A STUDY ON INDUSTRIAL HEAT SINKS FOR POWER ELECTRONICS

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ABSTRACT

The paper deals with an application of research over wavy fin channels to industrial devices for the cooling of high power-dissipating electronic component.

An industrial heat sink for forced air cooling of power electronics is tested to assess its performance under different working conditions: these include the layout of the power-dissipating sources, number of spacers between fan and heat sink, amount of power dissipated.

The velocity profiles at the exit of the channels are determined with a hot wire anemometer for two different number s of spacers. The temperature distribution is measured for the case of uniform heat flux through the component, when this is placed close to the edge where the fan is located or in the middle of the heat sink. The power dissipated is varied from 125 W to 500 W and the tests are performed for both one and five spacers between fan and heat sink. Thermal resistance is also calculated for all configurations which have been tested.

It is concluded that for the studied configuration the most profitable is the one having the heat sink placed in the middle and only one spacer.

INTRODUCTION

Finned heat sinks are largely employed in many engineering fields. The industry demands heat exchangers of ever increasing compactness and equivalent or improved performance, which spurs the researchers into devising and testing new geometries for the heat sinks. The use by the electronic industry of components dissipating more and more power has produced a large amount of studies on new models of heat exchangers, which must be able to accommodate large heat fluxes while keeping the same spatial dimensions. Both trends led the researchers to develop new profiles for the fins so as to optimise performance while decreasing the dimensions. [1-3]. The problem of profile optimisation for a heat sink in order to transfer the maximum amount of heat under the constraint of the least possible volume hasn't been completely resolved yet. The first to suggest an optimisation criterion was Schmidt (1926), who adopted a parabolic profile [4]. Many authors later opposed Schmidt's conclusions, as they hardly reproduced the physical occurrence of the phenomenon. Since then, many profiles for the heat sinks have been suggested, mainly parabolic or triangular in shape, but without an ultimately satisfying answer to the issue of optimisation [5]. Wavy profiles are of more recent origin [6], and it has been demonstrated that the superposition of a wavy profile to a parabolic one can increase the heat exchanger's efficiency remarkably. Studies on the performance of heat sinks with wavy profile have been presented [7,8]; these employed one- and two-dimensional models and demonstrated that substantial increase in effectiveness of the heat exchanger can be obtained by varying some form factors.

Since a few years fins with wavy profile have had a breakthrough in industrial application, and heat sinks with wavy fins are now available on the market for cooling electronic components both for computer hardware and power electronic applications. This study aims at a preliminary assessment of the performance of a commercial heat sink (manufactured by LDS System) with wavy channel walls under different operating conditions.

HEAT SINK DESCRIPTION AND EXPERIMENTAL SET-UP

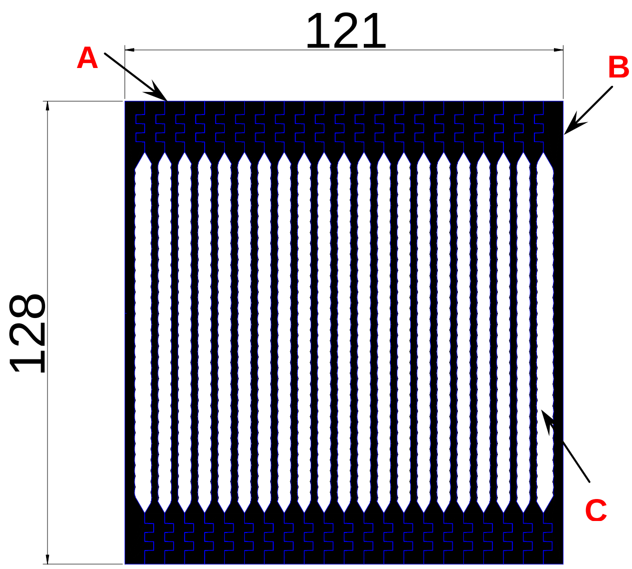

Figure 1 - Drawing of the Heat Sink's Front

The heat sinks studied consist of a series of corrugated aluminium plates produced through cold extrusion. The plates are assembled by packing the single elements (A in Fig. 1) and then pressing them together by means of a mechanical vice. The crests at the plate's top and bottom are squeezed into the adjacent grooves and expand, thus creating a permanent connection between the parts, with special elements (B in Fig. 1) being used to close the outermost channels. The modular assembly method

