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Thermal and Hydraulic Performance of Water/Glycol Mixture and the Application on Power electronics Cooling

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ABSTRACT

Numerical simulations were conducted to study thermal and hydraulic performance of liquid-cooled heat sink on power electronics cooling, in particular with water/glycol mixture. Considering non-uniform and discrete heat sources, geometry and number of cooling channels were analyzed. The results show that alternating rectangular channel has high thermal performance with a little penalty in hydraulic resistance. The number of cooling channels can be optimized to provide the best thermal performance, as 40 channels for the analyzed case. Performance difference resulting from working coolants was studied, with 100% de-ionized water, mixture of 50% ethylene glycol and 50% de-ionized water (EGW) by weight, and mixture of 60% propylene glycol and 40% de-ionized water (PGW) by weight. The variations of fluid physical properties with temperature were taken into account. It has been concluded that lower coolant temperature does not necessarily lead to better cooling capacity, i.e. a specific coolant has an optimum operation temperature to provide the maximum cooling performance.

Keywords: Electronics cooling, Liquid-cooled heat sink, Water/glycol mixture.

1. INTRODUCTION

Variable-frequency drive (VFD) is widely employed in motor applications, of which the unit power requirement has been upgraded to 30,000kW, with both heat generation and heat density being increased significantly [1]. Accordingly, the VFD cooling must be enhanced from air cooling to liquid cooling. In cold environment, water/glycol mixture is normally selected as working fluid to avoid freezing, with a penalty in thermal performance and hydraulic resistance [2].

Xie et al. [3] studied the effects of channel dimensions, channel wall thickness, bottom thickness and inlet velocity on thermal resistance, pressure drop and maximum allowable heat flux for a rectangular mini-channel heat sink. With a 3D conjugate heat transfer numerical model, Li et al. [4] found an optimal size for rectangular channel to obtain minimum thermal resistance and maximum pumping power saving. Using a novel multi-parameter optimization approach, Wang et al. [5] employed an inverse problem method to optimize the geometry for a micro-channel heat sink. Peng et al. [6] performed an experimental study on pressure drop and convective heat transfer (water as coolant) in straight rectangular microchannels. The results showed that in both laminar and turbulent flow regimes, cross-sectional aspect ratio had a great influence on flow friction and convective heat transfer efficiency. Using the sequential quadratic programming (SQP) method, Hu et al. [7] conducted an optimization study, focusing on the structure of micro-channel.

For a given heat sink, the paper gives optimal number of the channels, as well as width and height of the channels. Geometric construction of the channels has a huge impact on the heat transfer and flow performance. In the previous study conducted by the authors [8], five kinds of cooling channels were studied with a given cold plate. The results showed that alternating rectangular channel had the best thermal performance with a little penalty in pressure drop, and higher channel density could improve both thermal and hydraulic performances. Gunnasegaran et al. [9] studied heat sink performance of three different shapes of micro-channels, as rectangular, trapezoidal, and triangular with the Reynolds number ranging from 100–1000. It has been found that the heat sink having the smallest hydraulic diameter has the best performance in pressure drop and friction factor.

Type of coolants has a significant influence on heat sink performance. The coolant selection is quite difficult because many factors must be taken into the consideration, as corrosion resistance, regulatory constraints, thermal properties, cost, thermal stability, and environmental temperatures [10]. Water is an excellent heat transfer media, but freezing is a problem. Ethylene glycol and propylene glycol are colorless and odorless liquids that are miscible with water. Though pure glycols don't have the preponderance as a coolant, they can efficiently lower the freezing point of water when mixed, and hence the water/glycol mixture is widely used as antifreezer and heat transfer media. In applications, water/glycol mixture between 40% and 60% glycol percentage is typically recommended. Ethylene glycol solutions have better thermo-physical properties (such as viscosity) than propylene glycol solutions, especially at lower temperature. However, when the coolants are used for applications involving possible human contact or where mandated by regulations, the less toxic propylene glycol has its big advantage. In the applications for power electronics cooling, water and water/glycol mixture must be de-ionized. For a silicon based mini-channel heat sink, Li et al. [11] numerically studied the conjugate heat transfer and hydraulic performances with different kinds of coolants. The results showed that coolant thermo-physical properties had an influence on both the hydraulic and thermal performances. Dehghandokht et al. [12] studied a multi-port serpentine mesochannel heat exchanger with water and ethylene glycol-water mixture as coolant. The results showed that the serpentine meso-channel heat exchanger could be applied as an automotive radiator using ethylene glycol-water mixture.

