Review Study of Heat Transfer through Perforated Fin Arrays

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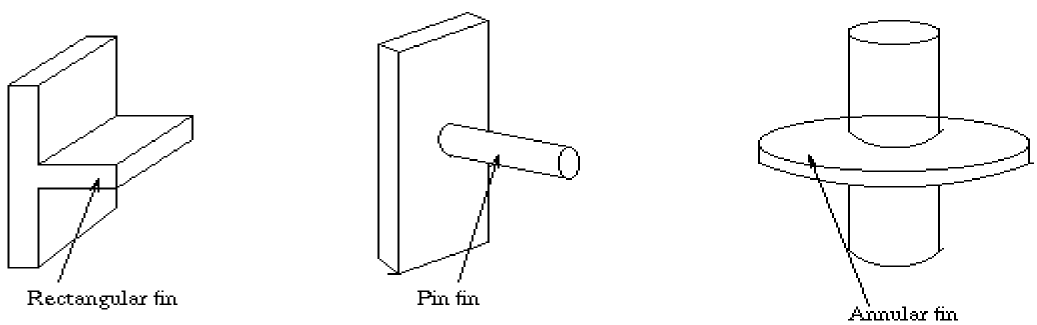
Abstract: *It was pointed out that most effective methods employ fins with appropriate length, geometry and material characteristics according to the specific application. The primary task of the fins employed in highly effective heat exchangers is to increase the heat transfer surface and heat transfer coefficient. Such a double effect can be achieved by a densely populated surface with elements in the form of interrupted lamellae or profiles with a smaller cross-section compared with their length. An interrupted element prevents boundary layer development in the flow direction and hence the thermal resistance of such fins is usually much lower than it would be in the case of continuous lamellae. Furthermore, the flow distribution and mixing of fluid streams with different temperatures are much better and hence the heat transfer from fins into fluid or vice versa is quite effective. There exist many different types of such fins, but most popular are wavy, strip and louvered fins. Wavy fins are widely used in the air conditioning, refrigeration and process industries. Generally, heat exchanger containing wavy fins can be encountered in gas to gas and liquid to gas heat transfer applications.*

I. INTRODUCTION

Heat dissipation is a drastic issue to tackle due to continued integration, miniaturization, compacting and lightening of equipment. Heat dissipaters are not only chosen for their thermal performance but also for other design parameters that includes weight, cost and reliability, depending on application. The present work analyses a study to investigate the heat transfer enhancement over horizontal flat surface with rectangular fin arrays called perforated fins with flat plate called non perforated fins by free convection along with conduction within the material for different thermal conductivities.

Different types of fins



Input Values

- Gravitational force (g) = 9.81
- Base temperature Tb = 300^0C
- Ambient temperature T0 = 30^0C
- Thermal conductivity of air kair = 0.024
- Prandtle number Pr = 0.714
- Width of the plate L = 0.05 m
- Length of the plate W = 0.2 m
- Thermal conductivity of the plate k = 50 - 300
- Number of perforation in X direction Sx = 0.001 - 0.005
- Number of perforation in Y direction Sy=0.001 – 0.005
- Side of the perforated hole b = 0.001 – 0.005
- Thickness of the plate t = 0.001 – 0.005
- Aspect ratio Ar = 1 – 5
- Gas constant R = 0.287
- Atmospheric pressure p = 101.325 N/m^2

II. REVIEW OF LITERATURE

Swee-Boon Chin, et al., (2013) Increasing miniaturization of high speed multi-functional electronics demands ever more stringent thermal management. The present work investigates experimentally and numerically the use of staggered perforated pin fins to enhance the rate of heat transfer in these devices. In particular, the effects of the number of perforations and the diameter of perforation on each pin are studied. The results show that the Nusselt number for the perforated pins is 45 % higher than that for the conventional solid pins and it increases with the number of perforation.

