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Investigation on Weight Consideration of Liquid Coolant System for Power Electronics Converter in Future Aircraft

Assel Sakanova*¹, Chin Fong Tong¹, Arie Nawawi¹, Rejeki Simanjorang², K.J. Tseng¹, A. K. Gupta²

¹School of Electrical and Electronic Engineering, RR@NTU Corp. Lab, Nanyang Technological University, Singapore 639798

²Advanced Technology Centre, Rolls-Royce Singapore Pte. Ltd

*Corresponding author: Tel.: +65-83133542, E-mail address: asse1001@e.ntu.edu.sg

Abstract

Cooling systems significantly contribute to the total mass and volume of power electronic systems. In the case of aerospace application, it will directly increase the operating cost of the aircraft. This paper experimentally and numerically investigates the weight contribution of the liquid cooling system for power electronics converter in future aircraft. In order to investigate, a cooling system of 2 and 6 pass cold plates is designed and its cooling performance is analysed. The weight and size contribution is discussed based on available coolants in the aircraft, flow rate ranges from 2-8LPM and 1-3% power loss dissipation. Water is added and examined for completeness of the studies. This paper concludes that oil is inappropriate coolant for this particular case. The optimum parameters ($Q=8\text{LPM}$ with 9.5 kg pump weight) for most promising coolant (fuel) that give high extraction rate with low weight contribution for the highest density cooling system are indicated.

Keywords: cold plate, CFD, thermal performance, power electronic cooling

Nomenclature

c_p	specific heat (J/kg K)
h	heat transfer coefficient (W/m ² K)
H	Pump head diameter (cm)
k	thermal conductivity (W/Km)
Nu	Nusselt number
Δp	pressure drop (Pa)
P_{pump}	pumping power (W)
q	heat flux (W/cm ²)
Re	Reynolds number

T	temperature (K)
U	velocity vector (m/s)
u, v, w	flow velocity (m/s)
x, y, z	Cartesian coordinates
<i>Greek symbols</i>	
μ	viscosity (N·s/m ²)
ρ	density (kg/m ³)
<i>Subscripts</i>	
f	fluid
i	inlet
o	outlet
s	solid

1. Introduction

More electric engine (MEE) technology can improve the fuel efficiency, reliability and reduce cost in operation of aircraft as this technology has already been proven in automotive sector. In aerospace sector, the MEE technology is implemented by replacing the mechanically driven engine accessories, oil pump, fuel pumps, hydraulic pump, etc. with electrically driven versions in aircraft. This replacement results in increase of electric loads compared to traditional aircraft system. For high efficiency system these electric loads is supplied from power buses by using power converter as an interface. The weight of power converter has to be considered because it will influence the consumed fuel in aircraft.

Therefore, a high power density converter is required. The power density reflected to volume and weight of power converter. The demand for cooling power converters in aerospace factor is progressively increasing [1] Size and weight are the main challenges in aeronautical industry [2]. Power dissipation density shows tendency to enhance therefore it results in more heat needs to be dissipated. Hence thermal management should not be neglected. The cooling system must able to cope with the increased thermal loads of the upgraded systems without adding too much burden to the weight and size. Natural and forced air cooled heat sink approach is the most

widely used cooling system due to simplicity and most primitive form. However, the stringent requirements of aircrafts make air cooling unsuitable for high power electronic devices. Liquid cooling is an attractive solution for the thermal management of power electronic components. Engine oil and fuel are readily available on-boards coolants. Water is another type of coolant with the relatively good thermos-physical properties. However its application will results in extra pump weight and size penalty as well as system complexity due to non-availability on the aircraft. However there are no studies in available literature regarding usage of oil and fuel in avionics. Although both the coolant is available coolants found in automotive sector. Oil and fuel are used where water is unsuitable. Oil has higher boiling point than water, it can be raised above 100 degrees Celsius without introducing higher pressures within the system

The most widely used and promising liquid cooling methods are mini/microchannel heat sink (MCHS), cold plate heat sink, jet impingement, pin fin heat sink, porous medium, liquid metal cooling and thermoelectric cooling. However some of them have cooling limitations which make them inappropriate for aircraft applications. Jet impingement has not received as much attention as compared with MCHS, due to its design difficulty [3]. Reliable analytical or numerical models for pin-fin heat sinks have not been developed yet due to the complex nature of fluid flow and heat transfer in its pin-fin arrays [4]. Porous medium requires high pump power and bulky packages [5]. Metal cooling (coolant) demands 7 times more pump power than water and leads to severe corrosion. Thermoelectric cooling is not efficient for high heat flux [6]. Due to ease of fabrication and being relatively cost-effective the cold plate heat sink is concluded as an attractive cooling technique for aircraft application.

