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Nature of velocity across the solid tube with Conical fins & Trapezoidal fins using CFD Altair hyper works (Acusolve)

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ABSTRACT

Natural convection from a heated pipe having fins of various configurations using CFD tool Acusolve has been carried out in this research paper. The material under consideration is aluminum and the free stream fluid is air. The heat transfer rate from the fins and maximum and minimum temperature has been calculated and compared for various fin configurations. Velocity contours for various fin configurations has been plotted and the motion of heated fluid is showing the convection loops formed around the heated pipe surface. The assumptions during the analysis have been taken considering the manufacturing and practical applications and working conditions. Hence the results obtained can be referred to while solving any such kind of problems in the practical field where only natural convection is under consideration. After comparing it is shown that the best configuration for this type of convective heat transfer of a heated pipe is a TRAPEZOIDAL fin as they have the highest total heat transfer rate.

Keywords: Altair (Hyper works) Acusolve (CFD), MATLAB.

INTRODUCTION

Heat exchangers are widely used in various, transportation, industrial, or domestic applications such as thermal power plants, means of heating, transporting and air conditioning systems, electronic equipment and space vehicles. In all these applications improvement in the efficiency of the heat exchangers can lead to substantial cost, space and material savings. Hence considerable research work has been done in the past to seek effective ways to improve the efficiency of heat exchangers. The referred investigation includes the selection of fluid with high effective heat transfer surfaces made out of high conductivity materials, high thermal conductivity and selection of their flow arrangements. For both single and two phase heat transfer effective heat transfer enhancement techniques have been reported. However in the present work only SINGLE PHASE STEADY STATE NATURAL CONVECTION technique has been considered. The heat transfer enhancement methods reported in publications be summarized in many forms but primarily they may be grouped as active enhancement methods.

OBJECTIVE OF RESEARCH

The exchange of heat energy is studied on a tube with circular cross-section and with specific inner and outer radius having outer disc shaped fins. The fins attached with the tube can be of variable shape and size. Two basic types of fins are considered and the transfer of heat energy from a tube with such fin configurations is estimated. The design calculations of the tube and the fin dimensions are done based upon equations suitable for the maximum heat transfer rate at low production costs. The material used for the calculations is considered to be ALUMINIUM. Both the tube and fins are considered to be made up of Aluminum and the fluid inside the tube is air. ALTAIR HYPERWORKS VERSION 11.0 version is used for the entire simulation processes. Experimental values of the working temperatures and corresponding properties for the fin and tube material along with air is considered and fed to the software. The convection type under consideration is NATURAL CONVECTION. The tube is vertically situated and vertical flow is considered for calculation. A very minimal fluid velocity is assumed and the entire heat transfer process is made to happen under the influence of gravity. The objective of the last part of the project is to plot various contours suggesting the ease of heat transfer with various fin cross-sections for example temperature contours, velocity contours across the length and cross section of the pipe. Various graphs suggesting the heat transfer rates also drawn by the software. The final objective of the work is to compare the results and to find out the best fin cross-section for the specified working conditions. Also the results are compared with that of different fin configurations (external and internal spiral fins) to find out the best fin configuration for the working conditions.

REVIEW OF PAPERS

The subject of heat transfer enhancement is of serious interest in the design of compact heat exchangers. The emphasis is given on minimizing the space occupied by the equipment for the desired rate of heat transfer. A large number of augmentation techniques have been developed in the last few decades and these are applicable to diverse areas such as, single phase flows, two phase flows and convective mass transfer. A number of review articles and handbooks by Bergles (1978, 1983 and 1985) deals with the enhancement of heat transfer for different applications.

