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Optimization of fluid flow and heat transfer using Taguchi method for flow around heated semicircular fin

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Abstract

Optimum design of cooling devices is an integrated part of any attempt to improve the efficiency of a thermal system. The use of extended surfaces is often more economical, convenient and trouble free, most proposed application of increasing surface area is adding fins to the surface in order to achieve required rate of heat transfer. In current paper, Optimization of semicircular fin is investigated by using Taguchi method. The parameter which is having either more or less impact on the heat flux of fin is to be determined and the performance of individual parameters is analyzed. In order to accomplish the required rate of heat dissipation, with the least amount of material, the most favourable combination of fin geometry and orientation is indispensable for the enhancement of heat transfer rate. To simulate the semicircular fin at diverse heat inputs and compare its heat transfer characteristics (convective heat transfer coefficient & Nusselt number) with vertical plate, vertical fin and V-shape fin. Also, parameters prediction like velocity, temperature and pressure sketches for constant heat flux on base-plate to obtain a correlation which estimates the optimum fin height values for maximum convection heat transfer rates from the fin is the aim of the work.

Key words: Semicircular fin, Taguchi method, Nusselt number, optimization.

1.0 Introduction

Devices generate heat while operating; the heat generated should be within the recommended working temperatures for the appropriate functioning of the system, because it may lead to faulty functioning and overheating. The space allocated for heat dissipation units reduced, as the devices are going to be diminutive in the upcoming days. Research is being done to increase the effectiveness in heat transfer by reducing the volume to weight ratio of the heat exchanging bodies. Optimization in its broad sense can be applied to solve any engineering problem. Optimization techniques are currently being used in a wide variety of industries, including aerospace, automotive, MEMS, chemical, electrical and manufacturing industries.

Misumi and Kenzo (1990) have reported an experimental work on enhancement of natural convection heat transfer from vertical plate having a horizontal partition plate and V-plates in the water ambience. They found that the heat transfer in the downstream region of the partition plate is markedly enhanced when the plate height exceeds certain critical values because of the inflow of the low temperature fluid into the separation region. For vertical plate with V-shaped fins, the heat transfer coefficient obtained was 40% higher than the conventional fins.

Edlabadkar et al. (2008) did experimental investigation on single V-type partition plate with different included angles, in air as ambience in the laminar air flow over a vertical base plate with length 0.3m, width 0.3m, and V shape fin (the fin limb length is 0.15m and width 0.05m) attached to it was numerically captured using FLUENT with laminar viscous model. Computations were performed for the geometrical configurations with fin included angles 90°, 120° and 60°, for equal base and fin areas dissipating heat under natural convection condition for temperature difference θ, varying from 30°C to 150°C. The results show that the 90° V fin gives least resistance to flow separation in the upstream region and most effective high heat transfer region in the downstream region of the base plate. It was observed that among the three V-type partition plates, the maximum increase in heat transfer enhancement is 12% for 90° V-partition plates as compared to vertical partition Plate and 15.27% as compared to horizontal partition plate.

Palande and Mahalle (2013) have worked on experimental investigation of enhancing heat transfer from vertical rectangular and V-fin under natural convection condition. It is observed that

effect of heat input on heat transfer coefficient, Nusselt number and percentage rise in heat transfer respectively. It is concluded that V-plate fin gives higher heat transfer performance than vertical or rectangular plate and heat transfer coefficient and Nusselt number increase with increasing heat input. The performance improvement for vertical fin is from 12.11% to 15% and for V- fin 13.52 to 15.64%.

Sable et al. (2010) investigated heat transfer enhancing technique for natural convection adjacent to a vertical heated plate with a multiple V- type partition plates (fins) in ambient air surrounding. They concluded that as compared to conventional vertical fins, the V-type partition plates work not only as extended surface but also as flow turbulator. The tall vertical fin array restricts the heat transfer enhancement from tall vertical base plate. This is because of the boundary layer thickening and subsequent interference developed over the height. The experiments were conducted with the width of the partition plate (fin height) varying from 20 mm to 38 mm for a plane vertical plate, vertical plate with vertical fins, vertical plate with V-fins with bottom spacing and vertical plate with different V-type fins. It is further observed that the base heat transfer coefficient (hb) of V-type fin array is better than all other configurations. The hb for plain plate is least among all. The V-type fin array better diminishes the stagnation high temperature fluid in upstream region of the plate. Thus, in this investigation work, a totally new heat transfer technique is found out to increase the rate of natural convection heat transfer on vertical heated plate. The V-type fin array can be seen as the combination of a horizontal and vertical partition plates. For the same surface areas, V-type partition plates gave better heat transfer performance than vertical rectangular fin array and V-fin with bottom spacing type array.

