Characterization of a Finned Heat Sink for a Power Inverter

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Abstract-
Heat is a by-product which is constantly being generated in the operation of a power inverter and if left unchecked will inevitably lead to the damage of the device. Hence a means to efficiently dissipate this heat has to be employed. In this research, a heat sink is mathematically modelled and its thermal performance was evaluated using ANSYS software and experimentally validated. The optimisation of the heat sink was done with the aid of the FMINCON optimization tool in MATLAB. A K-type thermocouple and a three channel temperature logger, MTM-380SD, with real time data logger were used to obtain temperature data of the heat sink for the purpose of experimental validation. The optimized heat sink parameters are heat sink length and width, number of fins, base thickness, fin height, thickness and spacing. Results show that the percentage deviation between the simulation and experimental temperature results for a pulse load of 300W is 8%, for a pulse load of 460W is 3%, for a pulse load of 600W is 8%, for a pulse load of 1015W is 2%. The maximum simulated and experimented temperatures are 84°C and 85.4°C. Thus the inverter can be safely and reliably operated.

Key words: Heat sink, Optimization, thermal performance, data logger, load, Simulation.

1. Introduction
Heat sinks are normally used for heat dissipation to reduce the temperature of heated surfaces. Analytical method for heat sink analysis is not easily obtained due to the flow field and temperature gradients surrounding the fin geometry [1]. The switches used in a power inverter dissipate heat which must readily be contained for efficient and reliability operation, and the desire to reduce the overall form factor necessitate optimization of heat sink designs to reduce size and material cost [2].

Heat sinks are essentially heat exchangers that help to dissipate heat generated from the switches into a fluid in motion. It is generally desire to enhance the surface area of the heat sink in contact with the cooling fluid. Figure 1 shows a heat sink with extruded rectangular fins.

Figure 1: Heat Sink with Continuous Rectangular Fins [3]
The fins that make up an integral part of the heat sink are defined as extended surfaces that use conduction-convection effects to enhance heat transfer between the heat sink and the adjoining fluid. In the design and optimization of heat sink parameters, it should however be noted that the parameters are kind of interdependent on each other [1].

Due to thermal energy generated arising from the switches operation, proper thermal management must be put in place to safeguard the switches, the switches commonly used are IRF 3205 MOSFET chips [4]. Thermal energy generated during the switches operation leads to increase in temperature of the IRF 3205 MOSFET chips which will ultimately affect its performance and can lead to thermally induced failure [5]. The heat sink can be optimized by variations to its geometry and material [6].

The dissipated heat from the switches in the power inverter is transferred to the heat sink through case and gap materials. Subsequently this heat is transferred from the heat sink to the ambient air. Hence there is need to optimized the heat sink geometry so that estimated junction temperature of switch is not greater than the prescribed junction temperature [7], [8].

Thus a general mathematical optimization of air cool heat sink was derived; the results were verified through numerical simulations and experiments. The heat sink optimization is performed on the basis of the heat sink geometry [9].

2. Mathematical Model of Thermal Dissipation and Heat Sink

The IRF 3205 MOSFET switches attached to the heat sink have junction temperature which is critical to its performance. The dissipated heat from the switches is transferred to the heat sink through the case and gap materials. Subsequently this heat is transferred from the heat sink to ambient air. It is the ultimate aim to design heat sink geometry so that estimated junction temperature of the switches is not greater than the prescribed junction temperature. The dissipated power forms an input to the thermal model of the heat sink.

2.1 Power Dissipation during Switch Operation and Conduction

Figure 2 shows the electrical to thermal transformation of the IRF 3205 MOSFET switch. The capacitor shown model the transient thermal response of the circuit which follows a resistance capacitance time constant.
Figure 2: Principle of Electro – Thermal Analogy

Temperature becomes an important factor when designing power inverter. Switching and conduction losses heat up the silicon of the IRF 3205 switch above its maximum junction temperature that causes performance failure, breakdown and worst case fire.

