Thermal Analysis of Fin: Comparative Thermal Analysis of Rectangular Fin and Rectangular Porous Fin

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ABSTRACT

The fins are extended surfaces that are made to increase the rate of heat transfer through the equipment/components. Generally, they used for faster cooling of the equipment/components. The comparative thermal analysis of rectangular fin and porous fin is done in the paper. Both the fins are subjected to same conditions i.e., both the fins have same room temperature, have same dimensions, and are exposed to same surrounding conditions i.e., the flow of air over the fins, the ambient temperature of the fins, etc. Of course, for calculations some assumptions are made. The heat transfer rate from both the fins are calculated and compared. The efficiency of fins and effectiveness is calculated and compared. The temperature distribution graph for rectangular fin is plotted in excels. The modeling software Solidworks is used to model rectangular non-porous fin and rectangular porous fin. The thermal analysis of both the fins is done in the software called Simscale.

Keywords: Fins, Heat transfer, Efficiency, Effectiveness, Solidworks, Simscale.

NOMENCLATURE

L	total length of fin
L_1	length of fin without holes
L_2	length of fin with holes
b	width of fin
t	thickness of fin
D	diameter of hole
d	distance between holes
n	no. of holes in x direction
m	no. of holes in y direction
P	perimeter across cross section area
A_c	cross section area of fin
h	heat transfer coefficient
K	thermal conductivity of Aluminum
T_L	temperature at root of fin
T_{amb}	ambient temperature
T_o	intermediate temperature

1. LITERATURE REVIEW

Mohammad Hamdan, Moh'd A. Al-Nimr [4], Presenting the numerical solution for steady, developing forced convection between two isothermal parallel-plate channel with porous fins. This study reports that heat transfer can be increased with the help of higher thermal conductivity porous fins, smaller Darcy numbers, and longer microscopic inertial coefficients.

Kota LeelaSaiBharath, V Pradeep Kumar,[5] conducted experiment over a different geometric shape and size. Thermal analysis is done for all cases. By observing the thermal analysis results, the overall heat flux is generated more in rectangular perforation fin. Hence, it is expected to be dissipating more heat from the surface and Temperature at the end of fin with rectangular perforation is minimum as compare to fin with other cases.

Kannan C, Sathyabalan P, Ramanathan S, [6] in this study the rectangular fins with different perforation are analyzed using numerical methods. The drop in temperature along solid fin is higher compared to a perforated fin. The heat dissipation rate can be increased with perforation on the fin surface. The perforation dimension and lateral spacing and number of perforations play a vital role in increasing heat dissipation rate.

GolnooshMostafavi, [7] Experimental, numerical and analytical studies were performed in order to establish optimized geometrical fin parameters for natural convection heat transfer from vertically installed interrupted rectangular fin arrays.The prepared samples were tested in the lab and collected data were compared with the numerical and analytical models developed in this study. The numerical and analytical results were successfully verified by experimental data; the mean relative difference found was 4.6% and the maximum relative difference was 14%.

D Merwin Rajesh,K Suresh kumar, [12]simulating the various engine cylinder models considering various parameters like material, geometry, height of fins and the number of fins at certain velocities. Heat transfer through the stepped fins is more than their Heat corresponding rectangular fin models. Fins with the length of 16mm showed better performance than their 13mm counterparts regardless of the other parameters. Aluminum Alloy 6061 is better since heat transfer rate, efficiency and effectiveness of the fin is more.

2. PROBLEM STATEMENT

Considered 2 fins one with half of length of fin has holes and other without holes, calculating the Rate of heat transfer from the fins, effectiveness and efficiency of both fins.

3. MATERIALS AND METHODS

The material used for the fin is Aluminum and the thermal conductivity of aluminum K is 225 W/m^2K

Assumptions:

1. Steady-State
2. 1-D x-direction conduction.
3. No internal heat generation.
4. Temperature is function of space in x-direction only.
5. Uniform cross section of fin.
6. Heat transfer via radiation is negligible.

7. Thermal conductivity is not function of temperature.
8. L_2 region will have fins with holes.
9. Heat transfer coefficient is constant for whole surface.
10. Fin is non-insulated with finite length.

