Available online at www.sciencedirect.com

ScienceDirect

Materials Today: Proceedings XX (2016) XXX–XXX

www.materialstoday.com/proceedings

PMME 2016

A Comparative Thermal Analysis of Pin Fins for Improved Heat Transfer in Forced Convection *

Saroj Yadav[a], Krishna M. Pandey[a]*

[a]*Department of Mechanical Engineering*
National Institute of Technology Silchar
Assam-788010, India

Abstract

The electronic thermal management industries are consistently working with a goal to reduce the overall dimensions of the fins. The aim is to improve the rate of heat transfer rate per unit weight of the system. In order to fulfil the requirements, researchers need to optimize various parameters of the existing fin geometries. Fins are the extended surfaces that helps in increasing the surface area. Use of an extended structure or a fin over a surface does not always assure suitable enhancement in the heat transfer rate. The rate of thermal diffusion in a fin is always affected by parameters like the size, the shape, the material, the relative arrangement and position of the fins, the working fluid and its velocity, etc. A working fluid with lower thermal diffusivity always put limitations in the heat transfer process. Moreover, the added surfaces increase the overall dimensions and the weight of the system. Therefore, effective utilization of surface area is an important factor. In this regard, the shape of the fin plays an important role in thermal diffusion process. In the present work, with air as the working fluid, a 3-D system of aluminium fin system has been numerically modelled, simulating conjugate heat transfer physics. The thermal analysis is performed for various input parameters considering different shapes of the fins.

Keywords: Extended surfaces; Conjugate heat transfer; Finite element method; Heat transfer co-efficient; Nusselt number.

* Corresponding author. Tel.: +91-9435173130; fax: +0-000-000-0000 .
E-mail address: kmpandey2001@yahoo.com

Selection and Peer-review under responsibility of International Conference on Processing of Materials, Minerals and Energy (July 29th – 30th) 2016, Ongole, Andhra Pradesh, India.

1. Introduction

Thermal management is an important aspect in design of engineering processes. A system while working always experiences heating. To avoid the damage of the components of such systems due to overheating, heat transfer in a controlled manner is desirable. Industries like aviation, heat exchangers, electronic equipment, IC engines, etc. are always in need of efficient heat dissipation techniques. In this regard, many researchers have already proven that the extended surfaces or the fins can be used effectively and extensively in many cases. Addition of fins to enhance the heat transfer rate increases the overall dimension and the weight of the system. This in turn increases the operating cost of the device. In order to have minimal production and operating cost, it is very important to focus on the compactness of the size and the weight of the fins with maximized rate of heat transfer.

The rate of diffusion of thermal energy is based on the active modes of heat transfer available for a particular thermal system. In case of a fin, the heat transfer to the surrounding fluid by convection and radiation remains a dominating mode over conduction. Therefore, in such systems, the rate of heat transfer is a function of the coefficient of convective heat transfer, the temperature of the system and the working fluid and the wetted surface area of the fins. In convective mode of heat transfer, the value of the heat transfer coefficient depends on the thermo physical properties of the fluid and the solid surface in contact, and the average velocity of fluid. Therefore, increase of surface area may provide an increased amount of thermal energy transfer. However, increase in the surface area of the fin results in increased dimension of the system, and adds to the cost of the material. Hence, the system becomes larger and inefficient.

Nomenclature

A	-	Area
c_p	-	specific heat
C_1	-	arbitrary constant
D	-	diameter
H	-	height
k	-	thermal conductivity
l, L	-	length
$\hat{n}$	-	unit normal vector
Nu	-	average Nusselt number
p	-	pressure
P_o	-	atmospheric pressure
Q_s	-	heat supplied
Re	-	Reynolds number
S	-	pitch
t	-	time
T	-	temperature
$\vec{u}_y$	-	y- component of velocity
$U, \vec{V}$	-	total velocity
x, y, z	-	co-ordinate axes