Hagote and Dahake (2014) did experimental study of natural convection heat transfer on horizontal, inclined and vertical heated plate by V-fin array. They were using V-shaped fin with 60^0 included angles because the maximum convective average heat transfer coefficient was obtained for 60^0 V-fin arrays. They conducted an experiment on base plate of V-fin arrays plate which is place at different orientation such as horizontal, vertical and inclined at 45^0 to horizontal. From the experimental analysis, it is observed that as compared to horizontal plate and inclined plate orientation the vertical plate orientation with V-fin array of 60^0 included angles and with surface area would give greater average heat transfer coefficient. They established a match of values between the experimental results and the results obtained by using CFD software.

2.0 Methodology

Semicircular fin is analyzed under constant heat flux condition by using ANSYS 17.2 software. Heat transfer characteristics of semicircular fin such as Nusselt number, heat transfer coefficient etc. would be compared with V-shaped fin. Semicircular fin height is to be optimized by varying between 5mm and 30mm with step of 5 under different heat inputs and other parameters are kept constant. Non-dimensional number i.e., Nusselt number, Prandtl number and Grashof number of all fin height would be calculated and correlation between Nusselt number and Rayleigh number with fin height is determined. Semi-circular fin geometrical parameters such as fin height, thickness, length and included angle, etc. are varied within certain limit to find out optimum shape of fin which would give better performance. Taguchi method is used for optimizing a semicircular fin by using orthogonal array L25 with 5 levels of each parameters. Output parameter (average surface temperature of fin) of each 25 operations would be calculated numerically by using CFD-POST software. Best combination out of 25 is obtained using larger-the-best and smaller- the-best approach. Effect of each of parameters is also obtained with help of analysis of variance (ANOVA) technique.

3.0 Governing equations

The governing equations for a convective heat transfer process are obtained by considerations of mass and energy conservation and of the balance between the rate of momentum change and applied forces. These equations may be written, for constant viscosity μ and zero bulk viscosity as:

Continuity equation:

$$\frac{D\rho}{Dt} = \frac{\partial \rho}{\partial t} + V.\nabla\rho = -\nabla\rho.V$$

Momentum equation:-

$$\rho\frac{DV}{Dt} = \rho\left(\frac{\partial V}{\partial t} + V.\nabla\mathbf{V}\right) = F - \nabla\mathbf{p} + \mu\nabla^2\mathbf{V} + \frac{\mu}{3}\nabla(\nabla.V)$$

Energy equation:-

$$\rho\, Cp\frac{DT}{Dt} = \rho\, Cp\left(\frac{\partial T}{\partial t} + V.\nabla\mathbf{T}\right) = \nabla.(k\nabla\mathbf{T}) + q''' + \beta T\frac{Dp}{Dt} + \mu\Phi v$$

where **V** is the velocity vector, *T* is local temperature, *t* is time, **F** is body force per unit volume, C_p specific heat at constant pressure, *p* the static pressure, ρ the fluid density, β the coefficient of thermal expansion of the fluid, Φ*v* the viscous dissipation (which is the irreversible part of the energy transfer due to viscous forces), and *q* the energy generation per unit volume.

In addition to conservation of mass, momentum and energy equations, Buoyancy forces are responsible for the fluid motion in natural convection. Viscous forces oppose the fluid motion. Buoyancy forces are expressed in terms of fluid temperature differences through the volume expansion coefficient.

$$\beta = -\frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_p = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p \quad \left(\frac{1}{k}\right)$$

For ideal gas

$$\beta_{\text{ideal gas}} = \frac{1}{T}$$

In the momentum equation, the local static pressure may be broken down into two terms: one, *pa*, due to the hydrostatic pressure, and other, *pd*, the dynamic pressure due to the motion of the fluid (i.e., $p = pa + pd$). If ρ∞ is the density of the fluid in the ambient medium, we may write the buoyancy term as

$$F - \nabla p = (\rho g - \nabla pa) - \nabla pd = (\rho\mathbf{g} - \rho\infty\mathbf{g}) - \nabla$$

