NUMERICAL ENHANCEMENT OF HEAT TRANSFER OF FIN AND TUBE COMPACT HEAT EXCHANGER USED IN AEROSPACE USING CFD

AnuragChoubey, Prof. D.S. Rawat[2]
M.Tech scholar[1], Assistant Prof. Mechanical department[2]
Jabalpur Engineering College, Jabalpur

Abstract

- In the present study, the hydrodynamic and heat transfer characteristics of compact fin andtube heat exchangers have been investigated numerically. The aim is to analyze the influence of operating conditions and the geometry parameters of compact heat exchanger to design more efficient heat transfer device. Fin spacing and fin thickness are the geometrical parameters.The over-fin fluid velocity or Reynolds number is used as the parameter of operation. Here in this work it finds the effect of different material, fin thickness and fin spacing on heat transfer enhancement of the compact heat exchanger. Here in this work four different Reynolds number that is 2622.96, 5248.98, 7873.25 and 10497.96 were considered to analyze the effect of different velocity of fluid. It also analyzed the effect of different materials used for the construction of Fin and tube, in this work three different steel alloys that is GH2132, GH3044 and S66280 were considered for the analysis in which GH3044 shows the better heat transfer as compared to other.For GH3044 material heat transfer rateis 15 % more than the GH2132 material. Whereas domain having 0.8 mm fin spacing shows 5 % increase in heat transfer as compared to 1.1 mm fin spacing.

1. Introduction

The speed of aircraft has increased from subsonic to supersonic over the last decades because of the advent and development of the aircraft jet engine. Recently, the technologic research on the aerospace plane made a higher demand of speed (up to 5 Mach) and as the result, the engine for hypersonic aircraft become a challenge of 21st century. As modern gas turbine used in aero engines, to achieve higher thermal efficiency, the inlet temperature of turbine increases and increasing pressure ratio inside the compressor are most commonly used methods. With the development of engine technology, current turbine inlet temperature is far beyond the allowable metal temperatures range, which is approaching near about 2000 K. As the material properties development lagged behind the demand of practical application, turbine inlet temperature increased further by using highly sophisticated cooling techniques. Fin-tube compact heat exchangers show potential applications in aero engines for their high efficiency. As compared with the traditional heat exchangers used on the ground, those used in aero engines are more compact and suffer higher temperatures and larger temperature differences. The temperature change over the exchanger depth and the temperature gradient in the near wall region are more conspicuous, they may reach several hundred degrees.

Plate fin-and-tube heat exchangers are employed in a wide variety of engineering applications such as air-conditioning equipment, process gas heaters, and coolers. Generally, the heat exchangers consists of a plurality of equally spaced parallel tubes through which a heat transfer medium such as water, oil, or refrigerant is forced to flow while a second heat transfer medium such as air is directed across the tubes in a block of parallel fins. In such types of heat exchangers, continuous and plain or specially configured fins are used on the outside of the array of the round tubes of staggered or in-lined arrangement passing perpendicularly through the plates to improve the heat transfer coefficient on the gas side. The heat transfer between the gas, fins and the tube surfaces is determined by the flow structure which is in most cases three-dimensional. In practical applications, the dominant resistance is usually on the air side which may be 10 times larger than that of the tube side. Hence to improve the overall heat transfer performance, the use of enhanced surfaces is very popular in air-cooled heat exchangers, although a continuous plain fin is still a commonly used configuration where low pressure drop characteristics are desired.

Here in this work, 3D numerical simulation of plane fin tube heat exchanger is performed and finds the effect of different material used for the construction of tube and fin to enhance the heat transfer rate. Here it also analyzed the effect of fin spacing and fin thickness on heat transfer of plane fin tube heat exchanger.

2. Model Specification

In order to achieve the above objective here first it develop the solid model of heat exchanger based on the geometry used given in Lingdong et.al [1] the geometric specification of heat exchanger used in the analysis is defining the tube bank configurations include the tube outside diameter (D), transverse tube pitch (P_t), longitudinal tube pitch (P_l), and number of tube rows (N). They are taken to be D = P_l = 3.0 mm, P_t = 6.0 mm and N = 12 in this research. The plain finned tube configuration involves additional parameters including the fin pitch (F_p) and fin thickness (d_f), which are specified to be F_p = 1.1 mm and d_f = 0.1 mm. due to the periodicity and symmetry of the heat exchanger geometry for numerical analysis it considered the two dimensional that is 2D airflow passage as shown in the fig. here the solid model of the heat exchanger is prepared in the design modular of Ansys.

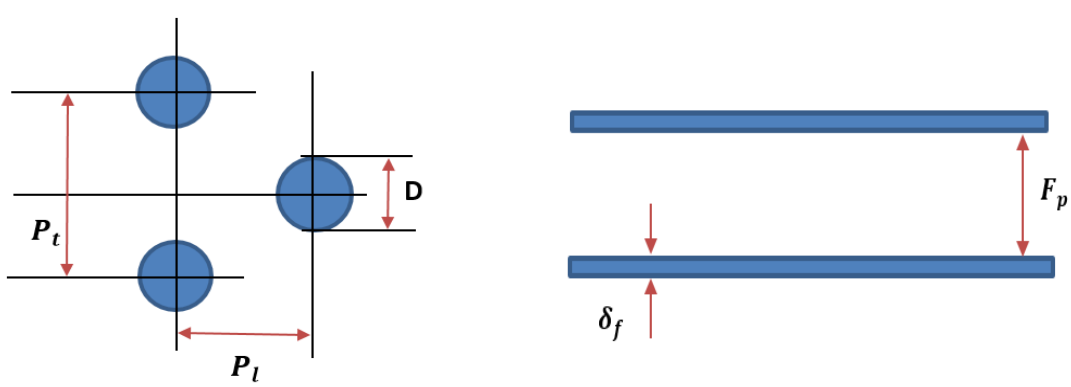

Fig.1 geometric condition of tube

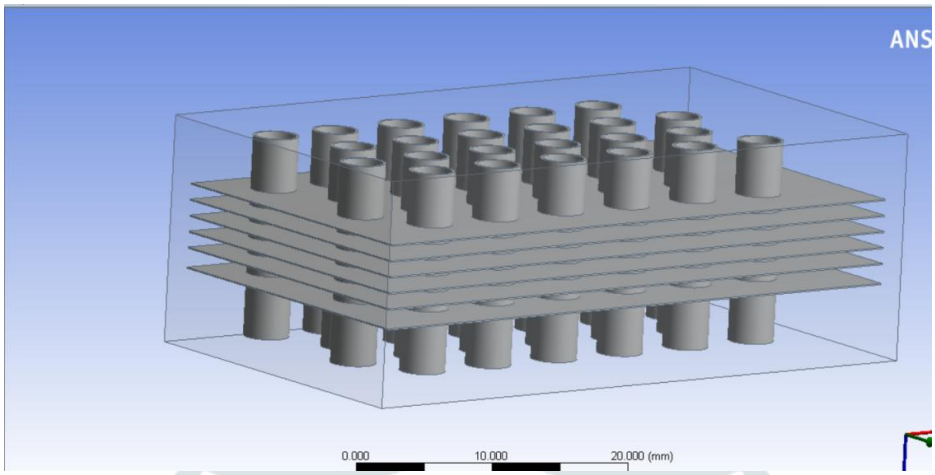

Fig.2 Solid model of complete computational domain

Figure 2 shows computational domain for plain-fin staggered configuration along with the co-ordinate system and the boundary conditions used. The boundaries of the computational domain include inlet and outlet boundaries, symmetry planes and solid walls. The computational domain extends farther than the heat exchanger to reduce the numerical oscillations.The fin inside this computational domain which used to increase the heat transfer rate. Considering the complete geometry for the numerical analysis increases the number of elements which increases the computational time required for the analysis, so here we have considered the small section in between the two rows of staggered tube arrangement. The computational domain used for the numerical analysis is shown in the below fig.3.