Greek symbols

ν	-	dynamic viscosity
ρ	-	density

Subscripts

b	-	base plate
D	-	diameter or diagonal
f	-	fluid
L	-	longitudinal
max	-	maximum
T	-	transverse
w	-	wall

The presence of a fin introduces a conductive resistance to heat transfer from the original surface. Moreover, in a fin, the entire heat transfer surface may not be efficiently utilized due to geometrical configuration of the system. A good number of works has been reported in the field of determination of the optimum profile and shape of a fin for maximized rate of heat transfer [1, 2]. Saha and Acharya [3] have performed a numerical study on a heat exchanger passage with rectangular-cross-section pin fins. Conjugate heat transfer physics has been used in this work to investigate the steady and unsteady flow regimes in the three dimensional system. Raaid R. Jassem [4] has studied five cases of vertical rectangular extended surfaces with and without perforations for examining the extent of heat transfer enhancement under natural convection. He has also done the comparison between different shapes of perforations like circular, square, triangular, and hexagonal. Yadav et al. [5] also studies and compare a system of three dimensional model of vertical fins with and without perforations mounted on a heated plate. Authors found that the use of a fin with holes instead of an un-holed surface helps in reducing the overall volume of the fins. With little compromise in the heat transfer rate, it reduces the cost of the material by reducing the weight of the system. Gururatana and Li [6] has applied a vibrational effect for small scale pin fin or heat sinks in order to improve the heat transfer enhancement for electronic applications. Sarma and Ramakrishna [7] has investigated splayed and hybrid pin fin heat sinks numerically for electronics cooling. The authors determined that splayed pin fins are more efficient than the hybrid pin fins. Agrawal et al. [8] explored the effectiveness of cylindrical pin fins for heat transfer enhancement in a rectangular channel cooled by air with translational oscillating plate. Sukumar et al. [9] investigated continuous and interrupted rectangular heat sinks, numerically. Further, the authors studied these models with holes for application in cooling of electronic components

Yadav et al. [10] has done a comparative analysis over different shapes of pin fins like square, cylindrical, conical, triangular, hemispherical and quarter-spherical under combined conduction-convection mode of heat transfer. They found that a triangular pin fin is comparatively facilitates higher heat transfer due to higher wetted surface area and thus higher value of fin effectiveness. Micheli et al. [11] presents a comparison of heat transfer performance of micro plate-fins and micro pin-fins under the circumstances of free convection where air has been used as working fluid and they concluded that micro pin-fins are better than micro plate-fins in terms of heat transfer coefficient. In the present work, different pin fin geometries are studied with laminar flow of air under combine conduction-convection mode of heat transfer. The considered shapes are: cylindrical, elliptical, sprocket, kite, channeled, and triangular. The pin fin system is modelled numerically using FEM based commercial solver COMSOL Multiphysics. The set of Navier-Stokes equations along with energy equation is solved for steady state conditions. The modelling is done in 3-D.

2. Formulations and Geometry

Thermal analyses of extended surfaces having different shapes of cross sections are performed in the present work. Consideration is given to a 3-D numerical model of aluminium pipe bundle placed in an air duct of dimension $L \times H \times L$ (figure 1a). The fins of diameter *D* and height *l*, arranged in a staggered formation are placed over a base plate of dimension $L_b \times L_b \times H_b$ (figure 1). The arrangement is located inside the air duct at a distance $\left(H - L_b\right)/2$ from the inlet. With a longitudinal and transverse pitch of S_L and S_T, the system is analysed for a forced convection scenario. The considered numerical model is validated prior to the analysis of different cases. Air is driven inside the duct at an isothermal room temperature T_f and velocity $\vec{V}$ in the positive y - direction. Numerically, the outlet is considered to be at atmospheric pressure (P_o) and at outflow condition $\left(\hat{n} \cdot k\nabla T = 0\right)$. The walls of the air duct is at no slip conditions $\left(|\vec{V}| = 0\right)$ and at an isothermal room temperature T_f (Figure 1a). The base plate of the fin system is supplied with a total uniform heat of Q_s.

