HEAT TRANSFER ANALYSIS OF WINGLET TYPE FINS ARRAY THROUGH NATURAL CONVECTION

MR. M. M. DEOGHARE
Department of Mechanical Engineering, Pankaj Laddhad Institute of Technology & Management Studies, Buldana-443001, (M.S), India
PROF. K. R. SONTAKKE
Department of Mechanical Engineering, Pankaj Laddhad Institute of Technology & Management Studies, Buldana-443001, (M.S), India

ABSTRACT:

Fins are the extended surfaces used for increase heat transfer rate by convection method. In method of natural convection generally fins used should be of rectangular or circular shape, for intension to dissipate more heat to surrounding and maintain valuable components, parts safe and cool. From study of research paper it should be clear that the fins used for the heat dissipation should not arrange in large number in regulated space. In natural convection the air move along the fins as per buoyancy concept so that the air move upward when it becomes comes in contact with fins and lighter, but logical thing is that the natural air movement carried out in all direction there is no particular direction of air to strike the fin wall that means , air inlet direction is not fixed but the outlet becomes fixed. In case of horizontal based fins array when air strike to fins and heat transfer with these air but hot air trapped in between two fins and transfer rate of heat decreases, also in case of vertical based fins there is problem introduced of boundary layer development, so that fins with large fin height should not be used. To overcome above difficulties in natural convection to transfer heat effectively from plate to introduced winglet type fins array, because of winglet shape of fins air strike to fin wall in any condition of movement of air because the fins are inclined to both horizontal and vertical direction so that the heat transfer from fns is logically more than two cases but it interesting to investigate winglet angle of fins for more heat transfer by natural convection method.

KEYWORDS: Natural Convection, Numerical analysis, winglet Fins, Heat transfer, etc.

I. INTRODUCTION

In many engineering application heat generate which is unwanted by-product may decrease the performance of the systems since almost every engineering system is designed to work in a certain temperature limit. If these limits are exceeded by overheating, this may even lead to total system failure. Therefore many engineering systems try to avoid this overheating problem as much as possible by using different methods for dissipation of heat away from the system to surrounding. Using fins is one of the cheapest and easiest ways to dissipate unwanted heat and it has been commonly used for many engineering applications successfully. Rectangular fins are the most popular fin type because of their low production costs and high effectiveness. Although rectangular fins can be used in two different orientations as vertical and horizontal, vertical orientation is used more widely since it is more effective than the horizontal one.

In this case winglet types of fin for heat transfer by convection method are used. Heat transfer takes place while dissipating heat from fins to surrounding. Since the entire fin configurations are made of aluminium alloys, which have low emissivity values, radiation heat transfer values are low. Therefore convection heat transfer is the dominant heat transfer mode while dissipating heat from fins. The heat transfer from fins has been the subject of experimental and numerical investigations.

II. PROBLEM STATEMENT:

The main objective of present work is to find out heat transfer rate of vertical plate with winglet fins array at four different angles (90^o, 70^o, 50^o and 30^o) by natural convection method and the optimum angle at which heat transfer rate maximum by taking fix spacing between two fins.

Under this aim, sequence of work is as given below,

1) Find out the heat transfer rate of vertical plate with winglet fins array numerically by using appropriate analysis software CFD tool.
2) Plot tables and graph for different fins angle result.
3) Check for optimum fin angle for more heat transfer rate.

III. NUMERICAL ANALYSIS OF FINS:

a. GEOMETRY CREATION:

The setup consists of vertical square (200 mm × 200 mm) base plates. The base plate considered with 3 mm thickness. The fins are attached to these base plates externally and are exposed to ambience. The fins are in eight pair, at one side of plate and to the other side of plate there is an insulated material of wooden block of 250 mm x 250 mm in dimension and thickness is 50 mm. The height of fins are 25 mm, length of fins are 52 mm and the fins are 3 mm in thickness, are attached to base plate on one side of plate. Plate and fins are both made up of aluminum material.

