2D Numerical Analysis of Natural Convection in Vertical Fins on Horizontal Base: Select Proceedings of NHTFF 2018

Chapter · January 2019

DOI: 10.1007/978-981-13-1903-7_48

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Abstract Natural convection heat transfer from a finned horizontal flat plate at a constant temperature has been studied in this work. It analyzes the fin performance and; natural convection behavior of the finned horizontal flat plate. A complete picture of heat transfer on the horizontal finned surface (temperature and velocity contours) is captured. Then behaviors of multi-number of fins (2, 4, 6, 8, 10 and 12 fins) were analyzed in the current progressed work. The base body is subjected to constant temperature difference from the surrounding $\Delta T = 40$ K for all cases in the laminar range, i.e., Raleigh number $5 < \text{Ra} < 10^8$. The types of plumes caused are pictorially viewed. This work is progressed by comparing the graphical relation between Q (heat transfer) to $S^* = S/L$.

Keywords Natural convection heat transfer · Constant temperature difference Raleigh number · Nusselt number

Nomenclature

A	Area of fins for convection m^2
G	Gravitational acceleration m/s^2
H_b	Height of base surface mm
H_{fin}	Height of the fin mm
h_c	Average heat transfer coefficient W/m^2 K
K	Conductivity of fin apparatus W/m K
L	Length of the cylinder mm
N	Number of fins
Nu	Average Nusselt number

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D. Srinivasacharya and K. S. Reddy (eds.), *Numerical Heat Transfer and Fluid Flow*, Lecture Notes in Mechanical Engineering,
https://doi.org/10.1007/978-981-13-1903-7_48

P	Pressure N/m^2
P_{atm}	Atmospheric pressure N/m^2
Q	Convected heat transfer W
R	Specific gas constant J/kg K
Ra	Raleigh Number
S	Spacing between fins mm
S/L	Nondimensional fin spacing
T	Thickness of the fin mm
T_s	Surface temperature K
T_∞	Ambient temperature K
u, v, w	Velocity components of fluid m/s
x, y, z	Cartesian spatial Coordinates m

Greek Letters

α	Thermal diffusivity m^2/s
β	Thermal expansion coefficient 1/K
ΔT	Temperature difference K
ν	Kinematic viscosity m^2/s
ρ	Density kg/m^3

1 Introduction

Longevity of devices made engineers to come up with the concept of fins. Researchers came up with ideas improving the performance of these fins. This study also deals with the thermal performance of the fins at different condition(s) varying various factors which usually affect the heat transfer rate of the fins. These factors are spacing between the fins; length of the fins; thermal conductivity of the fun apparatus made material; Alignment of the fins; Cross section of fins; temperature of the base surface, etc. Churchill and Chu [1] have developed a general correlation of Nusselt number (Nu) depending on the Raleigh number (Ra) and Prandtl number (Pr). This correlation can be used in both natural convection in laminar and turbulent regime. Leung et al. [2, 3], Sara [4] and Nada [5] encountered that natural convection by horizontal and vertical fin(s); for high range of Ra, heat dissipation Q from a flat surface in a rectangular channel flow by attaching array of staggered pin fins or continuous fins is varied keeping base area constant. And it was experimentally elaborated by modifying the array of such pattern, resulting decrement in clearance with enclosure and fin spacing on a constant base due to which Nusselt number was increased keeping every other factor(s) constant to constrained measures. An empirical relation was developed making Nu depending on C/H [4]. Some authors also gave a slight variation by giving interruption in the fin array resulting drastic changes in natural convection of fins on a surface. Narasimha et al. [6] explained the compact ratio of fin in an enclosure to its own base surface and the fin performance was stated in pictures. Senapati et al. [7–9] experimented numerically a finned cylinder on wide range of Ra in vertical and horizontal setup and modifying it by applying annular heating and eccentric fin arrangement from the base cylinder axis and correlation is developed to find the optimum spacing and annular displacement. Considering all covered areas of study done before by above authors, this led to addition to the curiosity of fin performance on fins installed on a horizontal base surface. This numerical study has encapsulated the study of vertical fins performance on the horizontal surface on constant temperature surface from the same base with visualized behaviors extracted numerically.

