Optimization of Horizontal Rectangular Fin Heat Sink: a CFD with Response Surface Analysis and Parametric Study

Article *in* Journal of the Institution of Engineers (India): Mechanical Engineering Division · August 2019

DOI: 10.1007/s40032-019-00527-9

J. Inst. Eng. India Ser. C
https://doi.org/10.1007/s40032-019-00527-9

ORIGINAL CONTRIBUTION

Optimization of Horizontal Rectangular Fin Heat Sink: a CFD with Response Surface Analysis and Parametric Study

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Received: 8 February 2019 / Accepted: 13 August 2019

Abstract In this article, a CFD analysis is performed on rectangular fin heat sink, which is used for electronic cooling applications. A steady-state heat transfer through natural convection is considered in this model. It focuses on thermal analysis of rectangular heat sink with aluminum as a material, by varying different parameters like temperature difference between fin base temperature (53–105 °C) and ambient temperature (20 °C), fin spacing (2–20 mm), fin height (25–100 mm), fin length (2–100 mm) and fin thickness (1–5 mm). The 3D elliptic governing equations of laminar flow and heat transfer are solved through the CFD tool ANSYS Fluent 15. Initially, a parametric study of fin is performed for obtaining orthogonal effect of input parameters on heat transfer coefficient (HTC) at fin base temperature 53 °C. Based on the parametric study, range of parameters is set for performing design of experiments (DOE) using ANSYS Design explorer 15. Response surface is generated between the geometrical fin parameters and HTC using DOE. Multi-objective genetic algorithm optimization is performed with the objective to maximize heat flux and HTC. Optimized values of geometrical parameters for maximum heat transfer are predicted using response surface.

Keywords Natural convection · Heat transfer through fins · Finite-volume method · Parametric study on fins · Response surface · Optimization of fin geometry

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List of Symbols

A Total heat transfer area (m^2)
C_p Specific heat (kJ/kgK)
g Acceleration due to gravity (m/s^2)
h Average heat transfer coefficient (W/m^{2}K)
H Fin height (m)
k Thermal conductivity (W/mK)
L Fin length (m)
P Pressure (Pa)
Q Total heat transfer rate (W)
Ra Rayleigh number
Gr Grashof number
Nu Nusselt number
Pr Prandtl number
S Fin spacing (m)
T Temperature (K)
t Fin thickness (m)
s Fin spacing (m)
u,v,w Velocity components
x,y,z Coordinates
μ Dynamic viscosity (Ns/m^2)
v Kinematic viscosity (m^2/s)
ρ Density (kg/m^3)
β Coefficient of volumetric expansion (K^{-1})
ΔT Temperature gradient (K)
IP Input parameter
OP Output parameter

Subscripts

a Ambient
b Fin base

Introduction

In large variety of cooling applications (processors, microchips, power transistors, optoelectronics, etc.), buoyancy-driven flow is used being low price, simple and quiet solution. The design of fin for heat sink plays an important role for enhancing heat transfer; if not properly designed, the flow rate could reduce. Increase in surface area helps to increase the heat transfer, but it may be a formidable task on electronics systems such as chips, transistors and printed circuit boards as these have space constraints. Within the given space limitations, the fin dimensions need to be accommodated. A proper material needs to be selected considering the expensive and applications of the electronics system. Therefore, perfectly designed fins are essential for engineering applications. Considering manufacturing ease, rectangular plate heat sink is the most general heat sink used for electronic cooling applications. The above facts motivate the authors to do this fin design problem. The present study focuses on phenomenon of natural convection due to buoyancy effects because of density change of air with temperature. The application with natural convection has lean construction as compared to forced convection setup because there is no need of fans or external power, as the flow generates by itself.