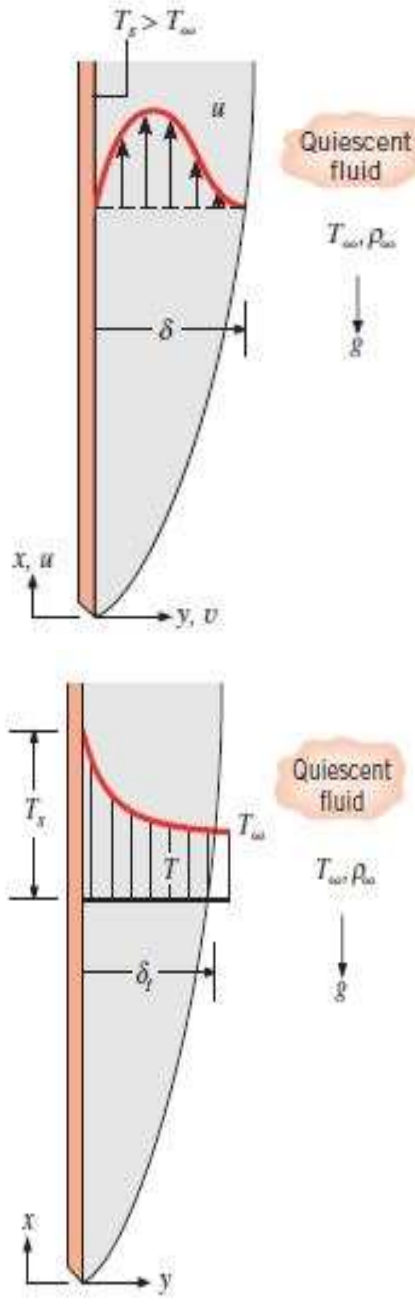
If **g** is downward and the *x* direction is upward (i.e., **g** = −**i***g*, where **i**is the unit vector in the *x* direction and *g* is the magnitude of the gravitational acceleration, as is generally the case for vertical buoyant flows shown in Fig. 1), then

$$\mathbf{F} - \nabla p = (\rho\infty - \rho)g\mathbf{i} - \nabla pd$$

The buoyancy term appears only in the *x*-direction momentum equation. Therefore, the resulting the momentum equation becomes

$$\rho\frac{DV}{Dt} = (\rho\infty - \rho)g - \nabla pd + \mu\nabla^2\mathbf{V} + \frac{\mu}{3}\nabla(\nabla.V)$$

…eq (3.4)



(a)

(b)

Figure 1: Boundary layer development on a heated vertical plate: (*a*) Velocity boundary layer. (*b*) Thermal boundary layer

These equations have to be solved simultaneously to determine the velocity, pressure, and temperature distributions in space and in time. Due to this complexity in the analysis of the flow, several

simplifying assumptions and approximations are generally made to solve natural convection flows.
For the numerical analysis, the following assumptions are imposed.

1) The flow is steady, laminar, and three-dimensional
2) Except density, the properties of the fluid are independent of temperature
3) Air density is calculated by treating air as an ideal gas
4) Radiation heat transfer is negligible

4.0 CFD Analysis:

Geometry of semicircular fin is made in design modular as shown in Fig. 2. For optimized height of fin, another parameter of fin is constant and varying fin height from 5mm to 30mm. Semi-circular fin radius and angle of semicircular fin are 60 mm and 180^0. Base plate dimension is selected depending on length of fin. Square base plate with 190 mm × 190 mm is used. Base plate and fin thickness are 3 mm and 1.5 mm respectively. Aluminum is material used for fin & base plate with its properties of specific heat C_p = 2800 J/kg K, thermal conductivity k=193 W/m K and density = 880 Kg/m^3

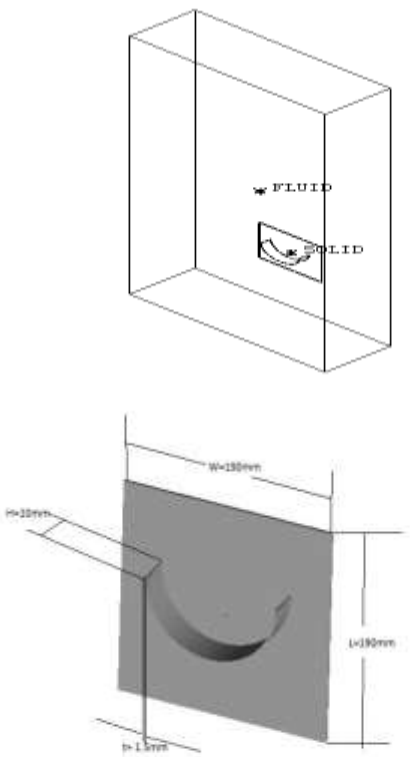


Figure 2: Geometry of fin and domain

Table 1: Parameters and levels of semicircular fin

Parameter		Level				
		1	2	3	4	5
A	Fin length (mm)	190	220	252	285	315
B	Fin height (mm)	10	15	20	25	30
C	Included Angle (degree)	180	157.5	135	112.5	90
D	Fin thickness (mm)	1	1.5	2	2.5	3

For optimum shape of fin, fin length, height, thickness and included angle parameters are chosen. The levels of each parameter are shown in Table 1.

Approach: SIMPLE (Semi-implicit method for pressure linked equation) scheme is employed as the laminar model because of the inherent laminar nature of the physics of the fluid flow of given problem.