2 Problem Description

On a 2D Rectangular horizontal surface of length L shown in Fig. 1 containing array of vertical fins equidistant to each other is taken for 2D analysis. The length (L) along horizontal axis is about 190 mm with fins of height ($H = 30$ mm) installed of thickness ($t = 3$ mm) with inter-fin spacing as (S). The fin is mounted on a base surface of thickness ($t_{\text{base}} = 30$mm). The fin apparatus is made of aluminum due to higher conductivity of material. The setup is encapsulated in an enclosure of height $5H_{\text{total}}$ and a width of $3L$ filled with air in it. For the enclosure, the behavior of the enclosure fluid is to be Bossenique Approximation. The analysis of the fluid is done in multi-fin of number of fins $n = 2$, 4, 6, 8, 10, 12 and 14. The inter-fin spacing is dependent on the number of fins on a constant horizontal surface. The main task of this project is Nusselt number (Nu) is a function of Raleigh number (Ra), S^* and H^*. Thereafter now, the base body surface is kept at a constant surface temperature (T_S) of 340 K. The operating condition of the enclosure fluid air (T_∞) is 300 K. The pressure at the edge of the enclosure is at P_{atm}. The project is progressed by analyzing the heat transfer of the fin setup on one side of the base body. The mesh contains 35,505 cells with 36,906 nodes when taken in symmetry form and mesh grid is aligned structured and fine to analyze the natural convection with very low skewness of grid of order of 10^{-10}.

2.1 Mathematical Model

For the type of flow around fins in the setup above is within Ra less than 10^8 that is laminar flow and the fluid properties are assumed under Bossenique's Approximation. The governing equation for this flow is listed below.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

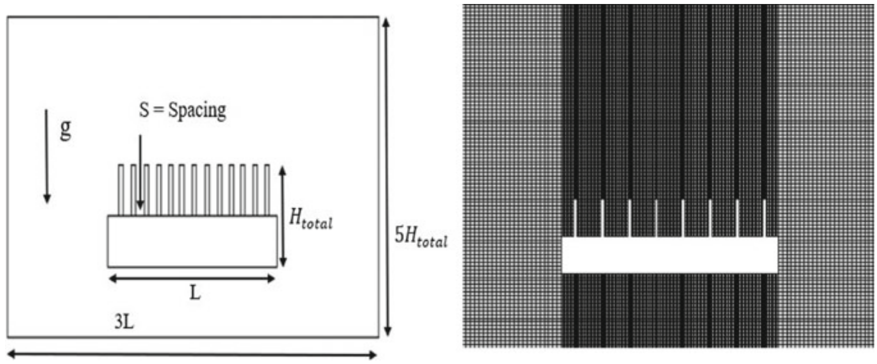

Fig. 1 (Left) Shows the geometrical view of fin setup; (Right) shows the computational grid of analysis of the fin apparatus setup in ANSYS R16

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{3}$$

$$Q = h_c A \Delta T \tag{4}$$

The above-listed equation(s) is the governing equation while Eq. (1) is continuity equation; Eq. (2) is the NS momentum equation for x-velocity; Eq. (3) is heat equation considering no heat generation and Eq. (4) is the convected heat equation.

Boundary Condition(*s*).

In the flow around fins, characteristic is found to be steady ($\mathrm{d}/\mathrm{d}t = 0$) no-slip boundary condition on the walls at the fin fluid interface $(u, v) = 0$. The numerical calculation is assumed to have the density characteristics as flow in Bossenique approximation(s). The flow encountered to be Laminar flow [$\mathrm{Ra} < 10^8$] The apparatus is set to enclosure with ambient pressure: P_{out} The temperature of base body set to $T_s = 340$ K. In a mesh design in Fig. 2 of the setup, fin wall and fin tip are coupled to the base and fluid region. Enclosure fluid temperature: $T_\infty = 300$ K. The equation of steady-state heat transfer in the base body taking no heat generation into account gives the temperature of the base body $\nabla^2 T = 0$. From the geometry perspective, the inter-fin spacing between the fins

$$S = \{L - (nt)\}/n \text{ and } A_{\mathrm{total}} = 2(L + H_{\mathrm{base}}) + 2nH_{\mathrm{fin}}$$

At the interface, the heat conducted by the fin body is the convected heat transfer

