CFD ANALYSIS IN A HEAT SINK FOR COOLING OF ELECTRONIC DEVICES

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Abstract

The reliability of electronic components is affected critically by the temperature at which the junction operates. As operating powers and speed increase, and as designers are forced to reduce overall system dimensions, the problems of extracting heat and controlling temperature can become crucial. The continuing increase of power densities in electronics packages and the simultaneous drive to reduce the size and weight of electronic products have led to an increased importance on thermal management issues in this industry. Plate fin heat sinks are commonly used devices for enhancing heat transfer in electronics components.. The choice of an optimal heat sink depends on a number of geometric parameters such as fin height, fin length, fin thickness, number of fins, base plate thickness, space between fins, fin shape or profile, material etc. Given a set of design constraints, one needs to determine the maximum possible performance of a heat sink within the constraints. Optimizing the above parameters to achieve low thermal resistance and low pressure drop is very difficult. In order to select the optimal geometric parameters of a heat sink for a particular application, a designer requires more design tools to predict heat sink performance. Therefore in this research work, to select an optimal heat sink design, preliminary studies on the fluid flow and heat transfer characteristics of a parallel plate heat sink have been carried through CFD modeling and simulations. The simulation is carried out with a commercial package provided by Fluent Inc. The geometric parameters considered in this study are fin height, fin thickness, base height and fin pitch. Experimental validation of simulation results also have been performed. In this study, the geometric parameters fin height, fin thickness, base height and fin pitch are found to be optimal at 48 mm,1.6 mm, 8 mm and 4mm respectively for an efficient heat sink design.

Keywords: Parallel plate heat sink, Thermal resistance, Pressure Drop, CFD

INTRODUCTION

Due to the rapid growth of electronic technology, electronic appliances and devices now are always in our daily life. Users prefer computers of higher performance and are therefore more willing to spend reasonably on them. This tendency has motivated the manufacturers to employ overdrive technology to improve their products. With this technology, these devices are capable of processing more data within a given period of time and the system performance is therefore regarded as higher. However, this capability is directly related to its heat generation. The larger the amount of data the system processes at a time, the greater the amount of heat it generates. The performance of these devices is directly related to the temperature; therefore it is a crucial issue to maintain the electronics at acceptable temperature levels. The rising demand for high performance and multiple functionality in electronic systems and devices continue to be the current great challenges in their thermal management. Under the condition of multifunction, high clock speed, shrinking package size, and higher power dissipations, the heat flux per unit area increased dramatically over the past few years. Besides, the working temperature of the electronic components may exceed the desired temperature level. Thus, the effective removal of heat dissipations and maintaining a safe operating temperature have played an important role in insuring a reliable operation of electronic components.

In the cooling enhancement of current electronic industry, heat sink is extensively used to provide cooling function for electronics components. Due to the circuit density and power dissipation of integrated circuit chips are increasing, the heat flux levels within these chips have increased. The accumulation of large amount of heat flux can create considerable quantities of heat stress on chips, substrate, and its package. Therefore, it is necessary for employing effective heat sink module to maintain the operating temperature of electronic components at a satisfactory level. If there is appropriate and effective heat sink design, it will critically affect the reliability and life span of chip function. Among various cooling techniques for electronic chips and/or modules, forced convective cooling with air features advantages of convenience and low cost The most common method for cooling electronic devices is by finned heat sinks made of aluminum. These heat sinks provide a large surface area for the dissipation of heat and effectively reduce the thermal resistance. . In order to design an effective heat sink, some criterions such as a large heat transfer rate, a low pressure drop, an easier manufacturing, a simpler structure, a reasonable cost and so on should be considered. Unfortunately, heat sinks often take up much space and contribute to the weight and cost of the product. Consequently, the need for new design and more effective ways to dissipate this energy is becoming increasingly urgent. Because of compelling market requirements, design optimization of new electronic devices, is often found to be prohibitive.

It is true that an undesirable phenomenon such as increasing in the pressure loss commonly takes place in the plate-fin type heat sinks in which fins are attached to the plate in order to enhance the heat transfer rate. Thus, high performance of heat sinks can be acquired through the design optimization which maximizes heat transfer and minimizes pressure drop. To achieve an optimum design of heat sink for an effective heat transfer, newer methodologies or techniques are to be identified.

