Verma et al. 2018. *Int. J. Vehicle Structures & Systems*, 10(5), 363-366
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.10.5.12

International Journal of
Vehicle Structures & Systems
Available online at www.maftree.org/eja

Comparison of Natural Convection Heat Transfer from a Vertical Cylinder Fitted with Annular Step Fins and Annular Triangular Fins

Shubham Verma[a] and Harishchandra Thakur[b]

Dept. of Mech. Engg., Gautam Buddha University, Greater Noida, India
[a]Corresponding Author, Email: vermashubham124@gmail.com
[b]Email: harish@gbu.ac.in

ABSTRACT:

Natural convection heat transfer from a vertical cylinder with annular step and triangular fins has been studied numerically at various Rayleigh numbers within the laminar range. The computations were carried at constant fin spacing to tube diameter ratio of 1. In the current study, numerical simulations of Navier-Stokes equation supported with the energy equation are conducted for a vertical cylinder with annular step fins as well as triangular annular fins using the algebraic multi-grid solver of Fluent 15. With an increase in Rayleigh number, we've discovered a trend that the surface Nusselt number goes on increasing with comparison from a simple rectangular fin. Apart from this, the material needed for the step and triangular fins has been reduced with enhancements in the heat transfers.

KEYWORDS:

Annular step fin; Annular triangular fin; Nusselt number; Rayleigh number; Heat transfer; Natural convection

CITATION:

S. Verma and H. Thakur. 2018. Comparison of Natural Convection Heat Transfer from a Vertical Cylinder Fitted with Annular Step Fins and Annular Triangular Fins, *Int. J. Vehicle Structures & Systems*, 10(5), 363-366. doi:10.4273/ijvss.10.5.12.

ACRONYMS AND NOMENCLATURE:

A	Area of convection surface (m^2)
A_{fin}	Area of annular circular fin (m^2)
d	Diameter of horizontal cylinder (mm)
D	Fin diameter (mm)
g	Acceleration due to gravity (m/s^2)
h	Average heat transfer coefficient (W/m^2-K)
k	Thermal conductivity of fin material (W/m-K)
L	Length of the horizontal cylinder (mm)
N_{fin}	Number of fins
Nu	Average Nusselt number
p	Pressure (N/m^2)
p_{atm}	Atmospheric pressure (N/m^2)
Q	Convection heat transfer (W)
Ra	Rayleigh number
S	Inter-fin spacing (mm)
S/d	Fin spacing to tube diameter ratio or non-dimensional fin spacing
t	Thickness of fin (mm)
T_∞	Ambient temperature (K)
T_{film}	Mean film temperature (K)
T_S	Temperature at solid surface (K)
T_w	Cylinder surface temperature (K)
x, y, z	Cartesian coordinates
u, v, w	Velocity components along x, y, z directions respectively
i, j, k	Direction vectors along x, y, z directions respectively
ΔT	Base-to-ambient temperature difference, K
α	Thermal diffusivity (m^2/s)
β	Thermal expansion coefficient (1/K)
ν	Kinematic viscosity (m^2/s)
ρ	Density (kg/m^3)

1. Introduction

Amongst each of the heat transferring strategies, natural convection systems are simple, easy and low cost because of no demand of any additional elements. Therefore, it makes the system obviate of any additional moving components, and this reduces the stress to add any further constraints of designing for the heat transferring system. Because of no or very less parts used, it makes the system free of noises and unwanted vibrations that additionally makes the system maintenance free. But, there is a little bit of drawback in using natural convection cooling strategies. The rate of the heat transfer is lesser when compared with forced convection techniques as explained by Bejan [1] and Kreith et al [2]. The study of this fin usage to boost heat transfer was studied in detail by Guvenc and Yuncu [3] and Yazicioglu and Yuncu [4].

Annular fins are majorly proved beneficial to be used once it involves the improvement of heat transfer rate from a cylindrically formed surface as they are quite easy in terms of production. Heaps of existing literature is there within the field of natural convection heat transfer. Senapati et al [5] gave a numerical investigation on the heat transfer from a vertical cylinder fitted with annular fins. They have summarised that on adding more fins to the heated cylindrical tube, heat transfer keeps on increasing for laminar flow. Apart from this, Churchill and Chu [6] studied the heat transfers through an experiment for natural convection for the case of a flat plate and a horizontal cylinder and a correlation was generated for average Nusselt number as a function of

Rayleigh number and Prandtl number. Senapati et al [7] studied the heat transfers from a horizontal cylinder fitted with annular fins.