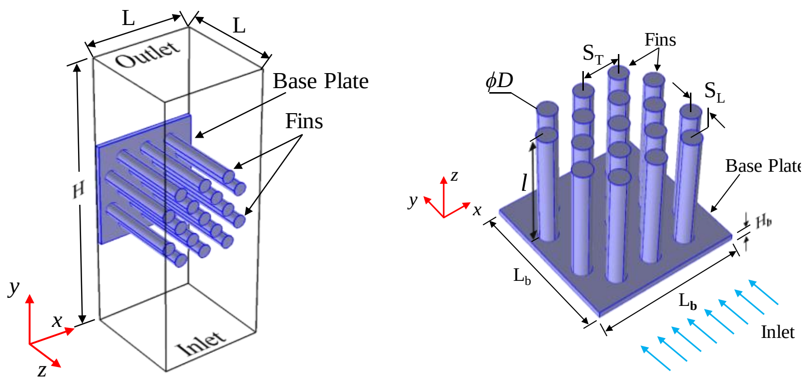
Generally, thermal behaviour of any system is determined by the participated modes of heat transfer and characteristics of the system. Considering the conductive and the convective modes of heat transfer, the energy equation given by

$$\rho c_p \left(\frac{\partial T}{\partial t} + \vec{V} \cdot \nabla T \right) = \nabla \cdot \left(k \nabla T \right) \tag{1}$$

where, ρ, c_p and k are the thermal properties of the system. With the known value of thermal properties, evaluation of temperature field of the system from Eqn. (1) requires the knowledge of the velocity field. Hence, apart from the energy equation, the governing equations of the considered system (Fig. 1a) also include steady form of continuity and momentum equations, given by

$$\nabla\cdot\left(\rho\vec{V}\right)=0 \tag{2}$$

$$\vec{V}\cdot\nabla\vec{V}=-\frac{1}{\rho}\nabla p+\nu\nabla^2\vec{V} \tag{3}$$



(a) (b)

Fig.1. Schematic diagram of the (a) computational domain and the (b) fin geometry

The governing equation (Eq. 1-3) has been discretized using finite element method (FEM). The discretization is performed using COMSOL multiphysics, a commercially available FEM solver. Solving Eq. 1-3 simultaneously using the above mentioned boundary conditions gives the velocity and temperature profiles in the computational domain. The average Nusselt number and the heat transfer co-efficient in such cases are calculated using,

$$Nu=\frac{Q_sD}{k_fA\left(T_w-T_f\right)}=\frac{h_fD}{k_f} \tag{4}$$

where A and T_w are the area of the base plate $\left(=L_b\times L_b\right)$ and the average surface temperature of the fin walls, respectively.

Fluid flow across a cylinder is a common in various engineering applications. There are many works that has proposed empirical co-relationship for convection heat transfer co-efficient. One of such special case is the flow over a cylindrical pin fin heat sink with application in electronic thermal management system. Khan et al. [12] worked on such cases and proposed co-relationship for convective heat transfer co-efficient of fin, h_f. With an approaching fluid velocity of $U\left(=|\vec{V}|\right)$, the Reynolds number for this flow is defined as

$$\mathrm{Re}_D=\frac{\nu U_{\max}}{D} \tag{5}$$

where ν is the kinematic viscosity of the flowing fluid and $U_{\max}$ is the maximum fluid velocity between the fins given by

$$U_{\max}=\max\left(\frac{S_T}{S_T-D}U,\frac{S_T}{2\left(S_D-D\right)}U\right) \tag{6}$$

where $S_D=\sqrt{S_L^2+\left(S_T/2\right)^2}$ is the diagonal pitch in the present case. The average Nusselt number for the fin is

$$Nu=\frac{h_fD}{k_f}=C_1\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3} \tag{7}$$

where $C_1=\dfrac{0.61S_T^{0.091}S_L^{0.053}}{1-2e^{-1.09S_L}}$ [12].