While surveying the literature, C.W. Leung and S.D. Probert (1988) have experimentally investigated and reported about the effect of optimal fin thickness, fin length and fin height for a better heat dissipation from an array of rectangular fins protruding from a base. Lee (1995) concluded that the cooling performance of a heat sink depends on a number of parameters including thermal conduction resistance, dimensions, location and concentration of heat sources as well as the airflow bypass conditions. These parameters make the optimal design of a heat sink very difficult. Also he has reported that the option of having excessive fins to improve the performance is a very dangerous way. Because, in most cases, having excessive fins induce a higher pressure drop across the heat sink, resulting in a severe reduction in flow velocity and/or a significant increase in flow bypass over the heat sink.

Teertstra et al(1999) have concluded that techniques, such as one-variable at a time is useful for simple designs, however often fall short for real-world applications. Copeland (2000) concluded that the traditional characterization method for a heat sink has been the topic of much debate as vendors have applied different standards or interpretations to determine the characterization of heat sinks. Analytical and empirical formulations for the fin efficiency, pressure drop and the heat transfer coefficient have also been used in the design process to determine the optimal heat sink design. Stewart and Stiver (2004) concluded that optimal heat sink design for an electronic system is extremely time-consuming.

Culhamand J. R and Muzychka Y. S (2001) have presented a procedure that allowed the simultaneous optimization of heat sink design parameters based on a minimization of the entropy generation associated with heat transfer and fluid friction. They have reported that all relevant design parameters for plate fin heat sinks, including geometric parameters, heat dissipation, material properties and flow conditions could be simultaneously optimized to characterize a heat sink that minimizes entropy generation and in turn results in a minimum operating temperature. In addition they have integrated, a novel approach for incorporating forced convection through the specification of a fan curve into the optimization procedure, providing a link between optimized design parameters and the system operating point.

Iyengar M and Bar-Cohen A (2001) have developed a well validated analytical model, using which the thermo fluid performance of the side-inletside-exit (SISE) heat sink has been characterized, parametric optimization carried out, and the maximum heat transfer capabilities for a range of operating points has been determined. A least-material optimization has been performed to achieve optimal material use. The analysis indicates the least-material design to provide significant mass savings for a moderate penalty in thermal performance

Bar-Cohen A and Iyengar M (2002) have explored the potential for the least-energy optimization of natural- and forced-convection cooled rectangular plate heat sinks. Guidelines for “sustainable” heat sink designs have been suggested. Cohen and Iyengar (2002) have concluded that air cooling in conjunction with traditional heat-sinks will continue be the method of choice for heat dissipation due to the ease of application, reliability and low cost.

Chen et al (2004) have estimated that the future cooling in computers and electronic applications, require more efficient compact heat exchangers employing active heat transfer methods in combination with ambient air. Hsien-Chie Cheng et al (2004) presented a mathematical expression of the chip junction temperature in terms of chip spatial location and size for effective thermal characterization of ball grid array (BGA) typed multichip modules (MCMs). The mathematical expression could not only effectively define the relation of the thermal performance and design parameters but also highlight their combinatorial effect.

John Parry et al (2004) have described a sequential global optimization methodology that could lead to better designs in less time, and have illustrated its use by optimizing the design of a heat sink for a simple system. With the results they have shown the need of a global approach, the insights that could be gained through automated design optimization, and also have illustrated the efficiency of the reported methodology in finding the optimum design. Yu-Tang Chen (2004) et al have performed experimental tests and theoretical analyses to investigate the characteristics of fluid flow and heat transfer in micro channel heat sink especially in the mechanism of bubble nucleation. Their experimental results in heat transfer indicted that forced convection in micro channel heat sink exhibited excellent cooling performance, especially in the phase change regime. It could be applied as heat removal and temperature control devices in high power electronic components. Seri Lee (2004) has developed an analytical simulation model for predicting and optimizing the thermal performance of bidirectional fin heat sinks. Optimization of heat-sink designs and typical parametric behaviors have been discussed based on the sample simulation results. The author has presented a simple method of determining the fin flow velocity and also has described the development of the overall thermal model.

Park et al (2006) have numerically performed the shape optimization of the plate-fin type heat sink with an air deflector to minimize the pressure loss subjected to the desired maximum temperature and geometrical constraints. In their study, the Kriging method, which is one of the metamodels, associated with the computational fluid dynamics (CFD) have been used to obtain the optimal solutions.