5.0 Results and Discussions

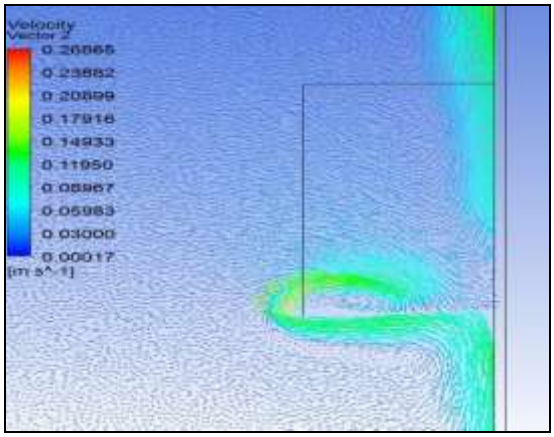
5.1 Comparison of Semicircular Fin with V-fin: -

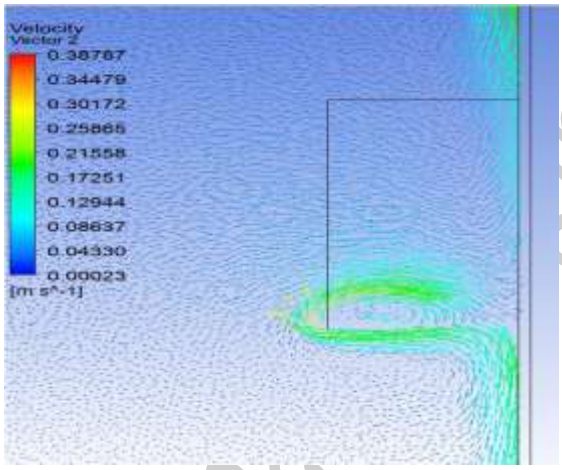
Simulations for semicircular fins are carried out under same condition and same surface area of V-fin for various values of heat input to the given plate and results are obtained to understand the behaviour of semi-circular fin under natural convection and compare its heat transfer characteristics with V-fin. The temperature, pressure and velocity contour for all heat inputs are plotted.

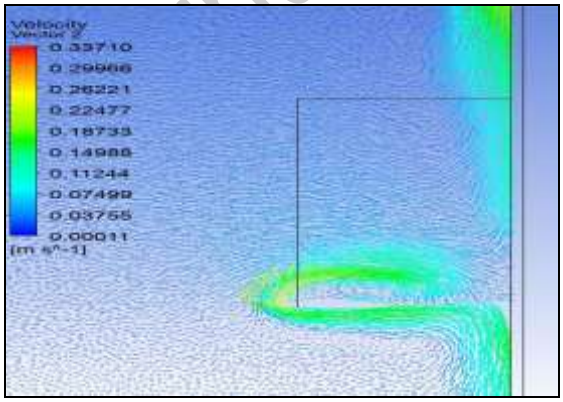
Velocity vector at plane along the centre of semicircular fin:

Nature of fluid flow under natural convection of semicircular fins is predicted by plotting velocity vector at mid plane of semicircular fin. Fluid near the bottom of base plate gets heat from plate and moves upward because of density difference. Then it sticks at bottom surface of then fin. Hot fluid move out from bottom surface.Denser cold fluid near the fin zone pushes the hot fluid and form circulation of fluid above the fin surface. More amount of heat extrated from fin occurred due to

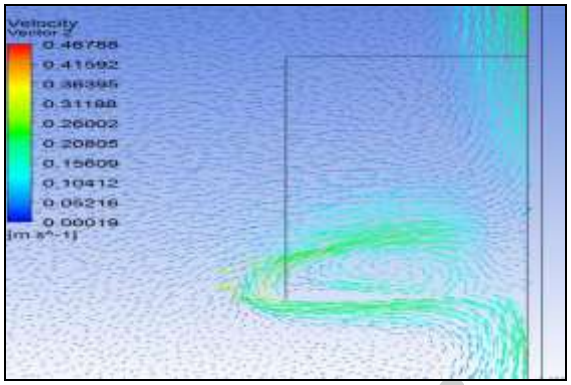
circulation of fluid. Centre of circulation of fluid shifted toward to baseplate and also increase the circulation zone when heat inputs are varied as shown in Fig. 3 (a) to (f). Flow velocity increase at end of fin height. it get decrese because of circulation occur at top surface of semicircular fin. New boundary layer of fluid form after circulation zone and it developed along length of baseplate. Maximum velocity reached at top of base plate.

