THERMO-HYDRAULIC PERFORMANCE COMPARISON OF STEADY LAMINAR FLOW CHANNELS

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ABSTRACT

Thermo-hydraulic numerical analysis has been carried out to compare the characteristics of various shaped channels with two kinds of coolants such as EGW (EthyleneGlycol/Water) and PAO (PolyalphaOlefin).

In this study, single-phase, constant property and three-dimensional laminar forced convections are considered, and the governing differential equations for continuity, momentum, and energy transfer are solved computationally using the finite-volume method and the commercial codes of ICEPAK and FLUENT.

The variations of the maximum temperatures as well as the temperature difference on the heat source surface with pumping power are compared depending on the coolants and channels.

NOMENCLATURE

A_C	[m^2]	Cross-sectional area
A_W	[m^2]	Wall area
c_p	[J/kgK]	Specific heat
D_h	[m]	Hydraulic diameter
D_f	[m]	Distance between fins
f		Darcy/Moody friction factor
H	[m]	Height of computational domain
Hc	[m]	Height of channel
h	[W/m^{2}K]	Heat transfer coefficient
k_l	[W/mK]	Thermal conductivity of liquid
k_s	[W/mK]	Thermal conductivity of solid
L	[m]	Flow length of computational domain
L_f	[m]	Length of fin
$\dot{m}$	[kg/s]	Mass flow rate
ΔP	[Pa]	Pressure difference
q''	[W/m^2]	Heat flux
Q	[m^3/s]	Volume flow rate
Re		Reynolds number based on hydraulic diameter
t_f	[℃]	Thick of fin
T	[℃]	Temperature
ΔT	[℃]	Temperature difference
u	[m/s]	x-directional velocity
v	[m/s]	y-directional velocity
w	[m/s]	z-directional velocity
W	[m]	Width of computational domain
Wc	[m]	Width of channel
x, y, z	[m]	Cartesian axis coordinate
Special characters		
μ	[Ns/m^2]	Dynamic viscosity
ρ	[kg/m^3]	Density
Subscripts		
A,B,C,D		Model A, B, C, D
f		Fin
i		Inlet
l		Liquid
s		Solid

INTRODUCTION

The indirect cooling method using a coldplate with liquid laminar flow channels is prevalent to keep the temperature of highly heat dissipating chips below the maximum absolute ratings while excluding a chip from direct contact to electrical conducting cooling liquid.

For the cooling performance of a flow channel can be enhanced by increasing the effective heat transfer surface area as well as decreasing or disrupting growth of the thermal boundary layer, various studies on the cooling performance of channels with different shapes have been carried out. Some examples of enhanced surface shape are offset-strip fins, louvered fin, perforated fins, and corrugated or wavy fins [1], and many researchers have studied their thermo-hydraulic performance characteristics.

Manglik et al. [2-4] have numerically studied laminar periodically developed forced convection in air (Pr=0.7) in two and three-dimensional wavy-plate-fin channels with sinusoidal wall corrugations, and the variation of Fanning friction factors and Colburn factors with Reynolds numbers presented to compare the performance. Ismail et al. [5,6] have analysed numerically turbulent air flow region of offset strip fin and wavy fin geometries, and presented in the form of Colburn factors and friction factors vs. Reynolds number curves. Kim K. W. and Kim S. J. [7] have carried out two-dimensional laminar numerical analyses for investigating the thermo-hydraulic characteristics of wavy channels with different shape parameters. PAO was considered as the cooling fluid.

Dewan et al [8] have investigated turbulent forced convection of tapered pin fins with circular cross section placed horizontally in the staggered arrangement on a heated base plate numerically under the steady-state conditions. Sahiti et al [9] have been numerically investigated six forms of pin cross-section in order to check how the form of pin cross-section influences the pressure drop and heat transfer capabilities. For airborne applications, experimental studies have been carried out over a wide range of geometric configurations and Reynolds numbers, and friction factor correlations [10] as well as Colburn j-factor correlations [11] have been obtained, and the correlations were applied to the design of pin fin coldwalls by Price et al [12,13].