Table 1. Table for dimensions of fin

Sr. No.	Dimensions	Value	Unit
1	L	50	mm
2	L1	25	mm
3	L2	25	mm
4	b	150	mm
5	t	2.5	mm
6	D	2	mm
7	d	2	mm
8	n	4	
9	m	25	
10	P	305	mm
11	A_c	375	mm^2

Table 2. Table for thermo-physical quantities

Sr. No.	Thermo-physical quantities	Value	Unit
1	h	50	W/m^2K
2	K	225	W/m^0C
3	T_L	1750	^0C
4	T_{amb}	27	^0C
5	T_o	1495.8736	^0C
6	M	13.444	1/m

Case-I: Fin without holes with same dimensions and same material.

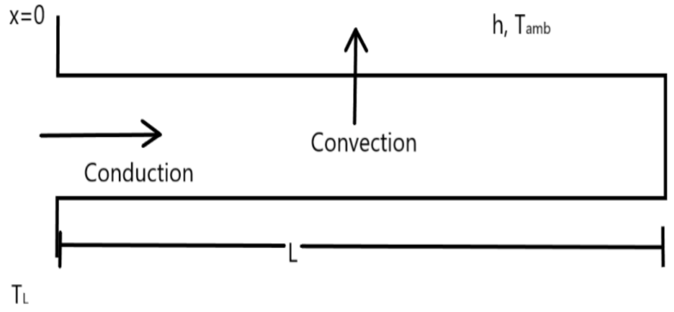


Fig.1: Schematic for energy balance of fin without holes

Energy Balance Equation:

$Q_{cond} = Q_{conv}$

$Q_{conv} = \sqrt{hPKA_c}(T_L - T_{amb}) \left[\frac{h/mk + \tanh(mL)}{1+\frac{h}{mk}\tanh(mL)}\right]$

Q_{conv1} = 1167.09566 W

Temperature distribution:

$$\frac{T - T_{amb}}{T_L - T_{amb}} = \frac{\cosh[m(L_c - x)]}{\cos h(mL)}$$

Below is the data for plotting graph.

Table 3. Table for temperature distribution

length (mm)	Cosh [m(Lc -x)]	cosh(mLc)	TL - Tamb (^{0}C)	Tamb (^{0}C)	T (^{0}C)
1	1.237	1.247	1723	27	1736.318
2	1.227	1.247	1723	27	1722.946
3	1.218	1.247	1723	27	1709.880
4	1.209	1.247	1723	27	1697.118
5	1.200	1.247	1723	27	1684.658
6	1.191	1.247	1723	27	1672.497
7	1.182	1.247	1723	27	1660.634
8	1.174	1.247	1723	27	1649.067
9	1.166	1.247	1723	27	1637.792
10	1.158	1.247	1723	27	1626.809
11	1.150	1.247	1723	27	1616.114
12	1.142	1.247	1723	27	1605.707
13	1.135	1.247	1723	27	1595.586
14	1.128	1.247	1723	27	1585.747
15	1.121	1.247	1723	27	1576.191
16	1.114	1.247	1723	27	1566.915
17	1.108	1.247	1723	27	1557.916
18	1.102	1.247	1723	27	1549.195
19	1.095	1.247	1723	27	1540.749
20	1.090	1.247	1723	27	1532.576
21	1.084	1.247	1723	27	1524.675
22	1.078	1.247	1723	27	1517.046
23	1.073	1.247	1723	27	1509.685
24	1.068	1.247	1723	27	1502.592
25	1.063	1.247	1723	27	1495.767

26	1.058	1.247	1723	27	1489.206
27	1.054	1.247	1723	27	1482.910
28	1.049	1.247	1723	27	1476.877
29	1.045	1.247	1723	27	1471.106
30	1.041	1.247	1723	27	1465.596
31	1.037	1.247	1723	27	1460.346
32	1.034	1.247	1723	27	1455.356
33	1.030	1.247	1723	27	1450.623
34	1.027	1.247	1723	27	1446.148
35	1.024	1.247	1723	27	1441.929
36	1.021	1.247	1723	27	1437.966
37	1.018	1.247	1723	27	1434.258
38	1.016	1.247	1723	27	1430.804
39	1.014	1.247	1723	27	1427.604
40	1.011	1.247	1723	27	1424.657
41	1.010	1.247	1723	27	1421.963
42	1.008	1.247	1723	27	1419.521
43	1.006	1.247	1723	27	1417.331
44	1.005	1.247	1723	27	1415.391
45	1.004	1.247	1723	27	1413.703
46	1.002	1.247	1723	27	1412.266
47	1.002	1.247	1723	27	1411.079
48	1.001	1.247	1723	27	1410.142
49	1.000	1.247	1723	27	1409.455
50	1.000	1.247	1723	27	1409.018