The total dissipated power in the switches is the sum of the conduction and switching losses [10], [11]. When the IRF 3205 MOSFET chip is turned on, the energy losses ($E_{onM}$) can be calculated as [12], [13]:

$$E_{onM} = U_{DD} \times I_{Don} \times \frac{t_{ri} + t_{fu}}{2} + Q_{rr} \times U_{DD}$$  \tag{1}

Where $t_{ri}$ is the current rise time, $t_{fu}$ is the voltage fall time, $I_{Don}$ is the drain current, $U_{DD}$ is the Inverter supply voltage (DC), $Q_{rr}$ is the reverse recovery charge. The voltage fall time can be calculated as a median of the fall times defined through the gate current as [14]:

$$t_{fu} = \frac{t_{fu1} + t_{fu2}}{2}$$  \tag{2}

Where

$$t_{fu1} = (U_{DD} - R_{DSon} \cdot i_D) \frac{C_{GD1}}{I_{Gon}}$$  \tag{3}

$$t_{fu2} = (U_{DD} - R_{DSon} \cdot i_D) \frac{C_{GD2}}{I_{Gon}}$$  \tag{4}

$$I_{Gon} = \frac{U_{Dr} - U_{Plateau}}{R_G}$$  \tag{5}

where $C_{GD1}$ is the gate-drain capacitance 1, $C_{GD2}$ is the gate-drain capacitance 2, $R_G$ is the gate Resistance, $U_{Dr}$ is the output voltage from driver circuit, $U_{Plateau}$ is the plateau voltage.

The turn off energy losses is expressed as:

$$E_{offM} = U_{DD} \times I_{Doff} \times \frac{t_{ru} + t_{fi}}{2}$$  \tag{6}
Where $t_{ru}$ is the voltage rise time, $t_{fl}$ is the current fall time. The voltage rise time can similarly be calculated as a median of the rise times defined through the gate current as:

$$\text{tru} = \frac{\text{tru1} + \text{tru2}}{2}$$  \hspace{1cm} (7)

Where

$$\text{tru1} = (U_{DD} - R_{DSon}.I_{D}) \cdot \frac{C_{GDI}}{I_{Goff}}$$  \hspace{1cm} (8)

$$\text{tru2} = (U_{DD} - R_{DSon}.I_{D}) \cdot \frac{C_{GDZ}}{I_{Goff}}$$  \hspace{1cm} (9)

$$I_{Goff} = \frac{U_{\text{Plateau}}}{R_{G}}$$  \hspace{1cm} (10)

The total losses due to switching in the chips are function of switching frequency and can be obtained as [15]:

$$P_{\text{swM}} = (E_{onD} + E_{offM})f_{sw}$$  \hspace{1cm} (11)

All silicon devices provide resistance to the flow of electric current that originates from the resistivity of the bulk semiconductor materials. When the device is operating in the ON state and the OFF state leakage current, the losses due to conduction in the chips is computed using the drain-source ON state resistance ($R_{DSon}$) [12], [6]:

$$U_{DS}I_{D} = R_{DSon}I_{D} \times I_{D}$$  \hspace{1cm} (12)

Where $U_{DS}$ is the drain source voltage, $I_{D}$ is the drain current.

Therefore, the instantaneous value of the IRF 3205 MOSFET conduction loss is [12]:

$$P_{C}(t) = U_{DS}(t) \times I_{D}(t) = R_{DSon} \times I_{D}^{2}(t)$$  \hspace{1cm} (13)

The total power dissipation due to switching ON and OFF and conduction of the chips is then obtained as:

$$P_{\text{d}} = P_{CM} + P_{\text{swM}} = R_{DSon} \times I_{Drms}^{2} + (E_{onM} + E_{offM})f_{sw}$$  \hspace{1cm} (14)