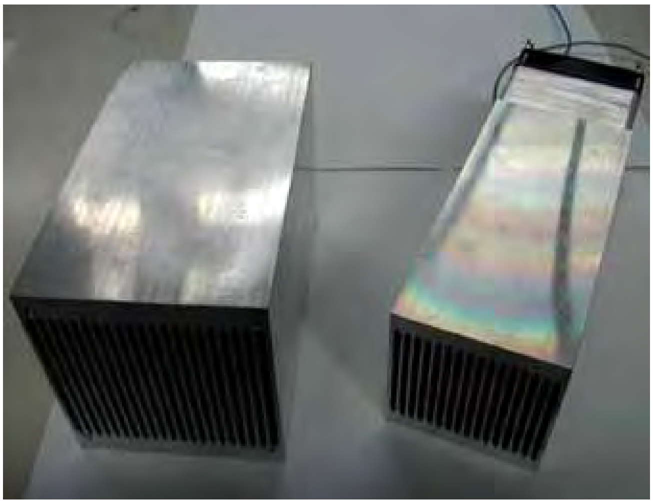

Figure 2 –Picture of two Models of Heat Sink

allows heat sinks of different length and breadth to be produced with the same operations, the die for cold extrusion being the only part being changed in the whole manufacturing process. By the same token, the pitch of the wavy corrugation can be changed at moderate cost, by manufacturing a die with different shape, which is a relatively economical undertaking, as the operational life of the dies never spans a whole batch of extrusion. The assembled heat sinks come in different sizes, as can be seen from Fig. 2, and can thus accommodate electronic components (e.g. IGBT) of varying shape and size. The component is secured to the heat sink by means of screws, with thermo-conductive paste spread between the two to enhance thermal contact. The heat dissipated by the component is conducted through the heat sinks, whose thermal conductivity is about 200 W/(mK), as the aluminium used for extrusion has characteristics similar to those of the pure substance. Forced convection of air through the channels (C in Fig. 1) is then used to remove the heat and keeping the temperature of the heat sink low. The fan, powered either by 220 V AC or 24 V DC, depending on the models, is bolted to one end of the heat sink, as shown on the right specimen in Fig. 2, and is kept at a set distance from the heat sink's end by means of spacers (either plastic or metallic). The spacers have the task of re-adjusting the swirled airflow that the fan's blades cause, albeit at the expenses of additional friction losses. Between one and five spacers are usually used, the number being determined mainly by the available space in the cabinet where the heat sink is mounted.

Experimental Set-up and Conditions

The heat sink which was chosen for preliminary investigation has a front section of 121x128 mm (as in Fig. 1) and length of 150 mm.

To simulate a heat-dissipating component, a 70x105 mm resistance heater was employed in order to generate a uniform heat flux over the whole area of contact. In order to be able to test the performance of the device when the heat source is placed in different locations, a load cell was used instead of the screws. This consists of a set of two bars, one placed under the heat sinks and one onto the heater and drawn together by means of two bolts. The upper bar touches the heater through a stem, which bears a set of springs. When the bolts are screwed tight, the springs are loaded till they eventually release a tag, thus indicating that the design pressure has been reached. In this way, there is no danger of damaging the ceramic parts of the resistance heaters and varying pressure can be applied, thus investigating the effect of the contact resistance between heat sink and component.

The temperature has been monitored for the heat sink by placing a series of T-type thermocouples into holes drilled on the side of the heat sink 1 mm under the top surface. The holes reach to the mid –plane and thus allow also transverse monitoring of the temperature by sliding the thermocouple along them.

The holes start at 10 mm from the face where the fan is placed and are spaced 10 mm apart, ending 10 mm from the opposite face.

The temperature of the ambient air is also monitored, as is that of the air at the exit of the central channel.

The heater is fed with 220 V AC and its power output is monitored and measured by means of a wattmeter, with an accuracy of ± 3 W in the range of interest.

The fan employed was supplied by the manufacturer of the heat sink and is powered by 220 AC, absorbing a total power of 13 W. Tests were run with both 1 and 5 spacers between the fan and heat sink, so as to assess if the different arrangement had any influence on the thermal performance of the device.

