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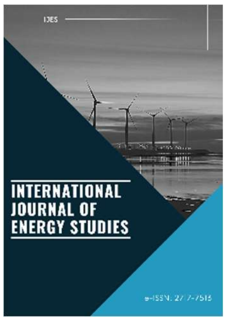

Research Article

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Thermal management with double layered heat sink produced by direct metal laser sintering

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Highlights

- Double Layered Heat Sink is useful for uniform temperature distribution.
- Heat treatmet is essential for Additive Manufacturing technology to have sufficient cooling with low flow rate.
- Surface roughness may have a considerable effect on the pressure drop as Dh goes to low values.
- AM technology can enable 0.3 mm fin space for hig flux cooling.

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ABSTRACT

In this study, the thermal management of a heat sink which is the heart of a cold plate is investigated considering Additive Manufacturing (AM) technology. Although AM enables many flexibilities, it also brings some limits to design and manufacturing. In general, a cold plate is developed to cool many heat sources with an appropriate heat sink design just under the sources to have a maximum cooling rate. Due to the nature of Direct Metal Laser Sintering (DMLS) AM technology, we have to use some supports (used as pin fins in this study) to avoid material collapse to have a rugged and fine product. In this study, straight rectangular channels with the production limit of AM technologies have been investigated in detail and the results are compared to investigate the heat treatment effect for the same design. Due to its cooling advantages in a confined space, a double-layered heat sink is chosen as the heat sink geometry to reach the goals. In order to succeed in electronic performance requirements, two heat sources on each side of the plate (heat sink) must be so cooled that their temperatures are close to each other. This is one of the requirements in radar electronics to have higher performance and to decrease the necessary calibration efforts. Deliberately, the heat load is selected as 100W/cm^2 using mini or smaller channel sizes to enforce the limits of AM technology. One of the most important aspects of a cold plate design is to distribute the fluid to the heat sources to collect it properly without sacrificing electronic components and use the space effectively. As a result; The Double Layered Heat Sink design allows temperature difference between the heat sources to be kept around 1°C with 100W/cm^2 heat flux.

Keywords: Double layered heat sink, Electronic cooling, Additive manufacturing, Selective laser melting

1. INTRODUCTION

Liquid cold plate design and manufacturing is a crucial stage in the thermal management of electronics due to possible leakage problems. Already, there are many bonding technologies such as brazing and friction stir welding that are frequently used to have a leakage-proof product to withstand the high-pressure load. The issue can be more critical as the number of cold plates is considreable. In the design of a phased array radar antenna, there can be many cold plates next to each other to accomplish electronic performance. After thermal and Radio Frequency (RF) performance are satisfied, compactness and lightweight become the most important issues. Since the radar rotates around itself, it requires a minimal pressure drop and flow rate, which is mainly transmitted through slip rings that have very tight limits to prevent leaks. Many studies [1-7] have investigated the double-layered (DL) fin both numerically and experimentally. Zhai et al. [1] conclued that By properly adjusting the channel height, DL-MHCSs can reduce their pressure drop. As DL-MCHSs have a lower irreversibility and uniform temperature distribution on the bottom wall, they can effectively eliminate microelectronic equipment's internal thermal stresses. DL-MCHS are therefore an alternative cooling method. Gongnan et al. [2] In their study, three types of liquid-cooled double-layer microchannel heat sinks, including rectangular straight microchannel heat sinks, parallel-flow wavy microchannel heat sinks, and counter-flow double-layer wavy microchannel heat sinks, have been designed, and numerical calculations have been conducted to determine the laminar flow and heat transfer. The counter-flow double-layer wavy microchannel heat sink performs better at a higher flow rate and produces a more uniform temperature rise. By coupling a 3D fluid-solid conjugated model with a multi-objective and multi-parameter genetic algorithm optimization method, Wang et al. [7] improved the performance of the previously proposed heat sink. The optimized design reveals a lower pressure drop and more uniform temperature distribution compared with the reference one. The thermal performance and pumping power are both significantly improved, where thermal resistance is decreased by 14.06% and 16.40% reduction is obtained in the pumping power. Ansari and Kim [8] used DL different types of heat sinks to cool the distributed heat sources with different heat loads. In their study, parallel flow, counter flow, and cross flow are investigated in detail. Thermally, crossflow MCHS (Micro Channel Heat Sink) overcomes the other two choices. Dupuis et al. [9] studied cold spray additive manufacturing technologies to investigate the thermal and flow structure in pin fin arrays. They have used three different pin fin shapes, cylindrical, diamond, and square. They also investigated the inline and staggered configurations for fin densities of 8 fpi (fin per inch) and 12 fpi. They have concluded that results (both thermal and hydrodynamics) had very varied with Re number. In their study [10], microchannel pin fin arrays were manufactured with Laser Powder Bed Fusion and compared to studies of pin fin arrays that were traditionally manufactured, which showed a much lower level of surface roughness on the pin and endwall surfaces. A comparison of smooth pin fin arrays from literature with rough pin fin arrays from the study revealed that high surface roughness had a stronger effect on friction factor enhancement than on heat transfer enhancement when compared to smooth pin fin arrays. Micro-scale fin arrays for compact offset strip fin heat exchangers produced with AM technology are examined by Oligee et al. [11]. Additional flow area is expected to reduce inertial pressure drop as the working fluid traverses between subsequent columns of strip fins by implementing small gaps between subsequent strips. By adding small gaps between columns of fins between columns of fins, pressure drop is reduced in AM compact heat exchanger designs while heat transfer performance is minimally affected. Insights from their numerical studies are expected to guide future experimental studies to design, print, and test AM heat exchangers for electronics cooling applications.

In this study, double layered heat sink manufractured using AM technology is used to cool phased array radar antenna where both space is limited and uniform temperature among the hot spots and in the heat sources are important. The novelty of this study is that the same liquid fort he primenary and secondary heat sink is used. The effect of warm liquid is compansated by means of using higher fin areas and smaller hydraulic diameter.

2. DOUBLE LAYERED HEAT SINK DESIGN WITH AM

In this study, a double-layered heat sink design is performed to cool heat sources on both sides of the plate to have a compact and lightweight design. Ethylene Glycol Water (EGW) mixture is used as the coolant which is mostly preferred due to its outstanding thermophysical properties [12]. However, it is an electrically conductive liquid, therefore leakage-free design is a must at the manufacturing stage. In the design of the heat sink, to have simplicity and high thermal performance with low-pressure drop, straight rectangular fins with a high aspect ratio are used. In Fig.1 & Table 1, it is illustrated that primary (upper heat sink, PHS) heat sinks cool the hot spot at the top surface, while secondary (lower heat sinks, SHS) heat sinks cool the hot spot on the other side (bottom) of the plate and allow a short return line. There are also tip fins at the returning region to have higher cooling performance, which causes an increase in the complexity level for other manufacturing but serves as additional support for AM technology. In PHS, fin space (Wc) is selected as 0.5 mm to have a small pressure drop and to produce it easily. However, for SHS, warm fluid is used to cool the same hot spot. Therefore, to increase the thermal performance, fin density is increased to a level that forces the limits of AM technology with a fin space of 0.3 mm.