Few parametric studies along with numerous experimental studies can be found in the literature. Elenbaas [1] first introduced heat dissipation of vertical square parallel plates with free convection in air. A few decades later, studies on fins were made by Starner and McManus [2]. They selected four different dimensions of fins and analyzed the effect of base with horizontal and vertical shapes and with inclinations. It was found that heat transfer could be enhanced or reduced based on the alignment of fins with base plate. Taguchi's method was used to optimize the fin parameters by Das and Dwivedi [3]. Triangular fins were taken for their investigation of natural convection heat transfer. Harahap and McManus [4] developed an experimental model to analyze the natural convection behavior of rectangular fins. The fins were made by smooth aluminum sheet metal. Proper heating coil provisions were made in the setup to generate the heat flux, and insulations were provided to avoid the heat loss. They estimated HTC coefficients and identified that shorter fins gave better performance. They finally develop correlations from the experimental outcomes. Jones and Smith [5] experimentally found the HTC and obtained optimum shape of fin array for achieving maximum heat transfer. Rahnama and Mehrabian [6] studied two concentric cylinders having radial fins with natural convection heat transfer. They predicted local and mean Nusselt number and concluded the heat transfer rate is increased on optimum value of fin height. Yuncu and Anbar [7] conducted an experimental analysis with a free convection application. Effect of fin space, height, horizontal and inclined base and temperature gradient between extended surfaces and air were investigated. A 3D numerical model was developed by Golparwar et al. [8] to optimize the fin spacing for the application of adsorption air-conditioning. Two types of modeling were selected, namely horizontal and circular fins. They optimized the fin spacing of 5–7 mm for rectangular fins and 5–6 mm for circular fins. Nada and Said [9] investigated the effect of annular fins with two different models, one without fins and other with fins. They identified the optimum fin thickness, fin gaps and other dimensions. It was reported that the increase in fin thickness increased the heat transfer rate. A similar analysis was performed by Borhani et al. [10]. Dialameh and Yaghoubi [11] performed computational study on free convection fins. The fins were made by thick horizontal rectangular with short lengths. They found the dependency of HTC on fin height, fin length, fin thickness and fin spacing. Correlations were proposed by them to predict average Nusselt number, and from that, HTC can be estimated. Jha et al. [12] studied thermal and hydrodynamic of fully developed natural convection flow with suction/injection in vertical parallel plate micro-channel. Parametric study was carried out between governing parameters of micro-channel affecting it both thermally and hydro-dynamically. They concluded that the rate of heat transfer decreases as suction/injection on channel surface increases. Campean and Kianaifar [13] presented the PermGA algorithm to find the optimized fin space. Their main objective was to get a maximum heat transfer with small number of fins. DOE analysis was made to confirm the number of trials. Liu et al. [14] presented an insight on multi-objective optimization using genetic algorithms (MOGA). They described different methods to find optimal solutions. Reliability of MOGA was analyzed for similar applications.

The above literature survey proves that very few numerical works [6, 7, 9] have been performed for 3D fluid flow over fin. DOE framework and response surface generation between fin geometric parameters and HTC are also not presented in any literature; moreover, in all previous studies, few parameters were kept constant in order to predict the behavior of fin geometric parameters with HTC. It is found that there are many studies based on parametric study of fins [3, 5, 7, 9, 15], but no studies were found on prediction of optimized value of all the parameters of fins and its related parametric study. Therefore, the main objective of this study is to find the optimized fin parameters using a numerical model assisted by DOE analysis, MOGA analysis and response surface methodology. Therefore, the essential steps to achieve the main objective are: (i) to model a 3D computational domain and perform simulation, (ii) to develop a numerical solution using ANSYS Fluent, (iii) to validate the results with the existing results from experiments, (iv) to perform the parametric study with DOE, so that every parameter can be varied with HTC using ANSYS Design explorer, (v) to generate response surface to fit the output parameters in terms of input parameters and (vi) to predict the optimized values using MOGA based on response surface in ANSYS Design explorer.

Methodology

Computational Domain

The rectangular fin model with coordinate system and appropriate dimensions is shown in Fig. 1. The computational model can be regarded as one quarter of one fin channel with extended top and open end. The computation domain has height extending up to 9H and length up to 0.75L in order to capture unmitigated effects of natural convection. It has been simulated and also verified from the literature [9] that extending length to L and H to 14 H did not have a significant effect on computational domain.

Assumptions followed in this study are: (i) it is uniform, steady and laminar ($Gr < 10^9$) natural convection flow, (ii) heat loss through radiation is considered to be negligible, (iii) density of air is considered to be piecewise linear function, while all other thermophysical properties of air are assumed as constant, (iv) material properties of fin are kept constant and (v) uniform base temperature of T_b = 326.15 K and ambient temperature of air is assumed constant, T_a = 293 K.

Governing Equations and Relations

The continuity, momentum and energy governing equations of 3D Cartesian coordinates are:

Continuity:

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial x}(\rho v) + \frac{\partial}{\partial x}(\rho w) = 0 \tag{1}$$

Momentum equation in x,y,z direction is:

$$\frac{\partial}{\partial x}\left(\rho u^2\right) + \frac{\partial}{\partial x}(\rho uv) + \frac{\partial}{\partial x}(\rho uw) = -\frac{\partial P}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

$$\frac{\partial}{\partial x}(\rho uv) + \frac{\partial}{\partial x}\left(\rho v^2\right) + \frac{\partial}{\partial x}(\rho vw) = -\frac{\partial P}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + g(\rho - \rho_a) \tag{3}$$

