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New Approach to the Heat Transfer through Pin Fin Heat Sink in Bypass Flow and Fin Optimization

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Abstract - *In this thesis a new formula is developed for Pin Fin Heat Sinks in Bypass Flow. In this work the flow for the heat sinks (fins) in the solar panels in bypass flow is treated as Low Reynolds Number. Flow, having very low velocity and pertaining to Isenthalpic flow. Hence the governing equations are obtained by combining Low Reynolds Number. Flow Equations with Isenthalpic Joule Thompson's Equation and Maxwell's and Tds relations and applying thermodynamic relations the pressure change for the Isenthalpic Flow are obtained and the substitutions are made in previous derivations for fins in bypass flow. The governing equations could now be solved by numerical methods and the exact and discretized data could be compared. An entropy generation minimization method is applied to study the thermodynamic losses caused by heat transfer and pressure drop for the fluid in a cylindrical pin-fin heat sink and bypass flow regions. A general expression for the entropy generation rate is obtained by considering control volumes around the heat sink and bypass regions. The conservation equations for mass and energy with the entropy balance are applied in both regions. Inside the heat sink, analytical/empirical correlations are used for heat transfer coefficients and friction factors, where the reference velocity used in the Reynolds number and the pressure drop is based on the minimum free area available for the fluid flow.*

In bypass regions theoretical models, based on laws of conservation of mass, momentum, and energy, are used to predict flow velocity and pressure drop. Both in-line and staggered arrangements are studied and their relative performance is compared to the same thermal and hydraulic conditions. A parametric study is also performed to show the effects of bypass on the overall performance of heat sinks.

In this dissertation, a graph is given of the dimensions in the bypass flow direction with respect to temperatures, to combat the bypass pressure drops and hence heat transfers so to design optimum fin dimensions.

Key Words: *Pin fin, Thermal conductivity, Heat transfer rate, Heat transfer coefficient, Fin efficiency, Temperature distribution, natural convection.*

1. INTRODUCTION

1.1 Pin Fin (As a Heat Sink)

Heat sinks are essential parts of most electronic assemblies, power electronic devices, and optoelectronic components. These passive heat exchangers dissipate heat generated by electronic devices to ensure that they are operating within the limits specified by manufacturers. Some of the key factors that should be considered when designing a heat sink include thermal resistance, material, fin configuration, fin size and shape, fin efficiency, heat sink attachment method, and thermal interface material. Geometries and parameters that always provide maximum heat dissipation are obtained by analyzing different heat sink models.

1.2 The Bypass flow

The presence of Bypass Flow decreases the fin effectiveness and fin efficiency. To compensate for the decreased efficiency, the spacing between fins should be continuously increased in the bypass direction, so that there would be less resistance in the flow, so there should be increments in the fin spacing in the bypass or perpendicular direction.

1.3 Heat Transfer from a Fin (Basic theory)

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.

A model configuration is shown in Figure 1.9. The fin is of length L . The other parameters of the problem are indicated. The fluid has velocity C_∞ 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 . These all parameters and notations are shown in the figure 1.1 given below: In the figure, inlet heat conducted into the differential element "dx" and outlet heat conducted out from that differential element are shown. Heat convected is shown in the vertical direction, the heat is actually convected from the air into the control volume and convected from the control volume to the air.

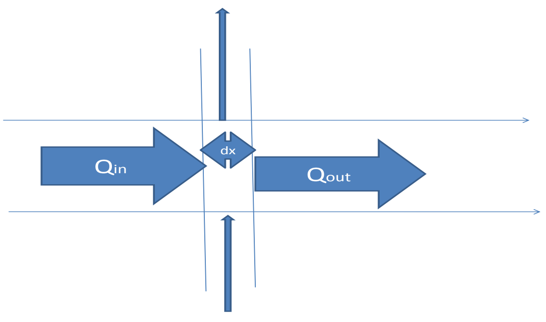


Fig 1.1: Heat Transfer by Fin

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 characteristic dimension in the transverse direction is A/P (For a circular fin, for example, A/P = r/2). The regime of interest will be taken to be

Bi = $\frac{A/P}{k}$<<1, that for which the Biot number is much less than unity, , which is a realistic approximation in practice.

1.4 A Pin Fin Heat Sink (Types)

A pin fin heat sink is a heat sink that has pins that extend from its base. The pins can be cylindrical, elliptical or square. A pin is one of the more common heat sink types available on the market. A second type of heat sink fin arrangement is the straight fin. These run the entire length of the heat sink. A variation on the straight fin heat sink is a cross cut heat sink. A straight fin heat sink is cut at regular intervals. A Pin Fin heat sink is a normal heat sink, but it differs from other heat sinks as it consists of pins that are extended from its base. These pins are in various shapes including elliptical, cylindrical and square shapes. It is the most common heat sink available in the market nowadays. Generally, a heat sink should work better when it has a large surface area, but this is not true in all the cases. The pin fin heat sink works on the concept of packing as much surface area as possible into any particular volume.