The authors gave an optimized value for fin spacing to extract the maximum heat transfers from an annular fin. The recent modifications done in the annular fins to improve or reinforce the heat transfers and at the same time to accommodate the restricted fin area accessibility includes the researches done by Kundu and Das [8]. They have analysed the performance of elliptic fins. Another theory was given by Senapati et al [9] who explained the utilization of eccentric fins by presenting a numerical study of heat transfer from a horizontal cylinder put in with such kind of a fin. They concluded that on increasing the eccentricity of the fins, the heat transfer decreases marginally. Then, some more modifications were given by Balram Kundu and Antario [10], where an optimum analysis on the design of an annular step fin is given. From this, we all know that annular step fins can be a bit advantageous in terms of production costs and material savings.

Now, on analysing the presented literature survey, we might get a lot of help in having a numerical investigation of natural convection heat transfer from a vertical cylinder fitted with annular fins having differently formed shape projections like step fins or triangular fins which could cut back the material used as compared to simple rectangular fins. In the present study, numerical simulation of full Navier-Stokes equation together with energy equation has been conducted for the natural convection over a vertical cylinder with annular fins having numerous forms of orientations of variable thickness for majorly laminar flow regimes. The calculations are performed at numerous Rayleigh number having a continuing inter fin spacing to diameter ratio (S/d) of 1. The results are obtained for the laminar range. For obtaining the solutions, the governing equations, a student version of ANSYS Fluent has been used. A comparison has been proposed for enhancing the heat transfers which could be helpful for industrial applications.

2. Mathematical modelling

A vertical cylinder having a diameter of 25mm is connected with various annular fins. The lengths of the fins used are 50mm each and have a relentless thickness of 1mm at the bottom. For the case of annular step fin and triangular fins, the shape changes are done as we tend go up. The material for the fins is chosen to be aluminium due to its high conductivity. The annular fin on a vertical cylinder is idealised by a 2-D axisymmetric geometry as shown in Fig. 1 and Fig. 2. The main target is to get the natural convection heat transfer rates for finned vertical cylinder having varied shape orientations. The fluid is assumed to have Boussinesq and laminar and steady as the Rayleigh numbers are in the laminar range. The governing equations used are:

$$\frac{\partial \mathrm{v_j}}{\partial \mathrm{x_j}} = 0 \tag{1}$$

$$\frac{\partial(\rho \mathrm{v_i v_j})}{\partial \mathrm{x_j}} = \rho \mathrm{g_i} \frac{\partial(\mathrm{p}+\frac{2}{3\mathrm{k}})}{\partial \mathrm{x_i}} + \frac{\partial[(\mu)\left(\frac{\partial \mathrm{v_i}}{\partial \mathrm{x_j}}+\frac{\partial \mathrm{v_j}}{\partial \mathrm{x_i}}\right)]}{\partial \mathrm{x_j}} \tag{2}$$

$$\frac{\partial(\rho \mathrm{v_j T})}{\partial \mathrm{x_j}} = \frac{\partial}{\partial \mathrm{x_j}}[\left(\frac{\mu}{\mathrm{Pr}}\right)\frac{\partial \mathrm{T}}{\partial \mathrm{x_j}}] \tag{3}$$

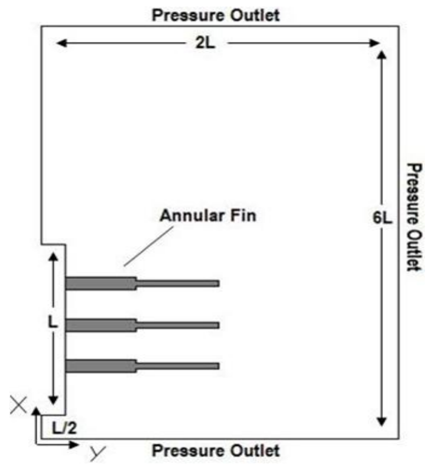

Fig. 1: 2D schematic diagram of step fin

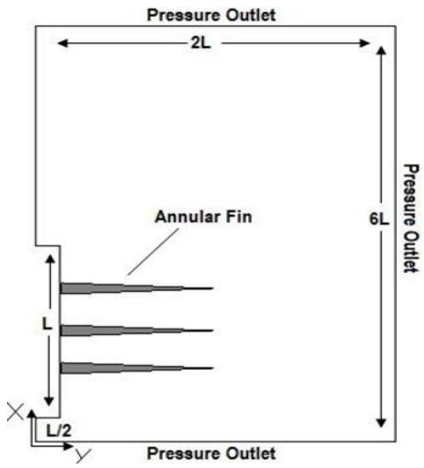

Fig. 2: 2D schematic diagram of triangular fin

The boundary conditions for the solution of equations are as follows:

At the axis:

$$\frac{\partial}{\partial y} = 0 \text{ and } \mathrm{v_y} = 0 \tag{4}$$

At the cylinder surface and the fin base:

$$V_x = V_y = 0, T = T_w \tag{5}$$

The temperature difference between the isothermal wall and the ambient temperature has been kept as $\Delta T = 50K$. With this temperature, the lengths of vertical cylinder are being set up on various Ra numbers. The rest of the parameters of air are kept constant at T_{mean} for the temperature ranges of 325K and 275K. As the temperature is obtained, the length is unknown. The lengths obtained are - 0.134m, 0.150m and 0.155m respectively for the various Ra numbers. The analysis is performed on these length segments separately and results have been obtained. Rayleigh number based on the tube length can be calculated using,

$$\mathrm{Ra} = \frac{\mathrm{g}\beta(\mathrm{T_w}-T_\infty)\mathrm{L}^3}{\nu\alpha} \tag{6}$$

The total heat transfer rate, Q is obtained from the numerical computations. The average heat transfer coefficient, h can be calculated using,

$$h = Q/A(T_w - T_\infty) \tag{7}$$

3. Numerical procedure

The differential governing Eqns. (1), (2) and (3) must be discretized by the finite volume methodology to seek out the needed algebraic equations after integrating them over control volume. To deduce the discretization equations, all the much-needed steps have been given within the paper by Senapati et al [7]. All the algebraic equations that resulted were resolved by an iterative methodology by using the boundary conditions Eqns. (5), (6) and (7) on the multi-grid problem solver of Fluent 15. For finding the momentum and energy equations, second-order upwind scheme was used for convective terms. For pressure equations, the body force weighted scheme was used. The relaxation sweeps were kept as 0.5 for pressure and 0.3 for momentum. For pressure correction equations, velocity and pressure were coupled using the simple algorithm. For the relative convergence criterion, 10^6 was kept for all the equations of energy, momentum and others.

The boundary-layer thickness has been neglected as compared to the diameter of the cylinder. Hence, on neglecting, the heat transfers are calculated with same relations as that of vertical plate. This is explained by Gebhart et al [11]. The meshing grid arrangements for the given computational domains of each of the annular step fin and triangular fins are shown in Fig. 3. In all, there are 3 fins to be placed on the vertical cylinder base and domain size is taken to be 2L × 6L (2L in y direction and 6L in x direction). From Fig. 4, it is clear that the maximum number of cells are made close to the base of the cylinder in comparison with the rest of the domain. This is done on purpose so as to capture the plume of natural convection around the base of the cylinder and fins more effectively with higher accuracy. Total of 56841 computational cells are created. Apart from this, a Quad mesh grid structure has been used for each of the computational domains. To capture the boundary layer phenomenon, the mesh is refined at the fin region and at the base of the cylinder.

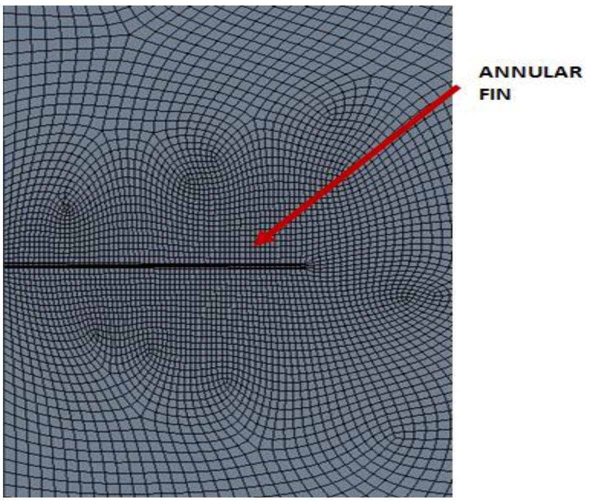

Fig. 3: A preview of the mesh component

4. Results and discussions

The equations have been resolved for annular step fins and triangular fins fitted over a vertically placed cylinder. The analysis is carried within the laminar ranges for the three Ra numbers (1.135×10^7, 1.617×10^7 and 1.773×10^7). The Nusselt number variation for the constant (S/d) ratio has been given in Fig. 4. The comparison is done between annular step fin and triangular fin. There is a small decrease within the Nusselt number for the case of annular step fin when compared to standard rectangular fin at the same Rayleigh number. There is a small increase in the Nusselt number for the case of triangular fins, when compared with the rectangular fins.