The aim of the study is to bring the temperature of both heat sources as much as close to each other. This value (ΔT=T2-T1) can be expected as ±1 °C. In Table 1, all dimensional details are given in a tabular form. SHS has a higher wetted fin area (A_w=960 mm^2>720 mm^2) and smaller hydraulic diameter, (D_h=3<3.75mm). Therefore, both the conduction and convection resistance of SHS will be smaller than that of PHS. One of the important issues of the present case is that the pressure drop value of the whole system should be as small as to keep the cooling power consumption low, (Pumping power, PP=*Flow rate x Pressure Drop)*.

In addition, the aspect ratio is determined as high as possible to have a higher Nu number in the rectangular channel for fully developed flow and low pressure drop. The total height of the cold plate is determined as H=10 mm, which is the height of heat sink. It can be said that it is an acceptable value for both thermal performance and structural integrity requirements. The cooling liquid is selected as EGW due to its superior thermophysical properties. The constant values: ($\rho = 1073\frac{kg}{m^3}, \mu = 0.00401302\frac{\mathrm{kg}}{\mathrm{m.s}}, C_p = 3180\frac{J}{kg.K}, k = 0.388\frac{W}{mK}$ of EGW at the inlet temperature of 30°C have been used for the analysis.

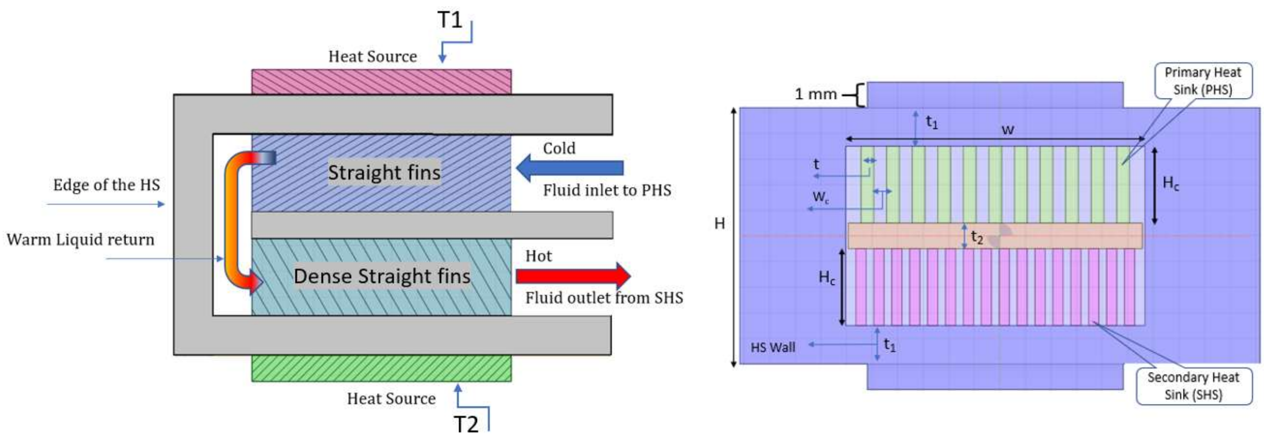


Figure 1. a) Illustration of DLHS for effective cooling in cut plane b) dimensional details of both upper and lower fin design for thermal analysis

Table 1. Double-Layered Heat Sink Parameters for the CFD Analysis

	W	t_1	t	t_2	Wc	H_c	D_h	AR (H_c/W_c)	N #fin	Base area	Total fin area+ Base area
Upper HS (PHS)	0,5	1,5	0,5	1	0,5	3	3,75	6	11	(11.5-11x0,5) x10=60 mm^2	30mm^2X11x2 + 60=720 mm^2
Lower HS (SHS)	0,5	1,5	0,4	1	0,3	3	3	10	16	(11.5-16x0,4) x10=51 mm^2	30mm^2X16x2+51 =960mm^2

3. THERMAL MODELLING AND CFD ANALYSIS

In the cold plate design, aluminum alloys are the most preferred material due to their easy machinability, high thermal conductivity, and lightweight [13]. For 3D AM, AlSi10Mg powder is used to fabricate the heat sink. The thermal conductivity of the material is assumed to be k=119W/mK without the heat treatment process according to the measurement [14]. To emphasize the necessity of the heat treatment process, analyses are also carried out for k=173 W/mK using the same reference. It is the thermal conductivity value reported after heat treatment. AM fabrication is done at the facilities of ALUTEAM ® company located in İstanbul. EOS M 290 machine is used with the DMLS technique with a layer thickness of 30µm.

3.1. Governing Equations and Boundary Conditions

All analyses are carried out for the smooth surface assumption by using the Autodesk CFD 2018® finite element CFD software. In Fig.2, 3D model details are given with boundary conditions. A rectangular prism is selected as the control volume to investigate the thermal and hydrodynamic behavior. At the flow inlet region, x=0, (Fig.2b), volumetric flow (Q_f) is varied from 0.25 to 2 LPM to find out the required flow rate where both temperature values of two heat sources are close to each other keeping the temperature below the target value. Due to the nature of the AM technology, pin fins both at the distribution and collecting pool regions, (31x2 cylindrical pin fins with a diameter of 0.5 mm) are added to prevent possible collapse during manufacturing.