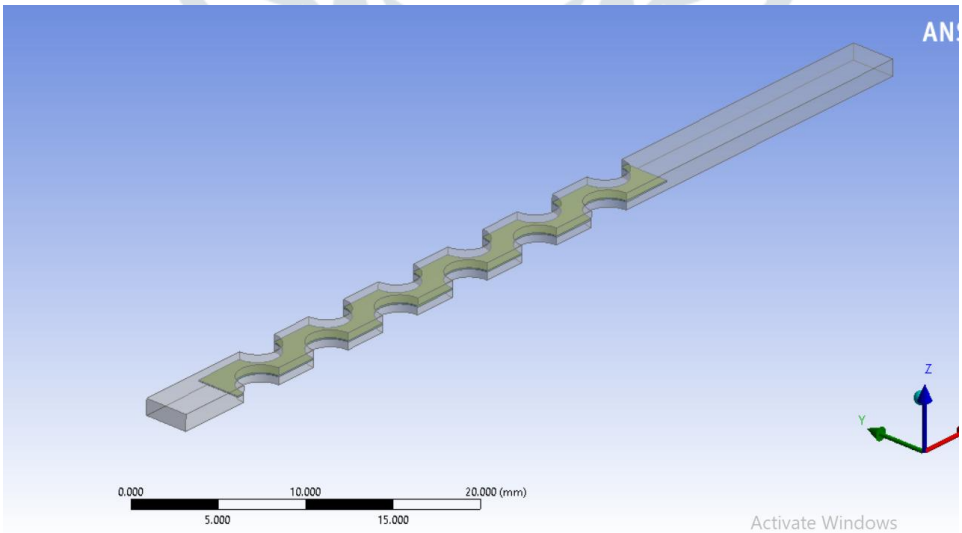

Fig.3Solid model for computational analysis

3. Meshing

After developing the solid model of given geometry, it is then discretizing in to number of elements and node because the numerical analysis is completely depends on the number of elements and number of nodes. The commercial computational fluid dynamics code Fluentemployed to carry out the present numerical investigation. The volume mesh consists of tetrahedral elements.

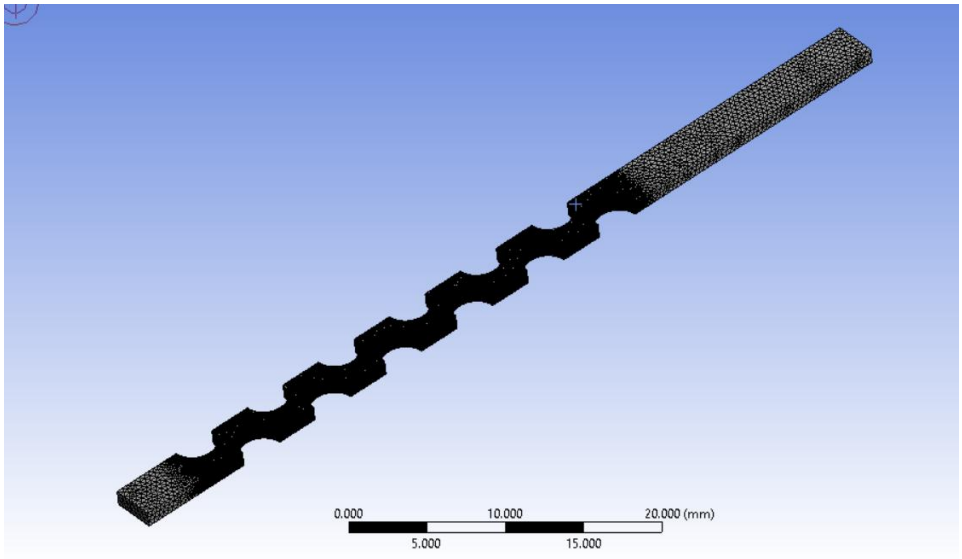

Fig.4 Mesh of the given geometry

3.1 Grid independence Test

To minimize the computation time with a desired level of accuracy, the grid independence test is required. The computational time increases with an increase in the number of elements. On the other hand, a coarser grid may give misleading results with steep gradients.

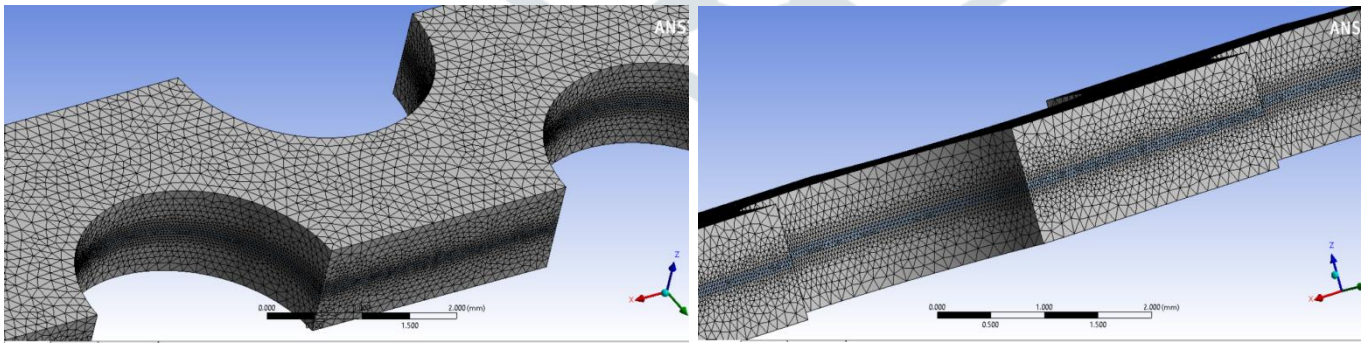

Fig.5 Front view of the meshed solid model

The computational mesh was chosen so that accurate results can be obtained and the computational time is minimum. It is necessary to perform the mesh independent test, it means that if it is increasing the number of element weather the property of system is increasing or not. Here in this analysis it used the temperature of air at the exit as a parameter to check the mesh independency.

Table.1 Grid selection

Number of elements	Temperature (K) of air at exit
473254	443.8584
432548	441.354
542259	445

3.2 Selection of Model

After performing the mesh, here it selects the model for doing the further analysis. To perform the numerical analysis it is necessary to select appropriate model according to the condition of problem. So in order to find out the appropriate model for the given problem, here it considered the six different model that is K-epsilon Standard, K-epsilon RNG, K-epsilon Realizable, K-omega Standard, K-omega BSL and K-omega SST. So to find out the appropriate model here it has taken the all six model and find out the temperature difference for each model and then with the help of numerical analysis it find out the coefficient of heat transfer for each case. The value of heat transfer coefficient for each case is shown in the table at four different velocities.