In recent years, maturation of computational fluid dynamics (CFD) software codes tailored for applications in the electronics industry and the availability of powerful low cost workstations have made possible the simultaneous solution of both the heat transfer and fluid dynamic problems in undertaking thermal design of electronic devices.

Therefore, in this work, by varying the geometric sizes of the heat sink, eighty one models have been created in Gambit and simulation have been be carried out using Fluent to determine the output parameters like base temperature, pressure drop . Best combination of the geometric parameters for the heat sink will be identified. This suggested heat sink could be cable of working efficiently at low base temperature, lop pressure drop with maximum heat transfer.

MODELING AND SIMULATION

In CFD calculations, there are three main steps.

1) Pre-Processing
2) Solver Execution
3) Post-Processing

Pre-Processing is the step where the modeling goals are determined and computational grid is created. In the second step numerical models and boundary conditions are set to start up the solver. Solver runs until the convergence is reached. When solver is terminated, the results are examined which is the post processing part.

Heat Sink Model

A typical parallel plate heat sink considered in this present work is shown in Fig. 1. The important geometric variables considered are fin height , fin thickness , base height and fin pitch.

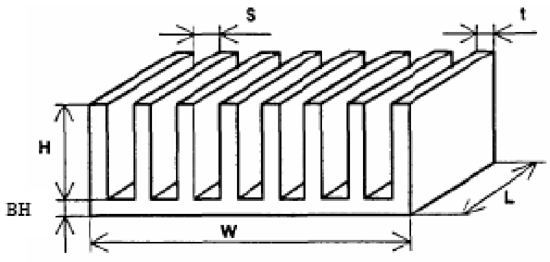


Fig.1 A typical Heat Sink

A typical computational model of the parallel plate heat sink created in Gambit is shown in Fig. 2.

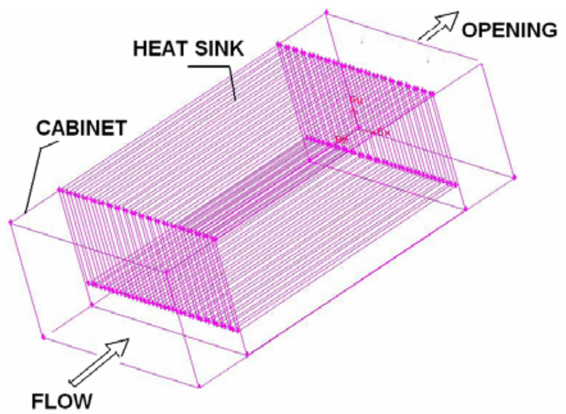


Fig.2 Computational Model

Governing Equations of Fluid Flow

The most general form of fluid flow and heat transfer equations of compressible Newtonian fluid with time dependency used in solver execution is given as follows:

Mass :

$$\frac{\partial \rho}{\partial t}+\nabla.(\rho\bar{V})=0$$

X – momentum :

$$\frac{\partial(\rho u)}{\partial t}+\nabla.(\rho u\bar{V})=-\frac{\partial p}{\partial x}+\frac{\partial \tau_{xx}}{\partial x}+\frac{\partial \tau_{yx}}{\partial y}+\frac{\partial \tau_{zx}}{\partial z}+S_{Mx}$$

Y – momentum :

$$\frac{\partial(\rho v)}{\partial t}+\nabla.(\rho v\bar{V})=-\frac{\partial p}{\partial y}+\frac{\partial \tau_{xy}}{\partial x}+\frac{\partial \tau_{yy}}{\partial y}+\frac{\partial \tau_{zy}}{\partial z}+S_{My}$$

Z – momentum :

$$\frac{\partial(\rho w)}{\partial t}+\nabla.(\rho w\bar{V})=-\frac{\partial p}{\partial z}+\frac{\partial \tau_{xz}}{\partial x}+\frac{\partial \tau_{yz}}{\partial y}+\frac{\partial \tau_{zz}}{\partial z}+S_{Mz}$$

Energy :

$$\frac{\partial(\rho h_0)}{\partial t}+\nabla.(\rho h_0\bar{V})=-p\nabla.\bar{V}+\nabla.(k\nabla T)+\Phi+S_h$$