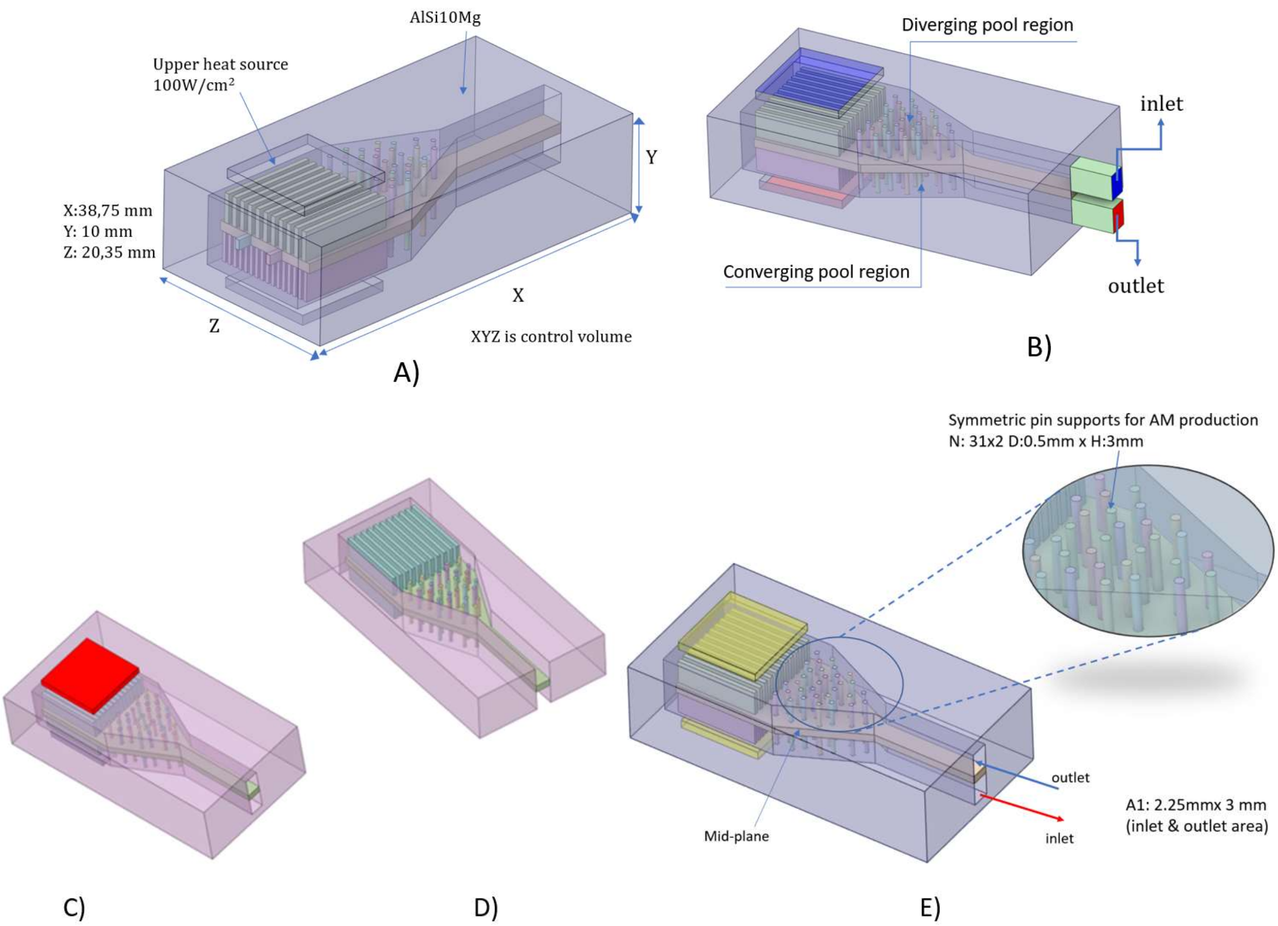


Figure 2. Illustration of DLHS with boundary conditions A) straight fins for cooling and cylindrical fins to avoid collapse and heat sources B) solid modal, D-E) and detailed transparent views

In the flow region, continuity, momentum and energy equation is given in Eq (1-3) respectively.

$$\nabla\vec{V} = 0, \tag{1}$$

$$\rho_f\left(\vec{V}\cdot\nabla\vec{V}\right) = -\nabla P + \nabla\cdot\left(\mu_f\nabla\vec{V}\right) \tag{2}$$

$$\rho_f c_p\left(\vec{V}\cdot\nabla T\right) = k_f\nabla^2 T, \tag{3}$$

Energy equation in solid is expressed as

$$k_s\nabla^2 T = 0 \tag{4}$$

There is no heat transfer through the solid outer surfaces due to the adiabatic boundary conditions. This condition is stated as:

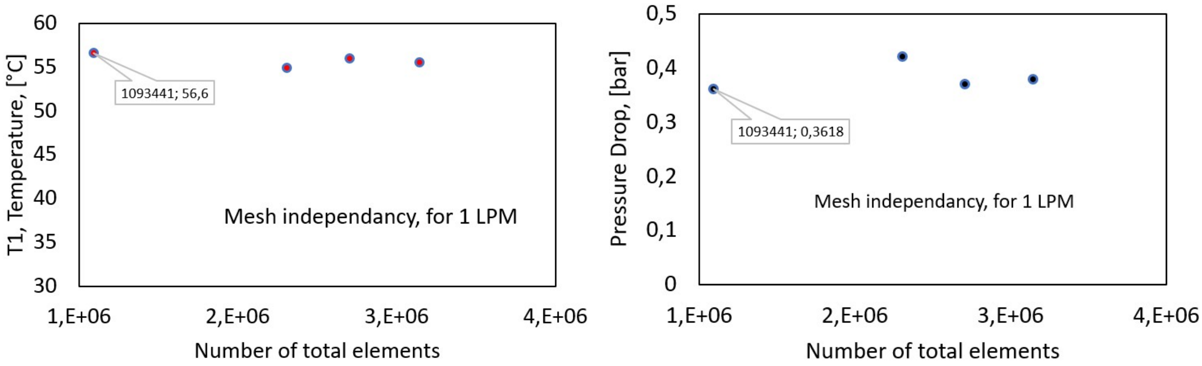
$$k_s \frac{\delta T}{\delta y} = k_s \frac{\delta T}{\delta x} = k_s \frac{\delta T}{\delta z} = 0 \quad (5)$$

In the fluid –solid interface on the domain, the following relationship is valid:

$$-k_s \frac{\delta T}{\delta n} = -k_f \frac{\delta T}{\delta n} \quad (6)$$

3.2. CFD Analysis

In the AM technique, we need to use cylindrical pins as supporters to avoid any possible collapse during the production stage. However, it may be beneficial in terms of heat transfer performance and also enables uniform flow distribution among the fins. However, the penalty of pins is expected to be a small increment in pressure drop. It is known that by using milling machines it is possible to obtain very high surface quality with Ra @ 1-2 µm. On the other hand, with AM technology, this value goes up and internal surfaces have higher roughness (around Ra=15-20µm) which has a direct effect on both thermal and hydrodynamic performance. There are some techniques to decrease the roughness value with post-processing. However, some roughness can be preferred to have higher thermal performance as far as reasonable pressure drop is allowed on the flow line. Upper Heat Source (T1) and Lower Heat Source (T2) are defined as the maximum temperatures of the heat sources on both sides respectively. Their allowed maximum case temperature is 60°C. This value is determined by considering the Mean Time Between failures (MTBF) values of all systems. The thermal conductivity values (k=119 W/mK & 173 W/mK) were analyzed with and without pin fins in order to fully understand the situation. There are four cases for six different flow rates. It is expected that high thermal conductivity will increase cooling performance. Indeed, heat treatment is an important stage in AM technology. It both provides higher strength and higher thermal conductivity. Therefore, in the analysis results it will be seen how essential it is. For CFD modeling, first of all, a mesh independence study is carried out to proceed well. In Fig.3, both the pressure drops through the heat sink and the temperature of the upper source are followed with respect to the number of elements.