$$Q = h_c * A * \Delta T = -k\frac{\mathrm{d}T}{\mathrm{d}x} \tag{5}$$

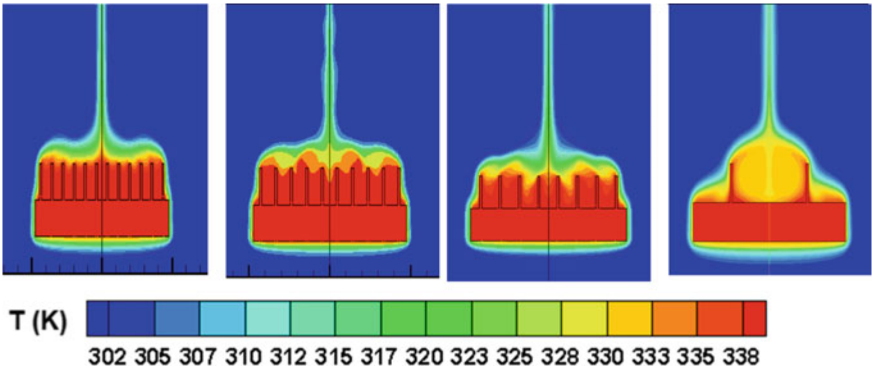

Fig. 2 Shows the temperature contour of 12, 10, 8, and 2 fins

The analysis is carried by nondimensionalizing the equations which rose to various nondimensional numbers. The flow in which regime laminar or turbulent is given by Raleigh number (Ra) where (Ra) $= \frac{\beta * g * \Delta T * H^3}{\alpha \nu}$. In the above analysis, the Raleigh number is less than 10^8 so it is in laminar range of flow condition. Nusselt number helps in knowing whether conduction is dominant on convection phenomenon or the vice versa. Nusselt number is given by (Nu) $= \frac{Q * S}{k * A * (T_s - T_\infty)}$.

Numerical modeling

The governing differential equations were integrated over a control volume and then discretized using the finite volume technique. The resulting algebraic equations were solved by the algebraic multigrid solver of FLUENT R16 in an iterative manner by imposing the boundary conditions. Second-order upwind scheme for the x-momentum, y-momentum, and continuity were considered for the momentum and energy equations. SIMPLE (Semi Implicit Momentum and Pressure Linked Equation) algorithm scheme was used for coupling the pressure and the velocity terms for the pressure correction equation. The relative convergence criterion for the energy equation was set to 10^{-6} and the continuity; x-momentum and y-momentum were set to 10^{-3}. The cells vertical along with fins are made smaller and also the fin till the end of the base body have been made of small cell. It results in very small computation calls near the fins and larger cell away from the fin setup. The case of computation grid for 8 fins is shown in Fig. 2 in the enclosure. The dense cells are in the vertical direction because the variation in heat flux temperature is encultured near the fins in vertical direction.

Analysis Parameters.

During analysis done in FLUENT R16 environment, it is very necessary to converge the solution to find a significant result. To converge it takes a minimum under-relaxation factors for pressure, density, body force, momentum, and energy. Table 1 indicates the set of under-relaxation factors were mostly used in all cases of multi-number fins having cases where $n = 2, 4, 6, 8, 10, 12, 14$.

Table 1 Under-relaxation factors used in FLUENT R16

Pressure	Density	Body force	Momentum	Energy
0.8	1	0.7	0.01	1

Table 2 Quantitative record of heat flux (Q) and S^* (S/L) for different cases of fins

No. of fins	S/L	Q (W)
2	0.4862	51.88
4	0.2342	53.48
6	0.1508	54.14
8	0.1089	54.28
10	0.08421	56.7
12	0.06754	51.62
14	0.05563	48.34

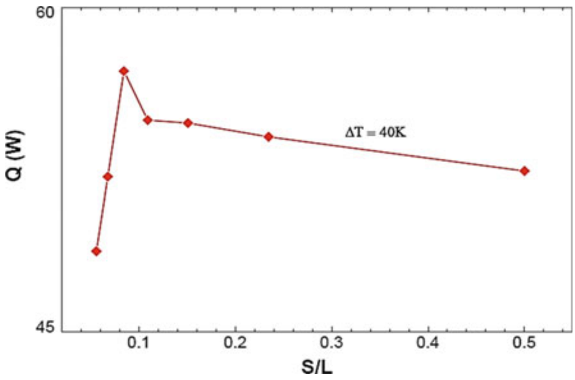

Fig. 3 Graph plotted of Q (heat Flux) and S^* (S/L)

3 Results

On a same base body surface, the number of fins is increased lowering the spacing in different cases. When at constant $\Delta T = 40$ K, less number of fins the heat transfer is low, as the fin number is increased, the heat transfer increases until, it is decreased to optimum spacing. Later if the spacing is decreased further, the heat transfer will decrease again. It is given by the graph in Fig 3 showing the variation of heat transfer with respect to spacing. The records of the heat transfer have been tabulated in Table 2. If observed keenly the contours presented below in Fig. 2, the red region indicated is conduction. It increases with increase in fin number later becomes responsible for less heat transfer (Fig. 3).

By constraining others factors like height, Ra, etc., graphically the optimum number of fins for this case is obtained to be 10 fins with a heat transfer of 56.7 W. 12 and 14 fins decrease the heat transfer from the base surface according to the information extracted from the graph.

4 Conclusion

The heat transfer first increases with the more fins and decreases with further added fins. The visualization of the temperature contour can make the reader know the behavior of the temperature distribution. It deduces the *S*/*L* (opt) among these cases is 0.08421. These fins performance can be tested when installed on both sides and different alignments and can be used in plate heat exchangers used in different thermodynamic cycles. This can be used as heat dissipation mechanism from ducts of large sides acting as horizontal base.

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