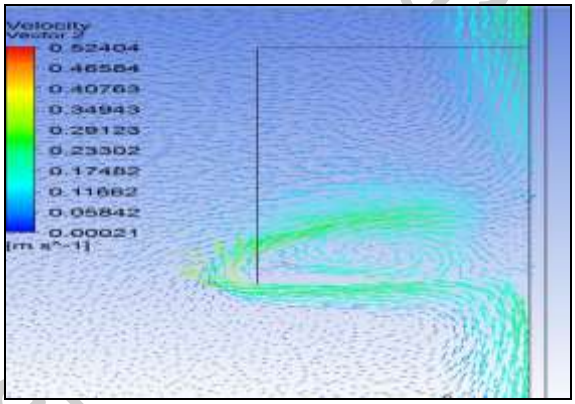


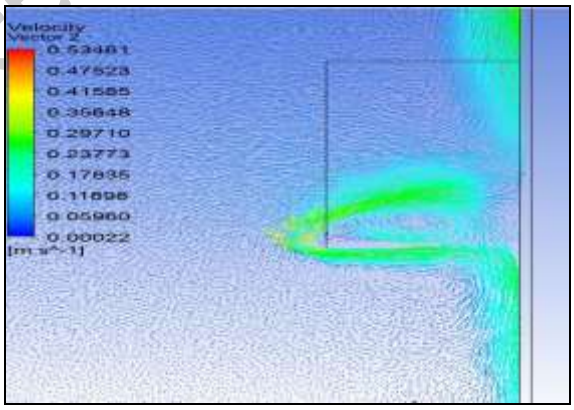




(b)







(e) (f)

Figure 3: Velocity vector for flow arround semicircular fin under natural convection at (a) 20 W (b) 40 W (c) 60 W (d) 80 W (e) 100 W (f) 120 W with fin height = 50 mm, fin thickness = 3 mm and length of fin =300 mm.

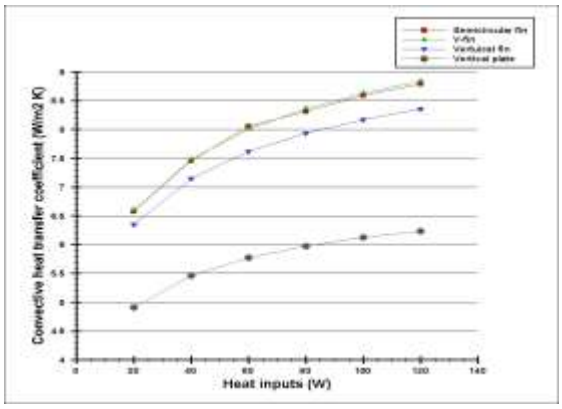


Figure 4: Variation of convective heat transfer coefficient with different heat inputs.

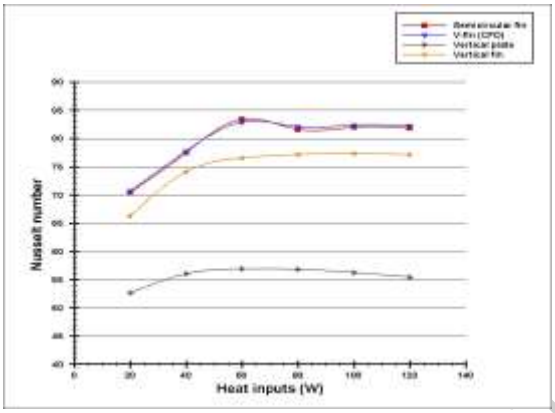


Figure 5: Variation of Nusselt number with different heat inputs.

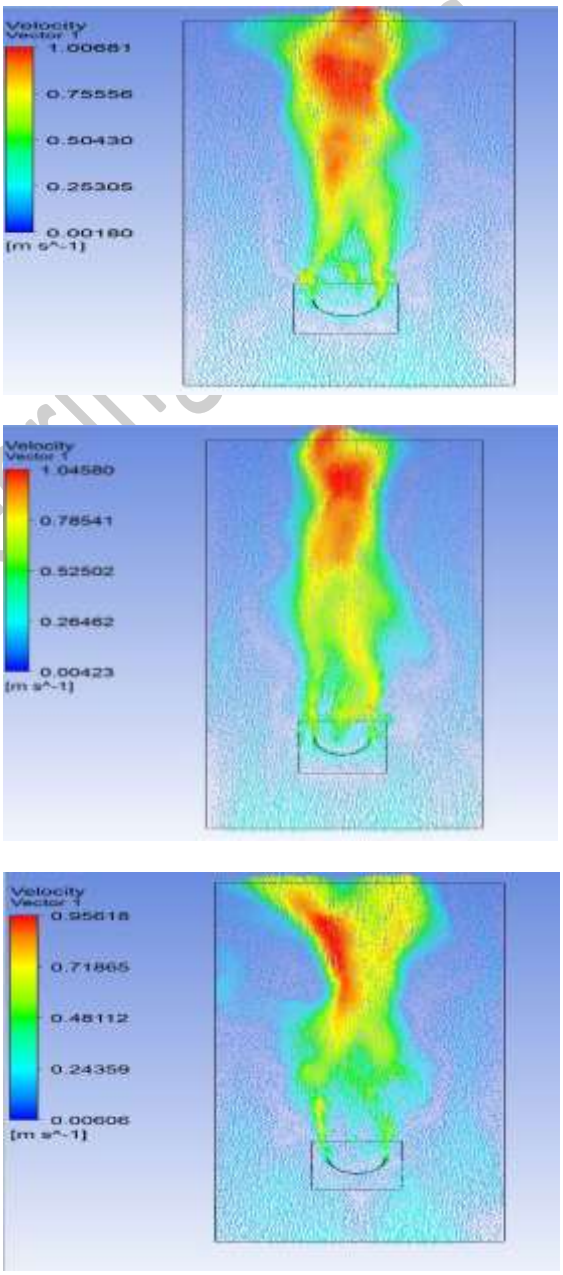
5.2 Heat input = 100W : -Temperature contour, velocity vector and pressure contour are captured for knowing the behivor of semicircualr fin as below.