Michael J. Nilles (1995) An introduction to the design, fabrication, and use of perforated plate heat exchangers is presented. Emphasis is placed on the numerical solution of the heat exchange equations, and methods are given for determining the characteristics of the perforated plate heat exchanger required for a given application directly from the parameters of the application, the properties of the perforated plate matrix material, and the properties of the working fluid. Topics addressed include pressure drop, plate conduction, and corrections for longitudinal thermal conductivity in the exchanger and entrance effects.

Bayram Sahin and Alparslan Demir (2007) The present paper reports on heat transfer enhancement and the corresponding pressure drop over a flat surface equipped with square cross-sectional perforated pin fins in a rectangular channel. The channel had a cross-sectional area of 100–250 mm2. The experiments covered the following range: Reynolds number 13,500–42,000, the clearance ratio (C/H) 0, 0.33 and 1, the inter-fin spacing ratio (Sy/D) 1.208, 1.524, 1.944 and 3.417. Correlation equations were developed for the heat transfer, friction factor and enhancement efficiency.

Zan Wu et al., (2013) Staggered pattern perforations are introduced to isolated isothermal plates, vertical parallel isothermal plates, and vertical rectangular isothermal fins under natural convection conditions. The performance of perforations was evaluated theoretically based on existing correlations by considering effects of ratios of open area, inclined angles, and other geometric parameters.

Andrea de Lieto Vollaro et al., (1999) The goal of this analysis has been the search for the optimal configuration for a finned plate (with rectangular and vertical fins) to be cooled in natural convection. Utilizing a simplified relation of the fins heat exchange some simple expressions for the determination of the optimum value of the fins spacing have been developed as a function of the parameters which feature in the configuration: dimensions, thermal conductivity, fins absorption coefficient and fluid thermo-physical properties.

Pardeep Singh et al., (2014) In this research, the heat transfer performance of fin is analyzed by design of fin with various extensions such as rectangular extension, trapezium extension, triangular extensions and circular segmental extensions. The heat transfer performance of fin with same geometry having various extensions and without extensions is compared. Near about ranging 5% to 13% more heat transfer can be achieved with these various extensions on fin as compare to same geometry of fin without these extensions. Fin with various extensions design with the help of software Auto CAD. Analysis of fin performance done through the software Autodesk® Simulation Multi physics.

Ramkrushna S. More et al. (2011) Fins are also called as extended surfaces the main purpose of that is to increase the heat transfer rate Fins offer an economical and trouble free solution in many situations demanding natural convection heat transfer. This fins are use for many applications such as variety of engineering applications, studies of heat transfer and fluid flow associated with such arrays are of considerable engineering significance. Geometry of fin arrays play an important role in heat transfer rate for that different types of fin arrays are used such as rectangular, circular, triangular and trapezoidal are used.

David J. Kukulkaa and Kevin G. Fuller (2010) Heat transfer enhancement has become popular recently in the development of high performance thermal systems. Much work has been done to gain an understanding of the fundamental flows that exist in arrays of smooth, parallel plates. A wide variety of industrial processes involve the transfer of heat energy. These processes provide a source for energy recovery and process fluid heating/cooling.

Dr. Aziz M. Mhamuad Thamir Kh. and Ibrahim Raid R. Jasim (2008) This work treats the problem of heat transfer for perforated fins under natural convection. The temperature distribution is examined for an array of rectangular fins (15 fins) with uniform cross-sectional area (100x270 mm) embedded with various vertical body perforations that extend through the fin thickness. The patterns of perforations include 18 circular perforations (holes). Experiments were carried out in an experimental facility that was specifically design and constructed for this purpose. The heat transfer rate and the coefficient of heat transfer increases with perforation diameter increased.

III. MATERIALS AND METHODS

Heat Transfer from a Fin

Fins are used in a large number of applications to increase the heat transfer from surfaces. Typically, the fin material has a high thermal conductivity. The fin is exposed to a flowing fluid, which cools or heats it, with the high thermal conductivity allowing increased heat being conducted from the wall through the fin. The design of cooling fins is encountered in many situations and we thus examine heat transfer in a fin as a way of defining some criteria for design.

