Heat Transfer Enhancement by Finned Heat Sinks with Micro-structured Roughness

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Abstract. We investigated the benefits of micro-structured roughness on heat transfer performance of heat sinks, cooled by forced air. Heat sinks in aluminum alloy by direct metal laser sintering (DMLS) manufacturing technique were fabricated; values of the average surface roughness $R_a$ from 1 to 25 microns (standard milling leads to roughness around 1 micron) under turbulent regimes (Reynolds number based on heating edge from 3000 to 17000) have been explored. An enhancement of 50% in thermal performances with regards to standard manufacturing was observed. This may open the way for huge boost in the technology of electronic cooling by DMLS.

1. Introduction

The ability of electronic devices to dissipate heat is an important aspect in electronics. Electronic cooling represents a commercial sector where a variety of solutions has been explored [1]. Skipping the very little performative solutions based on natural convection, thermal management of most of electronic devices, desktop and notebook computers to name a few, is based on forced convection of air as refrigerant, and usually rely on chip-attached or adhesively bonded extruded aluminum heat sinks [1, 2].

Concerning cooling of high power devices as high performance supercomputers, high thermal fluxes have already driven the development of liquid-based cooling loops. However, despite higher heat fluxes expected by these solutions, it is difficult to imagine a wide spread of the latter technology in notebook computers, which will remain dominated by cooling based on forced air convection. Thermal performance of the air cooled heat sinks must be further improved due to ever-increasing thermal challenges arising in next-generation electronics systems [3]. Hence, in the present paper, we focused on forced (single-phase) convection only.

In particular we investigated the capability of direct metal laser sintering (DMLS) manufacturing technique to produce micro-structured rough surface heat sinks [4].

Usually heat sinks have been produced by milling manufacturing technique; surface roughness of those heat sinks is very low, around 1 micron. Through DMLS we are able to produce heat sinks whose surface roughness can vary over a wide range of values. This point is very important because the roughness strongly influences the convective heat transfer. The ability of roughness to enhance heat transfer in fully developed flow has been studied for many years. Theory and experimental correlations have been developed for this phenomenon. Early works focused on close packed, granular...
surface roughness\cite{5,6}. On the other hand, artificial-roughness-based solutions designed to enhance heat transfer (e.g. repeated ribs) have been studied\cite{7,8,9}. In the latter cases, usually geometrical structured patterns are designed on the surface experiencing convective heat transfer. Recently it has become evident that the roughness has an even greater impact at the leading edge (or entrance region) of heat transfer surfaces, where the thermo-fluid dynamic structures (namely velocity and thermal boundary layer) start growing and they are very thin, thus they are more sensitive to structures that modify the flow field. This is the case of electronic cooling applications, because in this field geometric dimensions involved are usually very small and therefore the fraction of heat transfer at leading edge is far from negligible. Therefore we expect heat sinks produced by DMLS have better performance in heat dissipation than traditional those produced by milling.

In this work, through an experimental analysis in a wind tunnel, we compared the thermal performance of surfaces obtained by traditional milling to those obtained by DMLS. In particular two geometry will be tested, a flat heat sink and a finned heat sink (see figure 1). For each setup, the average convective heat transfer coefficient $h$ of two samples (with regards to the bare surface exposed to the air) was measured: A milled copper sample and an aluminum alloy (AlSi10Mg) sample produced by DMLS.

![Figure 1. Longitudinal view of (a) flat and (c) finned heat sink; isometric view of (b) flat and (d) finned heat sink.](image)

2. Direct Metal Laser Sintering process

The DMLS additive manufacturing process involves the use of a 3D CAD model of the part to product whereby a .stl file is created; then the required support structures are generated with the help of a dedicated software like Magics (from Materialise). The main functions of the supports structures are to hold unsupported geometries in place and to prevent the distortion of the part during fabrication, to fix the part to the building platform and to conduct excess heat away from the part. The complete built file, including parts and supports, is then sliced into layers with a chosen thickness (30 µm) and sent to the DMLS software. In an early stage the powders are sieved and put inside the dispenser in the building chamber filled with inert gas (Ar) in order to reach a level of oxygen lower than 0.1%. Once the process has started, the metallic powder is deposited with a stainless steel recoater blade on a building platform of similar material.