The heat sink was insulated at the bottom, the resistance heater placed on the top surface, first with one side aligned with the sink's edge (as customary in industrial application) wh the fan was bolted, then halfway between the edges: this wa order to investigate the effects of placement – if any – on the performance.

Tests were run with varying amounts of dissipated pow namely 125 W, 250 W, 375 W and 500 W.

It was verified that the thermal contact resistance could be considered independent of the pressure of the load cell, and equal to that generated by simply letting the cell's upper bar rest onto the heater, whereas failing to load the dissipating unit with any weight would produce temperature increases in the component of several degrees: this means that, although thermal resistance wasn't computed, its influence was the same throughout the tests carried.

Prior to testing the thermal performance of the heat sink, an estimate of the exit velocity of the air flow was made using a hot wire anemometer (uncertainty ± 0.05 m/s). The velocity profiles were obtained for 1 and for 5 spacers.

All thermal tests were run under steady state conditions, with the values of the quantities of interest being monitored via a program in LabView language and the data were acquired either by means of a Babuc Data Logging unit (in the case of the hot wire anemometer) or by a switcher and multimeter system in the case of power and temperature; as to the latter, the thermocouples' outputs were compensated by means of an ice point and the total uncertainty on the temperature measurement was estimated to be ±0.05 K.

After all the tests were run, the thermal resistance R_{th} was calculated in all the treated cases. The value thus obtained takes the effect of free convection between the uninsulated walls of the heat sink the load cell and the surroundings into account and is a realistic tool to assess the thermal performance of a heat dissipater during normal duty conditions; when it is not possible to have adiabatic conditions at the side and top of the device and contributions due to radiation also concur to the global amount of heat transferred.

