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Experimental and Computational Analysis of Different Types of Fins Using Convective Heat Transfer

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***Abstract*: This paper discusses about analysis of heat transfer rate and investigates the efficiency of various shapes of fins. Four different fins are used for analysis. The various shaped fins are circular, square and hexagonal and rectangular. The aim is to determine the heat exchange and efficiency of the same area with a different shape. The purpose of the study is to calculate the maximum heat transfer from the fin surface and efficiency with changed fin shapes. For this analysis, the CFD ANSYS 14.5 Fluent is used. Heat supplied to a solid rod is conducted to solid fin. At the same time, convection takes place with air which flows through a duct. Nichrome (band type) heater used for heat supply is mounted with the fin. Heater is the made up of aluminum material and the fins are aluminum which high thermal conductivity than other materials. Fin models are generated by the solid work software after that imported into ANSYS 14.5 fluent. The analysis is done in a procedural way. First, lay the experimental setup then take the readings from various shaped fins separately. Heat transfer rate and efficiency from the observed readings are calculated for different Reynolds' Number finally, the results are compared with other fins.**

***Keywords*: heat transfer, fins, thermal conductivity, CFD, ANSYS 14.5.**

1. Introduction

Heat exchangers are used in different processes ranging from conversion, utilization & recovery of thermal energy in various industrial, commercial & domestic applications. Increase in Heat exchanger's performance can lead to more economical design of heat exchanger which can help to make energy, material & cost savings related to a heat exchange process. The need to increase the thermal l performance of heat exchangers, thereby effecting energy, material& cost savings have led to development & use of many techniques termed as Heat transfer Augmentation. These techniques are also referred as Heat transfer enhancement or intensification. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger. Use of Heat transfer enhancement techniques lead to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of heat transfer rate & pressure drop has to be done. Apart from this, issues like long term performance & detailed economic analysis of heat exchanger has to be studied. To achieve high heat transfer rate in an existing or new heat exchanger while taking care of the increased pumping power, several techniques have been proposed in recent years. Extensions on the finned surfaces are used to increases the surface area of the fin in contact with the fluid flowing around it. So, as the surface area increase the more fluid contact to increase the rate of heat transfers from the base surface as compare to fin without the extensions provided to it. The need to increase the thermal performance of heat exchangers, there by effecting energy, material &cost savings have led to development & use of many techniques termed as Heat Transfer Augmentation. In the present work, we are supposed to use four types of aluminum fin out of which one is conventional one and other three of spiral surface with varying pitch to increase the convective heat transfer rate of pin fin. Fins are thin strips of metal attached to the heat transfer surface in order to increase heat transfer area. Since there is a certain relation between pressure and heat transfer rate, at low density situation the rate of heat transfer does not remain same as that at atmospheric pressure. To use fins for heat rejection at low density situation it is necessary to observe the heat transfer characteristics of the fin at that situation. Moreover, to reduce high waste heat, it is necessary to increase the heat transfer area of the radiator

2. Literature Review

Saroj Yadava et al.[1] suggested the thermal analysis performance of different shapes of the fins for various input parameters. Mehran Ahmadi et al. [2] have investigated numerically and experimentally to increase the heat transfer to added the vertically-mounted rectangular interrupted fins. R. Sajedi et al. [3] have suggested comparing the result of two common pin fin heat submerged with and without splitters in circular and square shape pin-fins. Pradeep Singh et al. [4] have compared the heat transfer performance of fin with same geometry having various extensions and without extensions. S. Kushwaha et al. [5] compared the heat transfer in electronic components of heat sink having fins of different profiles namely Rectangular, Trapezoidal and Parabolic. L. Prabhu et al. [6] has analyzed heat transfer performs by ANSYS workbench for the

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design of fin with various design configuration such as cylindrical, square and rectangular. Arun Eldhose et al. [7] has find out the Fin efficiency, Heat transfer Rate, Temperature distribution and Heat transfer coefficient of Pin fin by varying its geometry and material used. Muhammad Ferdous Raiyan et al. [8] have investigated the temperature distribution and heat flux through various fin surfaces. For this experiment, flared and rectangular fin arrays were considered Mukesh Didwania et al. [9] calculated maximum heat transfer rate of fin surface in duct due to change in shape fins. For analysis a three dimensional finite volume based CFD Tool ANSYS 12.0 Fluent was used.