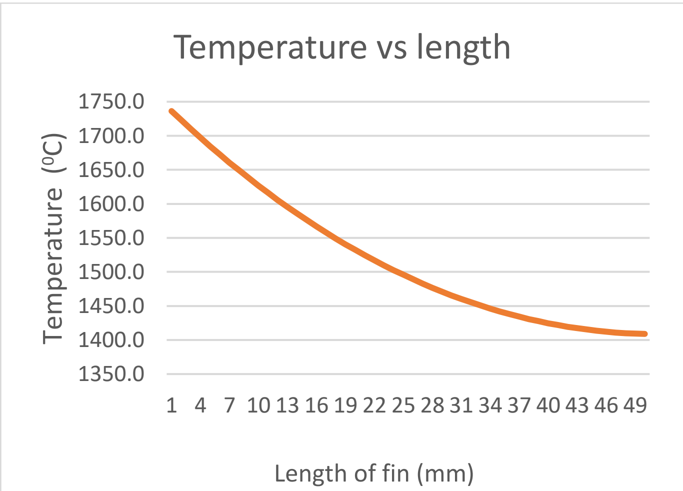


Fig.2: Temperature distribution of fin with length

Effectiveness of fin:

$$\varepsilon = \frac{Q_{conv}}{h\,(b*t)(T_L - T_{amb})}$$

$\varepsilon = 1167.09566/32.30625$
$\varepsilon = 36.2600$

Efficiency of fin:

$$\eta = \frac{Q_{fin}}{Q_{max}}$$

$\eta = 1167.09566/(1346.0937)$
$\eta = 0.86702$
$\eta = 86.702\%$

Case-II: Fin with holes (dimensions of fin remains same)

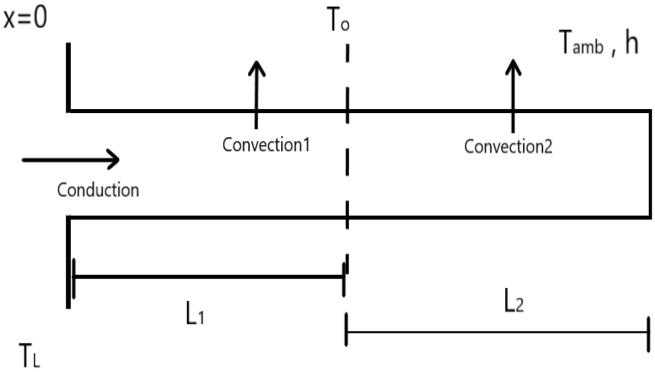


Fig.3: Schematic for energy balance of fin with holes

Energy Balance Equation:

$$Q_{cond} = Q_{conv1} + Q_{conv2}$$

Convection via fin without holes

$$Q_{conv1} = \sqrt{hPKA_c}(T_L - T_{amb}) \left[\frac{h/mk + \tanh(mL1)}{1+\frac{h}{mk}\tanh(mL_1)}\right]$$

Q_{conv1} = 661.970 W

Intermediate temperature:

$$\frac{T_o - T_{amb}}{T_L - T_{amb}} = \frac{\cos h(mL_1) + \frac{h}{km}[\sinh(mL_1)]}{\cos h(mL) + \frac{h}{km}[\sinh(mL)]}$$

$T_o = 1495.8736\ ^0C$

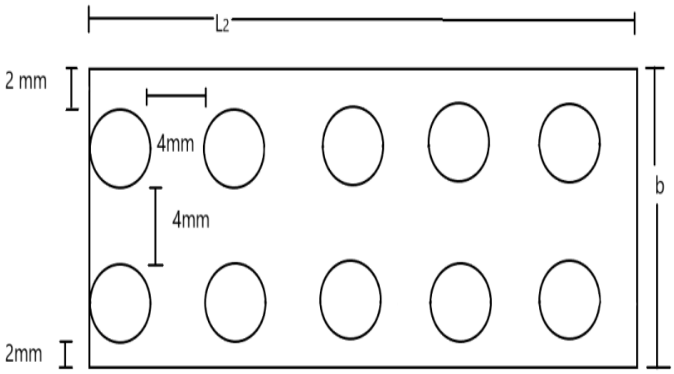


Fig.4: Rough sketch of fin with holes

The next region is with pores/holes so formula cannot be directly applied for heat transfer, as formula for heat transfer from fin with holes is unknown.