In this study a DMLS machine from EOS M270 Xtended machine was employed. This machine uses a 200W Yb-fiber continuous laser beam with the wavelength of 1060 – 1100 nm. The focused diameter of laser beam is 0.1 mm. The maximum scan speed is 7000 mm/s. The building volume of the machine is 250 mm x 250 mm x 215 mm. The high-powered Yb-fiber optic laser fully melts the powder particles of the sliced .stl file at each layer. The substrate platform then drops one layer thickness in the z axis before the material is recoated, and the process is repeated until the entire build is complete. At the end, the building platform with the parts is removed from the building chamber and it is put into a furnace for a stress relieving treatment: for the Al alloy employed in this work this means 2 hours at 300°C. Only after this the supports can be removed and the parts detached.
The part is fabricated with different exposure DMLS process parameters for core, downskin, upskin and contour respectively [4]. In detail, downskin refers to the first two layers at the base of the part, while upskin refers to the last three layers at the top of the part. Core represents the remaining layers between the downskin and the upskin of the part. For each layer, after the hatching of the core region is completed, contour exposure is applied with lower laser power in order to improve the surface finish of the sintered part. The scan lines of every layer within the core region are rotated by 67° with respect to the scan lines of the preceding layer, as illustrated in a previous study [10].

Starting from the results obtained on the optimization of process parameters on surface finish of AlSiMg sample produced by DMLS, values which can modify and increase the surface roughness were chosen and two samples were manufactured.

Two samples produced by DMLS are analysed in this study. First sample dimensions are 11.1mm x 11.1mm x 5mm. The upper face is the surface exposed to the air flow, and the one on which the convective heat transfer takes place, while the other surfaces are isothermal. Second sample dimensions are 11.1mm x 11.1mm x 5mm, with a fin built on the upper face. Fin measures 11.1mm x 10mm x 2mm. In figure 2 are shown the flat and finned heat sink obtained by DMLS and the corresponding copper ones. The samples were characterized by a 3D optical scanner ATOS Compact Scan 2M (GOM GmbH), and a numerical reconstruction of the flat heat sink surface is shown in figure 3. Surface roughness of copper samples is 1 micron, while surface roughness of flat and finned heat sink is 25 and 18 micron respectively.

![Figure 2](image1.png)  
![Figure 3](image2.png)

**Figure 2.** Picture of milled copper and DMLS Aluminum alloy heat sinks (a) flat (b) finned.

**Figure 3.** Numerical reconstruction of DMLS flat heat sink surface: z-axis has been magnified to show the nature of roughness obtained by DMLS process.
3. Experimental setup and procedure

We developed a sensor for the measurement of convective heat transfer coefficient. This sensor is based on the key idea of using a thermal guard. A heater in the bottom of the sample generates a controlled heat flux. The sample is surrounded by temperature controlled “guard” made of copper. This guard is kept at the same temperature of the sample, to force the heat to flows into the sample only.

This ensures that the heat flux through the sample is essentially one-dimensional, going from the heater to the air flow passing through the sample itself. Temperature are measured by thermocouples in the sample and in the upstream and downstream (referring to fluid flow) part of the guard. The upstream and downstream part of the guard experience different values of convective heat transfer coefficient, due to leading edge effect. Hence, the guard will be slightly non isothermal, and some (undesired) temperature gradient between the sample and the guard (of the order of 0.2 K) could arise. For this reason a thermal insulating shield made of Teflon is placed between sample and guard. Its function is to minimize parasite heat flux due to residual temperature gradients between sample and guard, in order to make them negligible.

