THERMAL ANALYSIS AND OPTIMIZATION OF L-SHAPE FIN HEAT SINK UNDER NATURAL CONVECTION USING ANOVA AND TAGUCHI

by

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Advancement in electronic systems resulted in miniaturization and high-power densities. Therefore, the rate of heat generation in circuit board increased dramatically. To overcome the problem of overheating, numerous heat sink designs are proposed including L-shape fins heat sink. The thermo-fluidic flow behavior and temperature difference are analyzed to get better understanding of heat transfer from the sink to ambient air. Governing equations for the model of conjugate heat transfer in 3-D environment are solved and discretized across the computational domain. Numerous experiments are carried out to validate the numerical results. The effect of fin numbers, height, and heat sink size at three different input power is reported. Furthermore, ANOVA and Taguchi statistical methods are used to predict parameters that affect the heat transfer. The study revealed that fin height affects the heat transfer rate the most, and accounts for 25.3% increase in heat transfer rate. Optimization of the heat sink is carried out to ensure better efficiency of the proposed heat sink. The optimized conditions for the sink are observed to be heat sink size of 90 mm, 9 number of fins, and 33 mm of fin height.

Key words: L-shape heat sink, thermal management, thermal analysis, novel heat sink design

Introduction

Due to compact structural design of the modern electronic devices, unwanted heat is produced in the circuit boards. High thermal densities decrease the life of the electronic chip, hence, proper thermal management is needed to extract unwanted heat from the system. Heat sinks made of aluminium alloy are commonly used cooling systems (active and passive) to extract heat from the system and dump into the environment. Heat transfer depends on the material properties and contact area between two materials. Many fins are mounted vertically on the horizontal base plate to maximize the area of contact. Currently, conventional heat sink with straight fins is commonly used due to the simple geometry, low cost of manufacturing.
and high heat transfer rate. Research have been carried out recently on straight fin heat sink to study thermos-fluid flow and optimize different input parameters.

Huang and Tung [1] studied the performance of wavy shape heat sink both experimentally and numerically. Further, Duan et al. [2] analyzed pressure drops and fluid-flow characteristics in heat sink with plate fins and bottom profile of elliptic shape. Alfellag et al. [3] used conjugate model of heat transfer and numerically studied the thermo-fluidic flow in mini channel heat sink with triangular shape fins. They claimed Nusselt number improved by 1.5 at angle of 55° for the full height slot. Further, they observed that hydrothermal factor of performance enhanced by 1.43. Abdelmohimen et al. [4] studied the effect of rod on heat transfer with various configurations in plate fin heat sink. They claimed that for the studied range of Reynolds number optimal case of heat transfer was achieved when four rods are placed through fins. Wiriyasart and Naphon [5] targeted liquid on different shapes of fin i.e. circular, rectangular, and conical to enhance heat transfer. However, results revealed a very small improvement in thermal resistance. Freegah et al. [6] proposed fillet profile in plate fin heat sink and numerically investigated improvement in heat transfer rate. Numerous studies are carried out to investigate and improve heat transfer in heat sink by using PCM [7-10]. Nair [11] proposed a method to enhance heat transfer through convection of plate fin heat sink by introducing additional shrouds to the top and bottom of the sink.

Conventional heat sink was studied by Jones and Smith [12]. They found inter fin distance and fin height are important parameters and affect the heat transfer rate drastically. Bar-Cohen [13] concluded that optimal fin thickness varies with combination of different parameters, i.e., material, environmental, and geometric constraints. Teertstra et al. [14] carried out experimentations and presented analytical model for pin fin heat sink under natural convection. They revealed that the presented analytical model could be used calculate heat transfer rate to choose suitable heat sink according to the specific specification. Baskaya et al. [15] carried numerical studies and investigated various input parameters, included inter-fin distance, heat transfer rate and temperature deference. They observed that increasing the fin height results in high heat transfer rate. Furthermore, de Lieto Vollaro et al. [16] numerically studied straight fin heat sink in vertical orientation. They focused on thermal conductivity, emissivity, dimensions of heat sink and other working fluid properties. Tari et al. [17] studied different combinations of input parameter in straight fin heat sink including fin length, fin height, heat sink orientation and spacing between the fins, under natural convection and suggested correlations.