Equation of state: $p=\rho RT$

where ρ is the density, u,v and w are velocity components, $\bar{V}$ is the velocity vector, p is the pressure, S terms are the source terms and τ terms are the viscous stress components which are defined for a Newtonian fluid as.

$$\tau_{xx}=\lambda\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+2\mu\frac{\partial u}{\partial x}$$

$$\tau_{yy}=\lambda\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+2\mu\frac{\partial v}{\partial y}$$

$$\tau_{zz}=\lambda\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+2\mu\frac{\partial w}{\partial z}$$

$$\tau_{xy}=\tau_{yx}=\mu\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)$$

$$\tau_{xz}=\tau_{zx}=\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)$$

$$\tau_{yz}=\tau_{zy}=\mu\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)$$

Here,μ is the dynamic viscosity, Φ is the viscous dissipation term and λ is the second viscosity and a good approximation for gases $\lambda=-\frac{2}{3}\mu$

It is always positive and represents the dissipation of mechanical energy into heat.

This dissipation term is usually very small except for high Mach number flows. Here,

$$\Phi = \lambda\left(\nabla.\vec{V}\right)^2 + \mu\begin{bmatrix} 2\left(\frac{\partial u}{\partial x}\right)^2 + 2\left(\frac{\partial v}{\partial y}\right)^2 + 2\left(\frac{\partial w}{\partial z}\right)^2 + \\ \left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2 \end{bmatrix}$$

h is the total enthalpy, which is defined as

$$h_0 = h_e + \frac{1}{2}\left(u^2+v^2+w^2\right) = i + p/\rho + \frac{1}{2}\left(u^2+v^2+w^2\right) = E + p/\rho$$

where h_e is the enthalpy defined as

$$h_e = \int_{T_{REF}}^{T} C_P\, dT,$$

i is the internal thermal energy, *E* is the total energy of the fluid which is the sum of internal thermal energy and kinetic energy.

CFD Simulations

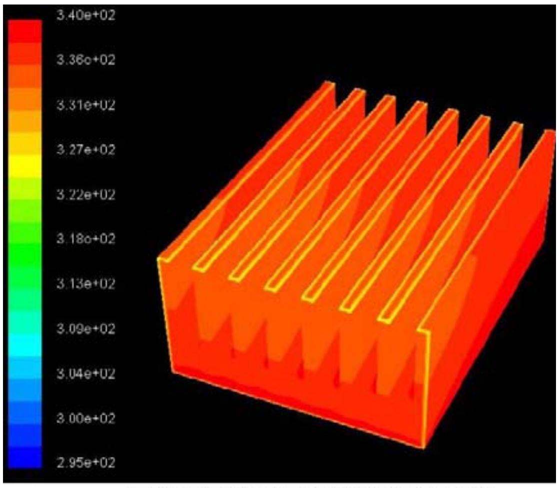
In the modeling and simulation, three levels in these input geometric parameters are chosen such that the differences are large enough to measure changes in the responses. If the levels are too close to each other, the change might not be large enough to detect and the opportunity for optimization could be lost. If the levels are too far apart, the design space of interest may not be able to be optimized due to non-linear or discontinuous response between the values. The simulations are carried out using fluent software with air flow at 12 CFM and a heat input of 100 W at the heat sink base.

The range of geometric parameters of the chosen heat sink for the present study is given in Table 1. Considering the various geometric variables present, 81 models are created in gambit and simulation are carried out using Fluent to determine the output parameters like base temperature, pressure drop.

Table 1. Range of geometric parameters

Parameter	Minimum value	Maximum value	Increment value
Fin Height (H)	16 mm	48 mm	16
FIN THICKNESS (T)	0.8 mm	1.6 mm	0.4
Base Height (BH)	4 mm	12 mm	4
FIN PITCH (S)	1.5 mm	4 mm	-

A typical temperature distribution in heat sink simulated using Fluent is shown in Fig. 3.



CONTOURS OF STATIC TEMPERATURE
Figure. 3

Heat sink model Validation

Experimental studies have been carried out with a parallel plate heat sink. These results obtained are compared with the simulation results. The experimental and simulation results are given in table 2 and are found identical in nature within acceptable limit of deviations.

Table 2.