A paper by D. Thornhill et al [2] works on proposing a variant for his formula to rectify this mistake, considering more fin geometries. Considering the heat transfer distribution around the circumference of the cylinder, for any particular fin geometry and flow condition, values are determined that are considered constant along the fin's length at any particular angular position around the cylinder surface, ignoring only the circumferential heat transfer by conduction around the cylinder and through the fins. Another basic flaw in the fundamental equations derived by Gibson and Thornhill was that they had a zero heat transfer at zero velocities, which is false. An attempt to correct this was made by Masao Yoshida and his group [1]

Biswas et al. (1989) numerically studied the developing laminar mixed convection in a rectangular channel with wing-type vortex generators. It was demonstrated that the passive generation of longitudinal vortices can be used to enhance natural secondary flows. Numerical calculations were performed at Reynolds numbers of 500 and 1815 with Grashof numbers = 0, 2.5 × 105, and 5 × 105. A single delta wing, with a unity aspect ratio and attack angles of 20° and 26°, was attached along its trailing edge to the bottom of the channel. The channel wall had no a hole under the wing, but wings are normally formed by punching them from the fin. So, Biswas and Chattopadhyay (1992) further extended this numerical model by including a hole under the delta wing for forced convection heat transfer.

Acusolve typically solves a given problem in the first attempt. Fully converged solutions are reliably obtained using Acusolve efficient steady-state solver. Nonlinear convergence remains strong even as solutions approach their final result. Two key components contribute to this robustness: the GLS finite element formulation, and a novel iterative linear equation solver for the fully coupled pressure/velocity equation system. This powerful iterative solver is highly stable and is capable of efficiently handling unstructured meshes with high aspect ratios and badly distorted elements commonly produced by fully automatic mesh generators. This linear solver yields significant stability and convergence advantages over the segregated solution procedures commonly found in many commercial incompressible flow solvers.

MATHEMATICAL FORMULATION

GOVERNING EQUATION

An important consideration in the design of finned surfaces is the selection of proper fin length L. Normally, it understood that, longer the fin, the larger the heat transfer area and thus the higher the rate of heat dissipation from the fin surface. But at the same time, with long fins, the weight, cost, and fluid friction increase. Therefore, increasing the length beyond a certain value cannot be justified unless the added benefits outweigh the increased cost. Further, the fin efficiency decreases with increasing fin length because decrease in temperature along the fin length. The fin lengths that cause the fin efficiency to drop below 60% cannot be justified economically and should be used.
With regard to limiting condition of fin length, when heat transfer does not increase with an increase in the length of fin can be recognized by:

$$\frac{dQ_{fin}}{dL} = 0$$

For the fins loosing heat by convection at its tip, the rate of heat transfer is given by:

$$Q_{fin} = \sqrt{h}PkA_c\ (T_0 - T_\infty) \times \frac{\sinh(mL)+\frac{h}{mk}\cosh(mL)}{\cosh mL+\frac{h}{mk}\sinh mL}$$

$= \sqrt{h P k A_c}\,(T_0 - T_\infty)\,\frac{\tanh(mL)+\frac{h}{mk}}{1+\frac{h}{mk}\tanh(mL)}$

Treating k, h, P, $A_c(T_0 - T_\infty)$ and m as constant quantities and differentiating above equation with respect to fin length L and equating it to zero;

$\frac{dQ_{fin}}{dL} = \sqrt{h P k A_c}\,(T_0 - T_\infty)\frac{d}{dL}\{\frac{\tanh(mL)+h/mk}{1+\left(\frac{h}{mk}\right)\tan mL}\} = 0$

Or $[1 + \frac{h}{mk}\tanh(mL)] \times m\,\sec^2 h\,(mL) - [\tanh mL + (\frac{h}{mk})] \times \frac{h}{mk} m\sec^2 h\, mL = 0$

The simplification of this equation leads to

$m^2 - \frac{h^2}{k^2} = 0$ Or $mk = h$

Or $\sqrt{\frac{kP}{hA_c}} = 1$ or $\frac{hP}{kA_c} = 1$

Introducing $P \approx 2w$, and $A_c = wt$.

$\frac{2k}{ht}=1$ or $\frac{1}{h} = \frac{t/2}{k}$

The term 1/h represents an external (convection) thermal resistance and $\frac{t/2}{k}$ represents internal (conduction) thermal resistance of a plane wall of thickness one half of a fin thickness.

The ratio of conduction resistance to convection resistance is known as the Biot number, that is

$Bi = \frac{ht}{2k}$

So we can draw the following conclusion with the help of equation.