The ON state resistance of the chip is highly temperature dependent, and it can be linearly obtained as:

$$R_{DSon} = R_{DSonSPEC} \times [1 + 0.005 \times (T_{jmax} - T_{SPEC})]$$  \hspace{1cm} (15)

Where $T_{jmax}$ is the operating junction temperature, $R_{DSonSPEC}$ is the ON state resistance value at $T_{SPEC}$. $T_{SPEC}$ is normally 25°C. The power dissipation varies with the duty ratio and equation (14) is re-written as:

$$P_{\text{d}} = \Delta t_{on}f_{sw}(R_{DSon} \times I_{Drms}^{2} + (E_{onM} + E_{offM})f_{sw})$$  \hspace{1cm} (16)

From equation (1) and equation (6), the power dissipated is obtained as:

$$P_{d} = (\Delta t_{on}f_{sw})\left[(R_{DSon} \times I_{Drms}^{2}) + \left(U_{DD} \times I_{Don} \times \frac{t_{ru} + t_{fl}}{2}\right) + Q_{rit} \times U_{DD} + U_{DD} \times I_{Doff} \times \frac{t_{ru} + t_{fl}}{2}\right]$$  \hspace{1cm} (17)
2.2 Thermal Resistance of Heat Sink

In order to determine the effectiveness of heat sink, a thermal performance metrics has to be introduced that take into account the overall heat dissipation, cost function as a weight of heat sink and the heat sink geometry of fin spacing, fin length, fin thickness, base thickness and number of fins. Figure 3 depicts a diagram of a straight fin heat sink.

![Heat Sink Thermal Resistance Network](image)

The hydraulic diameter, $d_h$, of the heat sink channel is defined [17]:

$$d_h = \frac{4A}{P}$$  \hspace{1cm} (18)

And the area, $A$, perpendicular to flow is expressed as:

$$A = (n - 1)s \cdot L_f$$  \hspace{1cm} (19)

The wetted perimeter, $P$, of the duct is obtained as:

$$P = 2(n - 1)(s + L_f)$$  \hspace{1cm} (20)

From equation (19) and equation (20), the hydraulic diameter of the fluid channel as a function of the fin spacing and fin height is then:

$$d_h = \frac{2sL_f}{s + L_f}$$  \hspace{1cm} (21)

For the given fin geometry, the total air flow through the channels creates a pressure drop along the heat sink length, $L$, for laminar flow as [18]:

$$\Delta P_L = 1.5 \frac{32\rho u L}{n(sL_f)d_h^3} \dot{V}$$  \hspace{1cm} (22)

Where $\rho$ is the density of air kg/m$^3$, $u$ is the kinematic viscosity of air m$^2$/s, $\dot{V}$ is volume flow rate m$^3$/s. The 1.5 correction factor takes into account the non quadratic channel shape characterized by $s \ll L_f$, where $L_f$ is the fin length.
For turbulent flow, the drop in pressure is as defined [18]:

\[
\Delta P_T = \frac{1}{2} \frac{\rho d^2}{\mu} \left( \frac{V^2}{2} \right)^2 \left( 0.79 \ln \left( \frac{2V}{\mu (S+L) A} \right) - 1.64 \right)^2
\]  

(23)

For appropriate fan selection the air flow static pressure characteristic curve is used as shown in figure 4.

Figure 4: Air Flow and Static Pressure Characteristics [19]

The air flow static pressure characteristic curves are found in the technical data of the fan manufacturer catalog. The Reynolds number, Re, as a function of the heat sink geometry as follows [18]:

\[
Re = \frac{\rho u d h}{\mu}
\]  

(24)

Where \( u \) is the velocity of flow, m/s, \( \mu \) is the dynamic viscosity, kg/ms. Now

\[
\nu = \frac{\mu}{\rho}
\]  

(25)

\[
u = \frac{V}{A}
\]  

(26)

Where \( V \) is the volume flow rate, m\(^3\)/s and \( A \) is the cross sectional area, m\(^2\). For the heat sink, the area of flow is obtained as:

\[
A = sL_f(n - 1)
\]  