The main portion of the power $P$ provided by the heater to the sample is transferred into the air flow by convective heat transfer, while a minor, but far from negligible, portion of $P$ is transferred to the wind channel walls through radiative heat transfer. Denoting by $Q_c$ and $Q_r$ the convective heat flux and the radiative heat flux respectively, the former could be calculated using the following equation:

$$Q_c = P - Q_r = P - A \varepsilon \sigma (T_s^4 - T_w^4)$$

where $A$ is the sample area, $T_s$ is the sample temperature, $T_w$ is the wind channel temperature, $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2/\text{K}^4$ is the Stefan-Boltzmann constant, $\varepsilon$ is the sample emissivity.

The sample emissivity has been estimated using an Infra-Red thermal camera. Values of 0.28 and 0.20 have been measured for Copper and aluminum alloy sample surface emissivity respectively.

The average temperature values of air and wind tunnel walls are 300 K and 299 K respectively. Their values are quite the same for all the tests. On the other hand $T_s$ assumes very different values in each test, varying in the range of 307 $\pm$ 335 K. Consequently also $Q_r$ assumes different values in each test. However, the maximum $Q_r$ experienced in the tests is less than 6% of the overall power $P$ provided by the heater.

The flow loop of the experimental system is shown schematically in figure 4. The sensor was mounted on one side of a horizontally oriented rectangular flow channel, which realizes a small open-loop wind tunnel. The channel has a smooth inner surface, a section of 228 mm x 158 mm (hydraulic diameter $D_h = 187$ mm) and an entrance length of 5 m (corresponding roughly to 26 hydraulic diameters). At the end of the channel, downstream from the test section, a vein anemometer was installed for measuring the axial velocity. The inner wall and air temperature are measured at the same location where the anemometer is located.
4. Results
Two different test cases were analyzed: case A and case B refer to the flat and finned heat sinks, respectively. For each test case, the average convective heat transfer coefficients (with regards to the surface exposed to the air flow) of the two samples was compared: A milled copper sample and an aluminum alloy (AlSi10Mg) sample produced by DMLS. Convective heat transfer coefficient was evaluated for different velocities, in order to explore the nature of the heat transfer phenomenon in a wide range of Reynolds numbers. To each value of mean velocity inside the wind tunnel $u$ we associated the following dimensionless Reynolds number $Re_L$, calculated according to the formula:

$$Re_L = \frac{uL}{v}$$

with $v = 1.493 \times 10^{-5} \text{ m}^2/\text{s}$ being kinematic viscosity of air and $L = 20 \text{ mm}$ being the characteristic length of heat source. Conversely, the average convective heat transfer coefficient $h$ was expressed in terms of dimensionless number $Nu/Pr^{1/3}$, whereas:

$$Nu = h \frac{L}{k}$$

is the dimensionless Nusselt number and

$$Pr = \frac{v}{\alpha}$$

is dimensionless Prandtl number with $\alpha = 2.224 \times 10^{-5}$ being the thermal diffusivity of air. To compare DMLS samples and milled samples we introduced the thermal performance enhancement defined as:

$$En = \frac{h_{DMLS} - h_{milled}}{h_{DMLS}} \times 100 \ [\%]$$

4.1. Test case A: Flat heat sinks
We ran 40 tests for DMLS sample and 15 tests for milled copper sample, and extrapolated from experimental data the following fitting curves:

$$Nu/Pr^{1/3} = 0.082 Re_L^{0.669}$$

for milled sample ($Ra = 1\mu$)

$$Nu/Pr^{1/3} = 0.093 Re_L^{0.732}$$

for DMLS sample ($Ra = 25\mu$). In figure 5 experimental points and interpolating curves are shown. Enhancement obtained is quite constant over the whole range of explored $Re_L$, with a minimum and maximum value of 47.7% and 56.4% respectively.
4.2. Test case B: Finned heat sinks