For some applications such as military phased-array radar, it is essential that the transmit/receive function device operate at similar temperatures to assure similar amplitude and phase operation. It means that it is required to cool the electrical components below a given threshold temperature as well as to maintain the electrical components within a given temperature range. John P. O. C. and Richard M. W. [14] carried out numerical as well as experimental study to control the surface temperature gradient by inserting multiple finstocks into the flow channel resulting in tailoring the convective thermal impedance. Kim K. W. and Kim S. J. [15] have been carried out numerical study to compare the thermo-hydraulic characteristics of channels over a wide range of geometric configurations to select the candidate channels design satisfying thermal requirement.

Up to now, many studies have focused on the channel without solid area and deal with the boundary condition of uniform temperature. Otherwise, this study considers solid area, that is conjugate heat transfer analysis, and uniform heatflux condition, and numerical analysis has been carried out for the several design models to select the candidate channel designs maximizing the cooling performance. In this study, thermo-hydraulic numerical analysis has been carried out to compare the characteristics of selected channels from the results of the previous study [15] with two kinds of coolants such as EGW (EthyleneGlycol/Water) and PAO (PolyalphaOlefin). Single-phase, constant property and three-dimensional laminar forced convections are considered, and the governing differential equations for continuity, momentum, and energy transfer are solved computationally using the finite-volume method and the commercial codes ICEPAK and FLUENT. For the comparison, the variations of the maximum temperatures as well as the temperature difference on the heat source surface with pumping power are compared depending on the coolants and channels.

MODELING AND ANALYSIS

Model Description

Figure 1 shows the front and top views of channels of candidate models. The four candidate channel designs are designated as model A, B, C and D. Model A is a general flow channel occupying the maximum cross sectional area. Model B is a channel with a thick fin in the middle of channel in which dissipating heat from heat source transfers to coolant directly by conduction as well as it has more heat transfer area than model A. Model C is a channel cutting off some part of the fin in model B to increase the heat transfer area as well as decrease the development of thermal boundary layer. Model D stands for a general plain finstock to maximize the heat transfer area. From the viewpoint of manufacturing, machining is needed to form the fins for the model B and C, while the plain finstock for the model D is usually available from a general fin producing company. Vacuum brazing is prevalent to make an enclosure and form a channel these days for the merits of extremely clean, flux-free braze joints of high integrity and strength.

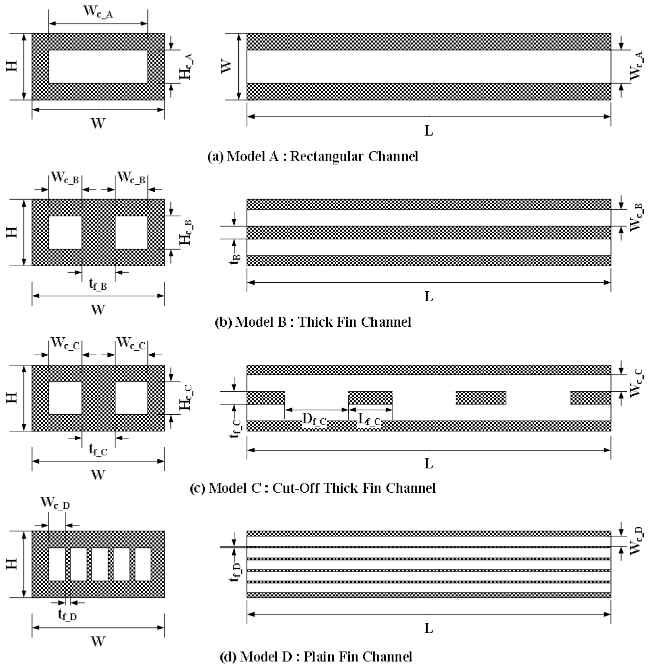

Figure 1 Configuration and nomenclature of the four models: Front and top views.

Table 1 Dimensions of models selected for analysis

Dimensions	Model designation A	B	C	D
L (m)	0.6			
W (m)	0.0156			
W_c (m)	0.007	0.002	0.0025	0.0009
H_c (m)	0.005	0.008	0.008	0.008
t_f (m)	-	0.003	0.002	0.0001
L_f (m)	-	-	0.04	-
D_f (m)	-	-	0.04	-

The dimensions of these models are shown in Table 1, which gives the computational domain length L, the computational domain width W, the channel width W_c, the channel height H_c, the thickness of fin t_f, and the length of fin L_f, of each model.