After attachment of fins to a surface, if external thermal resistance is equal to internal thermal resistance as in equation, i.e.

$$\frac{1}{h} = \frac{t/2}{k}$$

Bi = 1 *i.e.* $\frac{h}{mk} = 1$

Then equation, for find heat transfer rate yields to

$Q_{fin} = hA_c\,(T_0 - T_\infty)$,

Which represents the heat transfer rate from primary (root) surface without any fin. It suggests that as long as h/mk = 1, the heat transfer rate from the primary surface will not change by attaching fins as shown in fig()

When, internal resistance of fin is greater than the external resistance *i.e.*

$\frac{t/2}{k} > \frac{1}{h}$ or $Bi > 1$ or $\frac{h}{mk}>1$

The addition of fins (secondary surface) on the primary surface will reduce the heat transfer rate or the fins will act as insulating medium on the surface. It may happen when value of h is very high as for flowing liquids, and during change of phase. Therefore, the fins are not used on liquids side and on evaporating and condensing surfaces.

When internal resistance of fin is less than the external resistance *i.e.*

$\frac{t/2}{k} < \frac{1}{h}$ or $Bi < 1$ or $\frac{h}{mk}<1$

The use of fins will increase the heat transfer from the primary surface. In actual practice, the use of fins can only be justified, when the parameter $\frac{2k}{ht}$ has a value equal to or exceeding five. The use of fins on a surface can only be justified, if the parameter $\frac{2k}{ht}$ has a value equal to or exceeding five.

$\frac{2k}{ht} \geq 5$ or $Bi \leq \frac{1}{5}$

Further, it should be noted that the rate of heat dissipation beyond a certain length of fin is quite less as compared with the rate of heat dissipation with increase in parameter 2*k/ht*. Therefore, the use of shorter fins of higher conducting materials is more effective than longer fins.

Design Calculation

Solid Tube case:

R =25×10^-3, h = 10 , L = 0.15 , T_s = 380, T_∞=300
Now , $T_f = \frac{Ts + T_\infty}{2}$ 340
$B = \frac{1}{Tf}$ = 2.942×10^-3

Now, Surface area.
A_s = 2ΠRL = .02355
Q = hAs (T_s - T_∞) = 18.84

Velocity Contours for Solid Tube without fin

Bulk Temperature (T_b)
$T_b = \frac{Ts - T_\infty}{2} = 40^0C$

From Data Book
ρ = 1.029,ν = 20.02×10^-6, P_r = 0.694, K_{air} = 0.0296, g = 9.81

$Gr = \frac{g\beta(Ts - T_\infty)L}{\nu^2}$
Ra = GrPr = 13.6×10^7 , Then flow is laminar

At x = 0.15 at just Outer body of the Solid Tube.
U = 5.17× ν[Pr × $\frac{20}{21}$]^-0.5 ×[$\frac{g\beta(Ts - T_\infty)}{\nu^2}$]^0.5 × x^0.5

Therefore U = 2.38, which is velocity flowing around the solid tube

$U_{software}$ = 2.756
Percentage(%) Error = $\frac{\mathbf{2.756} - 2.38}{\mathbf{2.756}}$×100

Conical & Trapezoidal Fin Case:
Surface area of Conical fin
Surface area of non-fin area A_1 = 2ΠR×0.06
Surface area of one fin = Π($0.035^2 - 0.25^2$) × 2
Surface area of 15 fin
A_2 = 15Π($0.035^2 - 0.025^2$) × 2

Then, Total surface area $A_s = A_1 + A_2$
Then Heat transfer Rate(Q) = $hA_s(T_s - T_\infty)$
Surface area of Trapezoidal Fin
Surface area of non-fin area.
A_1 = 2ΠR×0.06
Surface area of one fin

A_2 = Π($0.035^2 - 0.025^2$) × 2×Area of Tip side
A_2 = Π($0.035^2 - 0.025^2$) × 2×0.035×2×10^-3
Surface area of 15 fin
$A_3 = A_2$×15