Shen et al. [18] experimentally and numerically investigated the effect conventional heat sink orientation on heat transfer rate. They suggested correlations for different orientations and Rayleigh numbers. Naik et al. [19] considered the fact that heat sinks are usually placed in enclosed electronic system and placed a straight fin heat sink under an adiabatic shroud. They experimentally investigated the heat transfer rate under natural convection. They concluded that the heat transfer rate is directly proportional to the clearance between the heat sink and shroud.

To enhance heat transfer rate both in active and passive convection various designs are proposed and studied i.e., straight fins with slope [20], straight fins with perforations [21], straight fins with square and circular cross-sections [22], straight fins with circular arrangement [23], and straight metal foam fins [24]. Furthermore, Kim [22] studied heat transfer rate with various cross-section designs mounted vertically on base plate under natural convection. Ledezma and Bejan [20] improved heat transfer performance of the conventional heat sink by introducing slope to the fins. Furthermore, Jang et al. [23] experimentally studied radial heat
sink and optimized its design in terms of thermal resistance. Costa and Lopes [24] introduced technique to obtain specific cooling with minimum material and mass usage. Jang et al. [25] and Li et al. [26] found that the heat transfer rate is improved for the radial heat sink by introducing a chimney. In addition, Elshafei [21] proposed hollow straight fins and claimed that improved heat transfer performance was achieved.

In a recent study, Chamkha et al. [27] investigated various parameters such as Rayleigh number, heat generation and volume fraction for slant porous cavity and nanoparticles. The author claimed that, addition of nanoparticles for several Rayleigh numbers reduced the heat transfer rate. Furthermore, Kargarsharifabad [28] numerically and experimentally investigated heat transfer rate for Cu-water nano-fluid under natural convection. The author concluded that, heat transfer is decreased when the volume fraction of nanoparticles is increased. In another study [29] the same author used genetic algorithm and numerical simulations and optimized the arrangement of both obstacles and conducting fins. An increase of 1.63 times in heat transfer for obstacles fins was noted when compared to conducting fins. Recently, Chamkha et al. [30] studies gamma-shaped porous cavity with Cu-water nanofluid and in the presence of magnetism. The study revealed that increase in nanoparticles volume fraction affect the entropy production more than enhancement in heat transfer rate. Bahmani and Kargarsharitabed [31] investigated horizontal flat plate heat exchangers under natural convection and constant heat flux. The results revealed that for Prandtl number higher than 1, low Nusselt number was achieved. However, the reverse trend was observed for Prandtl number less than 1 and another study [32] a correlation was derived from numerical simulation results.

The L-shape heat sink design is proposed in previous studies to enhance heat transfer rate. However, no study has been carried out on its optimization. In the present study, experiments and simulations are carried out to better understand fluid flow and heat transfer mechanism in L-shaped heat sink under natural convection. Furthermore, ANOVA and Taguchi methods are used to optimize the heat sink and analyze the effect of input parameters on the heat transfer rate. In addition, the effect of clearance between heat sink and shroud is also investigated by keeping the fact in mind that heat sink are usually kept in enclosures. A numerical model was validated through experimentations under natural convection.

**Materials and methods**

**Governing equations**

Following assumptions are made with computational model:

- The air-flow is incompressible, laminar, and steady.
- The fluid used in the model is considered to be Newtonian.
- Only heat transfer through conduction and convection is considered. However, heat transfer through radiation is neglected.
- Body forces and viscous dissipation effects are neglected.

Based on the assumptions, the energy, momentum and continuity equations in the three dimensions are [33]:

\[
\left[ \frac{\partial T}{\partial x}(u) + \frac{\partial T}{\partial y}(v) + \frac{\partial T}{\partial z}(w) \right] C_p \rho = \left[ \frac{\partial^2 T}{\partial x^2}(u) + \frac{\partial^2 T}{\partial y^2}(u) + \frac{\partial^2 T}{\partial z^2}(w) \right] k
\] (1)
Equations of momentum:

\[
\begin{align*}
\left[ \frac{\partial u}{\partial x} (u) + \frac{\partial u}{\partial y} (v) + \frac{\partial u}{\partial z} (w) \right] \rho &= \mu \nabla^2 \mathbf{u} - \frac{\partial p}{\partial x} \\
\frac{\partial v}{\partial x} u + \frac{\partial v}{\partial z} w + \frac{\partial v}{\partial y} v &= \nabla^2 uv + (T - T_\infty) \beta g \\
\left( \frac{\partial w}{\partial x} u + \frac{\partial w}{\partial z} w + \frac{\partial w}{\partial y} v \right) \rho &= \mu \nabla^2 wv - \frac{\partial p}{\partial x}
\end{align*}
\]