Mathematical Formulation

For the single-phase, constant property and three-dimensional steady laminar forced convections in the channels, the governing equations for mass, momentum, and energy conservation can be expressed as [16, 17]

For liquid region:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{\partial p}{\partial x}+\frac{\mu_l}{\rho_l}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{2a}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{\partial p}{\partial y}+\frac{\mu_l}{\rho_l}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \quad (2b)$$

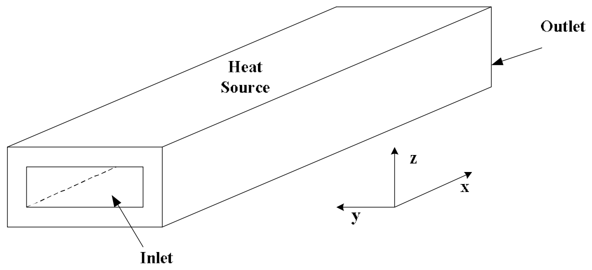

Figure 2 Schematic of coordinates and boundary definition for model A

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{\partial p}{\partial z}+\frac{\mu_l}{\rho_l}\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \quad (2c)$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{k_l}{\rho_l c_{p,l}}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad (3)$$

For solid region:

$$\frac{\partial}{\partial x}\left(k_s\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_s\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k_s\frac{\partial T}{\partial z}\right)=0 \quad (4)$$

The boundary definition and coordinates are shown in figure 2, and boundary conditions can be specified as follows. For the hydraulic boundary conditions, the velocity is zero at all boundaries except the channel inlet and outlet. A uniform velocity is applied at the channel inlet. The volume flow rate Q changes from 0.5 to 2.0 LPM. Pressure outlet condition is applied for outlet boundary condition.

$$u_i=\frac{Q}{A_c} \quad (5)$$

For the thermal boundary conditions, adiabatic boundary conditions are applied to all the boundaries of the solid region except the source surface, where a constant heat flux of 61W is assumed. At the channel inlet, the liquid temperature is equal to a given constant inlet temperature of 20℃.

Constant properties available in the literature are employed and summarized in Table 2, and the reference temperature for liquid properties is 20℃[18]. For liquid, PAO and EGW(50:50) were considered, and for solid, aluminium was used.

Table 2 Thermophysical properties of materials

Material	ρ	μ	k
PAO	770	0.009	0.137
EGW(50:50)	1087	0.0038	0.37
Aluminium	-	-	205

NUMERICAL METHOD

The commercial software of ICEPAK and FLUENT v15 [19, 20] employing the finite volume method was used to solve the governing equations with the predefined boundary conditions with the double precision. The liquid inside the channels was assumed to be steady, laminar and incompressible with the constant properties. The second-order upwind scheme was applied for the discretization of the governing equations, and the SIMPLE algorithm was adopted for the pressure-velocity coupling, while the conjugated heat transfer model was used for solving the energy equation. Under-Relaxation factors were set 0.3 for pressure, 0.3 for momentum, and 1.0 for temperature. As a convergence criterion, the residual of 10^{-4} was used for the continuity equation, momentum equations and 10^{-7} for the energy equation.

Validation of Numerical Models

A grid independency study was carried out to prove the accuracy of numerical solutions. The geometry and mesh were created using ICEPAK mesh for all models. Since the geometric characteristics were determined by taking the model A as reference, the grid independency study was performed for the model. Three cases were investigated: coarse mesh, normal mesh, and local fine mesh. There was no change between the results of the normal mesh and the local fine mesh, however, for the other models more complicate, the local fine mesh was applied. The detailed node numbers are listed in the table 3. The 64.5 of fRe is reasonable considering the entrance effect.

Table 3 Node numbers of four models

Model	Node numbers (local fine mesh level)	Remarks
A	95,760	Coarse mesh 11,550 Normal mesh 13,860 Local fine mesh 95,760
B	228,536	-
C	1,195,920	-
D	711,900	-

RESULTS AND DISCUSSION

In the view point of reliability as well as performance for phase array antenna, it is necessary to operate a high power amplifier below the given temperature in its datasheet, and also temperature difference among the high power amplifiers should be maintained within some range.