The calculated values of Nu and h_f obtained from the CFD solution, and the empirical relationship proves the correctness of the numerical modelling. Further, considering different cross-sectional shapes of pin fins the performances are checked and compared with the cylindrical pin fin. The considered shapes are arranged in staggered pattern are shown in Figure 2. The results are discussed in the following section.

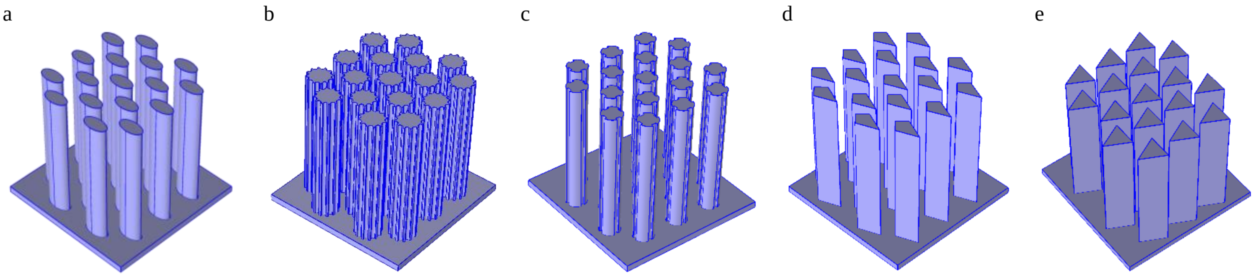


Fig. 2. Different shapes of pin fins considered for study (a) elliptical; (b) Sprocket; (c) Channeled; (d) kite; (e) triangular

3. Results and Discussion

In the following, we validate the formulations and the CFD model using the results obtained from the empirical relations given by khan et al. [12]. The 17 numbers of cylindrical Aluminium $\left(k = 237\,\mathrm{W/m\cdot K}, \rho = 2700\,\mathrm{kg/m^3}\ \text{and}\ c_p = 900\,\mathrm{J/kg\cdot K}\right)$ pin fins of are arranged in staggered fashion as shown in Figure 1(b). With $S_L = 19.6\,\mathrm{mm}$ and $S_T = 30\,\mathrm{mm}$, the fins (D = 100 mm, l = 100 mm) are fixed over a base plate of size 118 mm × 118 mm × 5 mm, maintaining a gap of 1 mm from both the side walls of the air duct. The dimensions of the air duct are 120 mm × 120 mm × 320 mm. A total uniform heat (Q_s) of 115 W is supplied at the bottom of the base plate in all the considered cases. The heated base plate conducts heat to the fins and finally the heat is dissipated into the air inside the duct. The forced air inside the duct carries the heat in upward direction using a fan.

Nusselt number is a parameter that depict about the strength of the convection heat transfer rate compared to the conductive heat transfer rate. The performance parameters (Nu and h_f) in the considered system are calculated using the empirical relations (Eq. 5-7) and computationally (Eq. 4). With tetrahedral grid to discretize the computational domain, the grid dependency tests were performed for different sizes of the grid. Using a grid having 1584685 numbers of elements, the governing equations were solved for a steady-state condition. Table 1 shows comparative values of average Nu and h_f for different cases of inlet velocity. A comparative plot of the same is also presented in Fig. 5a and 5b. From the results it has been observed that with increase in the inlet velocity of the air, the convective heat transfer coefficient increases. The increased value of h_f is due to elevation in the convective heat transfer rate. That also increases the value of average Nu. The compared results of average Nu and h_f are found within the acceptable accuracy. The deviation in the numerical result from the empirical is due to the ignored thermal radiation phenomenon in the mathematical modelling.

Table 1. Comparison of empirical and numerically obtained Nu and h_f for various conditions of Inlet velocity and temperature

Cases	Inlet Temperature T_f ºC	Inlet Velocity $\lvert\vec{V}\rvert$ m/s	Empirical		Computational	
			Nu	h_f	Nu	h_f
1	21.1	1.1	29.0	50.7	25.5	45.9
2	21.8	1.34	31.5	55.1	31.2	56.1
3	21.9	1.5	33.8	59.1	35.1	63.0
4	21.8	1.81	37.1	64.8	34.64	62.03
5	21.9	2.1	40.0	70.0	40.53	72.43

Having validated the numerical solver, the pin fins of different shape of cross sections are considered for study. The fins are modelled in such a way that the weight of the system is always maintained constant. In order to maintain uniformity in the modelling, the height of the fins are kept same for all the types, i.e., l = 100 mm. Further result has been generated for different case at different approach velocity is shown in the tabular form.