In the present study, focusing the application to power electronics cooling, thermal and hydraulic performances of a liquid-cooled heat sink will be numerically studied, with an emphasis on channel geometry, channel number, and coolant selection.

2. PHYSICAL AND MATHEMATICAL MODEL

2.1 Model description

Considering three diode rectifier modules (60 mm × 124 mm, A1, B1, C1) and four IGBT semiconductor modules (140 mm × 190 mm, D1, D2, E1, E2) installed on both the top and bottom surfaces of a heat sink, the analyzed heat sink dimension is given as Figure 1, length L = 450.9 mm, width W = 444.5 mm, and thickness t = 22.9 mm. The heat sink is made of copper, with one inlet and one outlet. In the model, each diode rectifier module is one heat source with uniform heat flux, q1 = 828W/m2, and each IGBT module is with uniform heat flux, q2 = 1710W/m2. Cooling channels are arranged as two passes with parallel flow. The physical model is focused on channel shape and channel number. Based on the previous study [8], two types of channels, straight rectangular channel and alternating rectangular channel are selected for analysis, as the alternating rectangular channel can provide the best thermal performance and the straight rectangular channel can be a baseline for comparison.

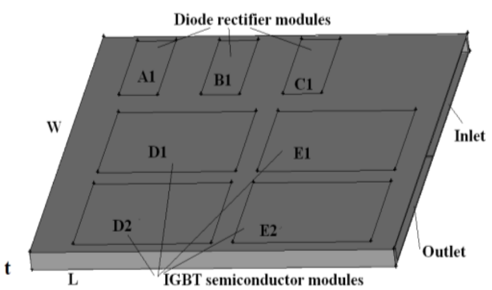


Figure 1. The scheme of the heat sink

2.2 Governing equations with boundary conditions

A three-dimensional numerical simulation model has been built for the heat sink. Governing equations for liquid coolant are,

$$\nabla \cdot (\rho \vec{V}) = 0 \tag{1}$$

$$\rho_f (\vec{V}.\nabla\vec{V}) = -\nabla P + (\mu_f \nabla^2 \vec{V}) \tag{2}$$

$$\rho_f c_{P,f} (\vec{V}.\nabla T) = k_f \nabla^2 T) \tag{3}$$

Energy equation for base solid copper wall region:

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

The boundary conditions for the governing equations are given as follows:

In the fluid:

Inlet:

$$\text{v} = \text{w} = 0;\ u = u_{in};\ \text{and T = Tin} \tag{5}$$

Outlet:

$$p = p_{out} \tag{6}$$

At heat sink surfaces covered by heat sources

$$q_w = -k_s \frac{\partial T_s}{\partial n} \tag{7}$$

At heat sink surfaces open to the ambient:

$$0 = \frac{\partial \mathrm{T}_s}{\partial n} \tag{8}$$

At fluid-solid interface:

$$T_f = T_s;\ -k_f \frac{\partial T_f}{\partial n} = -k_s \frac{\partial T_s}{\partial n} \tag{9}$$

2.3 Analysis consideration

Channel geometry, channel number, and coolant selection will be analyzed to study their effects on heat sink thermal and hydraulic performances.

Three coolant fluids are selected for the comparison, as 100% de-ionized water, 50% ethylene glycol/ 50% de-ionized water by weight (EGW50/50), and 60% propylene glycol/ 40% de-ionized water by weight (PGW60/40). Considering the temperature variations, cooling performances from the three coolants are compared under the same pumping power, thus the inlet velocity of the coolant, uin, is calculated according to the given pressure drop [13].

Thermal properties of the glycol solutions, especially viscosity, are sensitive to temperature. In the analysis, the changes in fluid properties with temperature are taken into account. The fluid physical properties are obtained from the ASHRAE Handbook [14].