(27)

Where \( n \) is the number of fins. Substituting equation 25, equation 26, equation 27 and equation 21 into equation 24, gives the Reynolds number as:

\[
Re = \frac{2V}{(n-1)(S+L)d}
\]  

(28)

The Reynolds number, Re, determine whether the flow is laminar or turbulent. For laminar flow Re is < 2300 and the Nusselt number, Nu, is defined as [18]:

\[
Nu = \frac{hL}{k}
\]  

(29)

Where \( h \) is the heat transfer coefficient, W/m\(^2\)K, \( L \) is the thickness of the heat transfer surface, m, and \( k \) is the thermal conductivity of the material, W/mK.
\[ \text{Nu}_{L} = \frac{3.657 \left[ \tanh(2.264X^{1/3} + 1.7X^{2/3}) \right]^{-1} + 0.0499 \tanh X}{\tanh(2.432Pr^{1/6}X^{1/6})} \]  

(29)

With

\[ X = \frac{L}{d_{h}RePr} \]  

(30)

\[ \text{Pr} = \frac{C_{p}\mu}{k} \]  

(31)

For turbulent flow \( Re > 2300 \) and the Nusselt number, \( \text{Nu} \), is expressed as [18]:

\[ \text{Nu}_{T} = \frac{[8(0.79\ln Re - 1.64)^{2}]^{-1}(Re - 1000)Pr}{1 + 12.7\sqrt{[8(0.79\ln Re - 1.64)^{2}]^{-1}(Pr^{2/3} - 1)(1 + (d_{h}/L)^{2/3})}} \]  

(32)

Properties of air are obtained at mean film temperature, \( T_{f} \), defined by [20], [21]:

\[ T_{f} = \frac{T_{b} + T_{c}}{2} \]  

(33)

The heat transfer coefficient is thus expressed as:

\[ h_{c} = \frac{\text{Nu}_{T}k}{d_{h}} \]  

(34)

From the geometry of figure 1, the thermal resistance of the heat sink is readily expressed as:

\[ R_{t hs} = \frac{1}{n} \left\{ \frac{1}{h_{c}A_{f}} + \frac{1}{\frac{1}{2}L_{kal}} \left( \frac{1}{h_{c}A_{f}} + \frac{1}{\frac{1}{2}L_{kal}} \right) \right\} + \left( \frac{d}{nA_{b}k_{hs}} \right) \]  

(38)

The governing equations in 2-D solved by the ANSYS Fluent code are defined by the continuity equation defined by equation (39), momentum equation defined by equation (40) and energy equation defined by equation (41).

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

(39)

\[ \frac{\partial u^{2}}{\partial x} + \frac{\partial vw}{\partial y} = \frac{v\partial^{2}u}{\partial y^{2}} \]  

(40)

\[ \frac{u\partial T}{\partial x} + \frac{v\partial T}{\partial y} = \alpha \frac{\partial^{2}T}{\partial y^{2}} \]  

(41)

Where

\[ \alpha = \frac{k}{\rho C_{p}} \]  

(42)
3. Methodology

The power dissipation of the IRF 3205 MOSFET was derived and fed as input to the generated heat sink model. The heat sink was optimized using MATLAB optimization tool, FMINCON, to obtain the optimal parameters of the heat sink. The optimized heat sink parameters are: heat sink length equal to 0.048m, base thickness equal to 0.0030m, fin length equal to 0.020m, fin width equal to 0.1m, fin spacing equal to 0.002m, fin thickness equal to 0.0030m and number of fins equal to 10. The thermal performance of the heat sink was done using ANSYS computational fluid dynamics. A test rig consisting of a 2.5 K.V. A, 12 VDC power inverter was assembled and experimental testing of the optimized heat sink was demonstrated at different load conditions. Figure 5 shows the experimental setup.