Equations of continuity:

\[
\frac{\partial u}{\partial x} + \frac{\partial w}{\partial z} + \frac{\partial v}{\partial y} = 0
\]

where \( u, v, \) and \( w, x, y, \) and \( z, \) are directions, \( C_p \) – the heat capacity, \( \rho \) – the density of fluid, \( \beta \) – the coefficient of thermal of expansion, \( T \) – the heat sink surface temperature, \( k \) – the thermal conductivity of material, and \( T_\infty \) – temperature of ambient air is specified with. However, heat loss due to radiation is not considered [34].

Equation for the coefficient of heat transfer for convection is given [35]:

\[
h_{avg} = \frac{Q}{(T_{surf} - T_{air})A}
\]

For the power input following equation is used:

\[
P_{in} = VI
\]

where \( V \) is voltage and \( I \) is current.

Equation for thermal resistance in convection:

\[
R_{Con} = \frac{1}{hA}
\]

In the eqs. (6) and (7), \( A \) is surface area and \( h \) is the heat transfer coefficient for convection.

Gauge pressure at inlet \((x = 0)\) is applied, inlet gauge pressure with uniform distribution was applied:

\[
T_i = T_{in} \]

\[
p = p_{in}
\]

At the heater power of constant value is applied at \( y = 0 \).

\[
-k_i \frac{\partial T}{\partial y} = q
\]
Adiabatic boundary condition is applied to the remaining surfaces:

\[
\frac{\partial T}{\partial z} = \frac{\partial T}{\partial y}
\]

(12)

**Computational domain set-up**

The L-shape fins heat sink with variation in fin number, fin height and heat sink sizes are considered in the current study. As shown in fig. 1, a computational model of length 1200 mm, and width and length of 120 mm is taken. Surface at \(X = 0\) mm is taken as inlet and surface at \(X = 1200\) mm is taken as outlet. In addition, remaining surfaces are considered to be adiabatic walls. The studied heat sink, L-shape finned heat sink with various configuration is placed at the center of the computational domain as shown in fig. 2. A constant fin thickness of 2 mm is considered with variation in fins height of 10 mm, 20 mm and 30 mm. Table 1 shows the properties of materials used in the current study. A constant ambient temperature of 300 K is considered with three different input power i.e., 20 W, 40 W, and 60 W.

**Grid independency and solution method**

In the current study, CFD commercial code of IcePack (ANSYS®) version 15.0 is used to carry out 3-D simulations numerically. Upwind scheme of 2nd order is considered for the convective terms. However, 2nd order difference scheme is considered for diffusion terms. Furthermore, SIMPLE algorithm is used for the pressure and velocity coupling. For momentum and continuity, a convergence criterion was set to less than \(10^{-6}\). Core i5 computer with RAM of 8 GB and processors of 2.5 GHz was used to conduct simulations.

It was ensured through mesh independency test that the generated solution is independent of mesh size. Element numbers varied from 325344 to 1119136 for the constant input power of 20 W. For any \(N\), relative percent error, \(e\%\), was calculated using eq. (13) between the course and finer mesh (\(N_1\) and \(N_2\)):

\[
e\%_N = 100 \frac{N_2 - N_1}{N}
\]

(13)

As illustrated in tab. 2, relative error, \(e\%\), less than 0.5% is observed in element number at 0.8 million based on \(\Delta T\) value. In the current study, Mesh number 3 is considered and is adequate to get acceptable results.