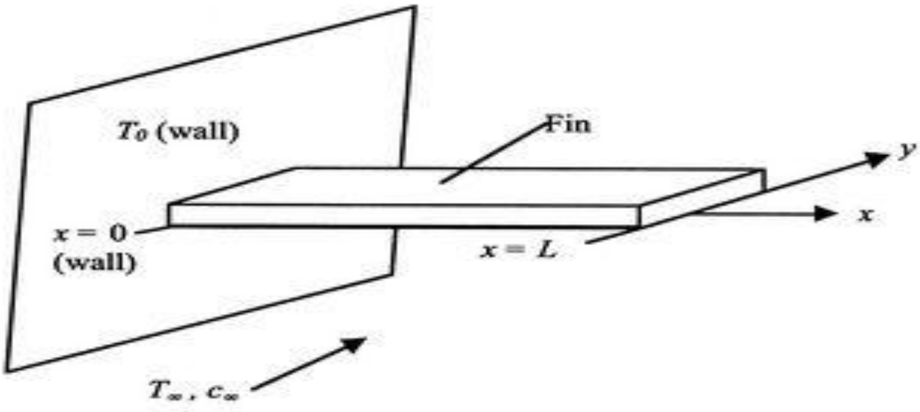


Figure 3.1: Geometry of heat transfer fin

A model configuration is shown in Figure 3.1. The fin is of length L. The other parameters of the problem are indicated. The fluid has velocityc_∞ and temperature T_∞ . We assume (using the Reynolds analogy or other approach) that the heat transfer coefficient for the fin is known and has the value h. The end of the fin can have a different heat transfer coefficient, which we can call h_L. The approach taken will be quasi-one-dimensional, in that the temperature in the fin will be assumed to be a function of x only. This may seem a drastic simplification, and it needs some explanation. With a fin cross-section equal to A and a perimeter P, the dimension in the transverse direction is $\frac{A}{P}$ (For a circular fin, for example, $\frac{A}{P} = \frac{r}{2}$). The regime of interest will be taken to be that for which the Biot number is much less than unity, $B_i = \frac{h\left(\frac{A}{P}\right)}{k} \ll 1$, which is a realistic approximation in practice.

The physical content of this approximation can be seen from the following. Heat transfer per unit area out of the fin to the fluid is roughly of magnitude $h(T_w - T_\infty)$ per unit area. The heat transfer per unit area within the fin in the transverse direction is (again in the same approximate terms)

$k\frac{(T_1 - T_w)}{\frac{A}{P}}$

Where T_1 is an internal temperature. These two quantities must be of the same magnitude. If $\frac{h\left(\frac{A}{P}\right)}{k} \ll 1$, then $\frac{(T_1 - T_w)}{(T_w - T_\infty)} \ll 1$.

In other words, if $B_i \ll 1$, there is a much larger capability for heat transfer per unit area across the fin than there is between the fin and the fluid, and thus little variation in temperature inside the fin in the transverse direction. To emphasize the point, consider the limiting case of zero heat transfer to the fluid, i.e., an insulated fin. Under these conditions, the temperature within the fin would be uniform and equal to the wall temperature.

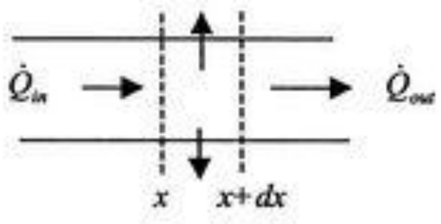


Figure 3.2: Element of fin showing heat transfer

If there is little variation in temperature across the fin, an appropriate model is to say that the temperature within the fin is a function of x only, T=T(x), and use a quasi-one-dimensional approach. To do this, consider an element, dx, of the fin as shown in figure 3.2. There is heat flow of magnitude Q_{in} at the left-hand side and heat flow out of magnitude $Q_{out} = Q_{in} + \frac{dQ}{dx}dx$ at the right hand side. There is also heat transfer around the perimeter on the top, bottom, and sides of the fin. From a quasi-one-dimensional point of view, this is a situation similar to that with internal heat sources, but here, for a cooling fin, in each elemental slice of thickness dx there is essentially a heat sink of magnitude Pdxh($T-T_\infty$), where Pdx is the area for heat transfer to the fluid.