STUDY	Parameters	Observations			
	Heat Input	25 W	50 W	75 W	100 W
	Air Flow	15 CFM	15 CFM	15 CFM	15 CFM
	Ambient temperature C	22.5	22.5	22.5	22.5
EXPERIMENTAL WORK	TEMPERATURE C	38.9	48.56	56.07	67.28
	PRESSURE DROP Pa	24	24	24	24
SIMULATION WORK	TEMPERATURE C	35.58	45.2767	54.194	67.188
	PRESSURE DROP Pa	23.78	23.78	23.78	23.78

RESULTS AND DISCUSSIONS

The results from the simulation of heat sink are shown in Table 3. For the selection of optimum geometric parameters for a low thermal resistance in the heat sink, many trade-offs have to be considered carefully. By keeping a minimum number of fins, maximum possible fin pitch can be maintained so that the pressure drop will be minimum and air flow will be maximum for an improved heat transfer in the heat sink. Similarly, to increase the heat transfer area of the heat sink, many aspects have to be considered. In the case of increasing the fin length, the flow length cannot be increased beyond the optimal length because of system level space constrains and possible pressure drops. Because of the manufacturability and flow velocity or flow bypass constrains, the designer cannot decrease the fin thickness beyond the limits to have excessive number of fins to enhance the heat transfer area.

After considering various constrains in the heat sink design, selecting a parallel plate heat sink with optimum geometric parameters such as optimum fin thickness, optimum fin height, and optimum base height and optimum gap between the fins, can improve the effective heat transfer.

From Table 3, it is evident that for a low thermal resistance and low pressure drop in the selected heat sink model, the geometric parameters namely fin height, fin thickness, base height and fin pitch are found to be optimal at 48 mm, 1.6 mm, 8 mm and 4mm respectively. These conclusions drawn in this present work are found to be in good agreement with conclusions drawn by Lee (1995), and C.W. Leung and S. D. Probert (1988).

Table 3

S.No	Fin Height	Fins thickness	Base Height	Fin Pitch	Max Temp	Thermal resistance	Pressure Drop
	mm	mm	mm	mm	k	C / W	pascal
1	16	0.8	4	4	351.90	0.2535896	44.1
2	16	0.8	8	4	348.43	0.215999	53.7
3	16	0.8	12	4	347.27	0.2025356	69.8
4	32	0.8	4	4	350.37	0.2451238	13.6
5	32	0.8	8	4	348.51	0.2221139	17.8
6	32	0.8	12	4	346.45	0.1962259	22.7
7	48	0.8	4	4	354.20	0.2829374	21.2
8	48	0.8	8	4	352.50	0.2598091	15.7
9	48	0.8	12	4	349.98	0.2307441	10.3
10	16	1.2	4	4	352.60	0.2653722	40.6
11	16	1.2	8	4	350.10	0.2346428	48.5
12	16	1.2	12	4	349.30	0.224305	54.29
13	32	1.2	4	4	351.70	0.257022	16.36
14	32	1.2	8	4	347.90	0.2175339	20.72
15	32	1.2	12	4	345.45	0.193409	22.4
16	48	1.2	4	4	344.96	0.1849861	11.8
17	48	1.2	8	4	344.28	0.1823272	16.58
18	48	1.2	12	4	343.67	0.1773628	23.5
19	16	1.6	4	4	351.40	0.2614525	35.65
20	16	1.6	8	4	348.70	0.2356021	43.38
21	16	1.6	12	4	347.04	0.2253962	54.5
22	32	1.6	4	4	344.20	0.1797753	23.7
23	32	1.6	8	4	346.78	0.2119641	29.74
24	32	1.6	12	4	344.20	0.1852644	34.92
25	48	1.6	4	4	346.80	0.2079646	16.5
26	48	1.6	8	4	342.90	0.1685728	12.8
27	48	1.6	12	4	344.60	0.1923523	23.8
28	16	0.8	4	2.5	356.70	0.3173024	59.5
29	16	0.8	8	2.5	353.59	0.2752753	70.4
30	16	0.8	12	2.5	350.20	0.2354189	78.9
31	32	0.8	4	2.5	355.87	0.311745	19.3
32	32	0.8	8	2.5	353.60	0.2819383	22.4
33	32	0.8	12	2.5	350.55	0.2428857	29.6
34	48	0.8	4	2.5	351.62	0.251523	16.7
35	48	0.8	8	2.5	349.50	0.2270567	21.38
36	48	0.8	12	2.5	348.28	0.2106356	25.84
37	16	1.2	4	2.5	355.73	0.3170592	67.93
38	16	1.2	8	2.5	352.86	0.2788559	78.35
39	16	1.2	12	2.5	350.29	0.2449603	84.7
40	32	1.2	4	2.5	347.45	0.2107215	36.5

Table 3 (Contd.)