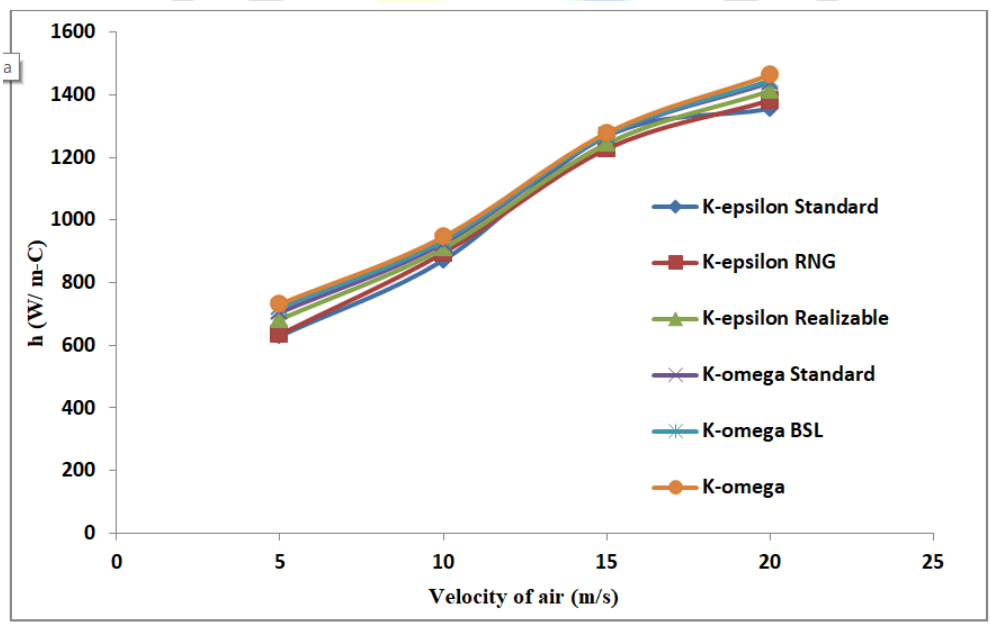

Fig.6Comparison of heat transfer coefficient for different model

Based on the above analysis it is found that there is not much difference in between the different model. But in the entire six models K-epsilon Standard model is showing the average values as compared to other model, so it

is better to prefer this model for further analysis. Here in this work it used the K-epsilon Standard model for the further analysis. The fig.6 shows the use of this model is

4. Material Used

During the analysis here, we have considered the three different steel alloys that is GH2132, GH3044 and S66280 for fin tube construction. The thermal properties of all steel alloys are mention in the below section. For the initial analysis it takes the material same as that of taken by Lingdong et.al [1]. Therefore, here it considering GH2132 alloy (Fe-25Ni-15Cr) as the fin tube material. The material properties are shown in the below table

Table.2 Properties of materials

Properties	material GH2132	material GH3044	Material S66280
Density	7.99 g/cm^3	8.89 g/cm^3	7.98 g/cm^3
Specific heat	447 J/kg-k	440 J/kg-k	460 J/kg-k
Thermal conductivity	14.2 W/m-C	11.7 W/m-C	12.2 W/m-C

4.1 Governing Equations and Boundary Condition

Here in this analysis the frontal air entering the heat exchanger is at different speed because it considered four different velocity of air that is 5, 10, 15 and 20 m/s. based on the velocity of air Reynolds number were calculated which are 2622.96, 5248.98, 7873.25 and 10497.96 from the Reynolds number value it is found that the flow is moving from transition state to turbulent state.

4.1.1 Boundary condition:

The upstream boundary (inlet)

$$u = constant, T = constant, v = w = 0$$

Fin and tube wall surface (no slip condition)

$$u = v = w = 0, T = const$$

The downstream boundary (outlet)

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = 0$$

Top symmetry boundary on the x-y plane

$$\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial T}{\partial z} = 0, w = 0$$

The laws governing the problem are explained in detail. The governing equations are subjected to the boundary conditions of the problem to formulate a solution. These expressions would then be discretized using the finite volume method to estimate the solution. The present problem involved thermal transport with convective heat transfer. Working fluid used is air with Prandtl number set to 0.736.

4.2Assumptions: During the mathematical calculation some assumptions were taken as follows

- The heat exchanger is a thin fin, horizontal compact heat exchanger.
- The fluid can be modeled as a three dimensional fluid flow through a computational domain.
- Viscous dissipation and viscous work were neglected.
- Body forces are neglected.
- The pressure drop along the domain caused by momentum change and viscous friction is negligible. Therefore, fluid pressure can be assumed uniform along the entire domain.
- Heat transfer from air to fin and tube through convection.
- Fluid properties taken at mean temperature of inlet and outlet.

➢ **For calculating the maximum velocity**

Maximum velocity of airflow inside the compact heat exchanger are calculated based on relation given for the staggered tube arrangement in Cengel. The relation used for calculating maximum velocity inside the computational domain is given below.

$$V_{max} = \frac{P_t \times V}{2(P_d - D)} \quad (1)$$

Where; D = diameter of tube, V = velocity of air at inlet, P_t = transverse distance in between the two tubes of same row, P_d = diagonal distance between the center of two tubes of adjacent row

➢ **For Calculating the Reynolds number**

To calculate the Reynolds number of air flowing inside the computational domain, maximum velocity of air flowing inside the domain were considered in a particular case of staggered fin tube type compact heat exchanger [29]. For calculation following relation were used

$$Re = \rho_{air} \times V_{\max\ air} \times L_c / \mu_{air} \quad (2)$$

Where;ρ_{air} = Density of air, $V_{\max\ air}$ = Velocity of air,μ_{air} = Dynamic viscosity of air

- **Heat transfer**

In order to calculate the heat transfer from hot air to fin and tube following relation were use. During the heat transfer from hot air to cold fluid flowing inside the tube, many researchers have found that the thermal resistance during the heat transfer from tube to cold fluid is less as compared to the thermal resistance, during heat transfer from air to tube and fins [6, 22, 29]. So during the calculation of heat transfer and heat transfer coefficient it mainly concern toward the heat transfer from air to fin and tube and neglect the thermal resistance toward the cold fluid domain which is very less as compared to the thermal resistant toward the air side. Due to this here it calculate the local heat transfer coefficient in between air and tube, and not calculating the value of overall heat transfer coefficient. To calculate the heat transfer rate at different velocity following formula used.

$Q = \dot{m}C_P\Delta T$(3)$\dot{m}$ = mass flow rate of air, C_P = specific heat of the air, ΔT= change in temperature between inlet to outlet.Mass flow rate of air and logarithmic mean temperature difference

$$\dot{m} = \rho_{air} \times V_{air} \times A_c \quad (4)$$

For calculating the mean temperature difference for particular staggered type arrangement of fin and tube type compact heat exchanger following relation were used as given in the base paper. To calculate ΔT_m following formula mention in the base paper and Cengel and Gajar used.

$$\Delta T_m = \frac{(T_{in} - T_w) - (T_{out} - T_w)}{ln[(T_{in} - T_w)/(T_{out} - T_w)]} \quad (5)$$

T_{in}= Temperature at inlet, T_{out} = Temperature at outlet, T_w =Temperature of the tube wall or fin.To calculate heat transfer coefficient

$$Q = hA\Delta T_m \eta_0 \quad (6)$$

Where, h = average heat transfer coefficient (W/m^2-k), A = surface area of domain, ΔT_m = logarithmic mean temperature difference, η_0 = surface efficiency or efficiency of computational domain. To calculate the surface efficiency following formula where used

$$\eta_0 = 1 - (1 - \eta_f)\frac{A_f}{A} \quad (7)$$

Where, η_0 = surface efficiency or efficiency of computational domain, η_f = fin efficiency, A_f = fin surface area, A = is the total surface area for heat transfer that is fin surface plus tube surface. To calculate the surface area of domain following calculation is use.

$$A = L \times W - (\frac{\pi}{8} \times D^2) \times N \quad (8)$$

Where, L = length of fin, W =width of fin, D = diameter of tube, N = no of tube in computational domain.