Temperature data were recorded with a K-type thermocouple and a three channel temperature logger, MTM-380SD, with real time data logger.

Figure 6 shows geometric model of the optimized heat sink using Design Modeler in ANSYS 15.0 Work Bench. While Figure 7 shows the Grid Refinement test image. The dimensions of the model are; heat sink length equals 0.048m, base thickness equals 0.003m, fin height equals 0.02m, heat sink width equals 0.1m, fins spacing equals 0.003m, fin thickness equals 0.003m and number of fins equals 10.

The flow is assumed to be steady, laminar and incompressible. The cooling fluid is air with constant thermo-physical properties. Heat transfer within the geometry is a combination of conduction and convection heat transfer. The mesh was generated automatically by the ANSYS mesh tool while the orthogonal quality and skewness were monitored to give better mesh quality than the expected limits. Grid refinement tests were carried out and results of maximum temperature on the heat sink base converged when the number of elements was 914,534. The governing equations in 2-D solved by the ANSYS Fluent code with their boundary conditions are defined by the continuity equation defined by equation (39), momentum equation defined by equation (40) and energy equation defined by equation (41).
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{39}
\]

\[
\frac{\partial u^2}{\partial x} + \frac{\partial w}{\partial y} = \frac{\nu \partial^2 u}{\partial y^2} \tag{40}
\]

\[
\frac{u \partial T}{\partial x} + \frac{v \partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2} \tag{41}
\]

Where

\[
\alpha = \frac{k}{\rho c_p} \tag{42}
\]

The cooling fluid is air with an inlet temperature of 27°C, symmetry boundary conditions are specified.

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### Figure 6: Modelled Heat sink Geometry

### Figure 7: Grid refinement image

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### 4. Results and Discussion

#### 4.1 Simulation Results using ANSYS FLUENT Computational Fluid Dynamics Software

Simulations results produced by ANSYS FLUENT solver are presented to determine the performance of the heat sink at different load conditions. The ANSYS model employed a dense mesh containing 12000 nodes, 234 temperature monitoring points and 56 time steps to ensure accurate results. The inputs to the solver are the dissipated power, air flow velocity, ambient temperature and pressure drop termed the boundary conditions.
4.1.1 Numerical Result at a Pulse Load of 300 W

Figure 8 shows the temperature contour plot as obtained from the simulation result of the optimized heat sink at pulse load of 300 W.

The thermograph represents the temperature distribution across the surface of the heat sink, it can be seen that the hottest point is at the heat sink base and has a maximum temperature of 307 K.

4.1.2 Numerical Result at a Pulse Load of 460 W

Figure 9 shows the temperature contour plot as obtained from the simulation result of the optimized heat sink at pulse load of 460 W in ANSYS.

The thermograph represents the temperature distribution across the surface of the heat sink, also it can be seen that the hottest point is at the heat sink base and has a maximum temperature of 319 K which is an increase of 3.9% when compared to the pulse load of 300 W.
Figure 9: Temperature contour plot at 460 W Pulse Load from ANSYS simulation

4.1.3 Numerical Result at a Pulse Load of 600 W

Figure 10 shows the temperature contour plot as obtained from the simulation result of the optimized heat sink at pulse load of 600 W.

Figure 10: Temperature contour plot at 600 W Pulse Load from ANSYS simulation
The thermograph represents the temperature distribution across the surface of the heat sink, again it can be seen that the hottest point is at the heat sink base and has a maximum temperature of 328 K which is a further increase of 2.8% when compared to the pulse load of 460 W.

4.1.4 Numerical Result at a Pulse Load of 1015 W

Figure 11 shows the temperature contour plot as obtained from the simulation result of the optimized heat sink at pulse load of 1015 W.

The thermograph represents the temperature distribution across the surface of the heat sink, clearly it can be seen that the hottest point is at the heat sink base and has a maximum temperature of 357 K which is 8.1% increase when compared to the pulse load of 600 W.