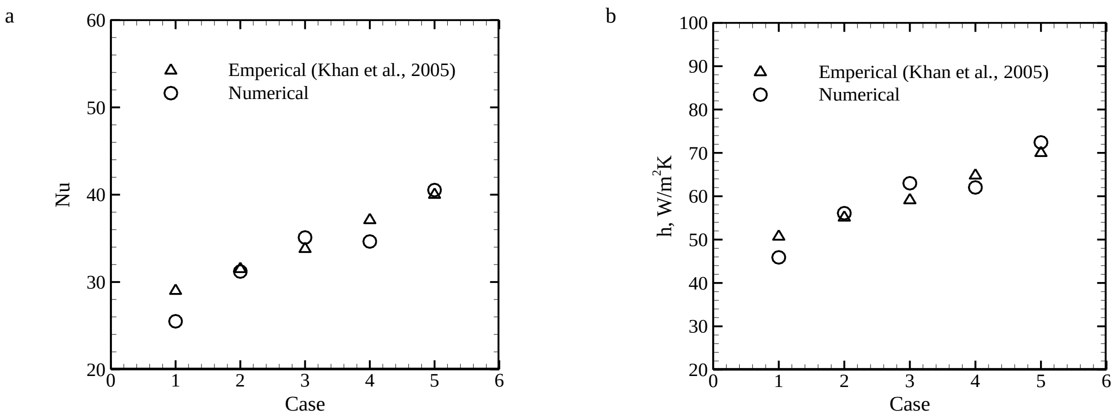


Fig. 5. Validation plot for (a) Nusselt number and (b) heat transfer coefficient

Table 2. Nu and h_f calculations for various shapes of fins at inlet velocity 1.1 m/s

Case	Characteristic Length(m)	Heat Flux (W/m^2)	Nu	h_f
Circular	0.0147	1177.8	29.08	52.43
Elliptical	0.0137	1120.2	44.95	86.52
Sprocket	0.0127	945.6	34.52	72.38
Kite	0.0140	997.3	49.88	94.23
Channelled	0.0114	935.0	31.15	72.19
Triangular	0.0133	890.6	31.31	63.11

Table 3. Nu and h_f calculations for various shapes of fins at inlet velocity 1.34 m/s

Case	Characteristic Length(m)	Heat Flux (W/m^2)	Nu	h_f
Circular	0.0147	1177.8	35.62	64.09
Elliptical	0.0137	1120.2	53.69	103.20
Sprocket	0.0127	945.6	41.53	86.90
Kite	0.0140	997.3	55.34	104.55
Channelled	0.0114	935.0	37.36	86.42
Triangular	0.0133	890.6	37.17	74.70

Table 4. Nu and h_f calculations for various shapes of fins at inlet velocity 1.5 m/s

Case	Characteristic Length(m)	Heat Flux (W/m^2)	Nu	h_f
Circular	0.0147	1177.8	40.08	71.98
Elliptical	0.0137	1120.2	59.43	114.05
Sprocket	0.0127	945.6	48.21	100.68
Kite	0.0140	997.3	61.04	115.11
Channelled	0.0114	935.0	41.43	95.70
Triangular	0.0133	890.6	41.01	82.25

Table 5. Nu and h_f calculations for various shapes of fins at inlet velocity 1.81 m/s

Case	Characteristic Length(m)	Heat Flux (W/m^2)	Nu	h_f
Circular	0.0147	1177.8	48.67	87.13
Elliptical	0.0137	1120.2	70.24	134.45
Sprocket	0.0127	945.6	54.93	114.37
Kite	0.0140	997.3	71.78	134.94
Channelled	0.0114	935.0	49.17	113.23
Triangular	0.0133	890.6	48.30	96.43