Total Surface area $A_s = A_1 + A_3$

$Q = hA_s(T_s - T_\infty)$
Now for Temperature Distribution

As we know that Q = 18.84 (conduction), K =280
Now, considering only one fin

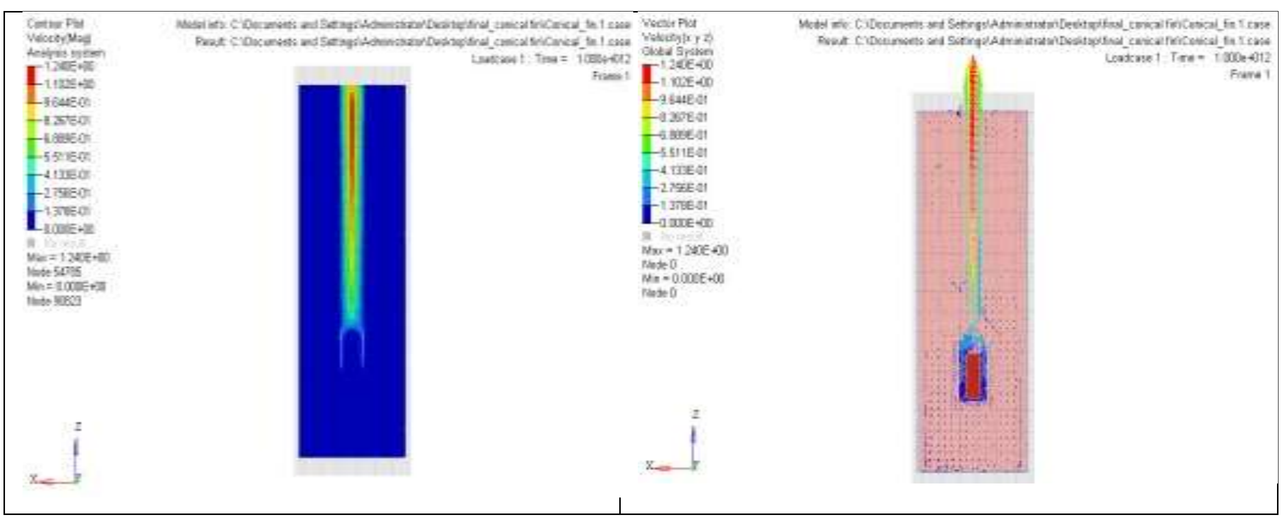
We get,

$$Q = \frac{T\text{surface} - \text{Tfintip}}{\frac{\text{L}}{\text{KAcross-section}}}$$

Fin length (L) =10mm
Across-section = 2ΠR×Thickness at base,

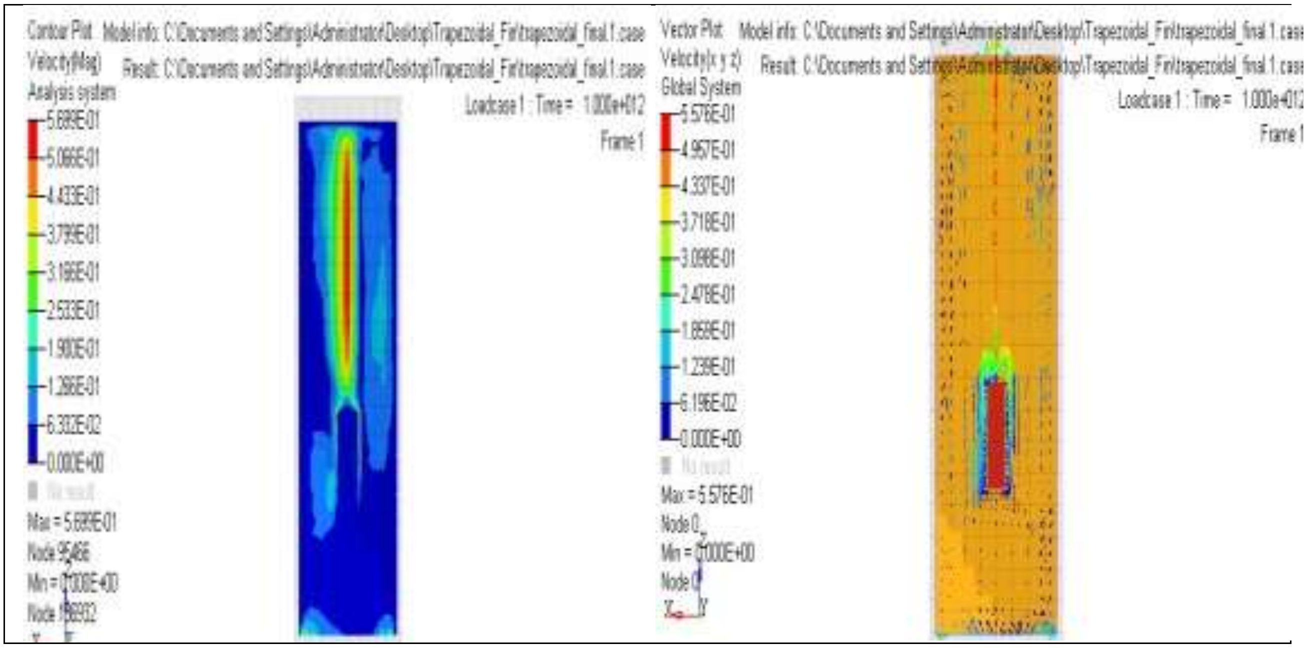
So from above relation we can get the fin tip temperature.
h = 241.7{0.0247-0.00148(l0.8/p0.4)} u0.73
l = fin length (mm)
p = fin pitch (mm)
u = air velocity (km/hr)
h = fin surface heat transfer co-efficient ($W/m^2\,^oC$)
by D. Thornhill
h = 2.11 * u^0.71 * s^0.44 * L^0.14
s = fin separation at middle fin length (mm)
h = 2.11 * u^0.71 * s^0.44 * L^0.14
so from above correlations we can find the velocity speed over the fins surface in CFD.