<table>
<thead>
<tr>
<th>Property</th>
<th>Unit</th>
<th>Ambient air</th>
<th>Heat sink</th>
</tr>
</thead>
<tbody>
<tr>
<td>(C_p)</td>
<td>[Jkg(^{-1})K(^{-1})]</td>
<td>1000</td>
<td>870</td>
</tr>
<tr>
<td>(K)</td>
<td>[Wm(^{-1})K(^{-1})]</td>
<td>0.0242</td>
<td>200</td>
</tr>
<tr>
<td>(P)</td>
<td>[kgm(^{-3})]</td>
<td>1.225</td>
<td>2700</td>
</tr>
</tbody>
</table>
Table 2. Mesh independency test

<table>
<thead>
<tr>
<th>Mesh number</th>
<th>Grid numbers</th>
<th>Temperature difference</th>
<th>ε%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>325344</td>
<td>54.79</td>
<td>3.59</td>
</tr>
<tr>
<td>2</td>
<td>519917</td>
<td>51.49</td>
<td>1.49</td>
</tr>
<tr>
<td>3</td>
<td>822371</td>
<td>50.09</td>
<td>0.48</td>
</tr>
<tr>
<td>4</td>
<td>919922</td>
<td>49.89</td>
<td>0.19</td>
</tr>
<tr>
<td>5</td>
<td>1118225</td>
<td>49.79</td>
<td>-</td>
</tr>
</tbody>
</table>

Experimental set-up

In the present study, free and forced convection apparatus, model TH320, manufactured by ESSOM (Thailand) was used to record experimental results. Different temperature sensors were mounted on various positions to record temperature i.e., inlet, outlet, ambient air and heat sink temperatures. Furthermore, voltmeter and ammeter were used to measure input power to the heater.

As shown in fig. 3, heat sink was mounted on heater in free and forced convection apparatus using thermal paste to ensure maximum heat transfer from heater to the sink without heat loss. An insulation made of rock wood was placed on the bottom surface of the heater to prevent heat loss. Thermal sensors (K-type thermocouples) at five different positions were mounted to record temperatures. Furthermore, two sensors were placed at inlet and outlet of the channel. All thermocouples were connected to TC-08 for data acquisition and the data was stored in a computer connected to the acquisition system. The heat power was displayed by ammeter and voltmeter connected to the heater, and varied through a knob in control box.

![Experimental set-up schematic](image)

Figure 3. Experimental set-up schematic

Experimentation in the current study was carried out in a laboratory with a room temperature and without HVAC system. Control unit was used to feed accurate input power of 20 W, 40 W, and 60 W. Ammeter and voltmeter were used to monitor voltage and current and calculate the input power to the heater. Data of all the thermocouples was recorded in the condition of steady state which was set to 0.5 °C temperature change per hour. Accurate data was ensured by repeating each experiment three times.

Aluminium alloy 6063 was used to manufacture L-shape fins heat sink. The alloy has thermal conductivity of 200 W/mK. A 100 mm by 100 mm single heater was used. The
sink was manufactured with 9 fins and fin height of 20 mm. However, the base of the sink was taken 3 mm.

Equations (14) and (15) is used to calculate measurement uncertainties in the experimental set-up [36]. According to the provided manual from the manufacturer, current and voltage reading accuracy is 0.1%. Using the concept of the uncertainty in [37]. Furthermore, the effect of radiation is neglected; therefore, the heat supplied to the system is assumed to be equal to heat transfer rate [38]:

$$\frac{\Delta q_{in}}{q_{in}} = \frac{\Delta P_{in}}{P_{in}} = \sqrt{\left(\frac{\Delta I}{I}\right)^2 + \left(\frac{\Delta V}{V}\right)^2}$$

(14)

$$\frac{\Delta q_{in}}{q_{in}} = \frac{\Delta R}{R} = \sqrt{\left(\frac{\Delta (\Delta T)}{T}\right)^2 + \left(\frac{\Delta q}{q}\right)^2}$$

(15)

Maximum uncertainty in the heat transfer rate is calculated to be 6% by putting both the $V$ and $I$ values in eq. (14). However, the maximum error in thermocouples is given 0.1 K by the manufacturer. Using this information error in temperature measurement, $\Delta T$, can be calculated. Average experimental uncertainties is calculated to be 3.5% for the thermal resistance.