3. Experimental Setup

Experimental Setup Specifications:

Duct size = 150x100mm
Coefficient of discharge Cd = 0.65
Number of thermocouples on fin = 5
Centrifugal blower with motor
Digital temperature indicator-range = 0-200 C.
Digital Voltmeter-range = 0 -200V AC
Digital Ammeter-range = 0-2 Amp., AC
Dimmer stat: open type = 0-2 Amp., 0-230V
Nichrome wire-Band type heater capacity- 250 Watt

4. Methodology

In this paper we followed the methodology given in the flow chart below first collect all the related information about the heat transfer and the fins and then collecting some of the literature review.

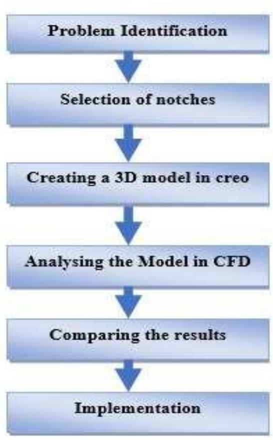

Fig. 1. Methodology

After collecting all the related data's, the fins are designed using CREO 2.0. After the model is created the analysis are done by using ANSYS – 14.5 (CFD – Fluent). Then the results of each analysis are compared and then the best fins are selected. The result from the ansys is compared with the theoretical calculation.

5. Material Characteristics

Fins are a way to obtain more heat transfer surface but the efficiency depends on fin materials and also fins design. The ideal material is to look for materials with high thermal conductivity and since heat dissipated from the fin by convection then need maximum surface area possible. No doubt the best material to select is the one with highest thermal conductivity to ensure heat transfer rates. Also, the material has thermal resistance; corrosion resistance and material weight are important factor especially at high temperature. Mostly, pure aluminum and copper is the best material due to its specific weight, high thermal conductivity to compare with other. In this case we have used aluminum material.

For aluminium fin:

1. Aluminum having the thermal conductivity value of 160 w/m-k.
2. Aluminum is mechanically soft material.
3. Good resistance to corrosion.
4. It has excellent thermal conductivity.

6. Fins Size, Geometry and CAD Model

4 types of fins geometry are used here as follows

1. Circular
2. Hexagonal
3. Square
4. Rectangular

Since the project objective is to calculate heat transfer coefficient and efficiency at different environmental conditions, with considering the blower capacity and structure of air conditioning system used the Equivalence diameter and dimensions in mm are decided as,