Figure 3 shows the maximum temperature distributions on heat source surfaces for four models with pumping powers. The models with liquid PAO yield higher maximum temperature as well as greater pumping power than those with liquid EGW due to its lower thermal conductivity and greater viscosity. As results, the models with liquid PAO needs more pumping power to obtain the same temperature with EGW.

The models of C and D show better thermal-hydraulic performance results. The model D has the largest heat transfer area. The model C has a cutting-off areas in the middle of fin, and it increases the heat transfer area as well as decrease the development of thermal boundary layer. At given pumping power, the maximum temperature of about 50 ℃ can be lowered depends on the channel shape and coolant.

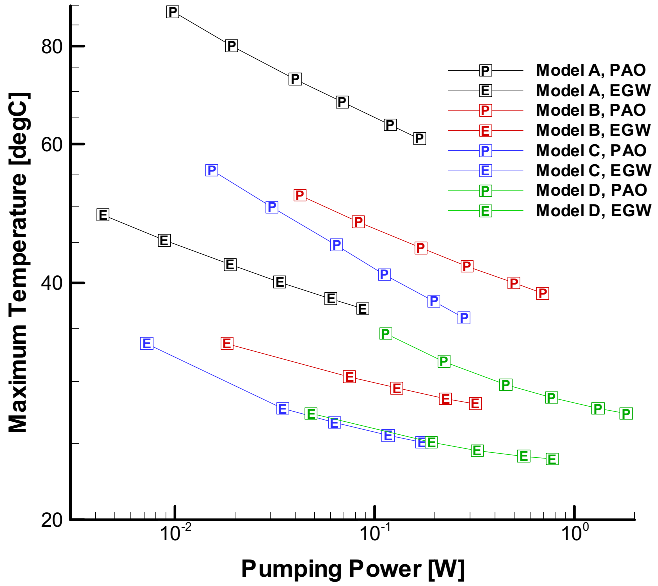

Figure 3 Comparison of the maximum temperatures on heat source surfaces for four models with pumping powers.

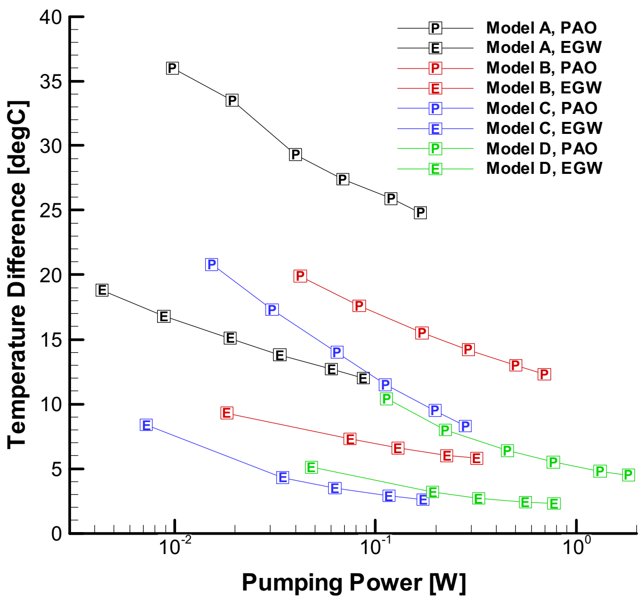

Figure 4 Comparison of temperature differences on heat source surfaces for four models with pumping powers.

Figure 4 shows the temperature difference distribution, which are the difference between the maximum temperature and the minimum temperature, on heat source surfaces for four models with pumping powers, and similar tendency with figure 3. The temperature difference of about 30℃ can be lowered depends on the channel shape and coolant, at given pumping power.

CONCLUSION

Thermo-hydraulic numerical analysis has been carried out to compare the characteristics of the four candidate channels with two kinds of coolants such as EGW (EthyleneGlycol/Water) and PAO (PolyalphaOlefin), and the models of C and D show better thermal-hydraulic performance results because of large heat transfer area and interrupting the growth of thermal boundary layer. The model C is prefer to the model D in the viewpoint of manufacturing, because machining process is better to manage and handle the quality than brazing process.

The authors believe that the model C can be more improved by optimizing the space length and interval between fins to minimize the maximum temperature as well as the temperature difference.

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