Table 6. Nu and h_f calculations for various shapes of fins at inlet velocity 2.1 m/s

Case	Characteristic Length(m)	Heat Flux (W/m^2)	Nu	h_f
Circular	0.0147	1177.8	56.47	100.91
Elliptical	0.0137	1120.2	79.92	152.74
Sprocket	0.0127	945.6	62.83	130.57
Kite	0.0140	997.3	81.39	152.72
Channelled	0.0114	935.0	56.13	129.00
Triangular	0.0133	890.6	54.79	109.12

3. Conclusions

The present work reported is the comparison of numerical data with empirical data. The empirical co-relationship for the calculation of thermal parameters is taken from the Khan et al. [12]. From the above work it has been concluded that heat transfer from pin fin increases with increase in inlet velocity and thus there is a good agreement of empirical and numerical results has been found. From the case studies, it has been found that kite and elliptical shape has greater effect on Nusselt number and heat transfer coefficient.

References

[1] R.S. Gorla, A.Y. Bakier, Thermal Analysis of Natural Convection and Radiation in Porous Fins. Int. Comm. Heat. Mass. 38 (2011), 638–645.

[2] S.F. Tsai, T.W.H. Sheu, S.M. Lee, Heat Transfer in a Conjugate Heat Exchanger with a Wavy Fin Surface. Int. J. Heat. Mass. Tran. 42 (1999), 1735-1745.

[3] A.K. Saha, S. Acharya, Parametric Study of Unsteady Flow and Heat Transfer in a Pin-Fin Heat Exchanger. Int. J. Heat. Mass. Tran. 46 (2003), 3815–3830.

[4] R.R. Jassem, Effect the Form of Perforation on the Heat Transfer in the Perforated Fins. SAVAP. Int. J. 4 (3) (2013), 198–207.

[5] S. Yadav, K. Das, K.M. Pandey, A Comparative Analysis of Heat Transfer in Extended Surfaces With And Without Holes. 5th International and 41st National Conference on Fluid Mechanics and Fluid Power, FMFP-2014.

[6] S. Gururatana, X. Li, Heat Transfer Enhancement of Small Scale Heat Sinks Using Vibrating Pin Fin. American J. App. Sci. 10 (8) (2013), 801-809.

[7] A. Sarma, A. Ramakrishna, CFD Analysis of Splayed Pin Fin Heat Sink for Electronic Cooling. Int. J. Engg. Re. & Tech. 1 (10) (2012).

[8] S. Agrawal, T.W. Simon, M. North, D. Bissell, T. Cui, Heat Transfer Augmentation of a Channel Flow by Active Agitation and Surface Mounted Cylindrical Pin Fins. Int. J. Heat Mass Trans. 87 (2015), 557-567.

[9] R.S. Sukumar, G. Sriharsha, S.B. Arun, D.P. Kumar, C.S. Naidu, Modelling and Analysis of Heat Sink with Rectangular Fins Having Through Holes. Int. J. Engg. Re. & App. (IJERA) 3(2) (2013), 1557-1561.

[10] S. Yadav, K. Das, K.M. Pandey, A Comparative Analysis of Heat Transfer in Extended Surfaces. 23rd National and 1st International ISHMT-ASTFE Heat and Mass Trans. Conference, IHMTC2015-480.

[11] L. Micheli, K.S. Reddy, T.K. Mallick, Experimental Comparison of Micro-Scaled Plate-Fins And Pin-Fins Under Natural Convection. Int. Comm. Heat and Mass Trans. 75(2016), 59-66.

[12] W.A. Khan, J.R. Culham, M.M. Yovanovich, Modelling of cylindrical pin fin heat sinks for electronic packaging. Proceedings of 21st IEEE Semiconductor Thermal Measurement and Management (SEMI-THERM) Symposium, San Jose, California. 2005, March 15-17.