1. Circular – 100*24
2. Hexagonal - 100*24.9
3. Square – 100*20
4. Rectangular – 100*20

A. *CAD Model*

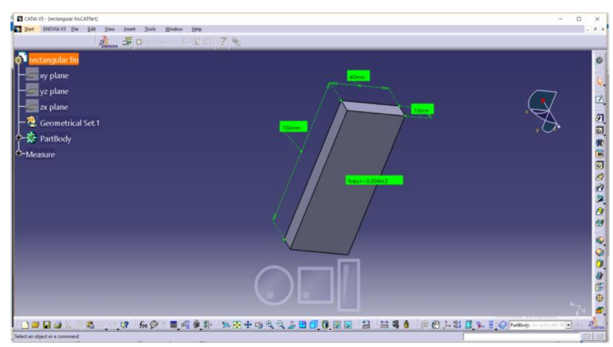

Fig. 1. Rectangular Fin

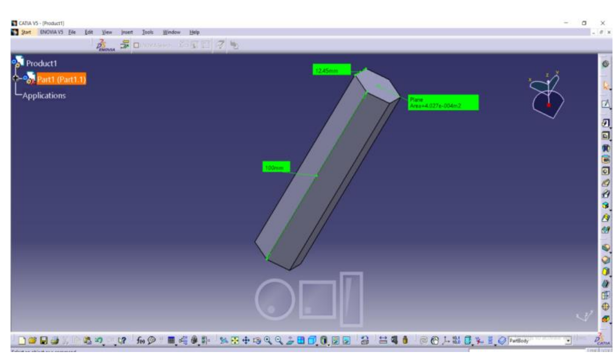

Fig. 2. Hexagonal Fin

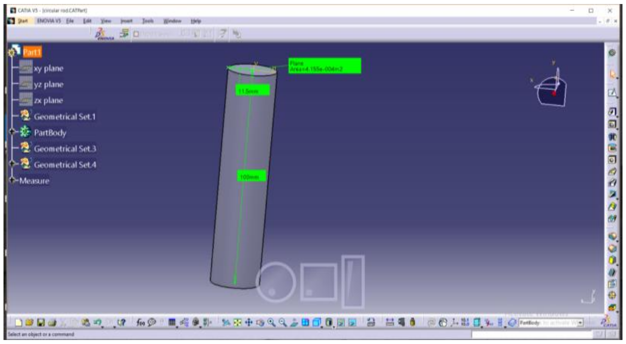

Fig. 3. Circular Fin

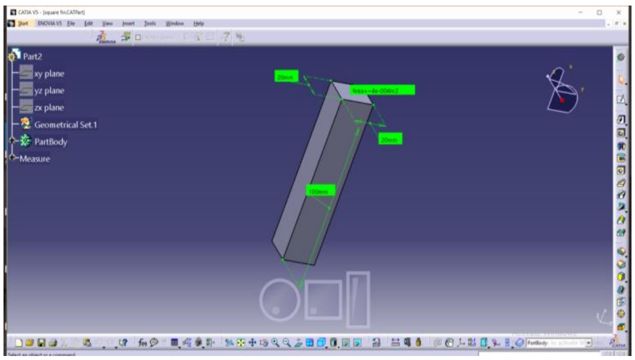

Fig. 4. Square Fin

B. Duct sizing

The fin assembly has to keep in a rectangular duct. The duct dimensions of air conditioning unit are 250 x 250 mm. Since the a/c unit has two compressors and with considering blower capacity (600 Cubic feet min) the fin duct is attached to a/c duct by a diffuser. So the duct dimension is decided as, Duct Dimension 150 mm x 100 mm. Length 1000 mm The finned type air heater of 150 W capacity is installed inside the circular tube.

7. Experimental Procedure

First blower and heater are started simultaneously. After starting the blower pressure differences due to the fins employed using manometer are noted. The reading of atmospheric temperature is also taken. Reading is taken for 4 pin. At the same time pressure drop in test section is also measured by using differential pressure transducer. The voltage, current and temperature at different points where thermocouples are attached are noted down.

Table 1
Experimental Data Readings

Trial No	FIN TYPE	Voltmeter Reading V (Volts)	Ammeter Reading I (Amps)	Manometer Reading Hw (mm)	Thermocouple Reading					
					T1	T2	T3	T4	T5	T6= Ta
1	Circular	96	0.46	215	185	140	130	105	90	40
2	Rectangular	95	0.48	220	187	140	150	123	95	40
3	Square	95	0.47	220	176	143	138	126	90	40
4	Hexagonal	96	0.48	205	180	114	103	105	93	40

7.1. Experimental Calculation

1) Room pressure Pa = 700mm Hg or $\frac{700 \text{ X } 101325}{760}$

Pa = 93325.65 N/m^2

2) Density of air at RTP = $\rho a = \frac{Pa}{RTa}$

$\rho a = \frac{93325.65}{287\text{X}308}$

= 1.055 Kg/m^3

3) Temp at base of fin

Tb= T1+2 = 185+2 =187 ^{O}C

Fins Calculation:

1) Circular Fin

Air head causing flow of air through orifice.