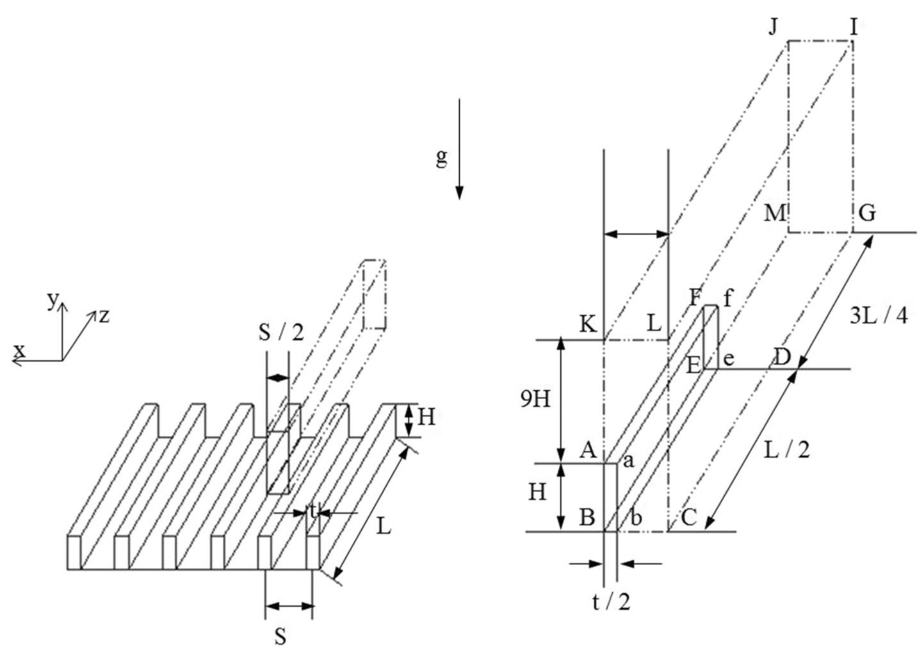
$$\frac{\partial}{\partial x}(\rho uw) + \frac{\partial}{\partial x}(\rho vw) + \frac{\partial}{\partial x}\left(\rho w^2\right) = -\frac{\partial P}{\partial w} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

Energy equation:

$$\frac{\partial}{\partial x}(\rho uT) + \frac{\partial}{\partial x}(\rho vT) + \frac{\partial}{\partial x}(\rho wT) = \frac{k}{C_p}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

Governing equation of fin:

Fig. 1 Computational domain



$$\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)=0 \tag{6}$$

Fourier heat conduction equation (Eq. 6) is used for modeling temperature distribution in fin material, where T is temperature (K), ρ is density (kg/m^3), g is gravity (m/s^2), u, v and w are velocity components, μ is dynamic viscosity (Ns/m^2) and ν is kinematic viscosity (m^2/s).

Solution Algorithm

A finite-volume-based CFD tool ANSYS Fluent 15 is used to simulate the computational domain. All governing equations are discretized by second-order upwind technique [16, 17]. A parametric study is carried out in ANSYS Workbench to predict the range of input parameters for DOE. DOE is carried out using ANSYS Design explorer followed by parametric study using optimal space filling scheme [13]. After DOE response surface is generated between input and output parameters in ANSYS Design explorer using kriging response surface algorithm [14]. Thus, optimization of input parameters is performed using MOGA [18] with the objective to maximize HTC and HF.

Optimization algorithm behaviors are the same for response surface optimization (RSO) and direct optimization. In the present study, RSO method is used with the help of multi-objective genetic algorithm (MOGA). The objective of optimization was set to maximize the heat transfer coefficient (HTC) and heat flux (HF).

If an engineer wants to optimize N objectives such that the objectives are non-commensurable and the engineer has no clear preference of the objectives relative to each other. Since this approach is based on a minimization problem, in the present case it is converted into a maximization problem by multiplying − 1. If a multi-objective decision problem with N objectives can be described by the following procedures. Given an n-dimensional decision variable vector $x = \{x_1 \ldots x_n\}$ in the solution space X, find a vector x^* that minimizes a given set of N objective functions $z(x^*) = \{z_1(x^*),\ldots,z_K(x^*)\}$. The solution space X generally has a series of constraints, such as $g_j(x^*) = b_j$ for $j = 1,\ldots,m$, and bounds on the decision variables. Most of the engineering problems lead to mistaken results, if the optimization is carried out by considering a single objective. This MOGA considers all the objectives presented in this problem and gives an optimized solution. It creates a solution and checks for each objective, and also it takes care that none of the objectives conflicts with the real solution.

Response surface of curve is generated for every output parameter. It is performed using ANSYS Design explorer, and it has two major steps. They are:

a. The output properties are solved for all design points.
b. Regression analysis is used to fit the output properties in terms of input properties.

In this study, kriging interpolation method [14] is used for the creation of response surfaces. Kriging is a meta-modeling algorithm that provides a better quality of response and fits with a suitable quantity of higher-order variations of the output property. Kriging in ANSYS Design explorer is a combination of a polynomial model in addition to departures of the form given by,

$$Y(x) = F(x) + Z(x) \tag{7}$$

Here, $Y(x)$ is the function that needs to be estimated, $F(x)$ is a polynomial function of x and $Z(x)$ is the realization of a normally distributed Gaussian random process with mean zero, variance σ^2 and nonzero covariance. While $F(x)$ approximates generally the design space, $Z(x)$ creates local deviations such that kriging model interpolates the N sample data points.