- **Fin Efficiency**

During the analysis, it found that the thickness of fin is a measure concern for the heat transfer. To find the effectiveness of different fin thickness here we have calculated the value of fin efficiency at different velocity of air. For calculating the fin efficiency following formula where used which is given in the base paper. The formula for calculating the fin is

$$\eta_f = \frac{\tan(mr\emptyset)}{mr\emptyset} \quad (9)$$

Where;

$$m = \sqrt{\frac{2h_0}{K_f \times \delta_f}} \quad (10)$$

$h_0$is the local heat transfer coefficient for fin surface which is calculated through numerical analysis , K_f thermal conductivity of fin material, δ_f fin thickness. Where $\emptyset$ can be find through the below formula

$$\emptyset = \left(\frac{R_{eq}}{r} - 1\right)\left[1 + 0.35 ln\left(\frac{R_{eq}}{r}\right)\right] \quad (11)$$

Where, r is the tube radius and

$$R_{eq} = r\left[1.27\frac{X_m}{r}\left(\frac{X_l}{X_m} - 0.3\right)^{1/2}\right] \quad (12)$$

Where;

$$X_m = \frac{P_t}{2}, \quad X_l = \left[\left(\left(\frac{P_t}{2}\right)^2 + {P_l}^2\right)/2\right]^{1/2} \quad (13)$$

$P_t =$ Transverse pitch, $P_l =$ Longitudinal pitch

4.3 Air Condition and Air Physical Properties

Air coming from the compressor and entering the heat exchanger in the aero engine operation assume to operate at altitude of 11 km and flies at mach 0.8 as given in Lingdong et.al [1]. During the analysis it considered the local atmospheric temperature and pressure that is calculated as 298 K and 22.63 kPa[1]. The inlet total pressure recovery coefficient is assume to be 0.97, the compressor compression ratio considered during the analysis is 25, the compressor efficiency is 0.90, and the air adiabatic index is 1.4, the air temperature and pressure at the inlet of heat exchanger is considered to be same as those at the compressor outlet, and it is considered as 653.99 K and 0.84 MPa[1]. The air velocities at the heat exchanger inlet (frontal air velocities) are set to range from 5 to 20 m/s, with the tube wall temperature being taken as 298 K.

5. Result

The numerical results presented in this work were obtained using commercial computational fluid dynamics (CFD) code Ansys Fluent. This code offers a wide range of flow analysis models, including laminar model and a number of turbulence models based on the eddy viscosity approach and the Reynolds stress approach. This feature of Fluent was particularly useful for the current investigation, which ranges from the transitional flow range to turbulent flow range. Accuracy of the current study was tested by comparing the results with the result, reported by etLingdong et.al [1]. In order to validate the CFD model of fin tube compact heat exchanger used in aero engines, here it first find out the temperature of air at the exit of heat exchanger for different velocity. In this work we have considered four different velocity that is 5, 10, 15, 20 m/s and in each case air exit temperature get calculated and based on the air temperature at the exit here we have calculated the heat transfer rate (q). The value of heat transfer rate calculated with the help of numerical method is then compared with the value of heat transfer rate given in the base paper Lingdong et.al [1]. In the initial case, it considered the material GH2132 for fin tube compact heat exchanger. The thickness of fins for this analysis is 0.1 mm, whereas the gap in between the two fins is near about 1.1 mm.

Here in this case velocity of frontal air is 5 m/s and the temperature of air at the inlet is 653.98 K. After applying the boundary condition it find out the air exit temperature. The contour plot of air temperature distribution for this case shown in fig.7.

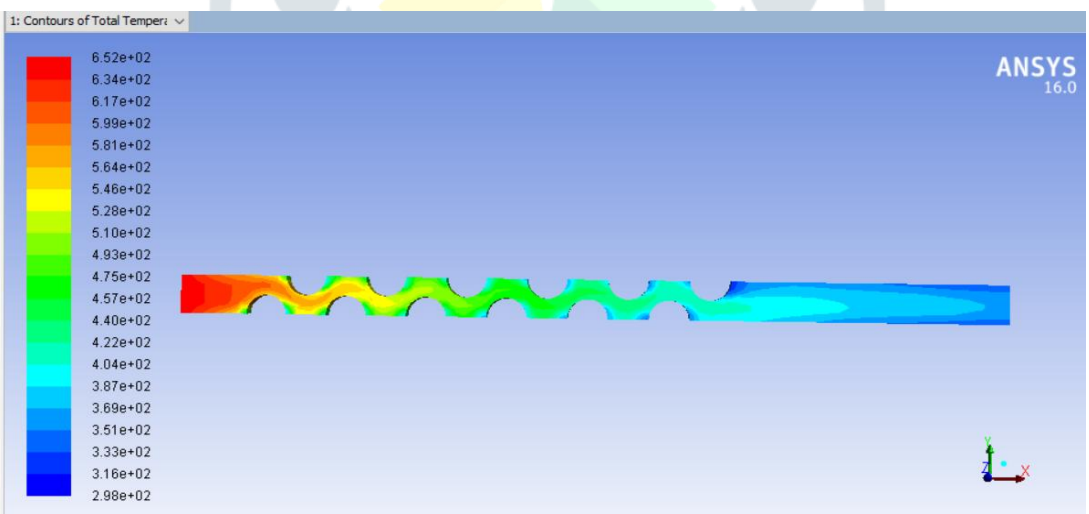

Fig.7 Temperature contour of heat exchanger at 5 m/s velocity

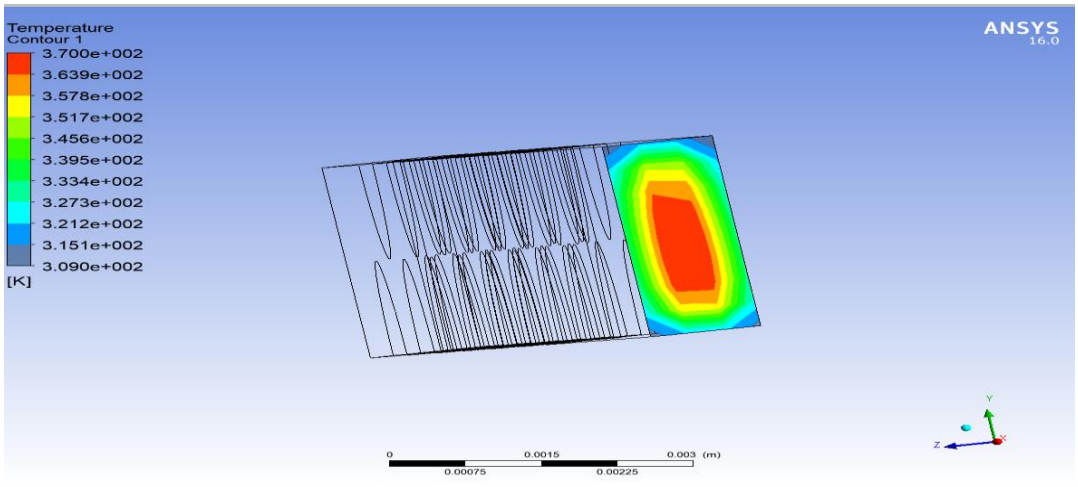

Fig.8 Contour of temperature distribution at the exit

Comparison of value of temperature of air at the exit and heat transfer coefficient calculated through numerical analysis with the value of temperature and heat transfer coefficient given in the base paper.