Velocity distribution through Conical fins



In the above figure flow of velocity is shown which is directing from vertical downward direction to vertical upward direction. It means through CFD (AcuSolve) flow is shown through natural convection heat transfer process which depict the nature of velocity across the solid tube with conical fins. In the above figure flow of velocity vector is shown which is directing from vertical downward direction to vertical upward direction. It means through CFD (AcuSolve) flow is shown through natural convection heat transfer process which depicts the nature of velocity across the solid tube with conical fins and vector shows the flow nature across the solid tube with conical fins which finally flow vertically upward direction by showing the directions and magnitude.

Velocity distribution through Trapezoidal fins



In the above figure flow of velocity is shown which is directing from vertical downward direction to vertical upward direction. It means through Cfd (AcuSolve) flow is shown by natural convection heat transfer process which depict the nature of velocity across the solid tube with trapezoidal fins. In the above figure flow of velocity vector is shown which is directing from vertical downward direction to vertical upward direction. It means through Cfd (AcuSolve) flow is shown through natural convection heat transfer process which depicts the nature of velocity across the solid tube with Trapezoidal fins and vector shows the flow nature across the solid tube with Trapezoidal fins which finally flow vertically upward direction by showing the directions and magnitude.

CALCULATION TABLE:

S.no	Solid tube with conical fin	Solid tube with trapezoidal fin
1	1.24	0.569
2	1.102	0.506
3	0.964	0.443
4	0.826	0.379
5	0.688	0.316
6	0.551	0.253
7	0.413	0.19
8	0.275	0.126
9	0.137	0.633
10	0	0

In the above given table comparative study of velocity distribution is shown between solid tube with conical & Trapezoidal fin using natural convection which is obtain by the Acusolve which determine the variation of velocity distribution and also summarises the comparison of velocity distribution between them.

CONCLUSION

Engineers can evaluate increasing heat exchanger performance through a logical series of steps. The first step considers if the exchanger is initially operating correctly. The second step considers increasing pressure drop if available in exchangers with single-phase heat transfer. Increased velocity results in higher heat transfer coefficients, which may be sufficient to improve performance. Next, a critical evaluation of the estimated fouling factors should be considered. Heat exchanger performance can be increased with periodic cleaning and less conservative fouling factors. Finally, for certain conditions, it may be feasible to consider enhanced heat transfer through the use of finned tubes, inserts, twisted tubes, or modified baffles. Velocity contours for various fin configurations has been plotted and the motion of heated fluid is shown successfully. The assumptions during the analysis have been taken considering the manufacturing and practical applications and working conditions. Hence the results obtained can be referred to while solving any such kind of problems in the practical field where only natural convection is under consideration. After comparing it is shown that the best configuration for this type of convective heat transfer of a heated solid tube is a trapezoidal fin as they have the highest total heat transfer rate.

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