Boundary Conditions

As mentioned in Fig. 1, ABbafeEFA is the quarter surface of a fin. A constant-temperature (T_b) boundary condition is imposed on surfaces of BbeEB and bCDeb. Surfaces AKJMEFA and BAFEB are applied with symmetrical conditions about at $x = 0$. A similar condition is imposed on surfaces of CLIGDC and BAKLCbB at $z = 0$. It is assumed that air entering and/or leaving from surfaces KLIJK (on the top side), MGDeEM (on the bottom side) and MGIJM (on the rear side).

Results and Discussion

Convergence and Mesh Independence Study

Convergence criterion is fixed for all the governing equations such as mass, momentum and energy equations as 10^{-4}. Overall, 850 iterations are considered in order to have converged solution. It is important that the solution is also independent of the mesh resolution. The initial simulation is run on the initial mesh (element size 0.3 million). After convergence criterion is met, mesh is refined globally so as to have finer cells throughout the domain. The values of temperature along X direction of finer mesh are compared with the initial mesh. A plot is drawn between temperature and X coordinate position as shown in Fig. 2. It was observed that there is no variation of temperature values along X direction when the mesh size in 0.7 and 0.8 million, so in the present study mesh size is considered to be 0.7 million elements in order to save computation time

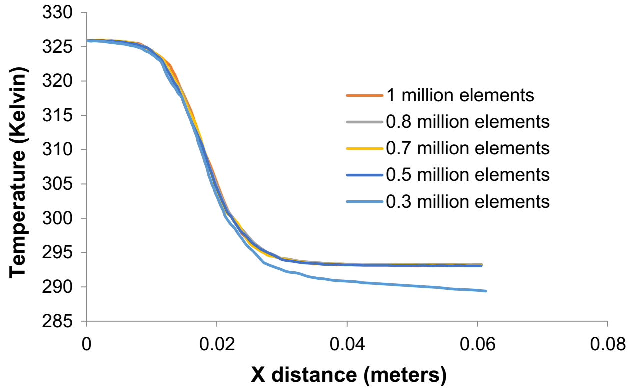


Fig. 2 Mesh-independent study

(with the Δx value of 0.272 mm, Δy of 0.136 mm and Δz of 0.335 mm).

The solution of the present result was compared with the experimental results of Harahap and McManus [4]. Aluminum array type fins were used in their study with the fin thickness of 1.3 mm and length of 127 mm. Different temperature gradients from 33 to 85 °C were considered for this analysis. Average HTC was estimated using,

$$h = Q/A\Delta T \tag{8}$$

where Q is net heat transfer rate from extended surface to air, A is wetted area of fin and ΔT is the temperature gradient between working fluid (air) and fin surface.

Figure 3 compares the present simulation value of HTC with measurements of Harahap and McManus [4] for a case when fin height of 38 mm, fin length of 127 mm, fin thickness of 1.27 mm, fin spacing of 6.35 mm and thermal conductivity of aluminum k—237 W/mK with various temperature differences between 33 and 85 °C. It is found that there is less than 5% error between experimental HTC and simulation HTC. Thus, it is concluded that the boundary conditions and domain selected are justifiable.

Figure 4a, b shows velocity and temperature contours of an air flow for fin with H—12 mm, L—50 mm, t—3 mm

