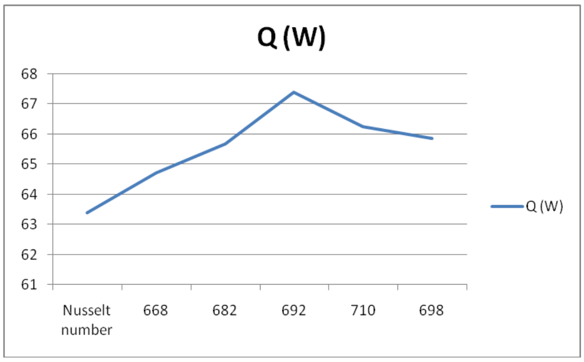


Figure 18: Relation between Nusselt number and heat transfer rate for k-omega flow model and 6.2m/s velocity magnitude.

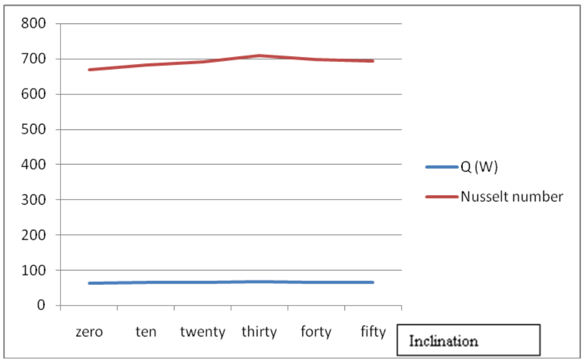


Figure 19: Relation between Nusselt number and heat transfer rate at various inclinations for k-omega flow model and 6.2m/s velocity magnitude.

V. Conclusion and Future Scope

Simulations with this study were carried out there following as closely as is possible the same operating condition and geometrical configurations in the two-row tube-fin heat exchanger, having tube collar diameter regarding 10. 23 mm along with fin pitch 2. 3mm, as presented within the paper by M. V. Ghori, R. K. Kirar [10]. This Reynolds number ranges coming from 330 to 7000, which match the frontal air velocity on the inlet ranging from 0. 3m/s to 6.2m/s. The work done with this project has shown that you possibly can make practical simulations regarding heat flow and heat transfer rate for a tube-and-fin heat exchanger utilizing fluent software, and validate the final results. Data resulting from the simulations need to be as accurate as possible, and therefore some considerations could be taken in future work to attempt to further improve the simulation conditions or calculations as well as the accuracy of the effects. In this dissertation just standard k-epsilon and SST k-omega turbulence model is used but in future realizable k-epsilon turbulence model and Reynolds stress model can be utilized for better result along with time variant solver they can double for a validation regarding simulation result.

REFRENCES

[1] CC. Wang *An experimental study of heat transfer and friction characteristics of typical louver fin-and-tube heat exchangers* International Journal of Heat and Mass Transfer Volume 41, Issues 4–5, February–March 1998, Pages 817-822

[2] CC. Wang, Chi-Chuan; Chang, Yu-Juei; Hsieh, Yi-Chung; Lin, Yur-Tsai (1996). "*Sensible heat and friction characteristics of plate fin-tube heat exchangers having plane fins*", International Journal of Refrigeration, Vol. 19, No. 4: pp. 223-230.

[3] Chi-Chuan Wang, Kuan-Yu Chi (2000). *Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, part I: new experimental data*, International Journal of Heat and Mass Transfer 43: 2681–2691

[4] Erek, Aytunc; Õzerdem, Baris; Bilir, Levent; Ilken, Zafer(2005). "*Effect of geometrical parameters on heat transfer and pressure drop characteristics of plate fin and tube heat exchangers*", Applied Thermal Engineering, Vol. 25: pp. 2421-2431.

[5] Rich DG, 1973, *The Effect of Fin Spacing on The Heat Transfer and Friction Performance of Multi-row, Smooth Plate Fin-and-tube Heat Exchangers*. American Society of Heating Refrigerating and Air conditioning Engineer, 81(1): 307-317.

[6] Lu C.W., Huang J.M., Nien W.C., Wang C.C.,2011, *A numerical investigation of the geometric effects on the performance of plate finned-tube heat exchanger*. Energy Conversion and Management, 52(3): 1638-1643.

[7] Tang L.H., Zeng M., and Wang Q.W., 2009, *Experimental and numerical investigation on air-side performance of fin-and-tube heat exchangers with various fin patterns*. Experimental Thermal and Fluid Science, 33(5): 818-827.

[8] He Y.L., Han H., Tao W.Q., Zhang Y.W., 2012, *Numerical study of heat-transfer enhancement by punched winglet-type vortex generator arrays in fin-and-tube heat exchangers*. International Journal of Heat and Mass Transfer, 55(21-22):5449-5458.

[9] Tao Y.B., He Y.L., Huang J., Wu Z.G., Tao W.Q., 2007, *Numerical study of local heat transfer coefficient and fin efficiency of wavy fin-and-tube heat exchangers*. International Journal of Thermal Sciences, 46(8):768-778.

[10] M. V. Ghori, R. K. Kirar, "*Numerical analysis of tube fin heat exchanger using fluent.*" ISSN: 2319-3182, Vol. 1, Issue 2, 2012.