The heat balance for the element in figure 3.2 can be written in terms of the heat flux using Q=qA, for a fin of constant area

$$qA=Ph(T-T_\infty)dx + \left(qA+\frac{dq}{dx}dxA\right) \qquad (3.1)$$

From the above equation we obtain

$$A\frac{dq}{dx}+Ph(T-T_\infty)=0$$

In terms of the temperature distribution, T(x)

$$\frac{d^2T}{dx^2}-\frac{Ph}{Ak}(T-T_\infty)=0$$

The quantity of interest is the temperature difference $(T-T_\infty)$, and we can change variables to put in the equation in terms of this quantity using the substitution

$$\frac{d}{dx}(T-T_\infty)=\frac{dT}{dx}$$

The above equation can therefore be written a

$$\frac{d^2}{dx^2}(T-T_\infty)-\frac{Ph}{Ak}(T-T_\infty)=0 \qquad (3.2)$$

The above equation describes the temperature variation along the fin. It is a second order equation and needs two boundary conditions. The first of these is that the temperature at the end of the fin that joins the wall is equal to the wall temperature.

$$(T-T_\infty)_{x=0}=(T_0-T_\infty)$$

The second boundary condition is at the other end of the fin. We will assume that the heat transfer from this end is negligible[18,1]. The boundary condition at x=L is

$$\left.\frac{d}{dx}(T-T_\infty)\right|_{x=L}=0.$$

The last step is to work in terms of non-dimensional variables to obtain a more compact description. In this we define $\frac{(T-T_\infty)}{(T_0-T_\infty)}$ as $\Delta\tilde{T}$, where the values of $\Delta\tilde{T}$ range from zero to one. We also define $\xi = x/L$, where ξ also ranges over zero to one. The relation between derivatives that is needed to cast the equation in terms of ξ is

$$\frac{d}{dx}=\frac{d\xi}{dx}\frac{d}{d\xi}=\frac{1}{L}\frac{d}{d\xi}.$$

The above equation can be written in this dimensionless form as

$$\frac{d^2\Delta\tilde{T}}{d\xi^2}-\left(\frac{hP}{kA}L^2\right)\Delta\tilde{T}=0.$$

There is one non-dimensional parameter in the previous equation, which we will call m and define by

$$m^2L^2=\frac{hPL^2}{kA}.$$

The equation for the temperature distribution we have obtained is

$$\frac{d^2\Delta\tilde{T}}{d\xi^2}-m^2L^2\Delta\tilde{T}=0.$$

This second order equation has the solution

$$\Delta\tilde{T}=ae^{mL\xi}+be^{-mL\xi}.$$

(Try it and see). The boundary condition at $\xi=0$ is

$$\Delta\tilde{T}(0)=a+b=1.$$

The boundary condition at $\xi=1$ is that the temperature gradient is zero or

$$\frac{d\Delta\tilde{T}}{d\xi}(L)=mLae^{mL}-mLbe^{-mL}=0.$$

Solving the two equations given by the boundary conditions for a and b gives an expression for ΔT in terms of the hyperbolic cosine or cosh

$$\cosh x = \frac{(e^x + e^{-x})}{2},$$

$$\Delta\tilde{T} = \frac{\cosh mL(1-\xi)}{\cosh mL}.$$

This is the solution to equation 3.1 for a fin with no heat transfer at the tip. In terms of the actual heat transfer parameters it is written as

$$\frac{T - T_\infty}{T_0 - T_\infty} = \frac{\cosh\left(\left(1 - \frac{x}{L}\right)\sqrt{\frac{hP}{kA}}L\right)}{\cosh\left(\sqrt{\frac{hP}{kA}}L\right)}.$$