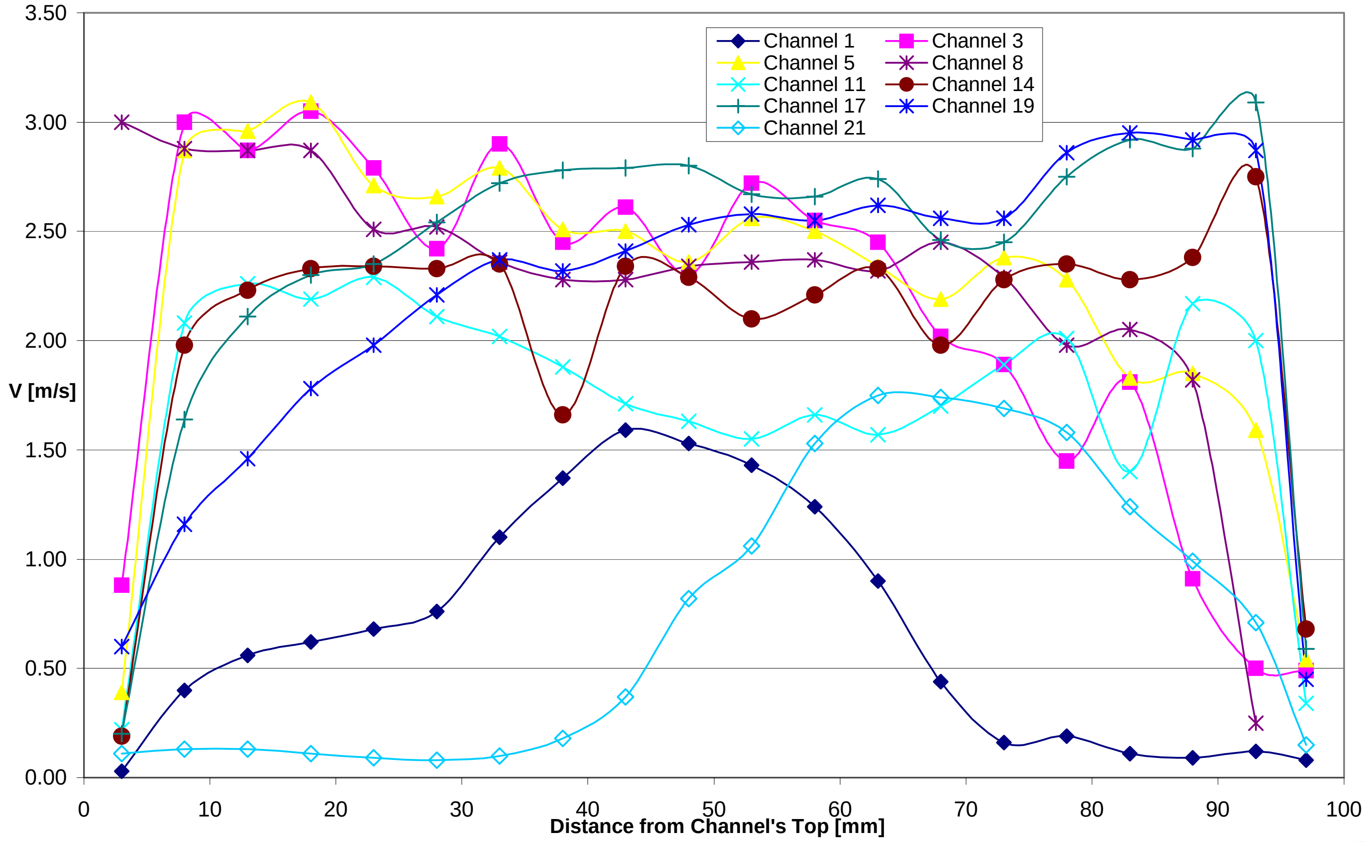

Figure 3 – Plot of the air velocity at the channel exit as a function of position

The uncertainty affecting the value of the thermal resistance was estimated in ±0.007 K/W.

RESULTS AND DISCUSSION

Air Velocity Profile

As previously stated, the velocity profile at the exit of the heat sink's channels was investigated first. Nine out of the 21 channels were considered and the values of velocity at the exit V was recorded for points located 5 mm apart, starting where the corrugation begins. The values showed a rather strong fluctuation (from ± 0.05 m/s to ±0.25 m/s) and were thus averaged over a period of time of 60 s, which corresponds to 60 readings. The measurements were taken for 5 spacers and 1 spacer and the results for the former case are presented in Fig. 3.

The profiles differ greatly from each other, which can be ascribed partly to the swirling motion of the flow when the air current impinges onto the face of the heat sink, but which is undoubtedly influenced by the presence of three bars connecting the central part of the fan (where the motor and shaft are located) to the frame. These are placed 120° apart and lie on the fan side facing the heat sink, thus breaking the swirled flow of the air exiting the fan.

It is also to be kept in mind that the spacers are square frames with a circular hole 110 mm in diameter, which means that air has an unobstructed path through the central channels only, whereas only a reduced section of those lying at the sides (1 and 21 in Fig. 3) is directly accessible, which explains the small value of V over the whole section for these.

One feature common to all channels, though, is the sharp drop in velocity at the top and bottom of the channels: this is due to their V-shaped profile at the ends, which is an undesirable characteristic if one wants to have an efficient heat transfer by convection, as the fluid is slowed down exactly there where higher velocities would be profitable.

The tests run with only one spacer yield similar results, a comparison between the two groups demonstrates how t

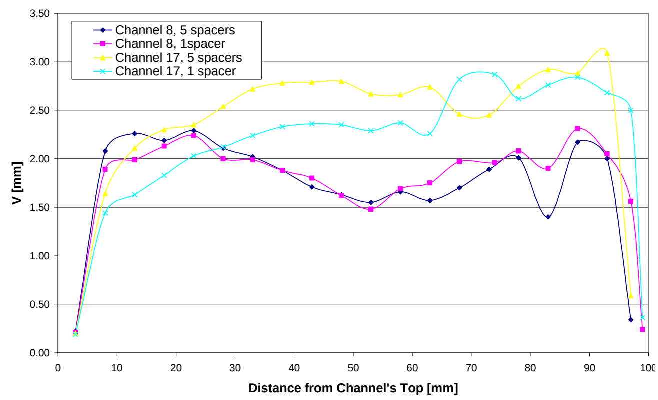

Figure 4 – Comparison of velocity profiles

number of spacers little affects the exit velocity. Fig. 4 shows the velocity profiles for channels 8 and 17 for both 5 and 1 spacers, and it is clear how only channel 17 exhibits any significant difference in velocity, which turns out to be about 20 % lower for the 1-spacer configuration, then again only over about half of the channel's length.

Temperature Distribution

The temperature was monitored along the centreline of the heat sink's upper face while varying the location of the dissipating heater, the power fed to it and the number of spacers between fan and sink.