Calculations for new area (surface area with holes):
Area without hole = P*L2 = 2(b+t) * L2
Area with hole = 2(b+t) * L2 + nmπD(t-D/2)
As the fins have more surface area for Heat Transfer, Area with hole > Area without hole
Therefore,
t-D/2 > 0
t > D/2
Therefore,
t = 2.5 mm
D = 2 mm
The condition holds the equation.

Calculations of number of holes:
m: no. of holes along width
n: no. of holes along x-direction
D: Diameter of hole
d: distance between consecutive holes
nD + (n-1)2d + d = 25
m*2d + nD = 150
Considering the D = d = 2mm
n = 4.5
therefore, taking n = 4
m = 25.

Area extra = Area with hole + Area without hole
Area extra =2(b+t) * L2 + nmπD(t-D/2) - P*L2 = 2(b+t) * L2
Area extra = nmπD(t-D/2)
i.e total number of holes * curved surface of a hole – 2 *circular area.

As mentioned earlier the formula cannot be applied directly, therefore making some considerations Whatever, area that will be increased assuming it to be on fin surface as shown in below figure.

Fig.5: Rough figure for extra area that will be generated.

Therefore, equating Area extra to linear surface area.
nmπD(t-D/4) = 2(b + t) *L new
L new = 3.09 mm
So, it can be assumed that due extra surface area the length of 3.09 mm (imaginary) is added to fin which can be considered without holes.
But considering that curved surface area would provide 50% effective cooling.
Therefore, heat transfer through L new linear area will be 0.5 times of calculated.

$Q_{conv2L} = \sqrt{hPKA_c}(T_o - T_{amb}) \left[\frac{h/mk + \tanh(mL_2)}{1+\frac{h}{mk}\tanh(mL_2)}\right]$

Q_{conv2L} = 564.336 W

$Q_{conv2Lnew} = 0.5 * \sqrt{hPKA_c}(T_o - T_{amb}) \left[\frac{h/mk + \tanh(mL_2)}{1+\frac{h}{mk}\tanh(mL_2)}\right]$

$Q_{conv2Lnew}$ = 0.5 * 97.24387 W

$Q_{conv2Lnew}$ = 48.622 W

Effectiveness of fin:

$$\varepsilon = \frac{Q_{conv1} + Q_{conv2L} + Q_{conv2Lnew}}{h\,(b * t)(T_L - T_{amb})}$$

$\varepsilon = 1274.928/32.30625$

$\varepsilon = 39.4638$

Efficiency of fin:

$\eta = \frac{Q_{fin}}{Q_{max}}$

$\eta = 1274.928/(1427.2858)$

$\eta = 0.89325$

$\eta = 89.325\%$

2.1 Design of Fin

Design is done in Solidworks.