The channel geometry determination and channel number optimization are taken as a design consideration. The design criteria are the inlet coolant temperature 308K, the coolant volumetric flow rate 5.05 × 10-4m3/s, the maximum allowable heat sink surface temperature 358K, and the maximum hydraulic pressure drop 40,000Pa. PGW60/40 is selected as working coolant, as it provides the worst thermal and hydraulic performances.

3. NUMERICAL SOLUTION

Meng et al. [15] conducted an experimental study on thermal and hydraulic characteristics of alternating elliptical channels. A threshold of Reynolds (Re) number defining flow states was given for alternating elliptical channel, as Re<500, the state is laminar flow and Re = 500 ~ 1×104 is low-Re flow. It has been found in Hu et al.'s study [8] that the Re number threshold can be as low as 200 for alternating rectangular channel, as the disturbance in it is greater than that in alternating elliptical channel. In the analyzed case, as the Reynolds number in the alternating rectangular channel is between 200 and 500, the flow state is considered in turbulence transition regime. Accordingly, low-Re k- ε turbulent model is employed for the CFD solution. The mesh for the alternating rectangular channel and the channel arrangement are shown in Figure 2 and Figure 3, respectively. The computational domain is discretized to control volumes, and grid points are located in the center of each control volume. The conservation equations are formulated with the parameters at each control volume and the surrounding control volumes. The flow field is solved using the standard SIMPLE algorithm.

Finite Volume Method (FVM) with hybrid differencing scheme is selected for solving the governing conservation equations with the boundary conditions executed on both fluid and solid regions. After a mesh independent study, 2.4 × 106 hexahedral grids are eventually applied for the numerical computation.

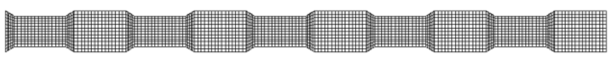


Figure 2.Shape of alternating rectangular channel with mesh

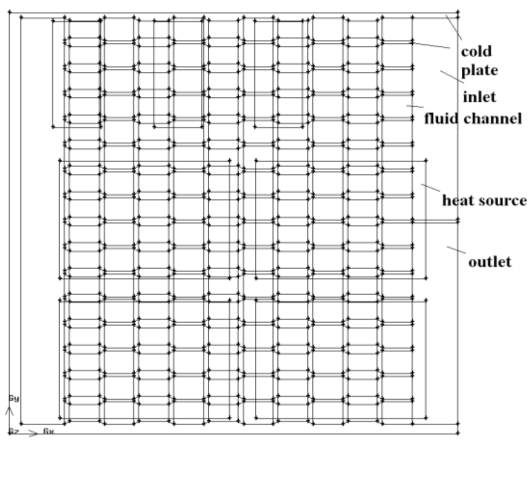


Figure 3. Channel arrangement in the heat sink (uniform arrangement)

4. RESULTS AND DISCUSSIONS

4.1 Channel selection

Channel geometry plays a very important role in improving heat sink performance. Five cooling channel geometries have been studied in Hu et al.'s work [8], as alternating rectangular channel shows the best heat transfer performance with a little penalty in hydraulic resistance. The comparison between the alternating rectangular channel and the straight rectangular channel is given as Table 1.

It can be found from Table 1 for the analyzed 16-channel heat sink that the straight rectangular channel results in the maximum temperature of 400K, much higher than the allowable 358K, while the alternating rectangular channel is only 367K. The two channels show about 50% difference in the heat transfer performance. The pressure drops are close, and both are quite within the design limitation. Therefore, the alternating rectangular channel will be selected for the analysis next on channel number optimization and coolant effect, mainly focusing on thermal performance.

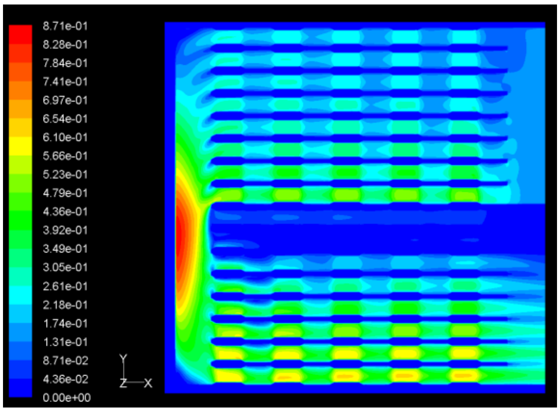
Table 1. The performance of the two kinds of channels (water as coolant)