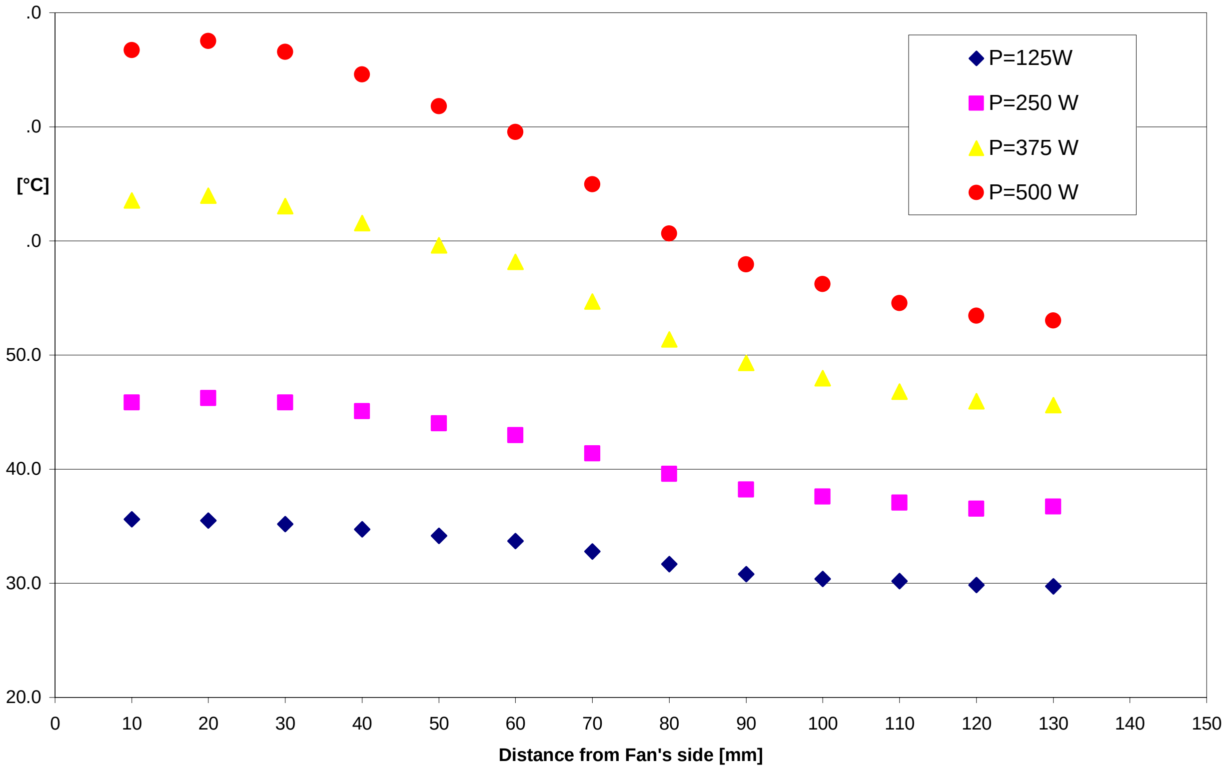

Figure 5 – Temperature distribution along the mid plane for heater located at the edge of the heat sink

The temperature distribution for the heater placed halfway from both ends of the sink is depicted in Fig. 6. In this case, the curve tends to flatten when moving towards the exit of the channels, while it decrease steadily when moving upstream with respect to the direction of the air current. The reason for this be-haviour can be explained by considering that the air enters t heat sink from the side corresponding to a value of zero for t abscissa and thus won't receive the bulk of the dissipated power