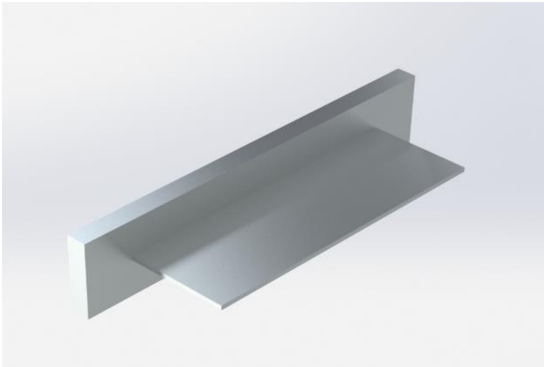


Fig.6: Design for Non-Porous Fins in Solidworks

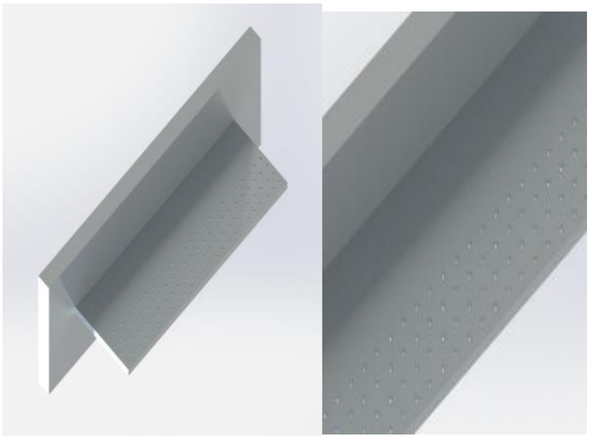


Fig.7: Design for Porous Fins in Solidworks

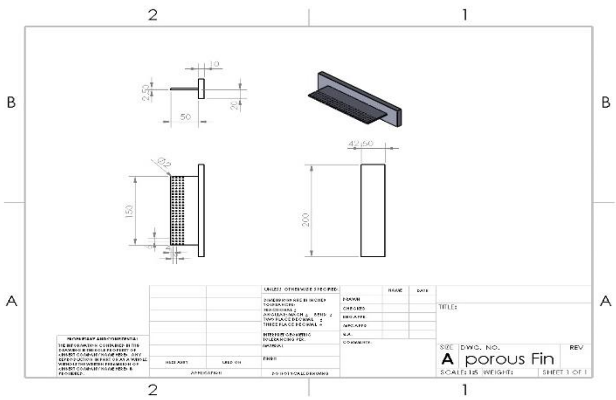


Fig.8: Drawing of Porous Fins

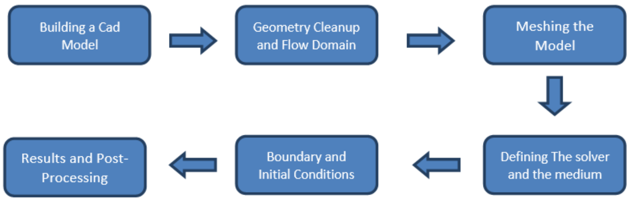
2.2Simulation Analysis

Steady State Thermal Analysis:

Computational Fluid Dynamics (CFD) is the process of mathematically modeling a physical phenomenon involving fluid flow and solving it numerically using the computational prowess.

In this project we have used the **Conjugate Heat Transfer (CHT)** analysis type. This allows for the simulation of heat transfer between **solid** and **fluid** domains by exchanging thermal energy at the interfaces between them. Typical applications of this analysis type exist as, but are not limited to, the simulation of heat exchangers, cooling of electronic equipment, and general-purpose cooling and heating systems.

Flowchart of Analysis:



The simulation analysis is done on software called Simscale, it is cloud simulation software. The results are as below.

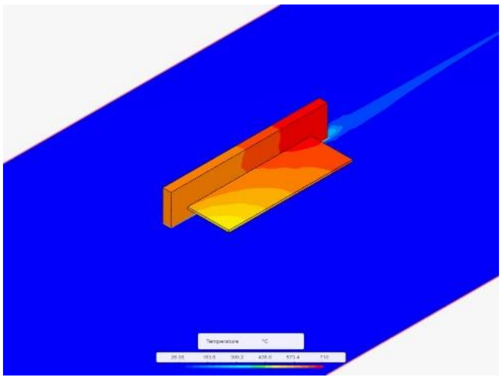


Fig.9: Simulation for Non-Porous Fins in Simscale

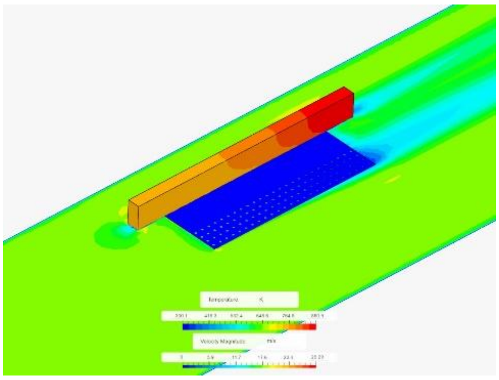


Fig.10: Simulation for Porous Fins in Simscale

The simulation of both the fins showed results that are other matter that the simulation of non-porous fin had some error during run-time.

The result of porous fin has shown temperature gradient in y as well as x direction. Whereas, the result for non-porous fin just have shown the result of fin surface in y-direction

4. RESULTS AND DISCUSSION

The temperature variation of the non-porous fin with its length has been plotted, and the effectiveness of both the fins i.e., the porous and non-porous has been calculated. The values of effectiveness are 36.26 for non-porous fins and 39.46 for porous fins. The efficiency of porous and non-porous has been calculated. The values of efficiency are 86.702% for non-porous fins and 89.325% for porous fins.

5. CONCLUSION

The rate of heat transfer from the fin with holes is more than the heat transfer from the fin without fin. The effectiveness and efficiency from the fin with holes are greater than the fin without holes. The fin with holes is good choice for better cooling purpose.

The result of porous fin has shown temperature gradient in y as well as x-direction. Whereas, the result for non-porous fin just have shown the result of fin surface in y-direction.

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