Liquid cooling system is composed of a chiller, a pump, connection tubes and the cold plate. The cold plate contacts with the heat generation device for to remove the heat from the hot spot [7]. A broad review of the cooling technologies in the component level as well as in the cabinet level is summarised by Chu et al [8].

Mu et al [9] numerically investigated a water-cooled mini-channel heat sink with different flow field configurations under high heat flux. They concluded that heat sink with circular turning configurations performs the best and could be adopted for high –heat flux cases. Xia et al [10] experimentally and numerically investigated the cooling performance of MCHS with complex structure. At the end, they came to the point that MCHS with complex structure is more economical for chip cooling system. Gong et al [11] numerically studied 4 types of MCHS (traditional MCHS, pin-fin MCHS, single-hole jet cooling and double-layer MCHS) with layout consideration. They claimed that jetting cooling heat sink possesses the best chip cooling followed by double – layer MCHS while traditional MCHS has a substantial development potential. Leng et al [12] presented a multi-objective optimization method for double-layer MCHS. Wang et al [13] introduced MCHS with micro-scale ribs

and grooves for chip cooling. They indicated that cooling effectiveness of rib-grooved MCHS is up to 1.55 times higher than the smooth one but leads to the high pressure drop at the same time.

Besides the cooling structure, the author would also like to acknowledge one method to improve the heat transfer which is done by applying nanofluids [14]. An optimal geometric structure for MCHS using nanofluids under different constraint conditions, and the impact of pumping power, volumetric flow rate and pressure drop were obtained by [15]. Passive technique to enhance the cooling performance is studied by [16]. The usage of nanofluids in aircraft application is not investigated in this work.

Many works studied the heat sink optimization and performance improvement with the target to achieve low thermal resistance. However key parameters in aircraft applications such as weight and size of the cooling system have been neglected. Moreover besides widely-used coolants such as water and air, the usage of the already existing fluids in aircraft haven't found due attention yet.

This paper experimentally and numerically investigates the weight contribution to the cooling system for a case study of 50kW power converter in aerospace sector which has not been reported earlier. 2 and 6 pass cold plates are designed and cooling performance is then analysed by applying the fuel, oil and water as the coolants. The cooling performance of these coolants are evaluated by the value of thermal resistance, pressure drop and pump power as well as the contribution to the weight and size of the cooling system. The research is done at the flow rate ranges between 2-8LPM and 1-3% power loss dissipation of 50kW maximum converter power.

2. Numerical method

(a)

(b)

(c)

Fig. 1. (a) 50 kW power converter, (b) structure of power module and schematic of (c) 6 pass, (d) 2 pass cold plates

The 50 kW power converter, structure of power module and schematic diagram of a 6 pass and 2 pass cold plates are shown in Fig. 1. The detailed parameters of the module are listed in Table 1. The cold plate coolant system is designed for three phase two level power converter. The dimensions of cold plates are presented in Table 2. The $L \times w$ size of cold plates fits three CAS100H12AM1 1.2kV, 100A silicon carbide half-bridge power modules (5cm $\times$ 8.8cm) [17]. The gap between power modules is 2 mm. The inlet coolant is defined with a uniform temperature of 70 °C. The outlet pressure is adopted with atmosphere pressure. 6 pass cold plate is designed in such a way that its copper pipe runs underneath the position of MOSFETs die inside the module to enhance heat extraction. The effective copper pipe length inside the cold plate is $6 \times w$ and $2 \times L$ for the 6 and 2 pass cold plates accordingly.

To investigate the impact of flow rate and types of coolant under 1-3% power loss of 50 kW power converter, flow rate ranging from 2LPM to 8LPM, and water, oil and fuel as the coolants are considered.

Name	Thermal conductivity (W/mK)	Height (mm)
SiC	420	0.18
Solder	58.7	0.15
Copper (DBC)	387.6	0.3
SiN	150	0.3
AlSiC	170	3
Thermal grease	5	0.1
Aluminum	170	7
Copper (cold plate)	387.6	6

Table 1 Height and material properties of the module

<i>Param</i>	<i>L</i>	<i>w</i>	<i>H</i>	<i>a</i>	<i>b</i>	<i>C</i>	<i>d</i>	<i>e</i>	<i>f</i>	<i>j</i>	<i>k</i>	<i>δ</i>
<i>eter</i>	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)
2 pass	268	50	7	25.4	50.8	25.4	6	-	12.3	2	88	3
6 pass	268	50	7	25.4	50.8	25.4	6	60.7	28.5	2	88	3