A) B)

Figure 3. Mesh independency study to check the CFD model, A) pressure drop vs element number, B) temperature vs element number. The smallest mesh number is selected for all analysis

It is seen that even a coarse mesh gives satisfactory results. Hence, it is selected to do parametric studies. As a turbulence model, k-epsilon is selected. 3 layers on the wall is selected with 0.45-layer factor. Tetra mesh is selected as default.

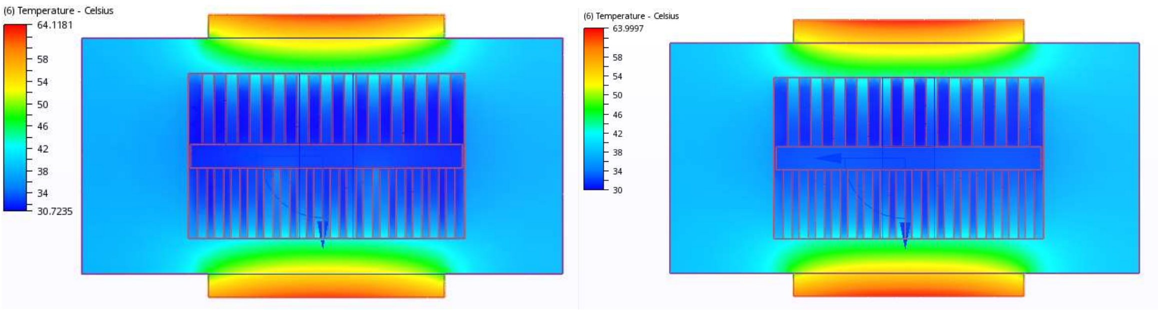
4. CFD RESULTS AND DISCUSSION

In Table 2, numerical analysis results are given for different flow rates to bring the T1 and T2 temperatures close to each other providing sufficient cooling.

Table 2. Maximum temperature values of two heat sources and pressure drop values of heat sink design with/without pin fins depending on the flow rate, k=119 W/mK

Flow Rate [LPM]	T2 [C]	T1[C]	ΔT [C] (T2-T1) WITH PIN FINS	T2 [C]	T1[C]	ΔT [C] (T2 -T1) WITHOUT PIN FINS	Pressure Loss [kPa] WITH PIN FINS	Pressure Loss [kPa] WITHOUT PIN FINS
0,25	79,4	73,7	**5,7**	77,75	73,1	**4,7**	6,69	5,38
0,5	69,5	66,8	**2,6**	68,49	67,1	**1,4**	16,10	13,08
0,75	66,0	64,4	**1,6**	65,22	65,05	**0,2**	27,40	22,41
1	64,1	63,2	**0,9**	63,34	63,96	**-0,6**	40,50	33,31
1,5	62,1	61,8	**0,3**	61,48	62,71	**-1,2**	71,22	58,64
2	60,9	61,1	**-0,2**	60,56	61,96	**-1,4**	116,06	88,77

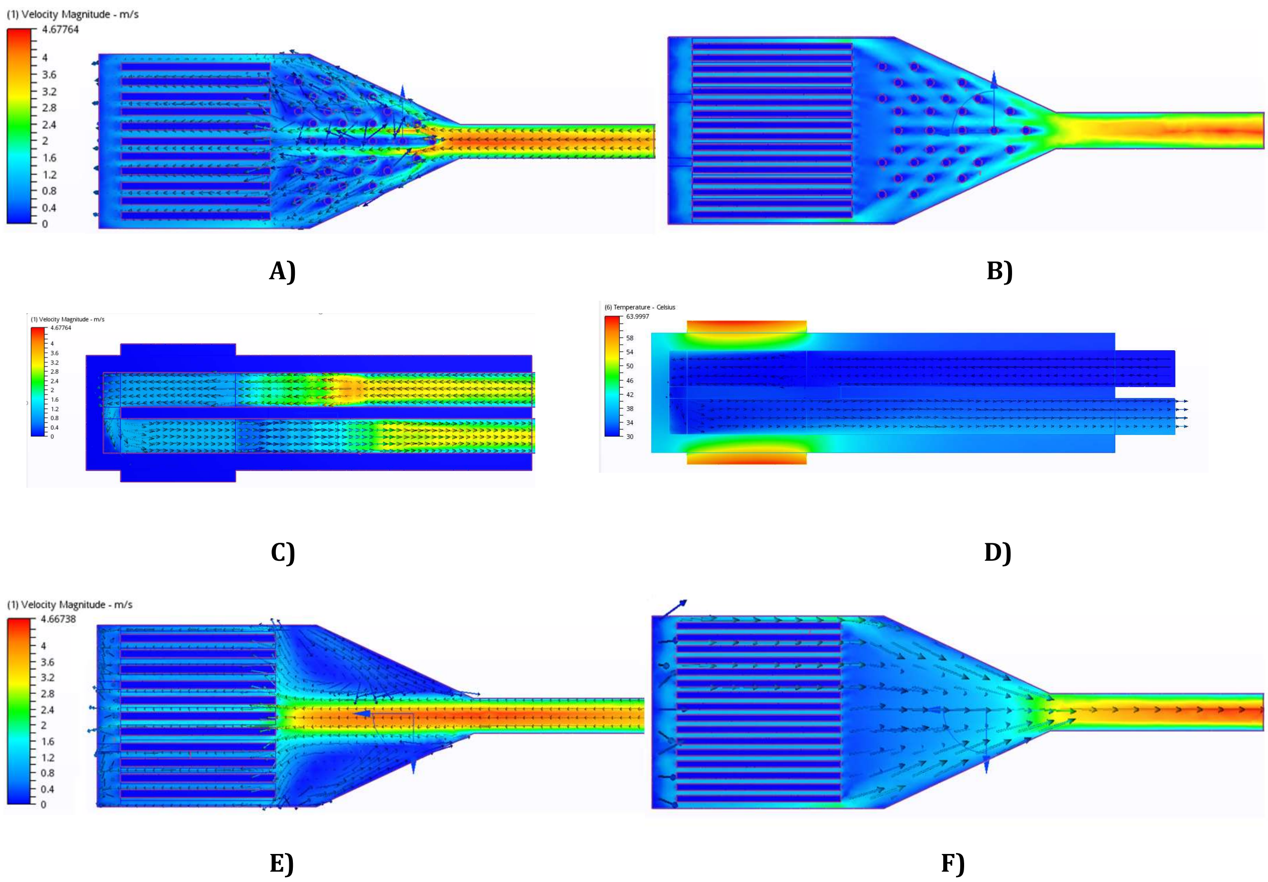
In Figure 4, the thermal distribution of heat sinks at 1 LPM flow rate is given in detail. The figure belongs to the midpoint of the heat source in the flow direction. It is seen that the flow rate of 0.75-1 LPM is good enough to have the smallest temperature difference between the two sources providing sufficient cooling as well.