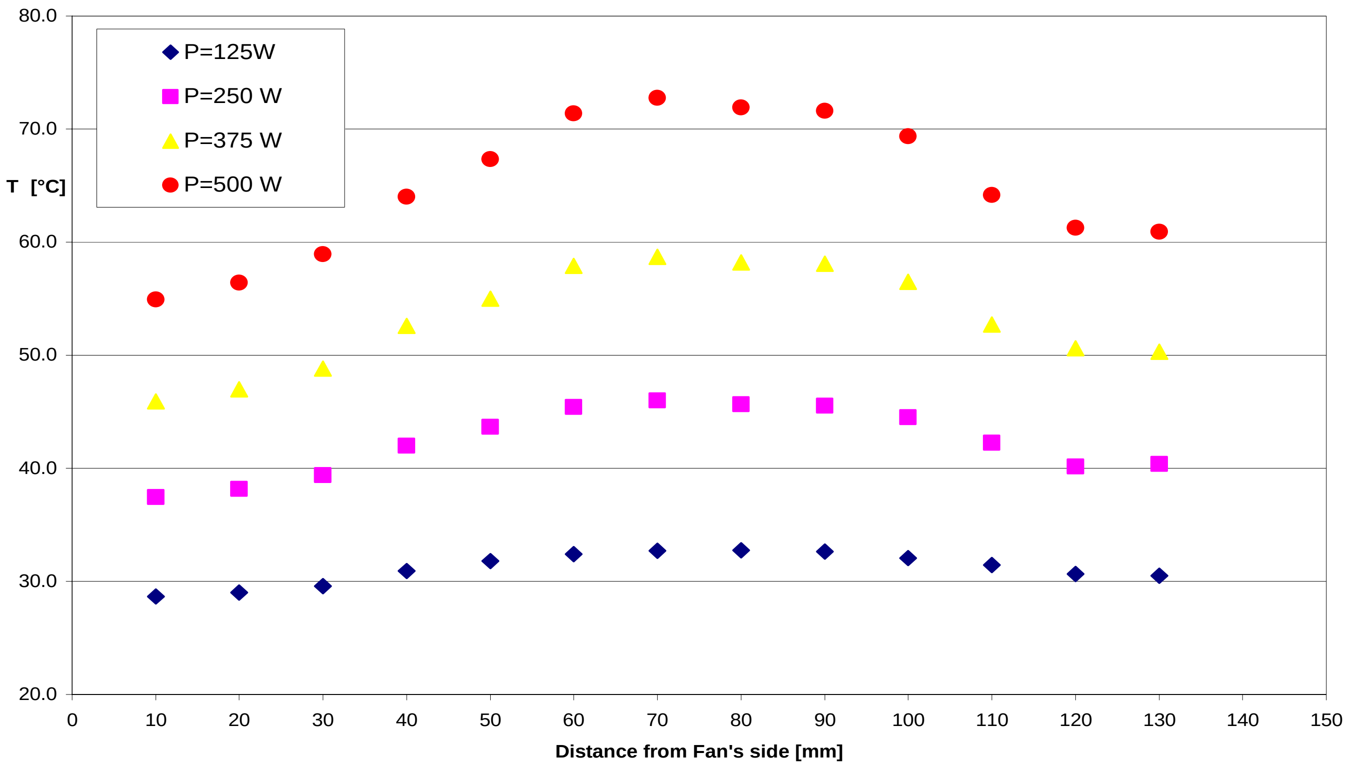

Figure 6 – Temperature distribution along the mid plane for heater located in the middle of the heat sink