Table.3 Comparison of numerical values and base paper value

Reynolds number	Velocity (m/s)	Average Heat transfer coefficient (h) (W/m^2K) calculated through numerical analysis	Heat transfer coefficient (h) (W/m^2K) from base paper	Error (%)	Heat transfer rate (W) calculated form numerical analysis	Heat transfer rate (W) values from base paper	Error (%)
2622.96	5	563	550	2.3	18.15	18	14.62
5248.98	10	989.115	985	4	33.24	31	7.2
7873.25	15	1351.3	1300	3.9	47.9	44	8.86
10497.96	20	1662.16	1600	3.8	54.059	52	3.95

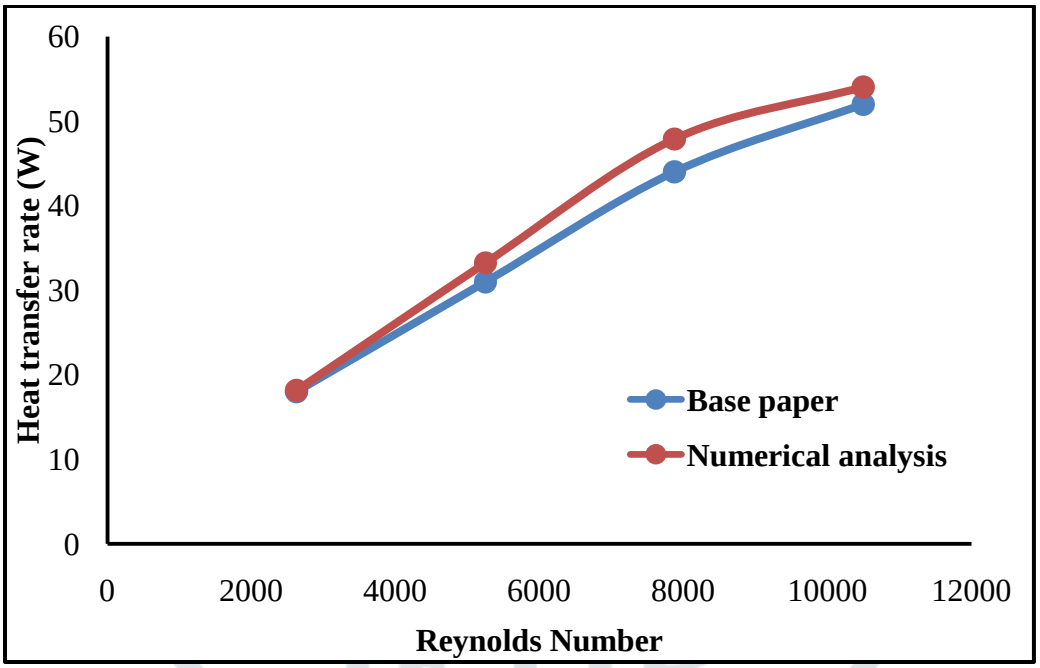

Fig.9Comparison of heat transfer rate at different velocity of air

From the above comparison it is shown that the value of temperature at the exit of heat exchanger obtained from the CFD analysis is closer to the value of temperature obtained from the base paper. It also analyzed that the value of heat transfer rate at different velocity of air obtained from the numerical analysis is close to value obtained from the base paper and follow the same trend as follow in the base paper. Therefore, after analyzing the graph it shows that the CFD model of fin tube compact heat exchanger develop in this work is correct. From the above fig. it also found that the value of heat transfer coefficient for all Reynolds number of air is near to the value of heat transfer coefficient at all velocity given in the base paper

5.1 Optimization of Material Used For Tube and Fins Manufacturing

For finding out the effect of different material used for tube and fins manufacturing, it has considered three materials that is GH2132, GH3044 and S66280. The value of heat transfer rate and heat transfer coefficient for different Reynolds number for different material is shown in the graph.

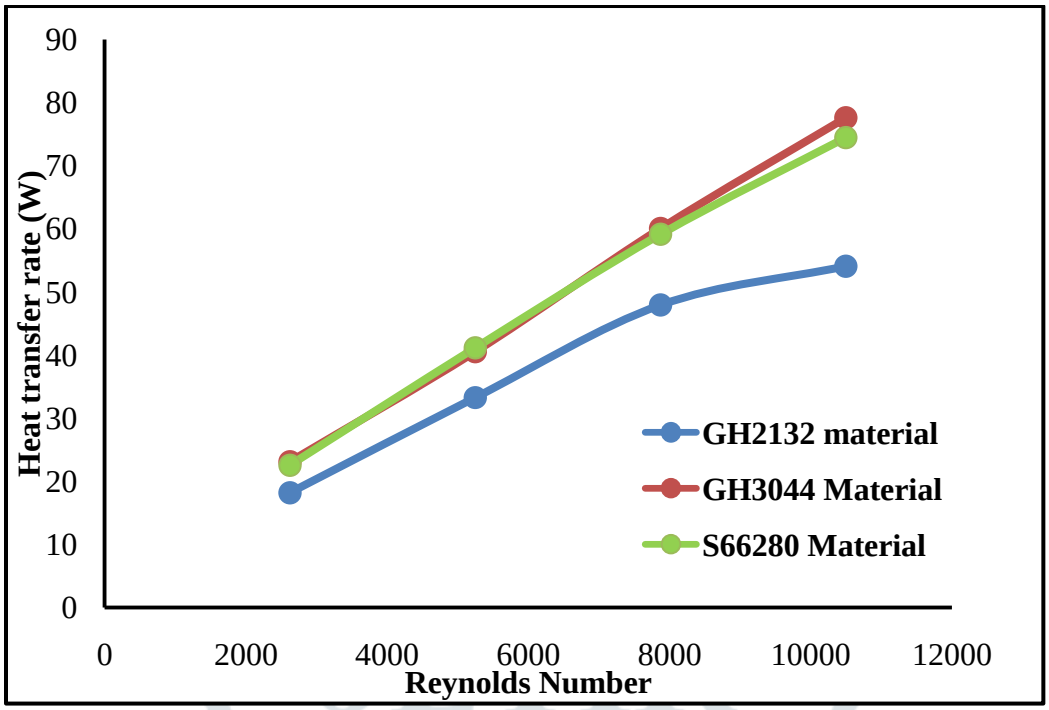

Fig.10 Comparison of heat transfer rate for different material at different velocity

From the above comparison graph, it is found that the temperature at the exit of heat exchanger is minimum for material GH3044 at every velocity of air. Through this analysis, it is found that as the material density, specific heat and thermal conductivity changes, the heat transfer capacity of the material also changes. From the analysis it is also observed that as the velocity of the air increases the rate of heat transfer also increases and it is high for material GH3044 at all velocity as shown in the comparison graph.Through graph it is also analyzed that as the velocity of the air increases the heat transfer coefficient also increases and it is maximum in case of GH3044 material.