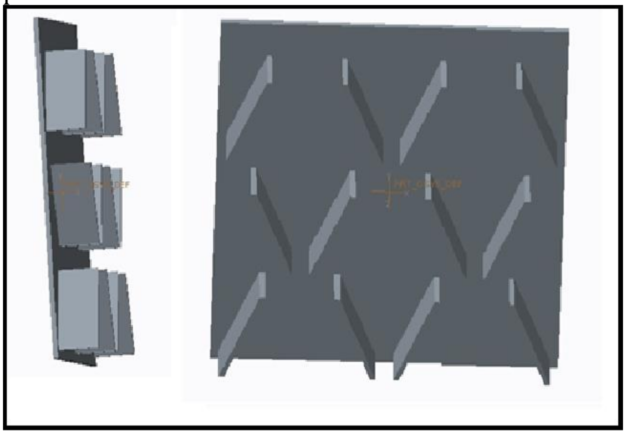


Fig.: Fins & base plate side & front view.

Figure shows geometry of fin and plate created by using CREO software, dimension of aluminum plate is 200 mm × 200 mm and on this plate aluminum fins are attached in pairs, there proper angle between each pair which is mansion below.

Height of fins (H) = 25 mm.

Length of fins (L) = 52 mm.

Total number of fins (N) = 12.

Fin spacing between two fins = 10 mm.

Complete base plate area = 200 mm × 200 mm.

Thickness of fins (t) = 3 mm.

Fin angles = (90^o, 70^o, 50^o, 30^o).

b. MESH GENERATION:

The Fluid and solid domain is meshed using unstructured grid with 'Tetra/mixed cells 'and mesh method is used robust (octree). The base plate-fin assembly is hung in the air surrounded by the domain (Enclosure). Eventually the set up is symmetrical about vertical plane.

Hence, only half set up is modeled. It is logical to assume that the behavior of the system in this half domain is similar to the behavior of the whole system. Modeling only half geometry reduces the total number of cell count, overall mesh size and thus also the computation time considerably.

The set up has two continuums. The solid continuum refers to base plate and fin(s). The fluid continuum consists of air surrounding the solid continuum (ambience) and is defined as the fluid domain. The domain has to be built around the base plate fin assembly, to study fluid mass flow and thus the heat flow from the base plate and fin.

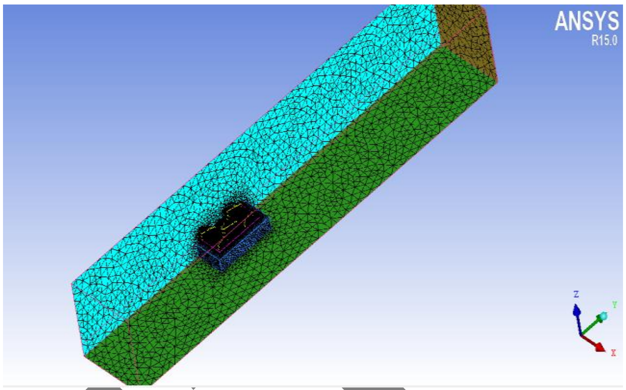


Fig.: Mesh structure of fins, plate and domain

Figure shows the actual meshing of Computational domain, wooden block, plate and aluminum fins. In this diagram we clearly observe that the mesh size of computational domain is large in size as compare to other component, also fins having finer mesh than other component so that it appears like in blackish color. We used finer mesh, where we have to calculate value by numerical calculation.

C. BOUNDARY CONDITIONS

The base plate & fins volume is given a uniform heat source in Kelvin. At the all surfaces of fins and base plate consider as uniform temperature heat source in Kelvin, ambient conditions are assumed by specifying free boundary condition in the form of pressure inlet. The top boundary is defined as pressure outlet allowing air to leave the domain. Symmetry is assumed along the mid boundary plane splitting the included winglet-angle as well as at the back bounding surface of the domain. All the cases are solved by specifying the heat source values at base plate &fins volume. The air enters through the pressure inlet boundaries at the ambient temperature T_a (T_{inlet}) and corresponding density ρ_0. Also the back flow of air (if exist) will be at the ambient temperature T_a and corresponding density $\rho_{0.}$ Here the ambient pressure is used as stagnation boundary condition with the incoming mass having the ambient temperature. A coupled boundary condition exists for the fin wall which transfers the heat from the base plate & fins to fluid.

For the purpose of numerical calculations in computer and as regarding to our project basis we take fins and base plate as complete geometry apply uniform temperature as about 318^0 Kelvin all over the geometry, because this analysis only about natural convection concept .In this platform consider worst condition of temperature at this level temperature of fins and base plate become same, and only considered convection method and their effect on temperature.

d. SOLVER

For segregated 3D second order steady and unsteady solver, the SIMPLE and PISO Pressure-Velocity Coupling algorithms for the pressure correction process are used respectively. The discretization scheme used is standard discretization for Pressure and second order upwind discretization for Momentum and Energy.