Parameter	straight rectangular channel	alternating rectangular channel
aximum surface T (K)	400	367
heat transfer ΔT(K)	92	59
p (Pa)	481	504

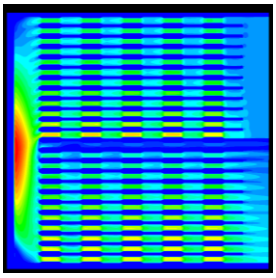
4.2 Channel number and arrangement

Uniformly located cooling channels are studied. The inlet temperature of coolants is 308K, and the volumetric flow rate is fixed 5.05×10-4m3/s. With an increase in channel number, the pitch between the adjacent cooling channels is kept constant. It means the channel size will be smaller, and the sum of channel surfaces directly facing the heat sources will become smaller. It can be said that the channel number will affect the entire heat convective transfer areas contacting with the coolant, as well as the ratio of cooling channel area and non-channel area at heat sink surface.

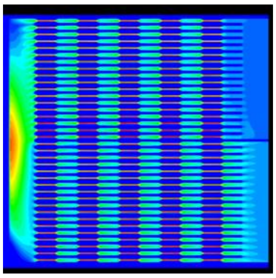
The channel number will also affect coolant flow distribution in the heat sink. Figure 4 shows the velocity profile at the cross-section of the half of the heat sink thickness, i.e. z=11.4mm. Each channel is numbered as channel 1, channel 2...channel N from top to bottom. For a 16-channel heat sink, the coolant flow distribution per channel is quite uneven, as the mass flow rate is increased significantly from channel 1 to channel 8 at Pass 1 and from channel 9 to channel 16 at Pass 2. With 24 channels in the heat sink, the mass flow rate is also increased from channel 1 to channel 12 and from channel 13 to channel 24, but the non-uniformity is not as significant as the 16-channel design. When 36 channels are applied, the coolant flow distribution has become quite uniform.



(a) 16 channels



(b) 24 channels



(c) 36 channels

Figure 4. The schematic velocity of surface z=11.43mm (the inlet temperature is 308.15K)

Another important factor affecting the heat sink performance is the heat transfer coefficient. The average heat transfer coefficient at cooling channel surface is defined as,

$$h = \frac{Q_c}{A_c \Delta T_m} \tag{10}$$

of which, $\Delta T_m = \dfrac{T_{f,outlet} - T_{c,outlet}}{\ln \dfrac{T_{c,inlet} - T_{f,inlet}}{T_{c,outlet} - T_{f,outlet}}}$, is the log mean temperature difference between the coolant and channels.

When the channel number is increased, the size of each channel becomes smaller. It causes an increase in the disturbance of the flow in each channel, and hence the heat transfer coefficient. The increase of the average surface heat transfer coefficient with channel number is shown as Figure 5.

The channel numbers take the thermal effects from coolant flow distribution, entire heat transfer area (including the ratio of heat convection area to heat conduction area), and heat transfer coefficient at cooling channel surface.

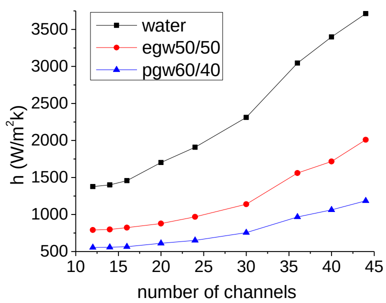
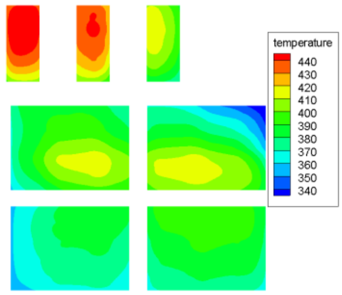
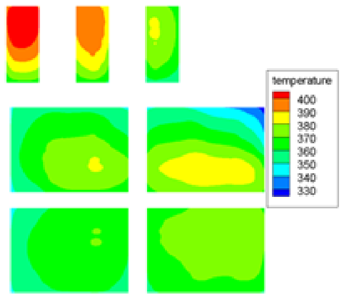