$$h_a = \frac{Pw \ x \ hw}{Pa}$$

$$= \frac{1000x0.215}{1.055}$$

=203.8m

Volume flow rate of air through orifice.

$$Q = Cd \times ao \times \sqrt{2g \times h_a}$$

$$Q = 0.65 \times \frac{\pi}{4}(0.02)^2 \times \sqrt{2 \times 9.81 \times 203.8}$$

Q = 0.0123 m^3/sec

Velocity of air flow through the duct

$$V = \frac{Q}{Area \ of \ duct}$$

$$= \frac{0.0123}{0.15x0.1}$$

= 0.82 m/s

Average temp of fin surface,

$$Ts = \frac{T1+T2+T3+T4+T5}{5}$$

$$Ts = \frac{185+140+130+105+90}{5}$$

Ts = 130°C

Mean flim temp of fluid

$$Tf = \frac{Ts+Ta}{2}$$

$$= \frac{130+40}{2}$$

= 85°C

Properties of air at mean film temp Tf= 80°C

From Heat and mass transfer data hand book

Kinematic viscosity of air ϑ = 21.09 x $10^{-6} m^2$/sec

Prantl's no. Pr =0.692

Thermal conductivity K = 0.03047 w/mk

Reynold's Number

$Re = \frac{VD}{\vartheta}$

$Re = \frac{0.82 \times 0.023}{21.09 \times 10^{-6}}$

Re = 894.26

Nusselt Number

Corresponding to this Reynolds number, lead the values of C,m and n in the equation

Nu = Nu = C x Re^{m} ×Pr^{n}

= 0.683 x $894.26^{0.466}$ x 0.692^{113}

Nu = 14.33

Heat transfer Coefficient

$= \frac{Nuxkair}{D}$

$= \frac{14.33X0.0347}{0.023}$

h = 21.61 w/m^2 k

$m = \sqrt{\frac{hP}{KA}}$

$= \sqrt{\frac{4h}{KD}}$

$= \sqrt{\frac{4X21.61}{160X0.023}}$

m = 4.84 m

Qfin = m x Kfin x Ao x (Tb – Ta) x tan h mL

= 4.84 x 160 x $\frac{\pi}{4}(0.023)^2$ x (187 – 40) x tan h (4.84 x 0.1)

= 21.25 watt

Efficiency of the fin

$nf = \frac{\tan h\, mL}{mL}$

$= \frac{\tan h\,(4.84\ X\ 0.1\ )}{4.84\ X\ 0.1}$

= 92.36%

Effectiveness of fin

$€ = \frac{Qfin}{hA\,(Tb-Ta}$

$= \frac{21.25}{21.61\,x\,\frac{\pi(0.023)^2}{4}\,x\,(187-40)}$

= 16.10

Temp at base of fin

Tb= T1+2 = 187+2 =189°C

2) Rectangular Fin

Air head causing flow of air through orifice

$ha = \frac{Pw\ x\ hw}{Pa}$

$= \frac{1000x0.211}{1.055}$

= 200 m

Volume flow rate of air through orifice

Q = Cd×ao×$\sqrt{2g\ \times}$$h_a$

Q = 0.65×$\frac{\pi}{4}(0.02)^2$×$\sqrt{2 \times 9.81 \times 200}$

Q = 0.0127 m^3/sec

Velocity of air flow through the duct

$V = \frac{Q}{Area\ of\ duct}$

$= \frac{0.0127}{0.15x0.1}$

= 0.84 m/s

Average temp of fin surface

$Ts = \frac{T1+T2+T3+T4+T5}{5}$

$Ts = \frac{187+140+150+123+95}{5}$

Ts = 139°C

Mean flim temp of fluid

$Tf = \frac{Ts+Ta}{2}$

$= \frac{139+40}{2}$

= 89.5°C

Properties of air at mean film temp Tf= 90°C

From Heat and mass transfer data hand book

Kinematic viscosity of air ϑ = 22.10 x 10^{-6} m^2/sec

Prantl's no. Pr =0.690

Thermal conductivity K = 0.03128 w/mk

Reynold's Number

$Re = \frac{VD}{\vartheta}$

$Re = \frac{0.84 \times 0.023}{22.10 \times 10^{-6}}$

Re = 874.20

Nusselt Number

Corresponding to this Reynolds number, lead the values of C, m and n in the equation