Velocity vector at plane along the centre of fin height:-

For all fin height, velocity vector at plane across the centre of fin height are also captured as shown in Fig. 6 (a) to (f). Because of density difference, velocity of hot fluid which comes from fin and base plate, get raised at top boundary. Fluid flow pattern get change from uniform to discrete as fin height increased due to near base plate more amount of heat dissipation and then its decrease. Velocity vector is also show that direction of vector separates at bottom surface due to stagnant of fluid. Then boundary layer develops along both side of bottom curve surface of fin.

5.2 Pressure contour at plane along centre of fin: -

High pressure zone created at bottom of fin increases up to some distance from baseplate as increase in the fin height. Suction pressure generated just above the fin surface and forms circular suction zone of pressure. For fin height 5 mm, high pressure is formed at the top of fin surface due to striking of hot fluid coming from corner of fin height. As the fin height is increased, the striking of this fluid caused is less since the cold fluid tries to accommodate the hot fluid zone.



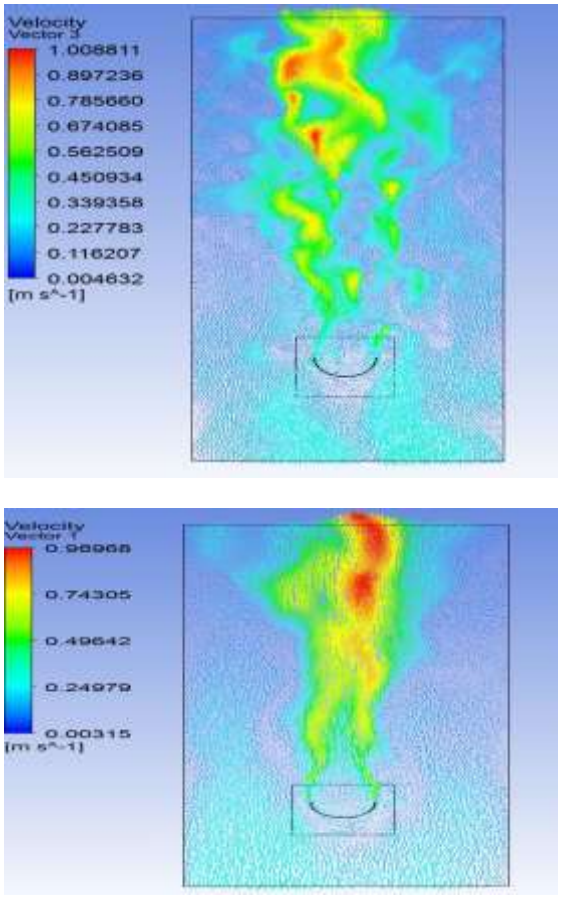
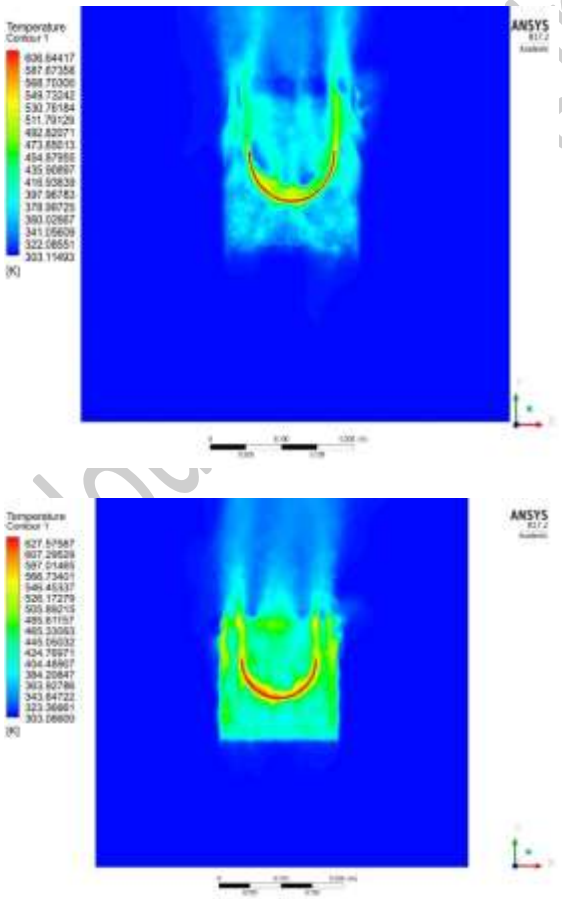