From the results of the numerical simulation, as the pulse load increases, the maximum temperature at the base of the heat sink increases. This is as expected as the dissipated power increases with pulse load due to the increase in the drain current in the IRF 3205 MOSFET, for all load conditions the maximum junction temperature of the switches are not exceeded.

4.2 Comparison between Experimental and Numerical Results

The experimental data plots were compared to the numerical results obtained using the ANSYS FLUENT solver. Figure 12 shows the experimental data plots of temperature against time at various inverter load conditions of 300 W, 460 W, 600W and 1015 W respectively. The MTM-380SD used for temperature data collection has a resolution of 0.1°C and an accuracy of ± 0.5% + 0.5°C.
Figure 12: Temperature -Time graphs at various Inverter Loads

In Table 1, a summary of results obtained numerically and from experiments are presented and from the tables one can see the percentage increase in temperature as the pulse loads increases. Results obtained show that the percentage deviation between the ANSYS simulation and experimental temperature for a pulse load of 300W is 8%, for a pulse load of 460W is 3%, for a pulse load of 600W is 8%, for a pulse load of 1015W is 2% as shown in Table 2. Thus the experimental data closely validates the simulated results. Table 2 also presents the relative error between the numerical and experimental results calculated relative to the ambient temperature.

In all test cases of pulse loads, the maximum junction temperature of the IRF 3205 MOSFET switch is not exceeded and the power inverter can thus be safely and reliably operated. This shows that the optimised heat sink was able to contain the heat generated during the switching and conduction cycle of operation of the switches.

Table 1: Comparison between temperature increase for different pulse loads

| S/N | Load /W | Numerical results | Experimental results |
|-----|---------|--------------------|---------------------|
|     |         | Steady-State Temp (°C) | % Increase in Temp. | Steady-State Temp (°C) | % Increase in Temp. |
| 1   | 300     | 34                 | 35.3               | 36.9                  |
| 2   | 460     | 46                 | 19.6               | 59.7                  | 25.9                 |
| 3   | 600     | 55                 | 52.7               | 85.4                  | 43                   |
| 4   | 1015    | 84                 |                     |                       |                      |
Table 2: Percentage difference in results obtained numerically and experimentally

| S/N | Load (W) | Numerical Steady-state temp. (°C) | Experimental Steady state temp. (°C) | % Deviation | Relative error (%) |
|-----|----------|-----------------------------------|-------------------------------------|-------------|-------------------|
| 1   | 300      | 34                                | 36.9                                | 8           | 34                |
| 2   | 460      | 46                                | 47.4                                | 3           | 7                 |
| 3   | 600      | 55                                | 59.7                                | 8           | 15                |
| 4   | 1015     | 84                                | 85.4                                | 2           | 2                 |

5. Conclusion

Power inverters are essentially for supplying energy for today’s society in a more efficient, sustainable and controllable manner. Thermally induced failure modes are the main cause of power inverter reliability issues. With the aim of efficiently dissipating the heat generated from the power inverter, a heat sink model was then developed and optimized using the MATLAB FMINCON optimization tool. The resulting optimized heat sink variables are: fin spacing equal to 0.005m, fin height equal to 0.02m, number of fins equal to 10, fin thickness equal to 0.003m, heat sink length equal to 0.1m, heat sink base thickness equal to 0.004m, from which width of the heat sink equal to 0.075m, fin spacing ratio of 0.6667. The static pressure of the fan is obtained as 21.7113N/m² and from the fan characteristic curve; the volume flow rate of air is 0.03065m³/s and the selected fan model is 2123HSL with dimensions 120mm square by 38mm. Thermal performance of the optimised heat is carried out using ANSYS Computational Fluid Dynamics (CFD) software. The maximum simulated and experimented recorded temperatures are 84°C and 85.4°C. The results obtained from simulations and experimental data are in good correlation with a maximum of 8 percent deviation at a load of 600W.

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