Ability to converge the results of numerical calculation means the imbalances in the iterative method have successfully fallen below the specified tolerance limits. The convergence can be associated in two ways, first one in which the solution not changing with the iterations and in second the solution not changing with the mesh. There are five residuals to be monitored in natural convection problem: continuity, X-velocity, Y-velocity, Z-velocity and energy. The default convergence criteria are 0.001 for all four of the above i.e. continuity and velocities and 10^{-06} for energy, for all the configurations. Same values are used for first order scheme. Once the solution is converged in first order; the convergence criteria is shifted to10^{-04} during second order for the three velocities. It is confirmed that beyond this limit, the changes in the average base plate-fin surface temperature, surface heat flux and base plate-fin surface heat transfer coefficient are negligible. For steady state solver, the solution does not converge for continuity, because of poor accountancy of flow separation, boundary layer separation and reattachment (secondary boundary layer formation) across V-fins. Unsteady formulations are used to get the steady solution. Unsteady solver gives solution closer to the experimental results. Use second order upwind scheme in energy and momentum for better accuracy in numerical calculations. There are two methods we can use in natural convection process, but now for these results use SIMPLE algorithm method.

IV. FLOW VISUALIZATION:

One of the most useful advantages of CFD is its ability to visualize the flow easily. In this section the variation of temperature of the flow with different spacing's investigated in previous is presented with the help of CFD visualization. Since there are different fin array angle configurations investigated in this study.

For 90° fin angle

For 70° fin angle

For 50° fin angle

For 30° fin angle

a. STATIC TEMPERATURE:

The Static temperature contours for various fins spacing are shown below, also we observe their flow differences for different fin angles.

For 50° Fin Angle:

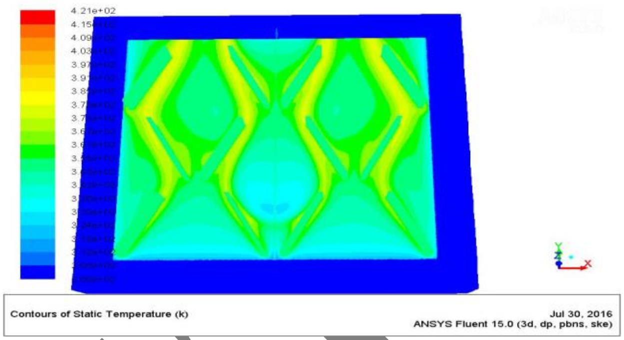


Fig.: Static temperature contours on fin and plate

Above image shows contours of static temperature on plate and fin for 50° fin angle. Static temperature scale has shown in left side of image, from this image it is clear that maximum temperature reaches to 361K.

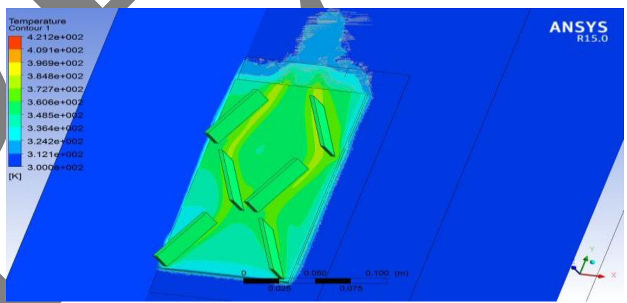


Fig.: Static temperature contours in computational domain

Above figure shows flow of static temperature contours of plate and fins. Static temperature is the temperature measured with the sensor traveling at the same velocity as the gas. In other word the temperature with no velocity effects. Lower temperature is display by blue color and higher temperature display by red zone.

b. VELOCITY VECTORS AND VELOCITY STREAMLINE:

The velocity vectors for various fins angles are shown below; also observe their flow differences for different fin angles.

For 50° fin angle

Below figure shows velocity vector of air movement in between 50° fin angles and there direction of movement, also image shows air velocity at different place.