Figure 5. Variations of surface heat transfer coefficient with number of channels (T= 308.15K)

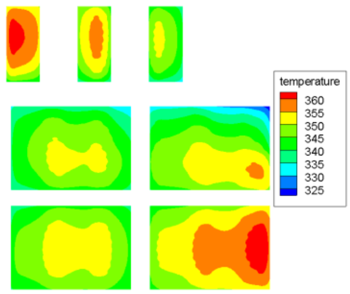
The heat sink surface temperature at the heat source areas is illustrated as Figure 6. It can be found that the maximum heat sink surface temperature is decreased with the channel number increasing up to 40. At the 16-channel heat sink, the hottest spot occurs under module A1 and the lowest temperature is under module D2. As the channel number increasing, the surface temperature reduction under module A1 is significant, while the effect under module E2 is little. When the channel number is increased to 36, the temperature under modules A1 and E2 are almost the same. At the 40-channel heat sink, the hottest spot exists under module E2. The results show the effects of the channel number increase on both uniformity of coolant distribution and enhancement of heat convective performance.



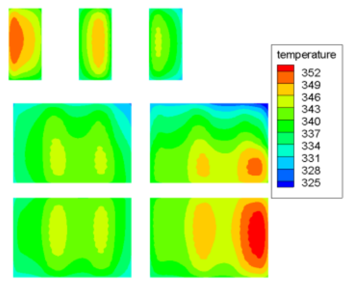
(a) 16 channels



(b) 24 channels



(c) 36 channels



(d) 40 channels

Figure 6. The temperature profile of different channels with PGW as coolant

The heat from power components is dissipated into the coolant with heat conduction and heat convection. As the channel number increases, the heat conduction area (i.e. the area between cooling channels) becomes larger. Because the conduction resistance is higher than the convection resistance for the analyzed case, a larger heat conduction area means an effect to increase thermal resistance. When the channel number is increased to 45, the effect of heat conduction resistance becomes significant, and cause the total thermal resistance increased and hence heat sink surface temperature increased. For a heat sink with smaller convection resistance, for example water vs. PGW60/40, the effect of the conduction resistance is greater. The conclusion can be found from Figure 7.

As shown in Figure 8, the pressure drop is always increased with the channel numbers in the heat sink. It is mainly because of the minor losses at branching points.

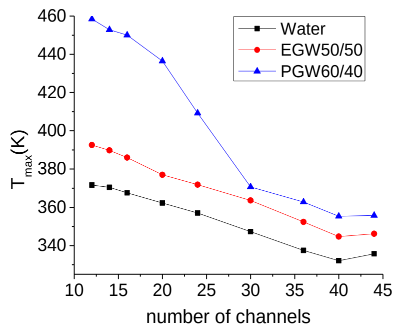


Figure 7. Maximum surface temperature of different numbers of channels (T= 308.15K)

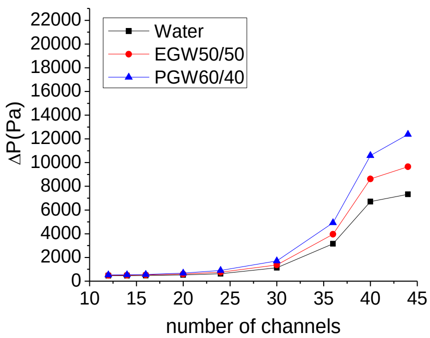


Figure 8. Pressure drop of different numbers of channels (T= 308.15K)

Figs. 4~8 show the effects from the cooling channel with uniform arrangement. Since the heat sources on the heat sink surface are discrete, the cooling channels can be located accordingly for a customized design. The non-uniform arrangement of the cooling channels, illustrated as Figure 9, can reduce the maximum surface temperature. Since the pressure drops mainly occur at each channel, the non-uniform channel arrangement has little impact on coolant flow distribution. Figure 10 shows the velocity profile at the cross-section of z=11.4mm for the non-uniform channels arrangement case.