Nu = Nu = C x Re^{m} ×Pr^{n}

= 0.683 x $874.20^{0.466}$ x 0.690^{113}

Nu = 14.17

Heat transfer Coefficient

$= \frac{Nuxkair}{D}$

$= \frac{14.17X0.03128}{0.023}$

h = 19.27 w/m^2 k

m = $\sqrt{\frac{hP}{KA}}$

= $\sqrt{\frac{4h}{KD}}$

= $\sqrt{\frac{4X19.27}{160X0.016}}$

m = 5.48 m

Qfin = m x Kfin x Ao x (Tb – Ta) x tan h mL
= 5.48 x 160 x $\frac{\pi}{4}(0.023)^2$ x (189 – 40) x tan h (5.48 x 0.1)
= 27.08watt

Efficiency of the fin

nf = $\frac{\tan h\, mL}{mL}$

= $\frac{\tan h\,(5.48X\,0.1)}{5.48X\,0.1}$

= 91 %

Effectiveness of fin

€ = $\frac{Qfin}{hA\,(Tb-Ta}$

= $\frac{27.08}{19.27\,x\,\frac{\pi(0.023)^2}{4}\,x\,(189-40)}$

= 22.70

Temp at base of fin

Tb= T1+2 = 176+2 =178°C

Fins Calculation:
3) Square Fin

Air head causing flow of air through orifice

ha= $\frac{Pw\ x\ hw}{Pa}$

= $\frac{1000x0.220}{1.055}$

=208.53 m

Volume flow rate of air through orifice

Q = Cd×ao×$\sqrt{2g\ \times}$$h_a$

Q = 0.65×$\frac{\pi}{4}(0.02)^2$×$\sqrt{2 \times 9.81 \times 208.53}$

Q = 0.0125 m^3/sec

Velocity of air flow through the duct

V = $\frac{Q}{Area\ of\ duct}$

= $\frac{0.0125}{0.15x0.1}$

= 0.83 m/s

Average temp of fin surface

Ts = $\frac{T1+T2+T3+T4+T5}{5}$

Ts = $\frac{176+143+138+126+90}{5}$

Ts = 134.5°C

Mean flim temp of fluid

Tf = $\frac{Ts+Ta}{2}$

= $\frac{134.5+40}{2}$

= 87.25°C

Properties of air at mean film temp Tf= 90°C
From Heat and mass transfer data hand book
Kinematic viscosity of air ϑ = 22.10 x $10^{-6}m^2$/sec
Prantl's no. Pr =0.690
Thermal conductivity K = 0.03128 w/mk
Reynold's Number

Re = $\frac{VD}{\vartheta}$

Re = $\frac{0.83\times0.023}{22.10\times10^{-6}}$

Re = 863.80

Nusselt Number
Corresponding to this Reynolds number, lead the values of C, m and n in the equation

Nu = Nu = C x Re^m×Pr^n

= 0.683 x $863.80^{0.466}$ x 0.690^{113}

Nu = 15

Heat transfer Coefficient

= $\frac{Nuxkair}{D}$

= $\frac{15X0.03128}{0.023}$

h = 20.4 w/m^2 k

m = $\sqrt{\frac{hP}{KA}}$

= $\sqrt{\frac{4h}{KD}}$

= $\sqrt{\frac{4X20.4}{160X0.02}}$

m = 5.04 m

Qfin = m x Kfin x Ao x (Tb – Ta) x tan h mL
= 5.04 x 160 x $\frac{\pi}{4}(0.023)^2$ x (178 – 40) x tan h (5.04 x 0.1)