Results and discussion

Model validation

To validate numerical model, experimentation has been carried out. The experimental and numerical results on the bases of temperature difference for L-shape fins heat sink are plotted as shown in fig. 4. Between experimental and numerical results, a slight difference of less than 9% is observed. The difference could be due to the error in sensors and input power to the heater. Furthermore, the difference could be due to heat loss to environment during experimentation, while in numerical solution all four walls were taken insulated. Overall, the numerical results show good agreement with numerical result. Hence, the model could be used for thermal analysis.

Effect of input parameters

Effectiveness of heat sink is dependent on the temperature difference between heat sink and surrounding. With various combination of input variables experimentation and simulations were carried out and temperature differences were recorded. As shown in fig. 5, the temperature difference increases as input power increases. At maximum input power high thermal energy is generated inside the chip and maximum heat is transferred to the heat sink which results in high difference in temperature.

A declination is observed in temperature difference when number of fins are increased from 5-9 as shown in fig. 6. The increase noted is due to the increase in the contact
area between heat sink and surrounding air which results in high heat transfer. Furthermore, steep declination of temperature difference can be seen when number of fins are increased from 5-7. However, further increase in fins from 7-9, the steepness reduces as the inter pin distance reduces, which results in drop in velocity and increase in fraction between the channel surface and air. Furthermore, the performance of \(L\)-shape heat sink increases when the number of fins increases. This is due to the fact that more number of fins offers more area for heat transfer and maximum heat is exchanged between the air and heat sink.

As shown in fig. 7, the temperature difference increases when the fins height increases. This is due to the fact that heat transfer is directly proportional the contact area. As the fin height increase extra material added which provides more area for heat to transfer to surrounding air through convection. With increase in fin height from 10-30 mm the heat dissipation mechanism improves and temperature deference drops. At high input power, little difference in temperature difference is noted.

The \(L\)-shape fins heat sink depicts similar gradual decreasing behaviour when the size of heat sink increases from 50-90 mm as shown in fig. 8. Furthermore, heat sink size offers extra material addition like number of fin and fin height, which results in increase in surface area of the sink and leads to high heat transfer rate.

**Effect of enclosure**

The heat sink are usually placed in enclosures as shown in fig. 1. In this study, simulations were carried out to understand the effect of shroud on heat transfer. As shown in fig. 9, the temperature at side A is slightly higher when compared to side B. At side A maximum
temperature of 53.15 °C is recorded. However, at side B the recorded temperature is 52.90 °C. The difference in both sides is due to the fact that clearance between side A and shroud is minimum which prevent fresh air circulation. Hence, results in low heat transfer. However, the clearance between shroud and side B is maximum which allow large amount of fresh air to come in contact with heat sink and result in high heat transfer.

Clearance between heat sink and top wall of the shroud is also considered in the present study and illustrated in fig. 10. Three cross-sectional planes show the temperature distribution in fluid domain at 10 mm, 20 mm, and 30 mm height. At 10 mm plane a small stagnation zone is formed, while the stagnation zone increases when the height increases. This is due to the top wall of the shroud which prevents the air to flow and result in large stagnation area.

Taguchi

Signals to noise ratios are shown in tab. 3 with input factors and their levels. Factors are ranked on the basis of their effect on average temperature. Smaller is the better model was used to rank input parameters according to their contribution. Power factor is ranked first which depicts that power has more influence on average temperature and accounts for 46.8%. In addition, fin height is the second most important factor and its contribution is 25.3%. While, heat sink size and number of fins are ranked third and fourth, respectively and have no significant effect. Collective effect of heat sink size and fin numbers accounts for 27.9%.

Table 3. Response table for signal-to-noise ratios

<table>
<thead>
<tr>
<th>Level</th>
<th>Heat sink size [mm]</th>
<th>Power [W]</th>
<th>Number of fins</th>
<th>Fin height [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>–44.43</td>
<td>–38.84</td>
<td>–44.06</td>
<td>–45.39</td>
</tr>
<tr>
<td>2</td>
<td>–43.41</td>
<td>–43.72</td>
<td>–42.90</td>
<td>–42.79</td>
</tr>
<tr>
<td>3</td>
<td>–41.45</td>
<td>–46.74</td>
<td>–42.33</td>
<td>–41.12</td>
</tr>
<tr>
<td>Delta</td>
<td>2.98</td>
<td>7.90</td>
<td>1.73</td>
<td>4.27</td>
</tr>
<tr>
<td>Rank</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>2</td>
</tr>
</tbody>
</table>