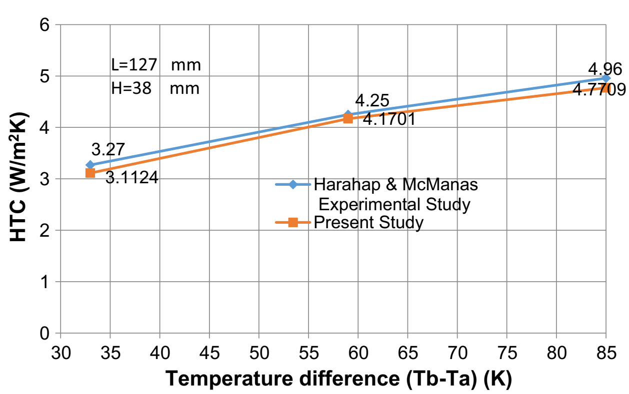


Fig. 3 Comparison of the present numerical study with experimental results

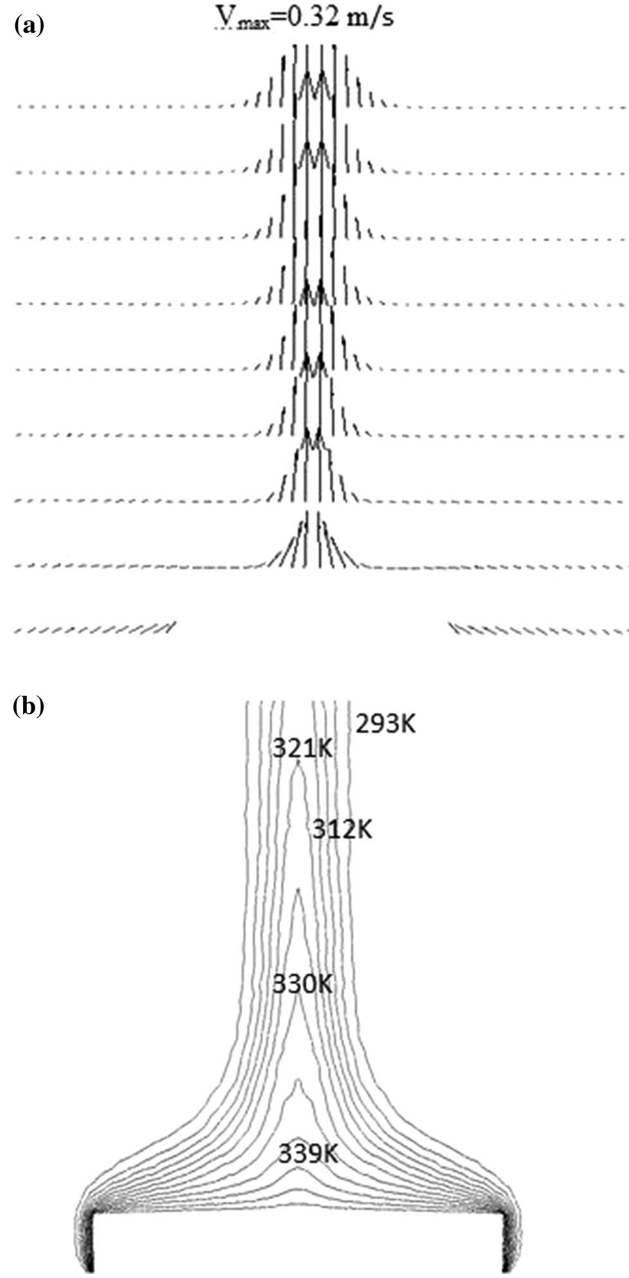


Fig. 4 **a** Velocity distribution and **b** temperature contours along symmetry plane $z = 0$

and S—12 mm. As a result of natural convection, there is a velocity in the flow which has a maximum value of 0.37 m/s and minimum value of 0.01 mm in vertical direction as the flow progresses. The parabolic dome region developed inside the rectangle can be observed in Fig. 4b; this shows decrease in temperature as the distance above the fin geometry increases. Therefore, the flow in this study is single chimney type. It is formed when cold air enters from two ends of fin arrays, traveling lengthwise and coalescing at the fin array center. This flow is more efficient from heat transfer standpoint [7].

Parametric Study

Initially, one of the parameters is varied, while other four parameters are kept constant in order to find independent behavior of particular parameter on HTC. The highlighted parameters (with *italic* font) are varied as shown in Table 1, while the other four parameters are fixed.

In the first case, fin space, length, height and thickness are kept constant and air and fin base temperature are varied (as mentioned in Table 1 by *Italic* font). From Fig. 3 and Table 1, it can be noticed that HTC increases with the increase in temperature. This is due to increased rate of natural convection with increased temperature.

From the initial study of variation of parameters with respect to HTC, it is concluded that HTC decreases with the increase in fin length as shown in Fig. 5a. This could be explained from boundary-layer phenomenon, thickness of boundary layer increases with the increase in fin length, thus decreasing local HTC along fin length which further decreases average HTC. From Fig. 5b, it can be observed that with the increase in fin thickness there is a decrease in HTC; as the fins become thicker, there is a reduction in fin spacing which leads to decrease in HTC.

It can be seen from Fig. 5c that HTC increases with fin spacing and reaches to some maxima because increased fin space makes a chance of fresh atmospheric air enter inside the channel. But increasing fin spacing beyond a certain point decreases HTC due to pressure drop between fins. From Fig. 5d, it can be concluded that HTC increases with fin height and reaches to maxima and then starts decreasing. This can be explained by considering buoyancy forces. Increase in fin height increases buoyancy producing high mass flow rate from open end, resulting in increased HTC. On further increasing fin height, the upper part of fin comes in contact with hot air since air entering from open ends flows long-distance upward resulting in decrease in HTC.