= 21.51watt

Efficiency of the fin

$$nf = \frac{\tan h\, mL}{mL}$$

$$= \frac{\tan h\,(5.04 x\, 0.1\,)}{5.04 x\, 0.1}$$

= 92 %

Effectiveness of fin

$$€ = \frac{Qfin}{hA\,(Tb-Ta}$$

$$= \frac{21.51}{20.4\, x\, \frac{\pi(0.023)^2}{4}\, x\,(178-40)}$$

= 18.39

Temp at base of fin

Tb= T1+2 = 180+2 =182℃

Fins Calculation:

4) Hexagonal Fin

Air head causing flow of air through orifice

$$ha = \frac{Pw\ x\ hw}{Pa}$$

$$= \frac{1000 x 0.205}{1.055}$$

= 194.31 m

Volume flow rate of air through orifice

$Q = Cd \times ao \times \sqrt{2g \times} h_a$

$Q = 0.65 \times \frac{\pi}{4}(0.02)^2 \times \sqrt{2 \times 9.81 \times 194.31}$

$Q = 0.0120\ m^3/sec$

Velocity of air flow through the duct

$$V = \frac{Q}{Area\ of\ duct}$$

$$= \frac{0.0120}{0.15 x 0.1}$$

= 0.8 m/s

Average temp of fin surface

$$Ts = \frac{T1+T2+T3+T4+T5}{5}$$

$$Ts = \frac{180+114+103+105+93}{5}$$

Ts = 119℃

Mean film temp of fluid

$$Tf = \frac{Ts+Ta}{2}$$

$$= \frac{119+40}{2}$$

= 79.5℃

Properties of air at mean film temp Tf= 80℃

From Heat and mass transfer data hand book

Kinematic viscosity of air $\vartheta = 20.02 \times 10^{-6} m^2/sec$

Prantl's no. Pr =0.694

Thermal conductivity K = 0.022966 w/mk

Reynold's Number

$$Re = \frac{VD}{\vartheta}$$

$$Re = \frac{0.8 \times 0.023}{20.02 \times 10^{-6}}$$

Re = 919.08

Nusselt Number

Corresponding to this Reynolds number, lead the values of C,m and n in the equation

$Nu = Nu = C \times Re^m \times Pr^n$

$= 0.683 \times 919.08^{0.466} \times 0.694^{113}$

Nu = 14.53

Heat transfer Coefficient

$$= \frac{Nuxkair}{D}$$

$$= \frac{14.53 X 0.022966}{0.023}$$

$h = 14.50 w/\boldsymbol{m^2}\ k$

$$m = \sqrt{\frac{hP}{KA}}$$

$$= \sqrt{\frac{4h}{KD}}$$

$$= \sqrt{\frac{4X14.50}{160X0.249}}$$

m = 3.81 m

Qfin = m x Kfin x Ao x (Tb – Ta) x tan h mL

$= 3.81 \times 160 \times \frac{\pi}{4}(0.023)^2 \times (182 - 40) \times \tan h\,(3.81 \times 0.1)$

= 13.07watt

Efficiency of the fin

$$nf = \frac{\tan h\, mL}{mL}$$

$$= \frac{\tan h\,(3.81 X 0.1)}{3.81 X\, 0.1}$$

= 95 %

Effectiveness of fin

$$€ = \frac{Qfin}{hA\,(Tb - Ta}$$

$$= \frac{13.07}{14.50\, x\, \frac{\pi (0.023)^2}{4}\, x\, (182-40)}$$

= 15.27

Table 2
Experimental calculation result

Fin Configuration	Mean Film Temperature °C	Reynolds's Number	Nusselt Number	Heat Transfer Coefficient w/m^2 k	m Parameter	Heat Transfer Rate Watt	Efficiency %	Effectiveness E
CIRCULAR	85	894.26	14.33	21.61	4.84	21.25	92.36	16.10
RECTANGULAR	89.5	874.20	14.17	19.27	5.48	27.08	91	22.70
SQUARE	87.25	863.80	15	20.04	5.04	21.51	92	18.39
HEXAGONAL	79.5	919.08	14.53	14.50	3.81	13.07	95	15,27