One can also find straight fin heat sinks, but Pin Fin heat sinks work better than them as the fluid flows along the pins axially instead of flowing tangentially. The main advantage with Pin fin heat sinks is that they perform in a better manner when they are placed in a tilted position. When compared to other heat sinks, they show a better performance as well. They are perfect for spot lights that are adjustable and they can also be used for shop lighting and down lighting.

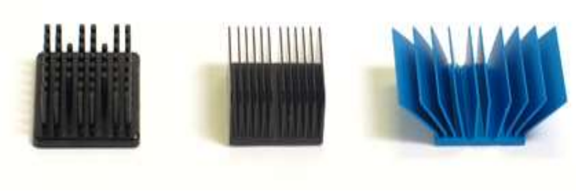


Fig 1.2 Heat sink types: Pin, Straight and Flared Fin

In general, the more surface area a heat sink has, the better it works. However, this is not always true. The concept of a pin fin heat sink is to try to pack as much surface area into a given volume as possible. As well, it works well in any orientation. Kordyban has compared the performance of a pin fin and a straight fin heat sink of similar dimensions. Although the pin fin has 194 cm^2 surface areas while the straight fin has 58 cm^2, the temperature difference between the heat sink base and the ambient air for the pin fin is 50 °C. For the straight fin it was 44 °C or 6 °C better than the pin fin. Pin fin heat sink performance is significantly better than straight fins when used in their intended application where the fluid flows axially along the pins rather than only tangentially across the pins.

Table: 1.1 Comparison of a pin fin and straight fin heat sink of similar dimensions.

Comparison of a pin fin and straight fin heat sink of similar dimensions.

Heat sink fin type	Width [cm]	Length [cm]	Height [cm]	Surface area [cm^2]	Volume [cm^3]	Temperature difference, $T_{case}-T_{air}$ [°C]
Straight	2.5	2.5	3.2	58	20	44
Pin	3.8	3.8	1.7	194	24	51

2. FORMULA DEVELOPMENT FOR FIN IN BYPASS FLOW

2.1 Formula Development

In the heat sink region, the entropy generation associated with heat transfer and frictional effects serves as a direct measure of the ability to transfer heat to the surrounding cooling medium. In the bypass regions, entropy generation is associated with the fluid flow only. A model that establishes a relationship between the entropy generation rate and heat sink design parameters can be optimized in such a manner that all relevant design conditions combine to produce the best possible heat sink for the given constraints. The total entropy generation rate can be written as given by Khan [48]

$$S_{gen} = S_{gen,hs} + S_{gen,bp} \tag{3.1}$$

where the entropy generation rate in the heat sink can be obtained by following Khan [28] and Khan et al. and applying the laws of conservation of mass and energy with the entropy balance and can be written as

$$S_{gen,hs} = \left[\frac{Q^2}{T_a T_{bp}}\right] Rh_s + m\frac{\Delta P_{hs}}{\rho T_a} \tag{3.2}$$

Similarly, the entropy generation rate in the bypass region due to the fluid flow can be written as

$$S_{gen,bp} = 2\Delta P_1 + \Delta P_2$$

$$S_{gen,bp} = m \tag{3.3}$$

where Pbp is the total pressure drop in the bypass regions and can be written as

$$P_{bp} = 2\Delta P_1 + \Delta P_2 \tag{3.4}$$

With

$$\Delta P_1 = \frac{1}{2}\rho U^2 f_1 \frac{L}{D_{h_2}} \tag{3.5}$$

$$\Delta P_2 = \frac{1}{2}\rho {U_2}^2 f_2 \frac{L}{D_{h_2}} \tag{3.6}$$

$$f_1 = \frac{24}{R_e D_{h_2}} \tag{3.7}$$

$$f_2 = \frac{24}{R_e D_{h_2}} \tag{3.8}$$

Now we know that the velocity in bypass flow is very less, so the flow approximated to be Throttling or Isenthalpic Flow

Hence applying Joule –Thompson relation for the isenthalpic flow in the

$$\left[\frac{\Delta P}{\Delta T}\right] h = Cp\left[T\left(\frac{\Delta V}{\Delta T}\right) p - v\right]^{-1} \tag{3.9}$$

This $(\Delta P)_h$" is nothing but ΔP_2 in the Eq. 3.4

Now substituting $(\Delta P)_2$ from Eq (3.9) above and substituting in Eq (1.4), for Δp_2 we have

$$P_{bp} = 2\,\Delta P_1 + \Delta P_2$$

$$P_{bp} = 2\,\Delta P_1 + (\Delta T)_h\, Cp\,[T\tfrac{\Delta v}{\Delta T})_P - v]^{-1} \tag{3.10}$$

(Δp_2 is nothing but P_{bp})

New Formula to Design the Fins in the bypass flow is thus developed as follows:

The second term in the equation (3.10) can be solved as:

$$\delta p = (\delta T)\, Cp[Tv\beta - v]^{-1}$$

$$= \frac{(\delta T)\, Cp}{[Tv\beta - v]}$$

$$= \frac{Cp\delta T}{v[\beta T - 1]}$$

$$\Rightarrow \int \delta p = \int \frac{\rho Cp\delta T}{[\beta T - 1]}$$

$$\Rightarrow \Delta P = \frac{1}{\beta}\rho C_p l_n(\beta T - 1) \tag{3.11}$$

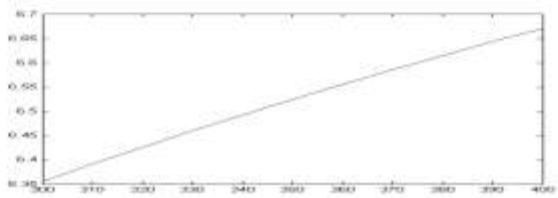
4. Results

To show that in the expression of pressure drop in the bypass flow direction, the air density may be taken constant Now we will prove that the above relation doesn't depend upon and hence can be taken as constant .Now to prove this, we will plot various curves of 'pressure drops' Vs 'temperature' for different (volume expansivity) and we will see that pressure drops are approximately same and the curves are very same, so we can say that the pressure drops in the above equation are with constant density.

These curves with respect to different temperatures are given below :

Temperature T V_s Δp Plots for β = 1.0

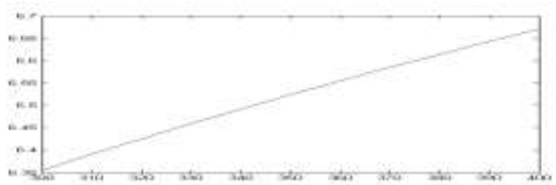
↑ Δp (pascal) (β = 1.0)



TEMPERATURE (centigrade) →

Temperature T V_s Δp Plots for β = 1.5

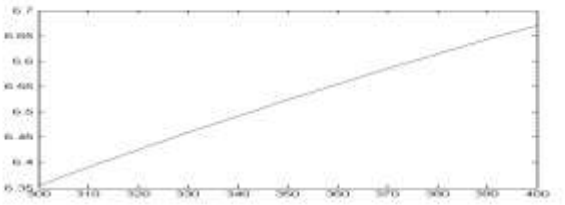
↑ Δp (pascal) (β = 1.5)



TEMPERATURE (centigrade) →

Temperature T V_s Δp Plots for β = 2.0

↑ Δp (pascal) (β = 2.0)



TEMPERATURE (centigrade) →

Now, the heat transfer will also be decreased as per the pressure drops in the bypass Flow direction, i.e., perpendicular to the main flow, hence the lengths or width of the fins spacing should be gradually increased in the bypass flow direction, so that the heat transfers should also be increased in perpendicular (bypass flow) direction. The Increased Fin Spacings "Δl" will be in the same proportion as that of pressure drops in the bypass flow direction. The Chart of "Δl" Vs "T" is prepared on the next pages to obtain the optimum fin design. Also the experimental graph will also be provided which will be in accordance with the theoretical graphs obtained by Eq. 3.11, in the previous pages.

↑ Δl (mm)

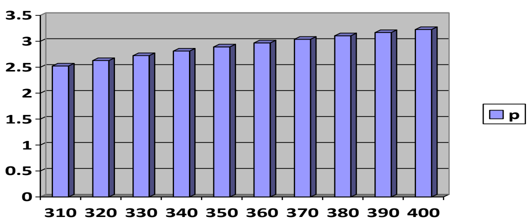


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Fig: "Δl" Vs "T" PLOT, (Bar Chart)

↑ **Δl (mm)**

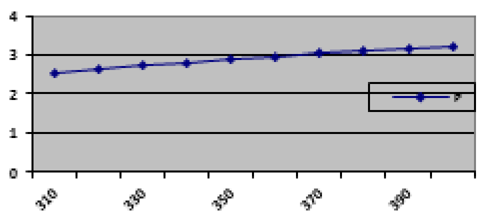

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Fig : The Length/Width increment
(Δl Vs Temperature curve)

3. CONCLUSIONS

1. First, the derivation is observed between pressure drops in the bypass region versus temperature.

2. It was also proved and shown that the air density can be taken constant for above relation.

3. The graph between pressure drops versus temperature is also plotted, which agrees the formula derived and the curves got by software's by the above relational formula.

4. The work relates the reduced pressure drops in bypass flow with the reduced heat transfers.

5. Now the fins could be designed accordingly, taking increments in the dimensions, to achieve larger surface areas to combat the reduced heat transfers.

6. This optimization will help to save the electronic devices from getting overheated and damaged.

7. Heat sinks are essential parts of most electronic assemblies, power electronic devices, and optoelectronic components. These passive heat exchangers dissipate heat generated by electronic devices to ensure that they are operating within the limits specified by manufacturers.

8. Some of the key factors that should be considered when designing a heat sink include thermal resistance, material, fin configuration, fin size and shape, fin efficiency, heat sink attachment method, and thermal interface material.

9. Geometries and parameters that provide maximum heat dissipation are obtained by analyzing different heat sink models.

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