Table 1 Parametric study of input parameters with respect to HTC

Fin spacing (mm)	Fin length (mm)	Fin height (mm)	Fin thickness (mm)	Fin base temperature (K)	HTC (W/mK)
6.35	127	38	1.27	*326.15*	3.11
6.35	127	38	1.27	*352.15*	4.17
6.35	127	38	1.27	*378.15*	4.77
6.35	*63.5*	38	1.27	326.15	4.54
6.35	*127*	38	1.27	326.15	3.11
6.35	*191*	38	1.27	326.15	2.20
6.35	*254*	38	1.27	326.15	1.69
6.35	127	*19*	1.27	326.15	2.60
6.35	127	*38*	1.27	326.15	3.11
6.35	127	*48*	1.27	326.15	3.16
6.35	127	*57*	1.27	326.15	3.12
6.35	127	*76*	1.27	326.15	2.93
6.35	127	*95*	1.27	326.15	2.69
3.15	127	95	1.27	326.15	0.42
6.35	127	95	1.27	326.15	3.11
9.45	127	95	1.27	326.15	4.88
12.6	127	95	1.27	326.15	5.40
15.74	127	95	1.27	326.15	5.58
18	127	95	1.27	326.15	5.642
20	127	95	1.27	326.15	5.64
22	127	95	1.27	326.15	5.61
24	127	95	1.27	326.15	5.56
6.35	127	38	*1.016*	326.15	3.32
6.35	127	38	*1.27*	326.15	3.11
6.35	127	38	*1.905*	326.15	2.52
6.35	127	38	*2.54*	326.15	0.86

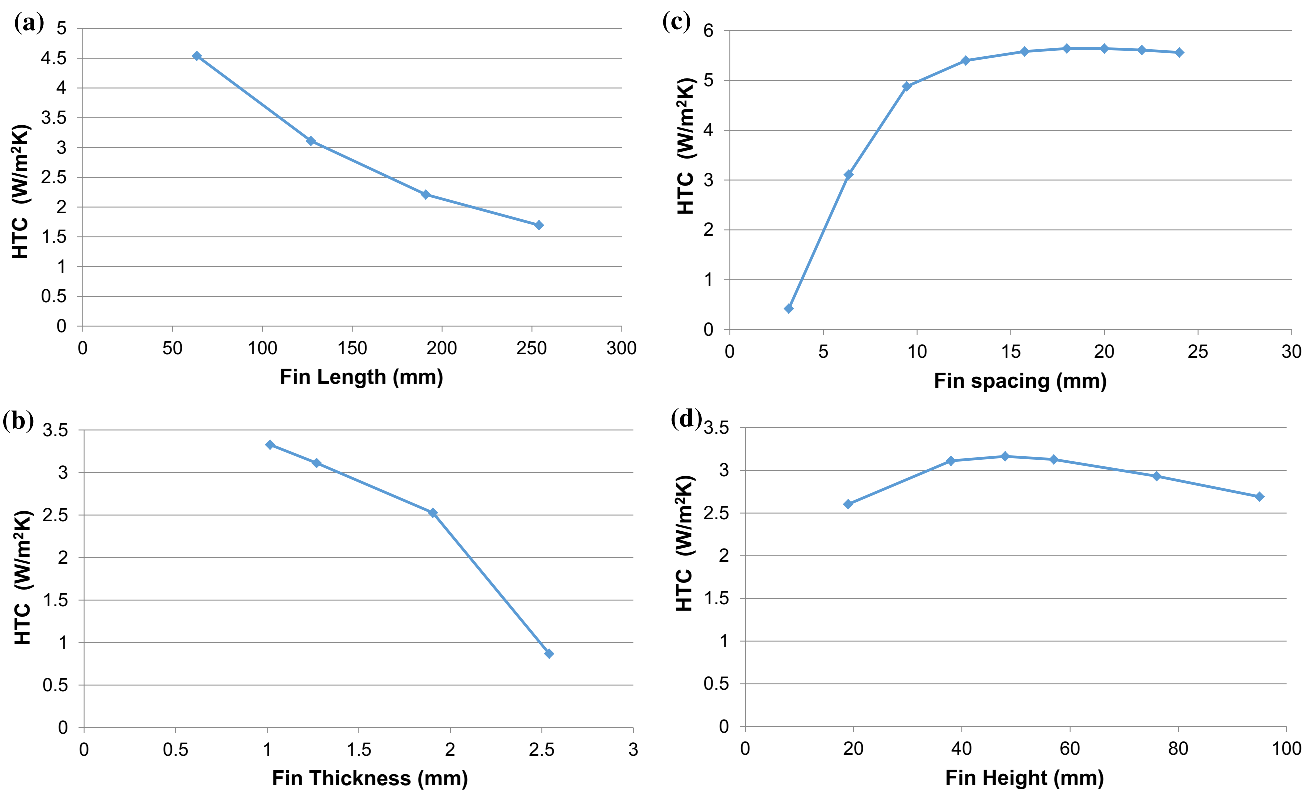


Fig. 5 Variation of HTC with respect to **a** fin length, **b** fin thickness, **c** fin spacing and **d** fin height

Design of Experiments

Based on the above parametric study, the range of parameters is set for DOE, fin height 25–65 mm, fin length 20–100 mm, fin thickness 0.5–1.5 mm and fin spacing 6–24 mm for computational model and DOE is performed. The solver used in DOE is optimal space filling (OSF). An OSF scheme distributes the design parameters equally throughout the design space [13]. Figure 6 shows OSF scheme used for DOE. The objective is to gain the maximum insight with the fewest numbers of design points.