8. CFD Analysis

A. CFD Modelling

Computational Fluid Dynamics (CFD) is the science of determining numerical solution of governing equation for the fluid flow whilst advancing the solution through space or time to obtain a numerical description of the complete flow field of interest. The equation can represent steady or unsteady, Compressible or Incompressible, and in viscid or viscous flows, including non-ideal and reacting fluid behaviour. The particular form chosen depends on intended application. The state of the art is characterized by the complexity of the geometry, the flow physics, and the computing time required obtaining a solution. Computational fluid dynamics (CFD) is a computer-based simulation method for analysing fluid flow, heat transfer, and related phenomena such as chemical reactions., since experiments have a cost directly proportional to the number of configurations desired for testing, unlike with CFD, where large amounts of results can be produced at practically no added expense. In this way, parametric studies to optimize equipment are very inexpensive with CFD when compared to experiments

B. Methodology

In CFD calculations, there are three main steps:

Pre-Processing, Solver Execution, Post-Processing.

Pre-Processing is the step where the modelling goals are determined and computational grid is created. In the second step numerical models and boundary conditions are set to start up the solver. Solver runs until the convergence is reached. When solver is terminated, the results are examined which is the post processing part.

Procedure:

- Step 1: Draw the cylindrical fin, rectangular fin, hexagonal fin and square fin in ANSYS Workbench or Space Claim 3D Modellers.
- Step 2: Using the created fins in ANSYS software for further simulations.
- Step 3: Use the data for analysis and calculating efficiency and other parameter.