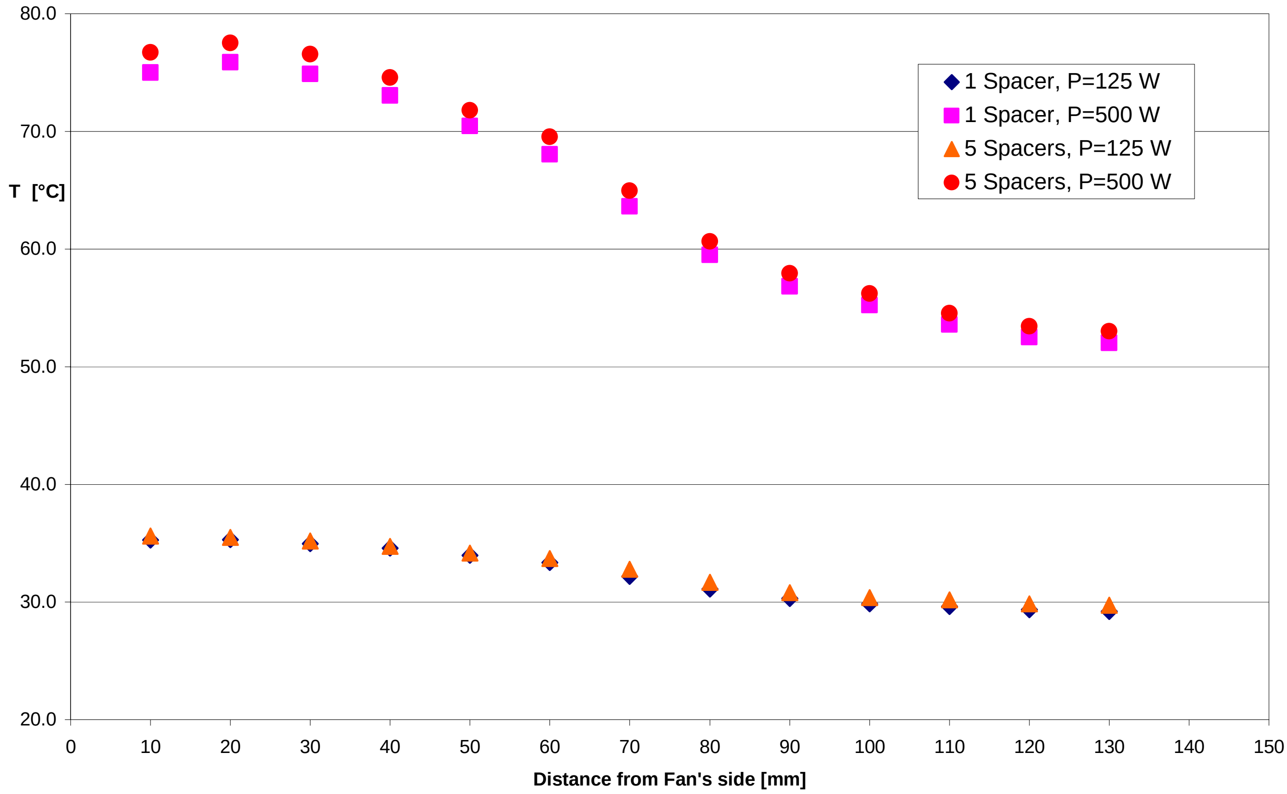

Figure 7 – Comparison of temperature profiles for 1 and 5 spacers, heater at the edge

immediately, but only the part which is conducted from the middle of the device; although the temperature difference between entering air and aluminium walls is lower, the heat transfer mechanism as a whole is more efficient, as the maximum temperature of the heat sink is lower in this case (about 6.8 K) than when the heat sink right at the edge. Conversely, the temperature after the heater is higher than before it, owing to the decrease in temperature difference between the air and heated walls caused by the massive heat transfer in the cent portion of the heat sink.

On the whole, the average temperature is higher for the lat case, but the peak temperature is somewhat lower (which is also reflected in the maximum temperature of the component).