Results from Response Surface

The response surface generated used kriging algorithm as mentioned in methodology. The surface generated is available in 2D and 3D also. Figure 7 shows the 3D response curve generated after applying kriging algorithm.

From Fig. 7a, it is observed that maximum value of HTC occurs when fin length is 10 mm and fin spacing is around 7 mm. From Fig. 7b, it can be noted that HTC is maximum at fin height of 25 mm and fin length of 10 mm. From Fig. 7c, maximum value of HTC occurs at fin length 10 mm and fin thickness around 1 mm; similarly, from Fig. 7d HTC is maximum at fin thickness of around 1 mm

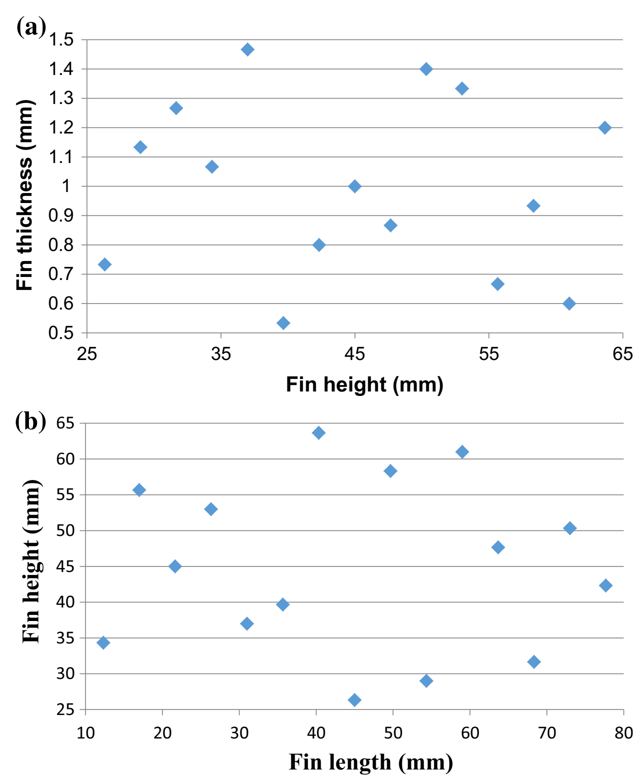


Fig. 6 Optimal space filling DOE: **a** fin thickness versus fin height and **b** fin height versus fin length