Thermal conductivity \( k \) of copper is more than AlSi10Mg one. This gives no effect on flat heat sinks, while in case of finned samples it is necessary to check if fin efficiency, due to conduction into the fin, may spoil the heat transfer enhancement or not. Fin efficiency \( \eta \) is calculated by means of the following formula [2]:

\[
\eta = \frac{\tanh(m \times L_{fin})}{m + L_{fin}} \tag{8}
\]

\[
m = \left( \frac{2h}{k} \right)^{-1} \tag{9}
\]

where \( L_{fin} = 10 \text{ mm} \) is fin length, while \( t = 2 \text{ mm} \) is fin thickness. \( k_{Cu} = 388 \frac{w}{m \cdot ^{\circ}K} \) while \( k_{AlSi10Mg} = 150 \frac{w}{m \cdot ^{\circ}K} \). To calculate fin efficiencies in equation (9) we chose the maximum value of \( h \) we obtained in the experiment. Therefore, considering the maximum value of \( h \), we calculated the minimum value of the efficiencies of the fin and the maximum gap between the efficiencies of the two finned samples (\( \Delta \eta \)):

\[
\eta_{copper} = 98.47\% \; \eta_{Aluminum\;alloy} = 96.18\% \; \Delta \eta = 2.29\%
\]

This choice of \( h \) allow us to consider the most conservative case. Since \( \Delta \eta \) is very small (2.29%), we can assume that the conductivity disparity of the two materials under study does not affect heat transfer performance of samples. We ran 13 tests with both DMLS and milled sample, then we extrapolated from experimental data the following fitting curves:

\[
Nu/Pr^{1/3} = 0.124 Re_L^{0.722} \quad \text{for milled sample (} R_a = 1\mu \text{)} \tag{10}
\]

\[
Nu/Pr^{1/3} = 0.153 Re_L^{0.720} \quad \text{for DMLS sample (} R_a = 18\mu \text{)} \tag{11}
\]
In figure 6 we report the experimental results, the fitting curves and the thermal performance enhancement. Clearly the latter is nearly constant all over the explored Reynolds number range, and it amounts to 20%. It is very important to point out that fin efficiency is a parameter concerning conduction into the material only. Therefore it is plausible that the values of convective heat transfer coefficient measured on two samples are very different although the fin efficiency of the two samples is approximately equal.

4.3. Comments

It is important to know the range of velocity and Reynolds number values for industrial application heat sinks. Comparing the latter with the range of the Reynolds number explored in our experimental campaign, it is possible to predict if the enhancement in heat transfer showed in our study will be achieved also when DMLS rough heat sinks works under fluid dynamics conditions typical of industrial applications.

Length of flat heat sinks or finned heat sinks basis plate usually vary from 2 to 15 cm, depending on the heat flux to dissipate. On the other hand velocity of cooling air for finned heat sinks electronic devices usually ranges from 1 to 5 m/s [11]. Consequently, according to equation (2), in the most of industrial electronic cooling devices $Re_L$ varies from $1.3 \times 10^3$ to $5.0 \times 10^4$. Hence we can argue that the experimental results shown in this work have been ran in a range of fluid dynamic conditions which are of interest for electronic cooling devices for industrial applications. Consequently it is possible to suppose the enhancement in heat transfer through DMLS manufacturing showed in the present work will be achieved also if DMLS rough heat sinks will be applied to common industrial electronic cooling devices.