Figure 6: Velocity vector at plane along centre of fin height at heat input 100 W for fin height (a) 5mm (b) 10mm (c) 15mm (d) 20mm.



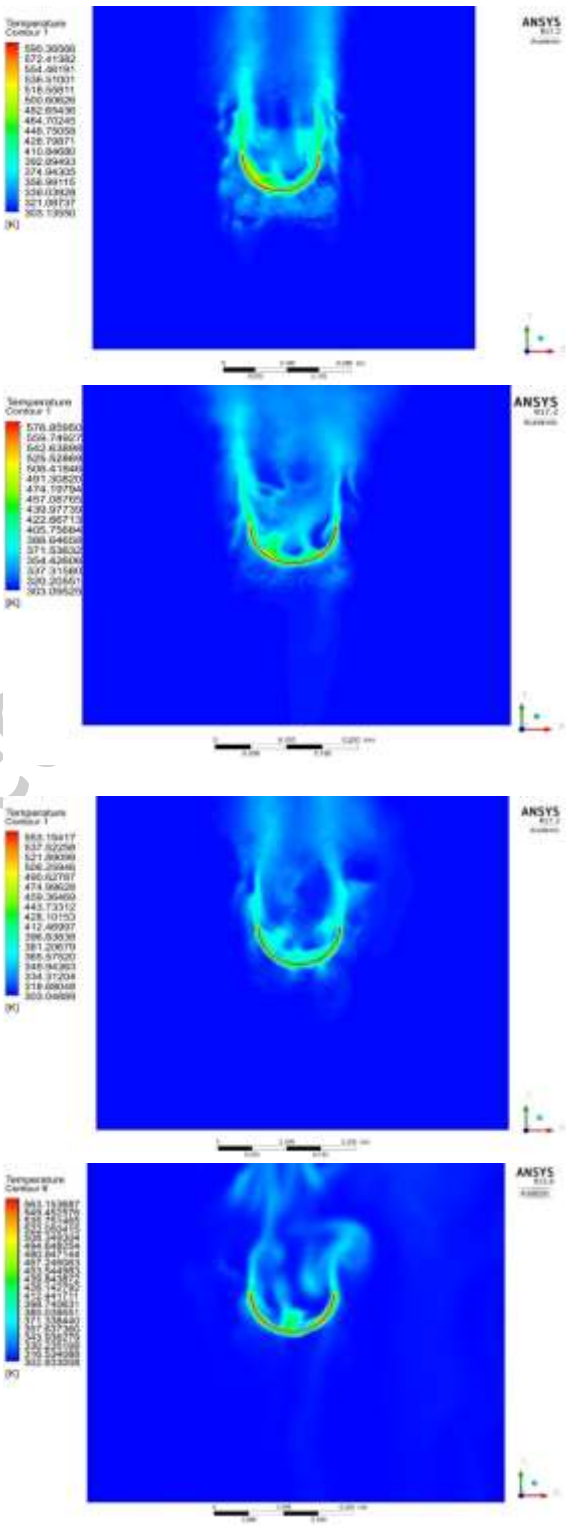


Figure 7: Temperature contour at plane along centre of fin height at heat input 100W for fin height (a) 5mm (b) 10mm (c) 15mm (d) 20mm (e) 25mm (f) 30mm.