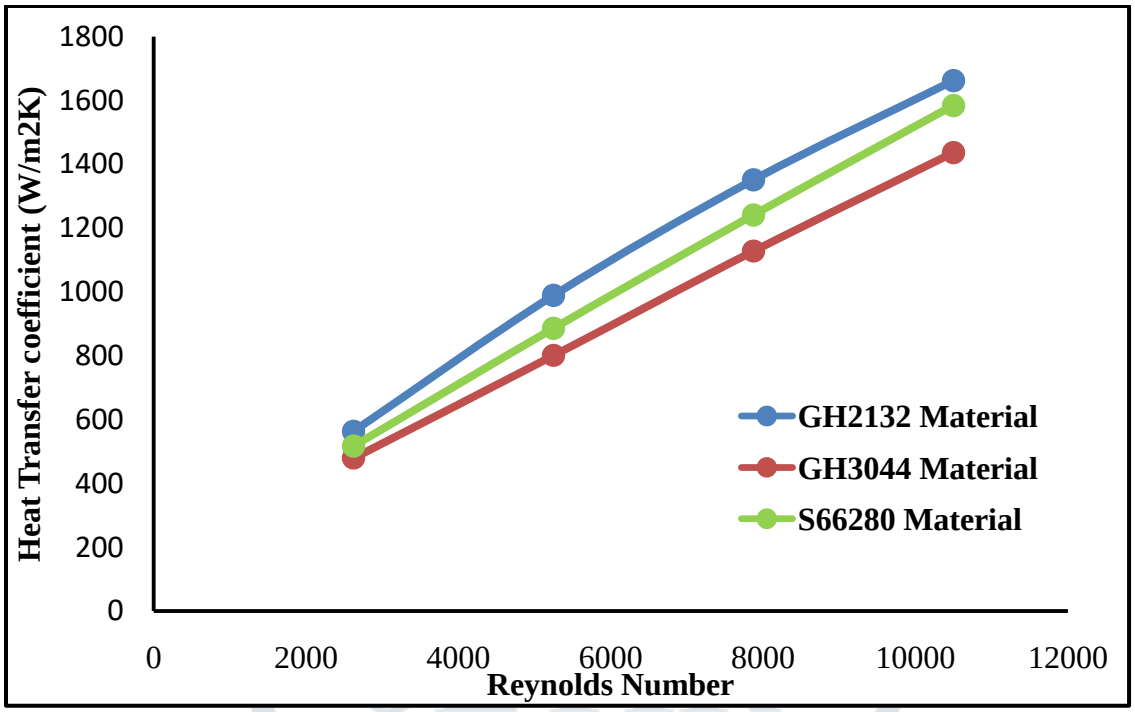

Fig.11 comparison of heat transfer coefficient for different material at different velocity

So it is found that the material GH3044 shows the better heat transfer as compared to the material GH2132 and S66280. Therefore further analysis of heat transfer rate for GH3044 material at different fin thickness and at different fin spacing.

5.2 Effect of Fin Thickness

After finding out the effect of material on the heat transfer rate, here it has also analyzed the effect of tube fin thickness on the heat transfer rate and the temperature of heat exchanger. In order to find out the effect of fin thickness.

Table.4Values of different parameters and heat transfer rate at different fin thickness

Velocity of air (m/s)	Temp. (K) at the exit for fin thickness 0.08 mm	Temp. (K) at the exit for fin thickness 0.1 mm	Temp. (K) at the exit for fin thickness 0.2 mm	Temp. (K) at the exit for fin thickness 0.3 mm	Heat transfer rate (W) for thickness 0.08mm	Heat transfer rate (W) for thickness 0.1 mm	Heat transfer rate (W) for thickness 0.2 mm	Heat transfer rate (W) for thickness 0.3 mm
5	502	357	365	369	11.839	23.10	24.511	22.199
10	505	386	384	389	23.2118	40.52	42.063	41.234
15	509	397	398	402	33.883	60.05	61.8228	58.88
20	515	405	404	406	43.3078	77.58	77.8912	77.27

Here it considered the three different fin thicknesses solid model and find out the temperature of air at the exit. Here it considered 0.08, 0.1, 0.2 and 0.3 mm thickness fin during the numerical simulation. The comparison of value of heat transfer coefficient and heat transfer rate for different fin thickness are shown in the fig.

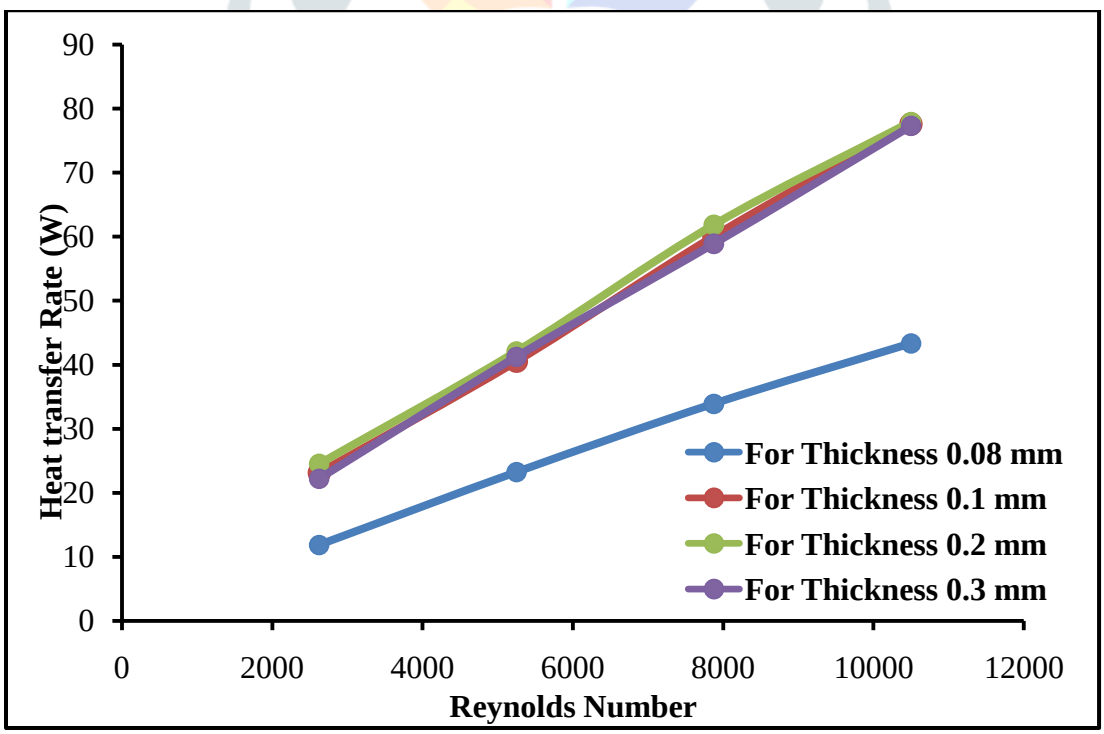

Fig.12 Comparison of heat transfer rate for different fin thickness

Velocity of air (m/s)	Reynolds Number	Heat transfer coefficient (W/m^2K for 0.08 mm	Heat transfer coefficient (W/m^2K) for 0.1 mm	Heat transfer coefficient (W/m^2K) for 0.2 mm	Heat transfer coefficient (W/m^2K) for 0.3 mm
5	2622.96	108.56	479.25	484.35	399.91
10	5248.98	208.366	800.25	809.458	614.37
15	7873.25	296.99	1127.53	1215.357	852.41
20	10497.96	382.074	1436.03	1445.658	1084.27