A) B)

Figure 4. Illustration of temperature distribution for the DLHS design A) with pin fins B) without pin fins at 1 LPM flow rate with k=119W/mK

In Fig. 5, the velocity distribution of the heat sink is given as a sample for both designs with (A-D) and without pin fin (E-H) configuration. Pin fin provides uniform flow distribution for both converging and diverging pool regions. As illustrated in Fig.5A&B, pin fins behave as resistance to flow not to directly entering all fluids into the center zone. Therefore, it seems that flow is more uniformly distributed than that of Fig.5E&F, where pin fins are not used.



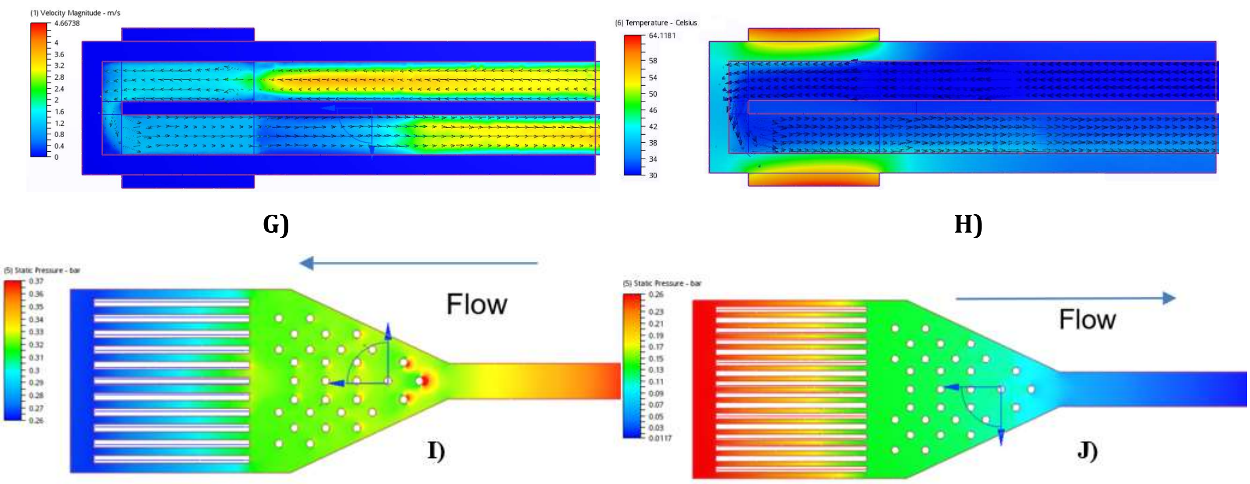
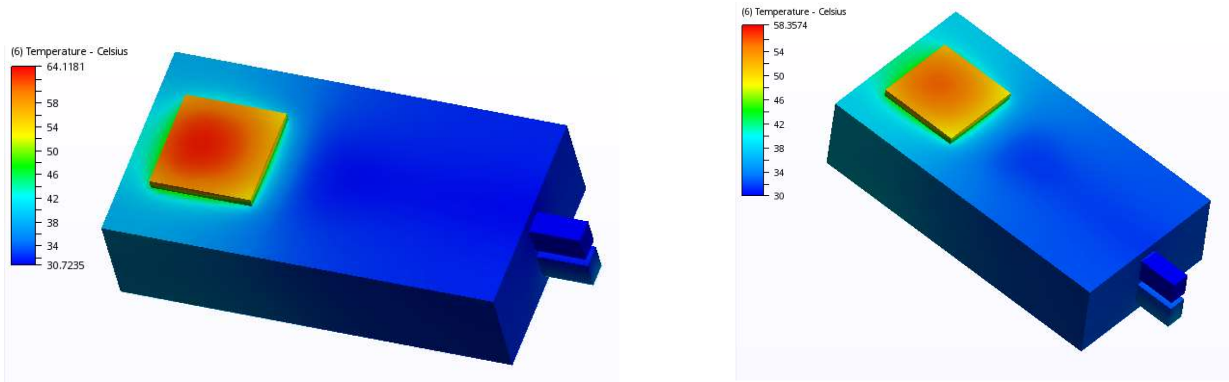


Figure 5. For 1 LPM, Illustration of velocity contour of DLHS A, B, C, D) with pin fins at the inlet midline & return line and side view of the channel, C, D, F, G) without pin at the inlet midline & return line & side view of the channel, and static pressure distribution H) from inlet to the return region and I) from the return region to outlet region

As no pin fin is used, then flow distribution becomes not uniform and some separation for the diverging inlet zone of the pool is inevitable. In Figure 5 I & J, static pressure distribution along the flow line is given from the inlet to the outlet of the heat sink. At the inlet, due to the high speed, static pressure rises in from of the pin fins.

In Fig. 6, the temperature distribution of all bodies for 1 LPM flow rate is given. In Fig.7, the pressure drops of the DLHS design manufactured with smooth surface assumptions have been shared to see the effect of pin fin configuration on both thermal and hydrodynamic performance. As the flow rate increases, the pressure drop through the liquid line goes up and the difference between the pin fin and without fin cases goes to higher values. This was one of the expected features during the design stage.



A) B)

Figure 6. Illustration of temperature distribution for 1 LPM with pin fin configuration, A) k=119W/mK B) k=173 W/mK

In Figure 8, thermal performance curves are illustrated with different interpretations. Temperature difference values are improved with higher flow rate due to the diminishing of caloric heating resistance (Eq.7) and higher convection heat transfer rate (Eq.8) For laminar flow, Nu number (Eq.9) of a rectangular channel for thermally fully developed flow is between 3.96-4.44. Re number (Eq.10) in the fin channel changes between Re=232-2184 depending on the flow rate. This means that for this type of mini channel, the flow regime is laminar and therefore the friction factor does not depend on the surface roughness. However, the situation for parts produced with AM technology can be different due to high relative roughness value. According to Collins et al. studies [13→15], Nu number depends on Re number for Re number lower than Re=2300 and fanning friction factor increases as the Re number increases at Re=600. Therefore, in reality, numerical pressure drop values and thermal performances of the whole heat sink are expected to be higher than calculated.