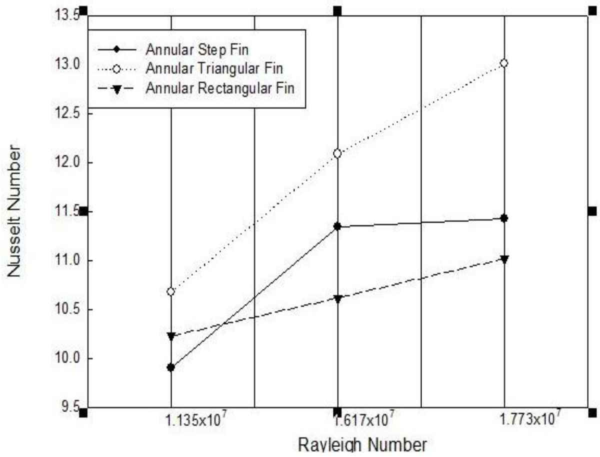

Fig. 4: Comparison of Nusselt number at various projections

The variations in the temperature contour of each of the annular step fin and triangular fin can be seen from Fig. 5 and Fig. 6 respectively. The plume has attained different shape profile for the case annular step fin after the first fin. In case of triangular fins, the temperature contour is seen to be uniform in all the fin spacing.

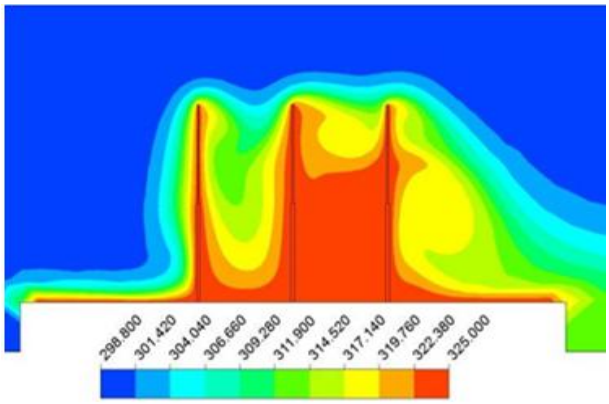

Fig. 5: Temperature contour in step fin

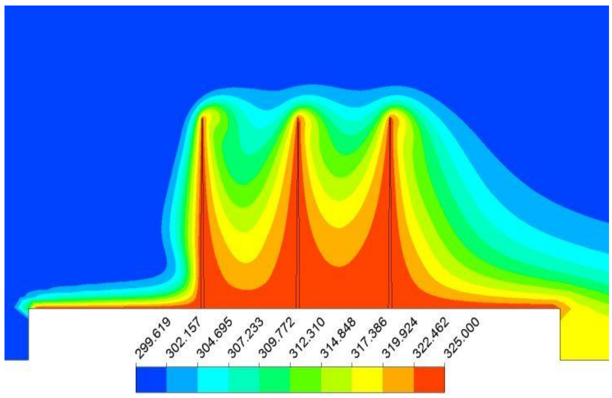

Fig. 6: Temperature contour in triangular fin

The results of annular step fins and triangular fins on the flow field vary a bit. From Fig. 7 and Fig. 8, it is clear that the flow plume is minimum at the tip of the fin. Each of the figure make it clear that that flow vectors are not fully developed in the inter fin space regions. The step fin surface is not smooth and uniform throughout the length. Hence, it gave an in-consistent flow plume.