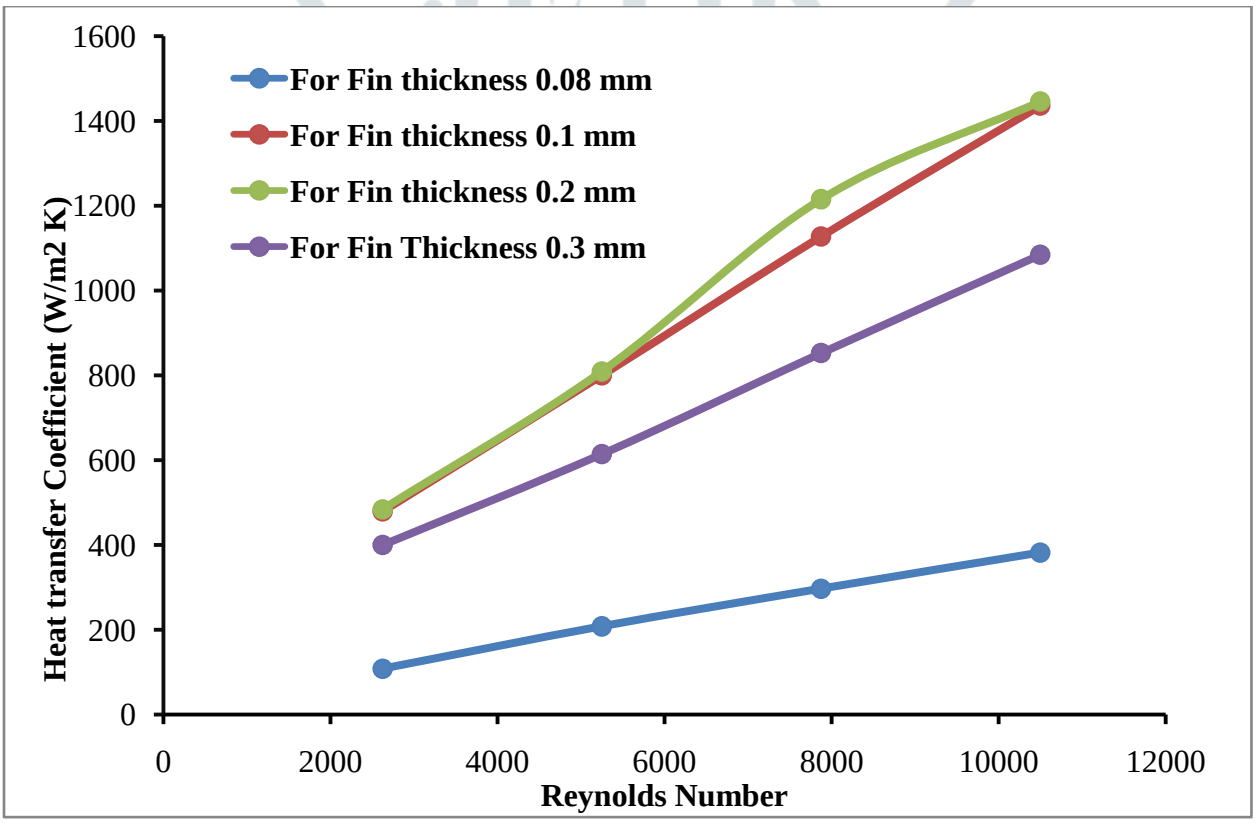

Fig.13 Comparison of heat transfer coefficient for different fin thickness

5.2.1Fin Efficiency

As the fin thickness play an important role during the heat transfer, it is necessary to calculate the effect of fin thickness on the heat exchanger. To calculate the effect of fin thickness here we have calculated the value of fin efficiency for different fin thickness that is 0.08, 0.1 and 0.2 mm. the comparison of fin efficiency for different fin thickness at different velocity shown in below table

Table.5Comparison of fin efficiency for different fin thickness

Velocity (m/s)	Fin Efficiency for thickness 0.08mm	Fin Efficiency for thickness 0.1mm	Fin Efficiency for thickness 0.2mm	Fin Efficiency for thickness 0.3 mm
5	0.9205	0.7852	0.8786	0.9618
10	0.8631	0.7025	0.8283	0.9449
15	0.8290	0.6411	0.7848	0.9285
20	0.7857	0.5961	0.7484	0.9142

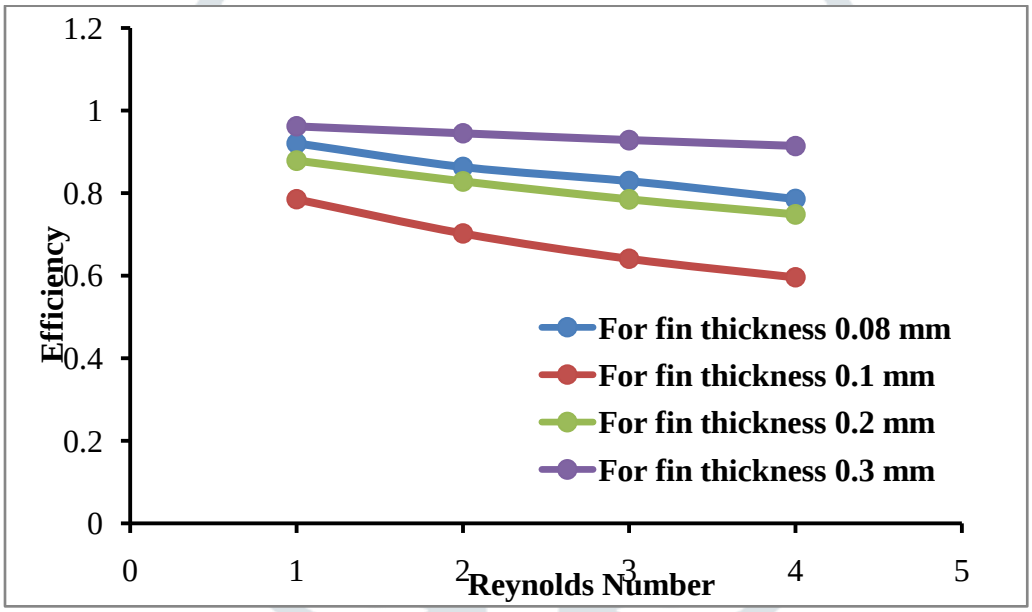

Fig.14 Comparison of Efficiency for different fin thickness

5.3 Fin Spacing

To analyzed the effect of change in gap in between the two adjacent fins here we have considered the four different type of fins spacing solid modal. Here it considered 0.5, 0.8, 1.1 and 1.6 mm distance in between the two adjacent fins during the numerical simulation.