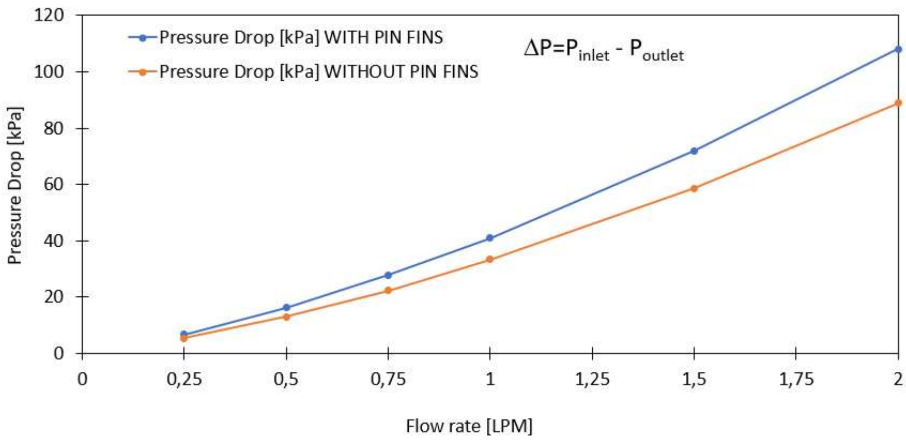


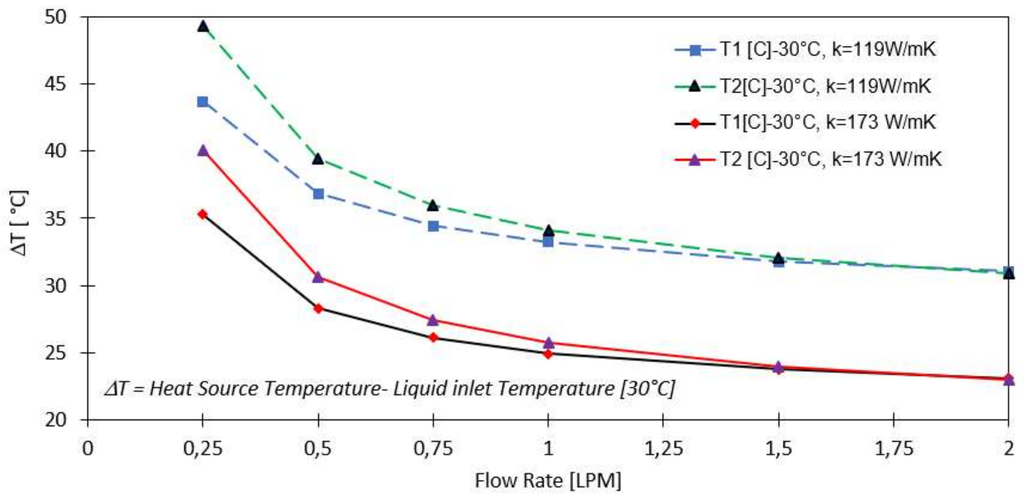
Figure 7. a) Illustration of flow rate vs pressure drop of DLHS with/ without pin fins

$$Q = \dot{m}c_p\Delta T \rightarrow \frac{1}{\dot{m}\,c_p} = \frac{\Delta T}{Q} \tag{7}$$

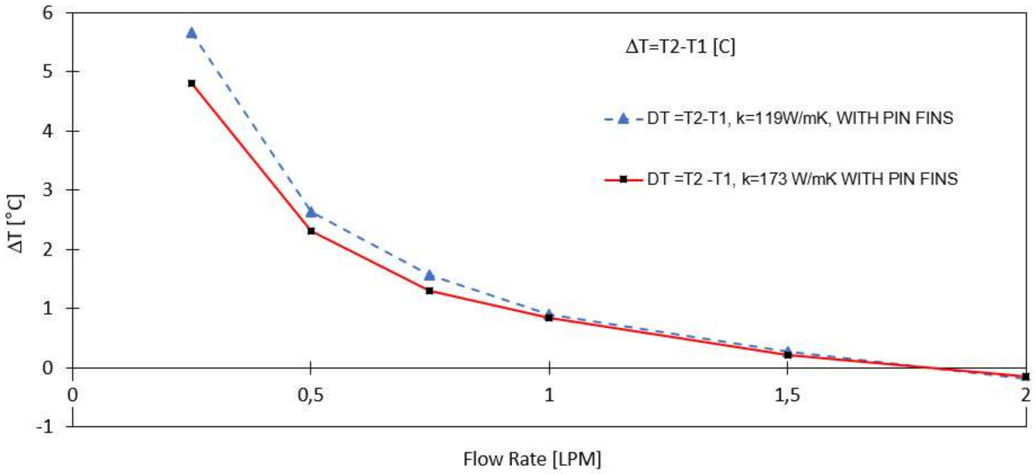
$$Q = hA\Delta T = hA(T_s - T_f) \tag{8}$$

$$Nu = \frac{hD_h}{k_f} \tag{9}$$

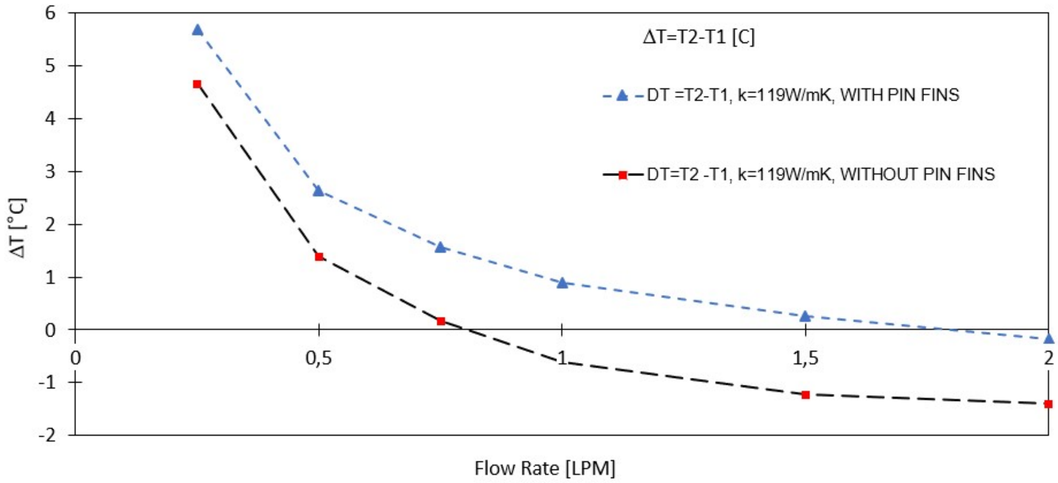
$$Re = \frac{D_h \rho V}{\mu} \tag{10}$$



A)



B)



C)

Figure 8. Illustration of flow rate vs temperature difference A) between the T1 &Tin and T2&Tin for k=119W/mK and k=173W/mK with pin fins used in DLHS B) between T1 &T2 with/without pin fins. C) between the cases with/without pin fins for k=119W/mK

5. FABRICATION

In Figure 9, pictures of DLHS design with measured surface roughness values are illustrated in detail. It is seen that compared to traditional techniques (CNC machine production) where surface roughness is around Ra=1-2 µm, this value reaches Ra=17 µm for AM technology. This is a very high value that can affect the pressure drop considerably. Microscopic view of the channel details is given in Figure 10. It seems that 0.3 mm fin space is a promising limit size that can be chosen as the heat flux is high as AM technology with AlSiMg10 is selected as the powder material.