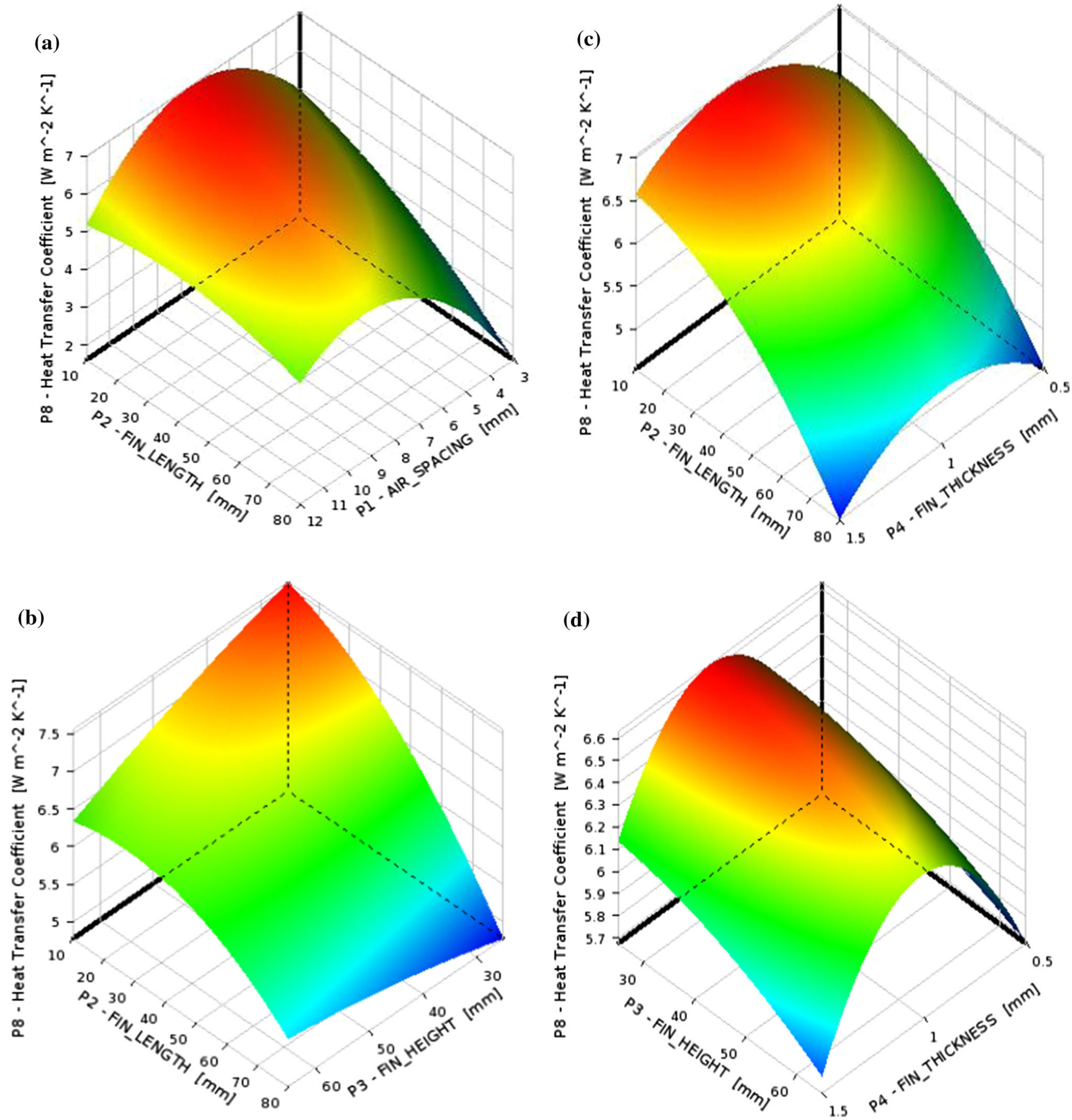


Fig. 7 Variation of HTC with **a** fin length and fin spacing, **b** fin length and fin height, **c** fin length and fin thickness, **d** fin height and fin thickness

and fin height of 25 mm. Now based on response surface, final optimization of input parameters will be done.

Optimization using Response Surface

Using multi-objective genetic algorithm (MOGA), an optimized solution is found using response curves [16]. The objective of optimization was set to maximize the HTC and heat flux (HF). Using MOGA, two suitable candidate points are generated which are then verified by actual simulation result. Out of these two candidate points, the best candidate is chosen with maximum HTC and HF as shown in Table 2 where input parameter (IP) gives value for maximum output parameter (OP).

It can be seen from Table 2 that candidate point 2 is the most suitable optimized value for maximum heat transfer. Point 1 and point 2 are verified points with simulation for any error, and there was only 2% error for HTC when

Table 2 Optimized candidate points for maximum heat transfer

Name	Fin spacing (mm) IP	Fin length (mm) IP	Fin height (mm) IP	Fin thickness (mm) IP	HTC (W/m^{2}K) OP	HF (W/m^2) OP
Point 1	7.8	10.03	25.04	1.08	7.57	243.7
Point 1 (verified)	7.8	10.03	25.04	1.08	7.73	251.8
Point 2	7.4	10.07	25.01	1.07	7.59	243.5
Point 2 (verified)	7.4	10.07	25.01	1.07	7.76	252.4

compared to underlined simulated value and less than 4% error for heat flux when compared to simulated value. Therefore, the present calculated optimized values are justified.

Conclusions

Heat transfer and flow over a 3D computational model of horizontal fin arrays were predicted numerically using a finite-volume-based computational fluid dynamics (CFD) code followed by parametric study, DOE, response surface generation and MOGA using ANSYS Design explorer tool. The contours of velocity and temperature generated in the present study showed that it is a single chimney flow pattern that further enhances the heat transfer. The simulation result was compared with experimental results of Harahap and McManus [4] and found that good accordance.

There were five fin parameters such as fin spacing (6–24 mm), thickness (0.5–1.5 mm), fin length (20–100 mm), height (25–65 mm) and fin base temperature (326–378 K) which are taken and varied for this analysis. It is concluded that heat transfer varies nonlinearly with fin spacing, fin height, fin length and fin thickness. Every parameter contributes toward heat transfer. Fin spacing and height were increased; heat transfer coefficient (HTC) initially increased and later decreased. When fin base temperature was increased, HTC increased. Fin length and thickness had given reverse effect on HTC, as these were increased and HTC decreased. A method was developed for automated optimization of rectangular fin heat sink within a defined range of inputs for maximum heat transfer. An optimal space filling (OSF) solver was used to do the design of experiments (DOE) analysis. Multi-objective genetic algorithm (MOGA) was used to optimize the parameters. The maximum heat transfer from the above study takes place at the optimum value of fin length—20 mm, fin height—25 mm, fin spacing—14.8 mm and fin thickness—2.14 mm within a given range. The optimized values are doubled because the model is symmetric except fin height, which was considered as it is.

Two more DOEs were performed with fin base temperature as 352.15 K and 378.15 K, and it was observed that optimum value in the present study was same for both the cases although the HTC had increased due to temperature rise, but the heat transfer was maximum at the same geometric parameters as was found to be in the present study.

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