The amount of heat removed from the wall due to the fin, which is the quantity of interest, can be found by differentiating the temperature and evaluating the derivative at the wall, x=0;

$$\dot{Q} = -kA\frac{d}{dx}(T - T_\infty)\bigg|_{x=0}$$

Or

$$\frac{\dot{Q}L}{kA(T_0 - T_\infty)} = -\frac{d\Delta\tilde{T}}{d\xi}\bigg|_{\xi=0} = \frac{mL\sinh(mL)}{\cosh(mL)} = mL\tanh(mL),$$

$$\frac{\dot{Q}}{\sqrt{kAhP}(T_0 - T_\infty)} = \tanh(mL).$$

The solution is plotted in figure 3.3, which is taken from the book by Lienhard. Several features of the solution should be noted. First, one does not need fins which have a length such that m is much greater than 3. Second, the assumption about no heat transfer at the end begins to be inappropriate as m gets smaller than 3, so for very short fins the simple expression above would not be a good estimate. We will see below how large the error is

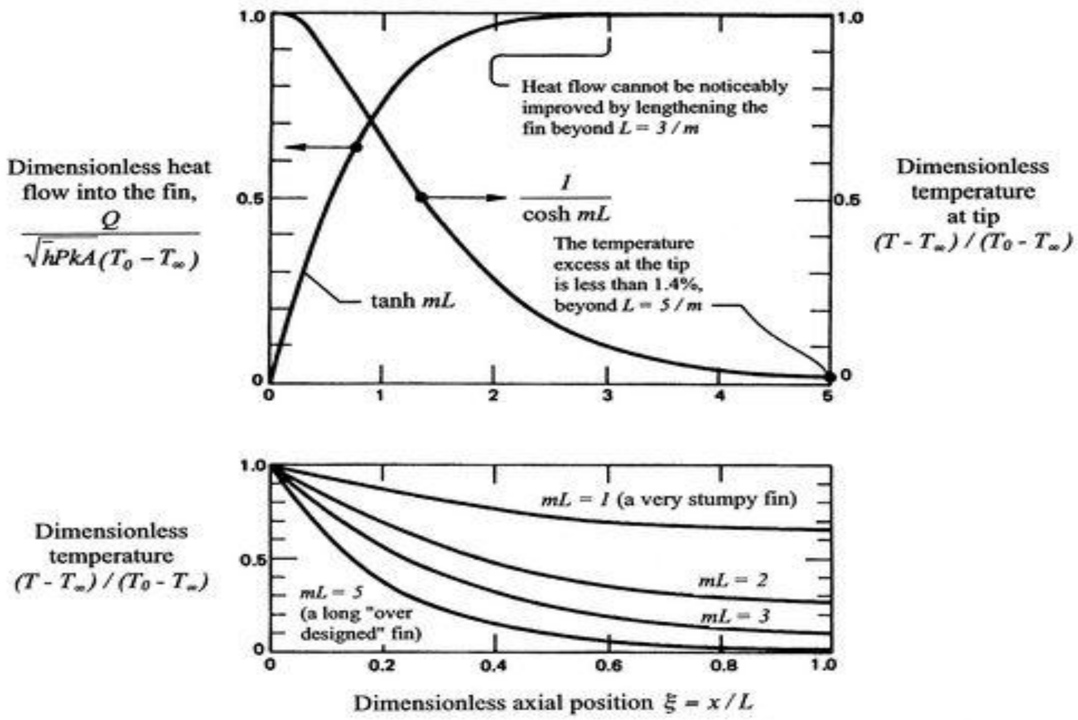


Figure 3.3: The temperature distribution, tip temperature, and heat flux in a straight one-dimensional fin with the tip insulated. [From: Lienhard, A Heat Transfer Textbook, Prentice-Hall publishers]

IV. CONCLUSION

As the fins are very important part hence its study is very important for improved design and also improving the heat dissipation rate performance of the plate by using different fin geometry and fin array also by other parameters such as fin height, fin spacing, This concept is followed by number of researches for their application. But still lot many work remains to be carried out in the future. This paper provides the background of fin to carried out further research work in future.

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