C. CFD Analysis

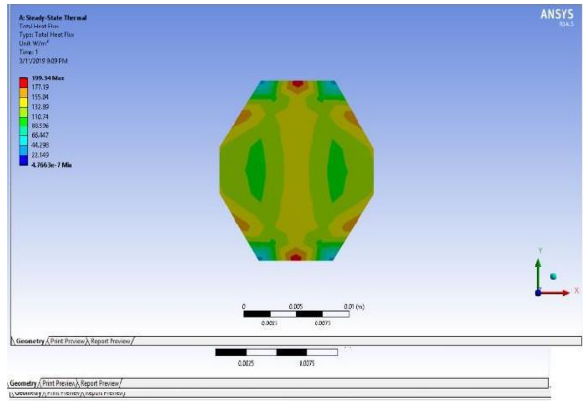

Fig. 5. Analysis of hexagonal fin

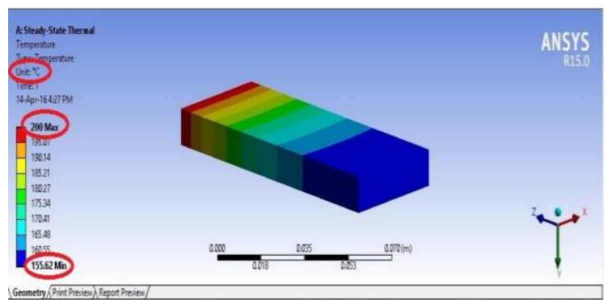

Fig. 6. Analysis of rectangular fin

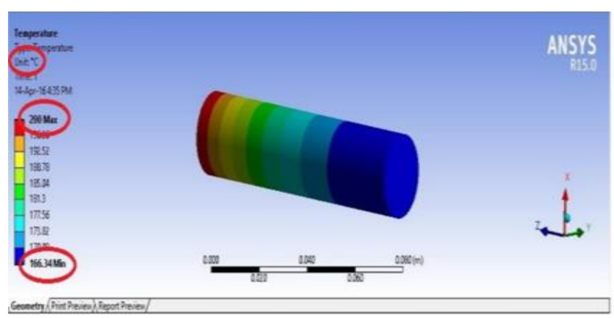

Fig. 7. Analysis of circular fin

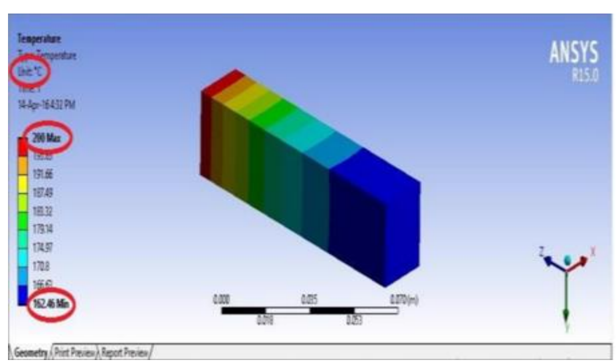

Fig. 8. Analysis of square fin

Table 3
Fins CFD analysis temp comparison

FIN CONFIGURATION	MAXIMUM TEMPERATURE	MINIMUM TEMPERATURE
RECTANGULAR	200	155.62
SQUARE	200	162.46
CIRCULAR	200	166.34
HEXAGONAL	200	150.21

9. Results, Discussion and Conclusion

In this present work, the first part of project includes the experimental investigation of fin effectiveness. We used four fins of different geometries i.e. circular, rectangular, Square and hexagonal of same material. The material which we have preferred is aluminum because of good thermal conductivity over the other material. We have find out the value of convective heat transfer coefficient, heat transfer rate, Reynolds's number, Nusselt number, Efficiency and Effectiveness.

In second part of project include the design and analysis of the fin by using ANSYS, CATIA and FLUENT software. The various input boundary condition is provided at the inlet of duct such as velocity of air, temperature and pressure, from the experimental investigation through the pressure drop, temperature drop, density difference and velocity difference for all the four fins.

From the output result of experimental calculation and analysis it is observed that hexagonal fin has better efficiency as compared to other 3 fins and also as per ansys analysis Temperature at the end of fin with hexagonal configuration is minimum, as compare to fin with other types of configurations. So the experimental and analytical computation for hexagonal fin is precisely equal.

According to the above data it is concluded that hexagonal fin is optimum for maximum heat transfer rate and efficiency due to maximum area of contact.

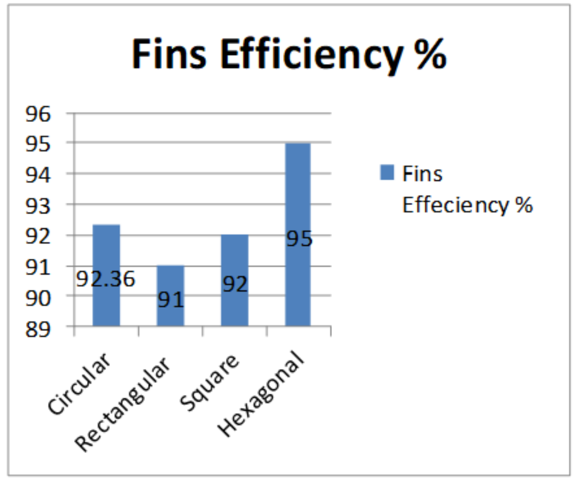

Fig. 9. Fins efficiency

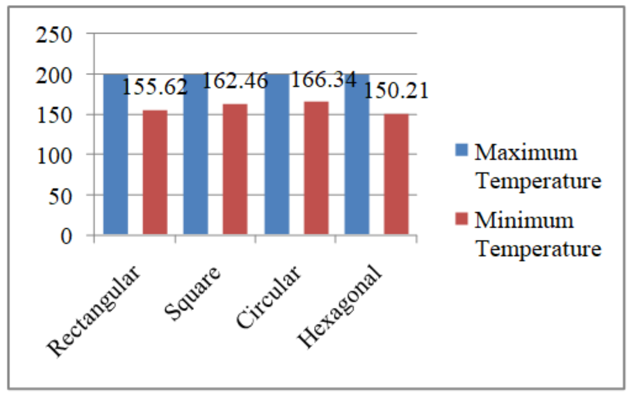

Fig. 10. Fins temperature difference in °C

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