It is appropriate to consider the problem of pressure losses in the electronic cooling device air circuits. Usually heat sinks are designed in order to obtain an optimum trade-off between heat transfer performance and pressure losses. In this study pressure losses associated to tested heat sinks have not been characterized. However it is reasonable to expect an increase in pressure losses when a rough heat sink is used instead of a smooth one. In spite of the presence of heat sinks is a source of pressure losses, it should be taken into account that a big portion of pressure losses in air circuits are due to other component rather than heat sinks. Pressure losses due to other component remain approximately constant, while varying type of heat sink. As usually done, pressure losses can be subdivided in distributed pressure losses along channel length and concentrated pressure losses due to fan intake, bends and restriction along the circuit, to name a few. Reynolds analogy states that in first
approximation $Nu_{HS} = f_{HS}$, where $Nu_{HS}$ and $f_{HS}$ are the dimensionless heat and momentum transfer coefficient on the heat sinks respectively.

Increasing $Nu_{HS}$ proportionally increase the dissipating heat flux, being heat sinks the component of the circuit which mainly experience heat transfer phenomenon, whilst increasing $f_{HS}$ proportionally increase only pressure losses associated to heat sinks, while pressure losses due to other component remain constant. Hence we can suppose that, when a rough heat sink is used instead of a smooth one, the benefit in enhancing heat transfer should be more effective than the negative effects of increasing pressure losses.

In this work, the effect of DMLS process surface roughness on thermal performance of air cooled heat sinks has been investigated. Nevertheless there are several technologies for electronic cooling, other than air cooled heat sinks. For example, confined jet impingement, heat pipe and so on. It is interesting to compare the presented technique with others, in order to prove the relevance of the obtained results in the electronic cooling research field. In general, air cooled heat sinks are designed to take advantage of extended surfaces, for example plate fins or pin fins. Jet impingement can also be incorporated into the design to further enhance the heat transfer. Because available space is limited and low power consumption is desired, the heat sink performance becomes the focus of many studies. In fact the latter is a widespread method in computers and CPU cooling due to its low price, effectiveness and reliability [12]. Recently, the class of techniques based on liquid or two-phase coolant fluids, as heat pipes to name one, became very attractive. Heat transfer mechanism in heat pipe combines the principles of both thermal conductivity and phase transition to tremendously increase heat transfer rate [12]. Despite this class of techniques allow to dissipate very high heat fluxes (compared to air cooled heat sinks) they are still relatively expensive. Hence, despite the common belief that the limits of air-cooling have been reached, yet a number of interesting approaches can be found in the literature. For example in [13] an innovative transient thermal management solutions that utilize PCMs (phase change materials) to design hybrid PCM-based air cooled heat sinks is presented. It provides a solution to deal with peak transient heat loads in portable applications exploiting the ability of PCMs to store energy in the form of latent heat.

We can reasonably suppose that a lot of electronic applications (like notebooks) will remain dominated by cooling based on forced air convection in the next future. Consequently it is important to find techniques, like the one exposed in this work, capable to improve thermal performance of air cooled devices to face the ever-increasing thermal challenges arising in next-generation electronics systems.

5. Conclusions
In this study average convective heat transfer coefficient of different heat sinks was measured using air as refrigerant fluid in turbulent regime. Copper heat sinks produced by traditional milling were compared to aluminum alloy (AlSi10Mg) heat sinks produced by DMLS. Effects of micro-structured roughness on thermal performance were shown. These are very huge because little geometrical dimensions are involved in heat transfer phenomenon (see leading edge), as usual in electronic cooling. Our results offer an evidence of the possible impact of DMLS on electronic cooling since a 50% and 20% enhancement (compared to milled samples) is observed for flat and finned heat sinks, respectively. We also show that the lower thermal conductivity of aluminum alloy plays a negligible role on heat transfer performances in our setup. Enhancement on flat heat sinks is larger than finned sinks, because average roughness of the former is higher than that of the latter. Building rough fins is not as easy as doing rough flat surfaces, but in future studies this will be one of the first improvements that will be pursued from the point of view of DLMS machining. These results open the way for a huge boost in the technology of electronic cooling by DMLS. As an outlook, based on the present results, we plan further studies focusing on investigation and optimization of micro-structured surfaces obtained by DMLS with regards to heat transfer.
6. References

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