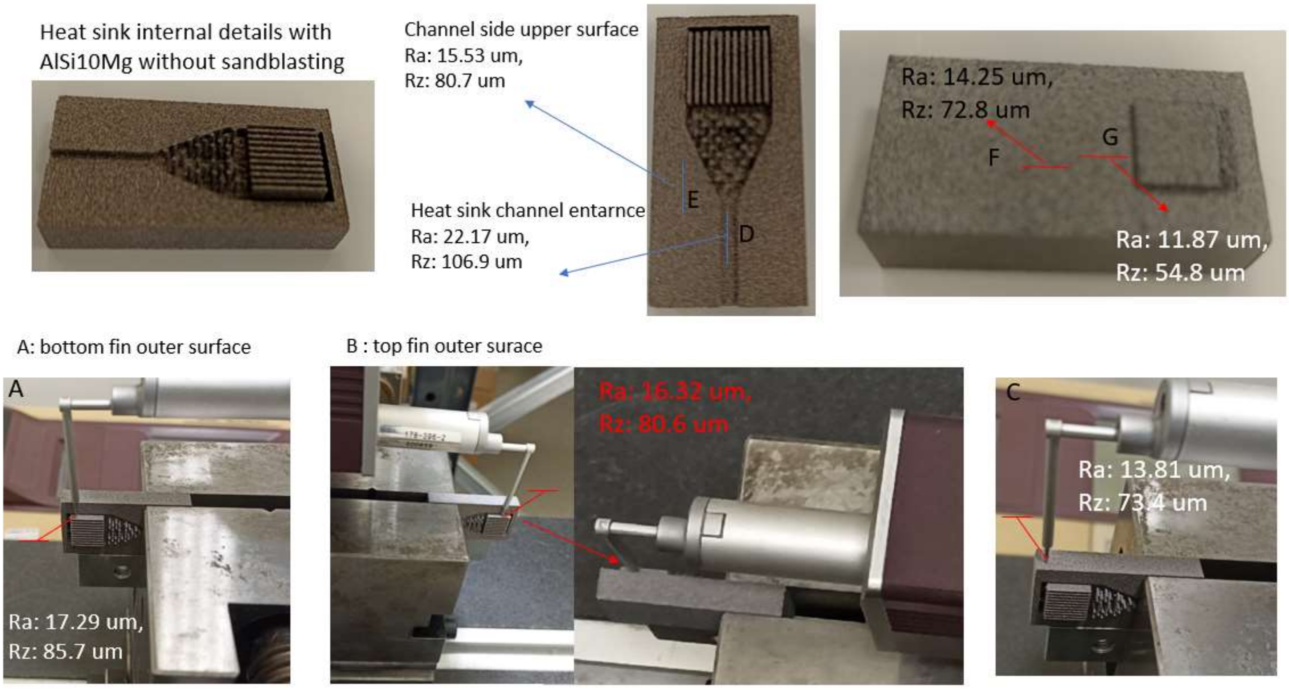


Figure 9. a) Illustration of heat sink samples produced with AM techniques to measure the surface roughness in µm

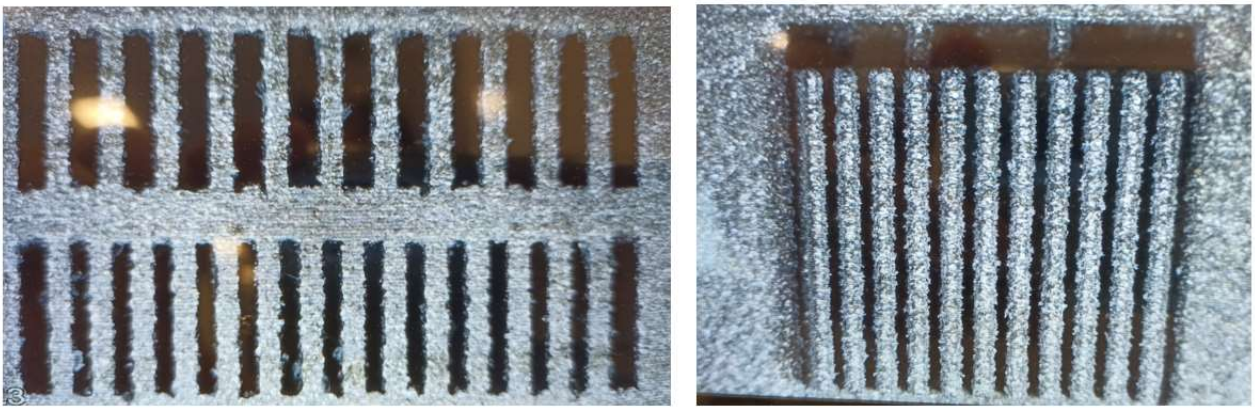


Figure 10. a) Cut view of the DLHS at the center line of heat sources, fin spaces 0.5 mm and 0.3 mm b) Top view of upper heat sink (cold liquid inlet region), fin space 0.5 mm

In Figure 11, additional details of DLHS are shown. At the center of the heat source, they are cut and the product is controlled to be sure that there is no blockage. After removing powder, fin spaces and thickness are examined to decide to accept the production. Fortunately, it is seen that both channels and pin fins are well printed.

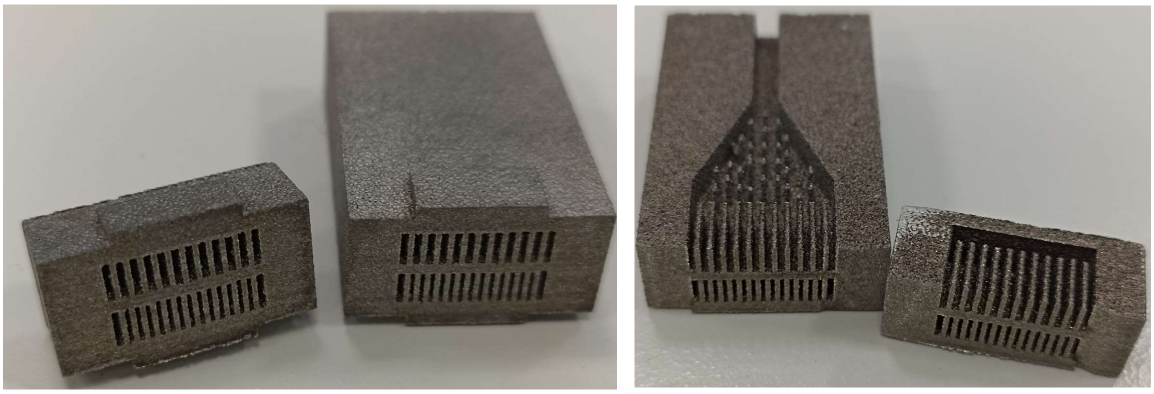


Figure 11. Cut view of the a) DLHS with all body, fin spaces 0.5 mm and 0.3 mm b) DLHS with removed top surface showing pin fins

In addition, it is well known that the thermal conductivity of a part produced with AM technology depends on whether heat treatment is applied or not. In order to uncover any doubt about it, a test is performed to find out the thermal conductivity variation of a specimen with respect to temperature with 50°C increment starting from room temperature at the facilities of Integrated Manufacturing Center (IMC) located in İstanbul, Turkey. In Fig.12, it is seen that there is a direct relation between the thermal conductivity and temperature of the specimen. It increases with temperature and at the working temperature which is around 60°C, it is %29 inferior to the one with heat treated. It is an important figure that one needs to consider when conduction and spreading resistance in a design is highly dominant resistance.