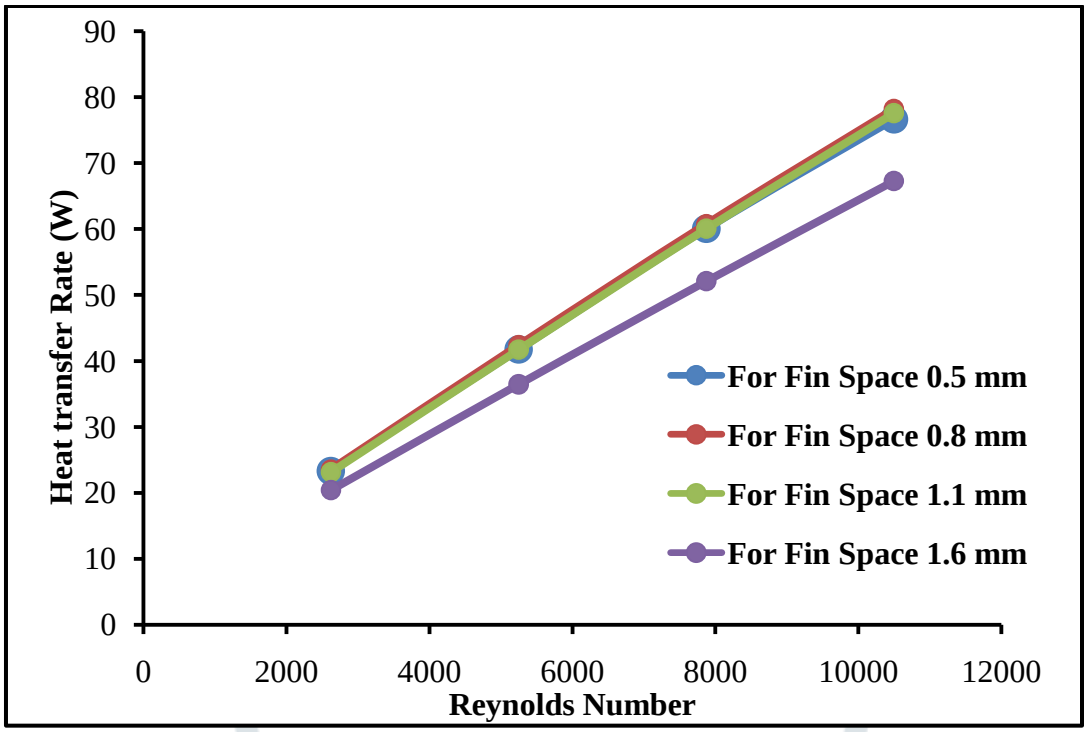

Fig.15 comparison of heat transfer rate for different fin spacing

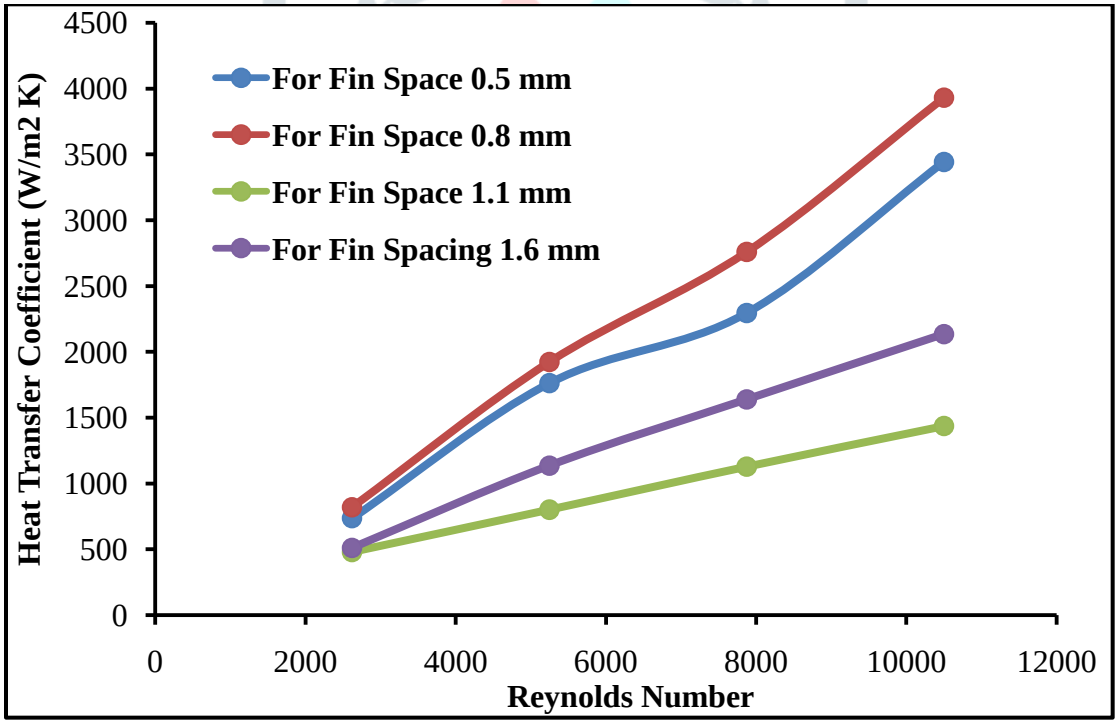

Fig.16 Comparison of heat transfer coefficient for different fin spacing

From the above analysis it is found that as the space in between the two adjacent fins get increases the temperature of air at the exit get increased which means that the rate of heat transfer get reduce due to the increased in fin spacing. Whereas with the decrease in fin thickness the temperature of air at the exit get also

decreases which means that the heat transfer rate get increased. However, after the particular distance in between the fins. If we reduce the thickness beyond that space the heat transfer rate, get reduced.

6. Conclusion

- The airside heat transfer and pressure variation characteristics of plain finned tube heat exchangers are numerically predicted with consideration of the air property variations caused by change in air velocity.
- Here it also find out the effect of material on the temperature of air at the exit, for analyzing the effect it consider the different steel alloy which is GH2132, GH3044 and S66820.
- From the graph it is found that as the Reynolds number increases the value of heat transfer increases for all the three material.
- GH3044 shows the maximum value of heat transfer as compared to the other material. From the graph it is conclude that the value of heat transfer for GH3044 is on an average 15 % more than the GH2132 material.
- it is found that as the thickness of fin increases from 0.08 mm to 0.2 mm the heat transfer rate increases, whereas beyond 0.2 mm thickness value of heat transfer again start decreasing.
- It is concluded from comparison graph of fin thickness that the use of fin thickness 0.2 mm is better as compared to the other fin thickness and shows 8 % increment in heat transfer as compared to 0.1 mm fin thickness.
- After analyzing the effect of different fin spacing it is found that as the fin spacing increases the heat transfer decreases.
- Heat transfer increases with decrease in fin spacing, but after 0.8 mm fin spacing the heat transfer start decreasing with decrease in fin spacing.
- Here computational domain having fin spacing 0.8 mm shows 5 % increase in heat transfer as compared to 1.1 mm fin spacing.
- After analyzing the effect of different material, fin spacing and fin thickness it is found that GH3044 material with fin thickness 0.2 mm having fin spacing 0.8 mm where the best combination to enhance the heat transfer rate on the air side in the computational domain.

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