The temperature profiles were also investigated for the case

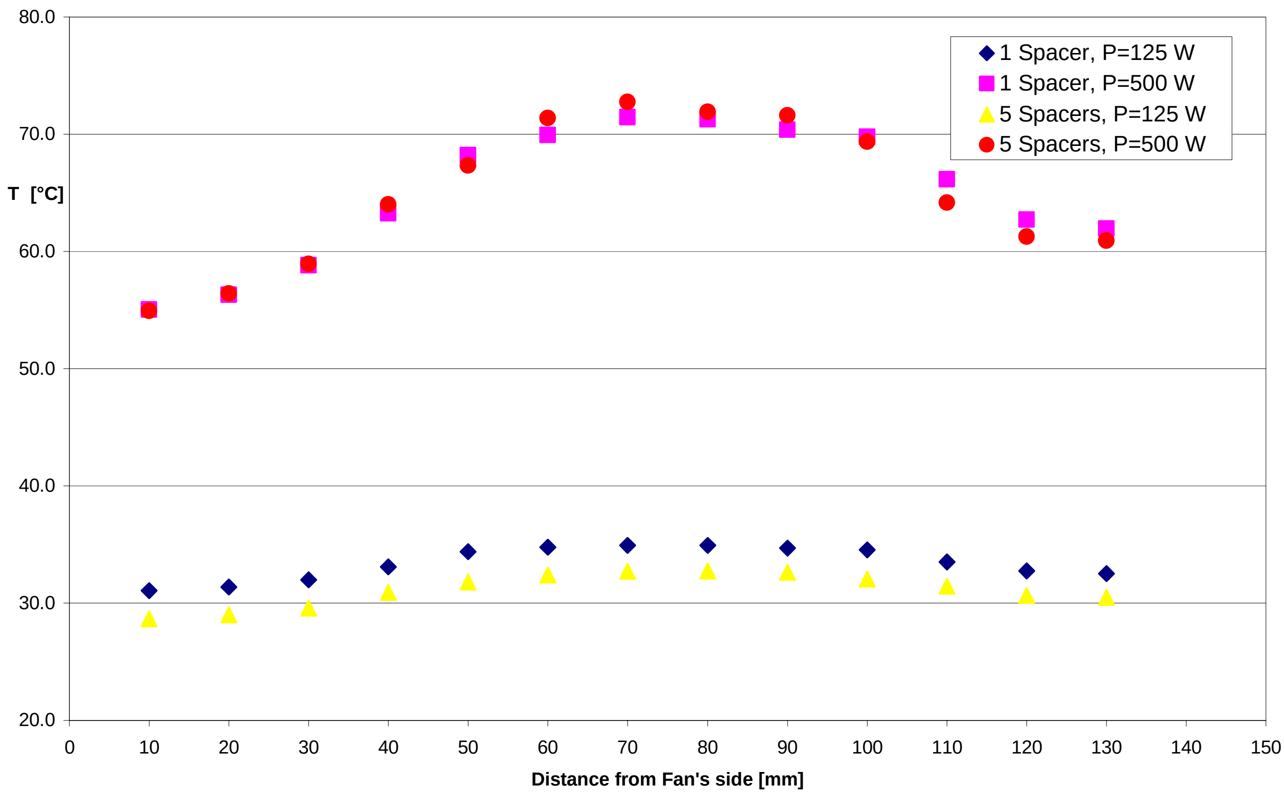

Figure 8 – Comparison of temperature profiles for 1 and 5 spacers, heater in the middle

of a single spacer for the extreme values of the dissipated power (500 W and 125 W, respectively). The results are reported in Fig. 7 and in Fig. 8 for the two locations of the heater. When the power-generating source is at the edge, the single spacer is seen to perform better than the assembly of five, with a maximum temperature about 1.6 K lower for a 500 W dissipation, while the two arrangements are basically equivalent when 125 W are dissipated. This behaviour is due to the higher turbulence of the flow when only one spacer is present, which increases the heat transfer: the increased frictional losses do slow down the flow but only then when the bulk of the heat exchange has already taken place.

When the heater is placed midway between the edges, the effect of increased turbulence is still felt where the temperatures are at their highest, but the friction losses slow the fluid down enough to give rise to higher temperature than with five spacers just downstream of the heat sink, as can be seen in Fig. 8. The enhancement in heat transfer is more marked for low dissipated powers, as temperature gradients along the heat sink are lower and conductive phenomena are thus less important.

The thermal resistance was finally calculated for each configuration. This parameter can only be defined for the device as a whole, since the heat flux isn't distributed over the whole are ot the upper surface.

The values of the thermal resistance are given in table 1 and are all rather close to each other and roughly contained within the eperimental uncertainty. Again, the configuration with one spacer and the heater placed midway is to be preferred as it does give a decrease in maximum temperature, which is especially felt at high thermal fluxes.

	Power [W]			
	125	**250**	**375**	**500**
Centre 5 spacers	0.102	0.103	0.103	0.104
Side 5 spacers	0.116	0.114	0.113	0.113
Centre 1 spacer	0.102	--	--	0.099
Side 1 spacer	0.115	--	--	0.109

Table 1 – Thermal Resistance [K/W] for the configurations

CONCLUSIONS AND FURTHER DEVELOPMENTS

The thermal performance of an industrial heat sink has been examined for two arrangements of the heat source, different number of spacers and varying dissipated power under steady state conditions and uniform heat flux through the component. The results and discussion have brought to the conclusion that placement of the component in the middle of the heat sink is more favourable, as is the use of a single spacer as opposed to several.

The analysis is to be further expanded by considering the case of a uniform heat flux over the whole heat sink and how the length of the heat sink influences the performance when components occupy only a fraction of the sink's surface.

A further step is the use of numerical codes which, starting from the experimental results, allow the determination of the heat sink's performance depending on its size, the placement of the power source, the amount of power dissipated. By the same token, a comparison with the performance of a heat sink with smooth channels and with wavy channels of different pitch should be possible with such code.

The study of the contact resistance and of contributions due to radiation and natural convection is also seen as worth to be investigated.

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