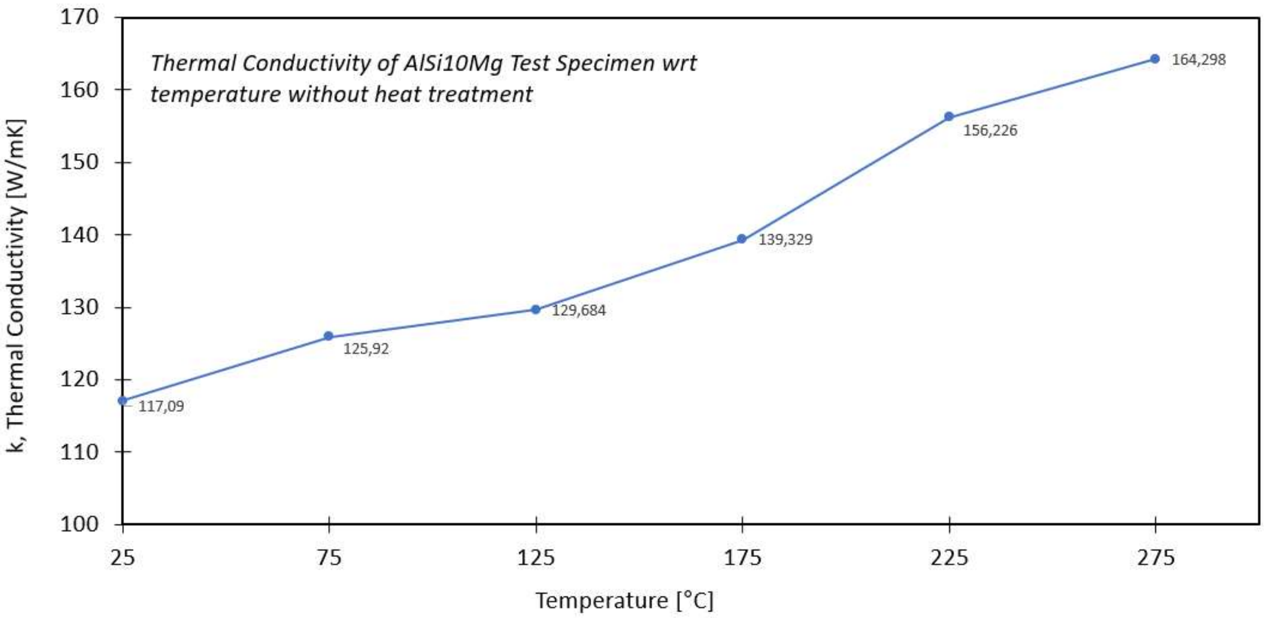


Figure 12. Thermal conductivity variation with respect to temperature of a specimen produced with AM without heat treatment

6. CONCLUSIONS

In summary, the design of the heat sink is done according to the smooth surface assumption. After the fin size is defined, it is produced with AM to measure the exact surface roughness to go further. It is well known that surface roughness values of inner surfaces are higher than those of outer sides where sandblasting is applied. It is most probably expected that thermal behavior will be better than calculated due to the smaller hydraulic diameter and some turbulence effect of higher roughness value.

- According to the present study, there are a few important conclusions drawn to note.
- Heat treatment has an effect of around 5°C improvement on the hot spot values. Therefore, it could be better to go on with the heat treatment process after AM.
- On the other hand, pin fins at the location at the inlet and outlet pools of the DLHS case have a negative effect both on the thermal and hydrodynamic performance of the DLHS. Pin fins or internal supports which are permanent lead to higher pressure drop values and temperature differences between the upper and lower heat sources.
- Although all analysis is carried out for smooth surface assumption, experimentally measured pressure drop will be expected to be much higher than that calculated due to the reaching early critical Re number. This part is left for future work.
- To produce with AM technology can force the designer to work with a smaller flow rate to have low-pressure drop and use the advantage of the surface roughness in thermal performance. The numerical results and picture of the sample promise that AM technology can be used as far as fin space is around 0.3 mm.
- For this specific case, 0.75 LPM seems to be a sufficient flow rate to succeed all requirements.
- DMLS AM technology produces aluminum parts with surface roughness values about 10 times higher than CNC machining
- The DLHS design allows temperature difference between heat sources to be kept around 1°C with 100W/cm^2 heat flux.

NOMENCLATURE

A	*Area, mm2*	*PP*	*Pumping power, W*
AR	*Aspect Ratio*	*Ra*	*Roughness, μm*
AM	*Additive Manufacturing*	*Re*	*Reynold Number*
CFD	*Computational Fluid Dynamics*	*s*	*solid*
DLHS	*Double Layered Heat Sink*	*Q*	*Heat load, W*
Dh	*Hydraulic diameterhydr*	*V*	*Velocity, m/s*
EGW	*Ethylene Glycol Water*	*T*	*Temperature, °C*
f	*Friction factor, fluid*	$\dot{m}$	*Mass flow rate, kg/s*
h	*Heat transfer coefficient, W/m2K*	T_{in}	*Inlet fluid temperature, +30°C*
H_c	*Fin height, mm*	*T1*	*Temperature of heat source at the top surface, °C*
k	*Thermal Conductivity, W/mK*	*T2*	*Temperature of heat source at the bottom surface, °C*
LPM	*Liter Per Minute*	*W*	*Fin thickness, mm*
LHS	*Lower Heat Sink (Secondary Heat Sink)*	W_c	*Fin space for flow*
MTBF	*Mean Time Between Failures*	*X, Y, Z*	*Axis, and Dimensions, mm*
N	*Number of fins for each heat sink*	ρ	*Density, (kg/m3)*
Nu	*Nusselt number*	μ	*Dynamic Viscosity, Pa.s, kg/ms*
P	*Pressure, kPa, bar*	Δ	*Delta*
PHS	*Upper Heat Sink (Primary Heat Sink)*		

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DECLARATION OF ETHICAL STANDARDS

The author of the paper submitted declares that nothing which is necessary for achieving the paper requires ethical committee and/or legal-special permissions.

CONTRIBUTION OF THE AUTHORS

Murat Parlak: The idea for the study was proposed by the author. A prototype was then produced by the author after the model was generated and solved using CFD tools. The manuscript was written and edited by the author, after all.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

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