NOVATEUR PUBLICATIONS
International Journal of Research Publications in Engineering and Technology [IJRPET]
ISSN: 2454-7875
VOLUME 3, ISSUE 1, Jan. -2017

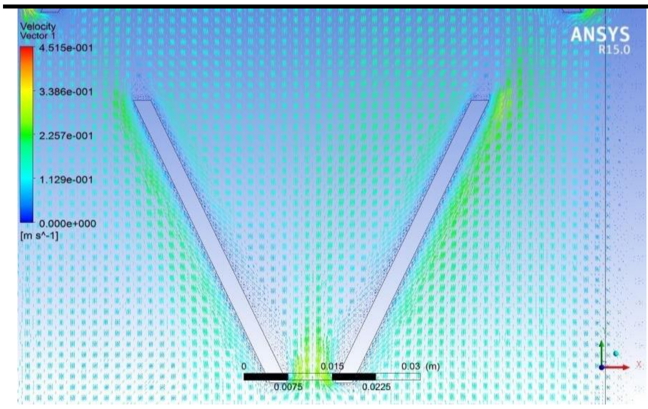


Fig.: Velocity vectors of 50º fin angle

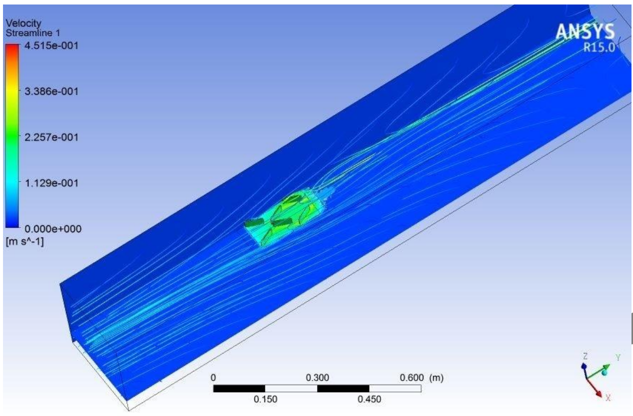


Fig.: Velocity streamlines of 50º fin angle

Above figure shows that the velocity of air increases when air strikes at hot part of plate fin because air become more lighter and move in upward direction.

V. RESULTS AND DISCUSSION:

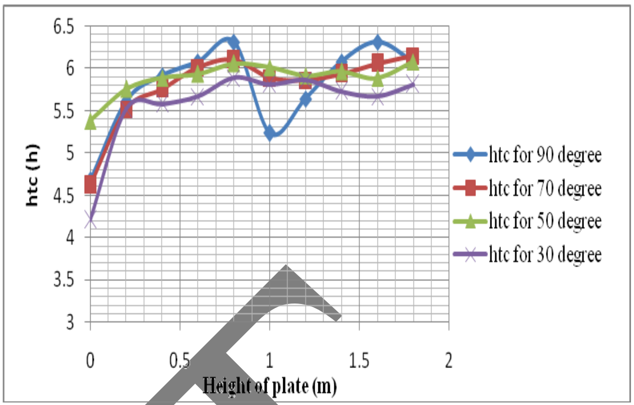
a. COMPARISON OF NUMERICAL 'h' VALUE USING CORRELATION:

Table Compare numerical values of 'h'

Sr. No.	Point Height (m)	'h' for 90º	'h' for 70º	'h' for 50º	'h' for 30º
1	0	4.675605878	4.617895037	5.381330502	4.210164247
2	0.2	5.610385327	5.516305675	5.755656634	5.516305675
3	0.4	5.914013646	5.755656634	5.888694057	5.57964907
4	0.6	6.080795712	6.011444534	5.938936295	5.670110456
5	0.8	6.31203103	6.103258216	6.058012263	5.888694057
6	1	5.233431504	5.888694057	6.011444534	5.81021872
7	1.2	5.640531406	5.862964389	5.914013646	5.862964389
8	1.4	6.080795712	5.938936295	5.963474497	5.727653192
9	1.6	6.31203103	6.058012263	5.888694057	5.670110456
10	1.8	6.080795712	6.147255908	6.080795712	5.81021872

Above table shows that value of heat transfer coefficent numbers which are obtained numerically from all four cases i.e (90º, 70º, 50º, 30º fin angles) which tabulated at one place for comparison purpose.

b. COMPARARISON GRAPH OF NUMERICAL 'h' VS HEIGHT OF PLATE:



Graph : Compare numerical 'h' Vs Height of plate

In above graph compare numerically calculated value of heat transfer coefficient in all four cases; for that plot the graph of heat transfer coefficient verses height of plate. On 'X' axis take value of height of plate and on 'Y' axis take value of heat transfer coefficient of four cases i.e (90º, 70º, 50º, 30º fin angles). From the above graph it is clear that, the average value of heat transfer coefficient for 50º fin angle is more better than other cases.