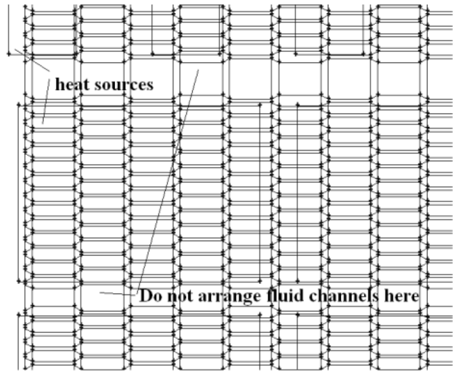


Figure 9. Non uniform arrangement of the fluid channels

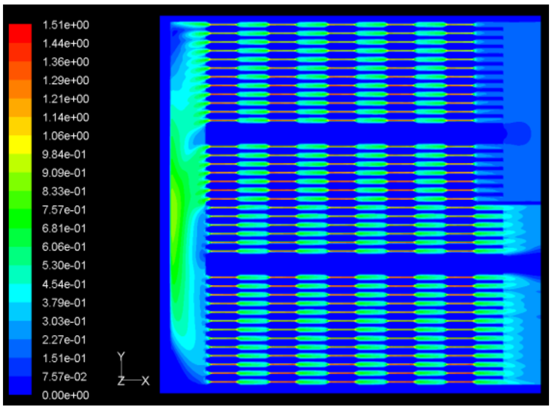


Figure 10. The schematic velocity of surface z=11.43mm of the non-uniform arrangement case.

4.3 Coolant effect

In a cooling design, it is in general considered that a lower coolant temperature gives higher temperature difference for the heat transfer from heat dissipating components to liquid coolant, and hence results in higher cooling capacity. However, the conclusion may not be accurate for the coolants which physical properties changes significantly with temperature, such as water/glycol mixture. Based on the analyzed heat sink case, the cooling ability of water/glycol mixtures is studied to investigate the variation with coolant temperature, under the same pumping power. The results are shown as Figure 11. It can be found that the maximum heat sink surface temperature is not always reduced with a decrease in coolant temperature. There exists an optimal inlet temperature for each coolant, as 288K for EGW50/50 and 268K for PGW60/40. The optimal temperature is resulted from the associated effect of temperature difference, heat convection performance, and coolant flow rate. As the inlet coolant temperature is decreased, the temperature difference for heat transfer is increased, while the heat convection performance at fluid/solid interface is decreased because of an increase in coolant viscosity resulting from lower temperature. In addition, if the same pumping power is considered, the coolant flow rate will be reduced with the coolant temperature, as shown in Figure 12. Without the consideration of the same pumping power, the optimal coolant temperature also exists because of the effect from coolant viscosity and hence heat convection performance.

5. CONCLUSIONS

Numerical simulations were conducted to study the heat transfer and fluid flow characteristics of a liquid-cooled heat sink with water/glycol mixtures as working coolant. The analysis was focused on channel geometric construction, channel number, and coolant effect. The conclusions have been summarized.

(1) Alternating rectangular channel can provide high heat transfer performance, with a little penalty in hydraulic resistance. Because the device temperature is always the major concern compared to pumping power, the alternating rectangular channel should be widely applied to power electronics cooling.

(2) The cooling channels in heat sink have an optimal number to provide the maximum thermal performance and hence the minimum surface temperature. It is resulted from the associated effect of heat convection resistance at channel surface and heat conduction resistance in heat sink solid part. The pressure drop is always increased with channel number.

(3) Water/glycol mixture has an optimal operating temperature to provide the maximum cooling performance.

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NOMENCLATURES

A	heat transfer area (m2)
D	equivalent diameter (mm)
k	thermal conductivity (W/(K·m))
L	length of the heat sink(mm)
P	pressure (Pa)
Q	total heat generation (W)
q	heat flux (W/m2)
Re	Reynolds number
T	temperature (K); thickness of the heat sink(mm)
u, v, w	velocity component along x, y, z (m/s)
V	volume flow of the heat sink(m3/s)
W	width of the heat sink(mm)
x, y, z	coordinate directions (mm)

Greek symbols

Δ	difference
ρ	density (Kg/m3)
μ	dynamic viscosity ((N·s)/ m2)
λ	thermal conductivity (W/(K·m))

Subscripts

f	fluid
in	inlet
max	maximum
out	outlet
s	solid
w	wall
c	channel