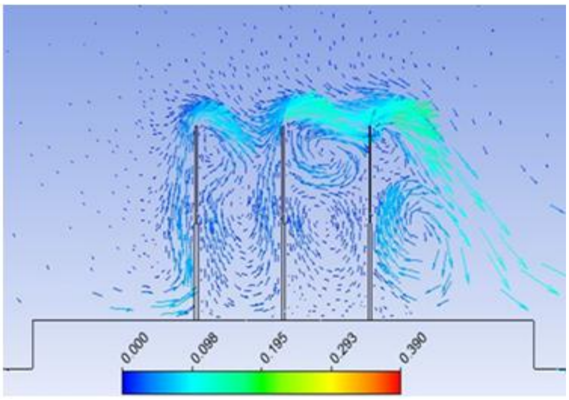

Fig. 7: Flow vectors in step fin

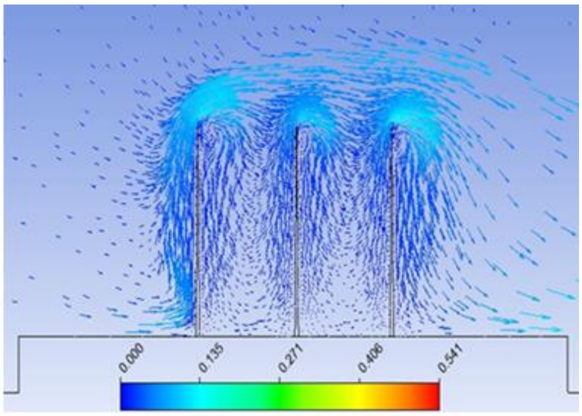

Fig. 8: Flow vectors in triangular fin

5. Conclusions

In this paper, annular fins of assorted projections have been studied for heat transfer underneath natural convection within the laminar ranges ($10^4 <$ Ra $< 10^8$). The changing effects of Nusselt number on temperature and velocity plumes by making annular step fins and triangular fins are recorded in the laminar regimes. The triangular annular fins result in more increase in Nusselt number variations than the annular step fin. Each of these new fins have higher Nusselt number than the standard rectangular fin. The temperature plumes of annular triangular fin are seen to be more uniform as compared with the annular step fin. The visualization of flow field concluded that the annular triangular fin has shown more consistent flow than the step fin.

REFERENCES:

[1] A. Bejan. 2004. *Convection Heat Transfer*, 3rd Ed., Wiley.

[2] F. Kreith, R.M. Manglik and M.S. Bohn. 2011. *Principles of Heat Transfer*, 7th Ed., Cengage Learning.

[3] A. Güvenç and H. Yüncü. 2001. An experimental investigation on performance of fins on a horizontal base in free convection heat transfer, *Heat Mass Transf.*, 37, 409-416.https://doi.org/10.1007/s002310000139.

[4] B. Yazicioğlu and H. Yüncü. 2007. Optimum fin spacing of rectangular fins on a vertical base in free convection heat transfer, *Heat Mass Transf.*, 44, 11-21.

[5] J.R. Senapati, S.K. Dash and S. Roy. 2017. Numerical investigation of natural convection heat transfer over annular finned vertical cylinder, *Int. J. Heat Mass Transf.*, 111, 146-159.

[6] S.W. Churchill and H.H.S. Chu. 1975. Correlating equations for laminar and turbulent free convection from a horizontal cylinder, *Int. J. Heat Mass Transf.*, 18, 1049-1053. https://doi.org/10.1016/0017-9310(75)90222-7.

[7] J.R. Senapati, S.K. Dash and S. Roy. 2016. Numerical investigation of natural convection heat transfer over annular finned horizontal cylinder, *Int. J. Heat Mass Transf.*, 96, 330-345. https://doi.org/10.1016/j.ijheatmasstransfer.2016.01.024.

[8] B. Singh and S.K. Dash. 2007. Performance analysis and optimization of elliptic fins circumscribing a circular tube, *Int. J. Heat and Mass Transf.*, 50, 173-180. https://doi.org/10.1016/j.ijheatmasstransfer.2006.06.043.

[9] J.R. Senapati, S.K. Dash and S. Roy. 2016. 3D numerical study of the effect of eccentricity on heat transfer characteristics over horizontal cylinder fitted with annular fins, *Int. J. Thermal Sci.*, 108, 28-39. http://dx.doi.org/10.1016/j.ijthermalsci.2016.04.021.

[10] B. Kundu, K.S. Lee and A. Campo. 2015. An ease of analysis for optimum design of an annular step fin, *Int. J. Heat and Mass Transfer*, 85, 221-227. https://doi.org/10.1016/j.ijheatmasstransfer.2015.01.111.

[11] B. Gebhart, Y. Jaluria, R.L. Mahajan and B. Sammakia. 1989. *Buoyancy-Induced Flows and Transport*, Hemisphere Publishing. https://doi.org/10.1137/1031037.

[12] J.P. Holman. 2010. *Heat Transfer*, 10th Ed., Mc Graw-Hill.