S.No	Fin Height	Fins thickness	Base Height	Fin Pitch	Max Temp	Thermal resistance	Pressure Drop
	mm	mm	mm	mm	k	C / W	pascal
41	32	1.2	8	2.5	346.48	0.2040185	45.72
42	32	1.2	12	2.5	345.26	0.1931729	54.83
43	48	1.2	4	2.5	348.53	0.2244206	17.5
44	48	1.2	8	2.5	346.29	0.2053081	20.3
45	48	1.2	12	2.5	345.61	0.1999478	29.6
46	16	1.6	4	2.5	348.97	0.2256605	96.1
47	16	1.6	8	2.5	347.69	0.2183411	101.47
48	16	1.6	12	2.5	346.30	0.2069667	104.69
49	32	1.6	4	2.5	349.68	0.2409692	34.86
50	32	1.6	8	2.5	344.64	0.1881385	40
51	32	1.6	12	2.5	343.57	0.1772541	56.957
52	48	1.6	4	2.5	351.65	0.2584699	17.83
53	48	1.6	8	2.5	348.20	0.225371	21.37
54	48	1.6	12	2.5	343.59	0.1800095	23.3
55	16	0.8	4	1.5	363.72	0.3667351	118.65
56	16	0.8	8	1.5	361.25	0.3455982	132.74
57	16	0.8	12	1.5	357.08	0.3055793	144
58	32	0.8	4	1.5	355.03	0.2870844	64
59	32	0.8	8	1.5	352.67	0.2575157	75.9
60	32	0.8	12	1.5	349.16	0.2193655	83.47
61	48	0.8	4	1.5	358.60	0.344207	24
62	48	0.8	8	1.5	355.16	0.2922688	39.6
63	48	0.8	12	1.5	351.84	0.254429	48.72
64	16	1.2	4	1.5	352.65	0.2628118	178
65	16	1.2	8	1.5	351.80	0.2508696	192
66	16	1.2	12	1.5	348.90	0.2174139	203
67	32	1.2	4	1.5	352.40	0.2681319	56.8
68	32	1.2	8	1.5	350.06	0.2389718	78.8
69	32	1.2	12	1.5	348.60	0.2185445	93.5
70	48	1.2	4	1.5	356.30	0.3165548	39.5
71	48	1.2	8	1.5	354.70	0.2950276	44.82
72	48	1.2	12	1.5	349.93	0.2402783	49.38
73	16	1.6	4	1.5	353.40	0.2809735	197.3
74	16	1.6	8	1.5	350.02	0.2366275	207.1
75	16	1.6	12	1.5	348.70	0.2147303	214.6
76	32	1.6	4	1.5	351.54	0.2662896	83.6
77	32	1.6	8	1.5	349.17	0.2370661	87.2
78	32	1.6	12	1.5	347.04	0.2104081	90.7
79	48	1.6	4	1.5	346.44	0.2059105	57.5
80	48	1.6	8	1.5	345.70	0.1940151	62.5
81	48	1.6	12	1.5	345.16	0.1837259	66.18

CONCLUSIONS

In this research work, optimal design of the heat sink have been carried out on a parallel plate heat sink considering the geometric parameters such as fin height , fin thickness , base height and fin pitch with a constant length and width of a heat sink using computational fluid dynamics study. The simulation is carried out with the Commercial software provided by fluent Inc. Experimental studies have been carried out with a parallel plate heat sink to validate the heat sink model. The results obtained in the experimental studies have been compared with the simulation results and found to be in good agreement..

These results and conclusions drawn in this present work are found to be in good agreement with conclusions drawn by Lee (1995), and C.W. Leung and S.D. Probert (1988). This study will benefit the design engineers involved in electronic cooling. Using the approach presented in this research work, the design engineers can carry out optimization of parametric CAD models, for the selection or design of heat sink for effective thermal management in their electronic assemblies.

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