VI. CONCLUSION:

The main aim of this project is to select the optimum fin angle for better heat transfer coefficient (h). For this to calculate the heat transfer coefficient at each point for different fin angles numerically and the optimum fin angle is selected by averaging the values of heat transfer coefficients at respective points for a particular fin angle. Tables and graphs shows the values of 'h' at various points, from this it is observed that the value of heat transfer coefficient for 50º fin angle is more than other fin angles values. So, finally it is conclude that for 50º fin angle is optimum for better heat transfer performance.

VII. REFERENCES:

1. R. L. Edlabadkar, N. K. Sane, G.V. Parishwad , "*Computational Analysis Of Natural Convection With Single V-Type Partition Plate*", Pvg's College Of Engg. And Technology, JSPM's College of Engineering, Govt. College of Engineering, Pune. 5th European Thermal-Sciences Conference, The Netherlands, 2008.

2. Burak Yazicioğlu, "*Performance Of Rectangular Fins On A Vertical Base In Free Convection Heat Transfer*", A Thesis Submitted To The Graduate School Of Natural And Applied Sciences Of Middle East Technical University, January 2005.

3. Rameshwar B. Hagote, Sachin K. Dahake, "*Enhancement of Natural convection heat transfer coefficient by using V-fin Array",* Student of mechanical

Engg. Department, MET's IOE, Adgaon, Nashik (Maharashtra, India), April, 2015.

4. Manuja Pandey, Utkarsh Prasad, Vineet Kumar, Nafish Ahmed, "*Experimental Analysis of Cooling Fins*" Assistant Professor, Mechanical Engineering Department, MGM'S COET, Noida, India, 02 May-2015.

5. Guei-Jang Huang, Shwin-Chung Wong, Chun-Pei Lin, "*Enhancement of natural convection heat transfer from horizontal rectangular fin arrays with perforations in fin base",* Department of Power Mechanical Engineering, National Tsing Hua University, Hsin-Chu 300, Taiwan, ROC 11 January 2013.

6. Umesh V. Awasarmol, Ashok T. Pise , "*An experimental investigation of natural convection heat transfer enhancement from perforated rectangular fins array at different Inclinations"*, a Department of Mechanical Engineering, Army Institute of Technology, Pune, MS 411015, India

7. Mehran Ahmadi, Golnoosh Mostafavi, Majid Bahrami, *"Natural convection from rectangular interrupted fins",* Laboratory for Alternative Energy Conversion (LAEC), Mechatronic Systems Engineering, Simon Fraser University (SFU), Surrey, BC, Canada 18 September 2013.

8. M.J. Sable, S.J. Jagtap , P.S. Patil , P.R. Baviskar & S.B. Barve, " *Enhancement Of Natural Convection Heat Transfer On Vertical Heated Plate By Multiple V-Fin Array",* Faculty Of Engineering, Department Of Mechanical Engineering, Rajarshi Shahu College Of Engineering. Tathawade, Pune-33, India. November 2010

9. Mahdi Fahiminia1, Mohammad Mahdi Naserian, Hamid Reza Goshayeshi, Davood Majidian, *"Investigation of Natural Convection Heat Transfer Coefficient on Extended Vertical Base Plates",* Department of Mechanical Engineering, Mashhad Branch, Islamic Azad University, Mashhad, Iran Energy and Power Engineering, 2011.

10. H. R Goshayeshi, F. Ampofo, "*Heat Transfer by Natural Convection from a Vertical and Horizontal Surfaces Using Vertical Fins",* Department of Mechanical Engineering, Azad University of Mashhad (IAUM), Mashhad, Iran Energy and Power Engineering, 2009.

11. Wankhade Sachin D. , Bhor S.K ,Nagraj A. M, *"Investigation of Thermal Performance in Natural Convection from Rectangular Interrupted Fins",* P.G. Student, Department of Mechanical Engineering, Dr.D.Y.Patil SOEA, Ambi, Maharashtra, India 1, January 2015.

12. Prof.S.A.Wani, Prof.A.P.Shrotri, Prof.A.R.Dandekar, *"Experimental Investigation of Natural Convection Heat Transfer from a Fin Array",* Assistant Professor, Padmabhooshan Vasantraodada Patil Institute of Technology, Budhgaon 1, 2016.

13. Starner K.E. and McManus H.N., "*An Experimental Investigation of Free Convection Heat Transfer from Rectangular Fin Arrays"*, Journal of Heat Transfer, 273-278, (1963).