The ANOVA

Experimental results at a confidence level of 95% is evaluated using ANOVA. Analyzing P-values in the analysis of variance shown in tab. 4, all the parameters have significant effect on the output result. Combination of heat sink size and no of fins, heat sink size and power and heat sink size and fin height have also significant influence on temperature difference. However, other combination of the input factors on temperature difference is found to
be insignificant. Further, observing F-value in ANOVA and signal-to-noise ratio in Taguchi, it is noted that power and fin height have great influence on temperature difference. In two factor combination, model F-value for power and fin height is maximum, which significantly affect the results when compare to other combinations.

Table 4. Analysis of variances

<table>
<thead>
<tr>
<th>Source</th>
<th>DF</th>
<th>Adj SS</th>
<th>Adj MS</th>
<th>F-Value</th>
<th>P-Value</th>
<th>Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat sink size</td>
<td>2</td>
<td>32098</td>
<td>16049</td>
<td>465.28</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Power</td>
<td>2</td>
<td>238081</td>
<td>119041</td>
<td>3451.19</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Number of fins</td>
<td>2</td>
<td>12129</td>
<td>6065</td>
<td>175.82</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Fin height</td>
<td>2</td>
<td>76812</td>
<td>38406</td>
<td>1113.45</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Heat sink size × power</td>
<td>4</td>
<td>2379</td>
<td>595</td>
<td>17.24</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Heat sink size × number of fins</td>
<td>4</td>
<td>326</td>
<td>82</td>
<td>2.36</td>
<td>0.097</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Heat sink size × fin height</td>
<td>4</td>
<td>401</td>
<td>100</td>
<td>2.91</td>
<td>0.055</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Power × number of fins</td>
<td>4</td>
<td>1853</td>
<td>463</td>
<td>13.43</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Power × fin height</td>
<td>4</td>
<td>6849</td>
<td>1712</td>
<td>49.64</td>
<td>0.000</td>
<td>Significant</td>
</tr>
<tr>
<td>Number of fins × fin height</td>
<td>4</td>
<td>181</td>
<td>45</td>
<td>1.31</td>
<td>0.309</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Heat sink size × power × number of fins</td>
<td>8</td>
<td>383</td>
<td>48</td>
<td>1.39</td>
<td>0.274</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Heat sink size × power × fin height</td>
<td>8</td>
<td>615</td>
<td>77</td>
<td>2.23</td>
<td>0.082</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Heat sink size × number of fins × fin height</td>
<td>8</td>
<td>265</td>
<td>33</td>
<td>0.96</td>
<td>0.498</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Power × number of fins × fin height</td>
<td>8</td>
<td>400</td>
<td>50</td>
<td>1.45</td>
<td>0.250</td>
<td>Insignificant</td>
</tr>
<tr>
<td>Error</td>
<td>16</td>
<td>552</td>
<td>34</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>80</td>
<td>373324</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 11. Main effect plot for signal-to-noise ratios

Optimization

Optimization of the heat sink is carried out through Taguchi to analyze effect of each parameter and to find optimal level for each factor as shown in fig 11. Smaller is better model for signal-to-noise ratio is considered to predict optimal conditions for the heat sink. The optimal condition for the heat sink is obtained at 90 mm heat sink size, 20 W input power, 9 number of fins, and 33 mm fins height. Combination of these level could be used to achieve maximum efficiency.

Conclusion

In this study, L-shape fins heat sink in investigated under natural convection with various input parameters. The energy, continuity, and momentum governing equations are discretised and solved for different input powers. The results are presented in the form of av-
verage temperature, temperature difference, $\Delta T$, and temperature contours. The mean temperature difference is the most important parameter to evaluate performance of the heat sink at different configurations. The results revealed that increasing the clearance between the shroud and sink increases the heat transfer rate. Furthermore, ANOVA and Taguchi results disclosed that fin height is the most influential input parameter and accounts for 25.3% of total contribution. In addition, fin height of 33 mm, heat sink size of 90 mm and 9 fin numbers is found to be the optimal configuration and results in maximum heat transfer.

References