Major inferences: - In Taguchi method, combination of fin parameters is used to find out performance of each parameters on response i.e., heat transfer coefficient, temperature and Nusselt number. Four geometrical parameters such as fin length, height, angle and thickness, are analyzed under natural convection. Fin surface temperature for each 25 simulations is calculated with help CFD-POST software. Velocity vector, streamline, temperature and pressure contour at fin centre and fin height centre plane are plotted. From all figures of contour, it is shown that fin height increases obstacle in boundary layer which starts from bottom of base plate. Because of obstacle, streamlines of fluid move along bottom surface and at corner, it bends and moves vertical along fin thickness. Body force of ambient fluid is more than hot fluid from corner of fin height so that above the surface of fin cold fluid is pushed down the hot fluid. Because of body force, circulation of flow occurs above fin surface. This circulation creates suction zone. This helps induction of low temperature fluid in the region. As increase in the fin height, this circulation zone enlarges and moves away from base plate. This circulation and destructing the boundary layer leading to establishment of high heat transfer zone in downstream side. Fin thickness helps to redirect the flow path and push up more hot fluid. As increase in fin thickness, circulation zone height is increased which help to enlarge the suction zone. Also, redirection of fluid flow occurs in proper way. Maximum pressure at bottom fin surface is created due to more striking of hot fluid. As decrease in fin angle, high pressure zone along bottom circumference of fin increases but maximum value of pressure is down. Also, suction zone become narrow and increase along top circumference of fin i.e., increase in circulation range. Boundary layer at bottom surface of fin becomes thicker with decrease in the fin angle. With increase in fin length, the effect of all the fin parameters (fin height, thickness and angle) becomes prominent.

Fin length and angle are more dominant for maximizing the heat transfer rate when Taguchi larger the better approach is used for average heat transfer coefficient of all L25 simulations. Heat transfer coefficient is inverse proportional to surface area and temperature of fin surface. Perpendicular to fin length, more heat transfer occurs. As increase in fin angle, stagnation of fluid at bottom surface is reducing. Also, more surface is in contact with suction zone. Taguchi smaller the better approach for average surface temperature of fin is used. Fin length and height are significant parameters for minimizing the surface temperature because maximum surface area is exposed to fluid domain. Circulation zone area also increases due to obstacle of fluid path. Nusselt number is also important non-dimensional parameter of heat transfer. So, Taguchi method with larger the better approach is used for Nusselt number. For optimum cost of material, weight of fin is also considering as response. Finally, Nua/ wt is used as response of larger the better approach. Fin height, thickness and length are more dominant for maximizing the Nua/ wt ratio with percentage of 42.20, 39.37 and 18.23% respectively. Optimum combination **A1 B1 C1 D1**is find out. Fin angle parameter is very less significant.

As contribution of fin length with range of 190mm to 315mm is more than other parameters of fin. Effect of other parameters is very less due to high range of fin length. To improve the effect of other parameters, range of fin length is changed from 190-315 to 220-285mm. Simulations of new combinations is performed. Taguchi approaches are also used for new simulation to evaluate the effect of fin parameters. Due to change in fin length range, the effect of other parameters is improved. For maximizing ha, fin length effect decreases from 78.94% to 34.87%. Fin angle parameter effect improved from 10.06% to 24.27% and also fin thickness from 7.45% to 15.57%.

For minimizing the average temperature of fin, improvement in other parameters is very less. It is revealed that fin length is only parameter that affects surface temperature of fin. Fin height contribution for maximizing Nua/ wt ratio get improve from 42.20 to 65.87%. Fin length and thickness effect decreases as decrease in range of fin length. Fin height is important parameter to maximize the heat transfer rate.

Table 2: S/N ratio for first five simulations

Simulations	A	B	C	D	Nu_a /wt	S/N ratio (dB)
1	190	10	180	1	149.3	43.46
2	190	15	157.5	1.5	67.15	36.53
3	190	20	135	2	37.11	31.39
4	190	25	112.5	2.5	23.79	27.50
5	190	30	90	3	16.63	24.41

First 5 simulation show that fin length is constant and other parameters of fin are varying. It helps to understand flow behaviour under natural convection. It shows that circulation zone increases as increase in fin height. u-velocity and w-velocity are increased with decrease in fin angle. V-velocity decreases as increase in fin angle. Pressure contour shows that high pressure zone created at bottom surface of fin increase and suction pressure reduces.

6.0 Conclusion

Natural convection heat transfer from semicircular fin is investigated numerically by using CFD software. The effect of heat input and fin parameters (fin length, height, angle and thickness) on semicircular fin has been discussed.From the analysis of the results in the semicircular fin using the conceptual signal-to-noise (S/N) ratio approach, regression analysis, analysis of variance (ANOVA), and Taguchi's optimization method, the following can be concluded from the present study.

1. Semicircular fin gives better performance than vertical fin and rectangular plate. But if compared with V- fin, values of convective heat transfer coefficient are close with difference about 0.3% to 0.6% as heat input is varied from 20W to120W respectively.
2. Velocity vector of semicircular fin at plane perpendicular to baseplate shows circulation zone above the fin surface. This circulation zone increases as heat input is increased.
3. The enhancement in base to environment temperature difference of fin decreases with increase in fin height and reach 20% when H=25mm compared with 5mm for 100W heat input.

Average convective heat transfer coefficient of fin is increased up to 25 mm fin height and graph of it with respect to fin height shows the asymptotic nature.

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