Table 2 Geometric dimensions

The boundary conditions are listed as below:

Channel inlet:

$$u = u_{in}, \quad v = 0, \quad w = 0, \quad T = T_{in} \quad (1)$$

Channel outlet:

$$p = p_{out} \quad (2)$$

Coolant – solid interface:

$$u = v = w = 0, \quad T_f = T_s, \quad -k_f \frac{\partial T_f}{\partial n} = -k_s \frac{\partial T_s}{\partial n} \quad (3)$$

Bottom wall of the heat sink:

$$q_w = -k_s \frac{\partial T_s}{\partial n} \quad (4)$$

Other solid walls and symmetric boundaries:

$$-k_s \frac{\partial T_s}{\partial n} = 0 \quad (5)$$

The assumptions below are taken to simplify the analysis:

- 1) Single-phase, incompressible laminar (oil) and turbulent flow (water, fuel). Since the studies are under constant flow rate, the Re number is different for each type of coolant. Because oil has very high viscosity the flow becomes laminar while under same flow rate the flow is turbulent in the cases of water and fuel.
- 2) The effect of gravitational force and heat dissipation caused by viscosity is negligible
- 3) Natural convection and radiation are neglected

Under these simplifications, 3D steady governing equations for the conjugated heat transfer can be written as follows [18]:

Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (6)$$

Momentum equation

$$\rho_f \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial P}{\partial x} + \mu_f \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (7)$$

$$\rho_f \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \mu_f \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \quad (8)$$

$$\rho_f \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial P}{\partial z} + \mu_f \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \quad (9)$$

Energy equation for the coolant

$$\rho_f c_{p,f} \left(u \frac{\partial T_f}{\partial x} + v \frac{\partial T_f}{\partial y} + w \frac{\partial T_f}{\partial z} \right) = k_f \left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right) \quad (10)$$

Energy equation for the solid region

$$0 = k_s \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right) \quad (11)$$

Based on ANSYS Fluent, the 3D conjugate heat transfer problem is numerically solved. Implicit solver option is used to solve the governing equations. The second order upwind scheme is adopted for both energy and momentum discretization. While for pressure discretization, standard interpolation scheme is adopted. The pressure-velocity coupling is implemented by SIMPLE algorithm. The convergence criteria for the x , y , and z directions velocity are set to 10^{-6} while the residuals of energy equations are restricted to 10^{-7} .

To evaluate the mesh density of the simulation model, 3 types of hybrid mesh with number of elements of 11286545 (mesh 1), 4876121 (mesh 2) and 1206623 (mesh 3) are investigated. Fig 2(a) shows the temperature along the base plate of the module A when phase A turns-on is captured every 10 cm. And pressure drop distribution along the pipe is displaced in Fig. 2(b). The difference in maximum temperatures is 1% between mesh 1 and mesh 2, while it is 2% mesh 1 between and mesh 3. With respect to pressure drop distribution, the maximum deviation occurs with 0.14 % between mesh 1 and mesh 2 and 0.25 % between mesh 1 and mesh 3. Finally, mesh 2 is selected as the best trade-off between accuracy and CPU time.

Fig. 2. Comparison of temperature (a) and pressure drop distribution (b) at different mesh densities.

3. Experiment

Fig. 3 shows the experiment set up in this study. It consists of a power supply, liquid and temperature measuring instruments, power module and cooling unit. The flow rate through the loop, inlet-outlet coolant temperatures, and pressure drop along the loop are some of the important parameters in the heat transfer and fluid dynamic research. The description of the equipment used in the experiment and its experiment conditions are presented below.

K-type thermocouples (PTFE Exposed Welded Tip Thermocouples) are inserted into the coolant in order to measure the inlet (T_{in}) and outlet temperatures (T_{out}). Two other K-type thermocouples (T_2 , T_3) are mounted on the power module base to test the wall temperature as shown in Fig. 1. The data logger (Hioki LR 8400-20) is used to monitor, log and analyse the output of the thermocouples. The pressure drop across the loop is measured by the pressure gauge (YEA/THEI).

The cooling unit consists of the chiller machine (Altech Industrial chiller HT-01A), K-type thermocouples and cold plate. The allowable inlet temperature ranges between 5 to 40 °C. The flow rate can be adjusted by a valve at the inlet. TDK Lambda current source power supply GEN600/8.5 along with Matrix MPS -3005TK-3 voltage source supply are used to provide the power supply. Fluke 8840A digital multimeter is used to measure the voltage.

The experiment process can be explained as follows. A 32 °C water from chiller machine is supplied to the cold plate in order to cool the device down. It absorbs the heat from the module and then returns to chiller machine to be cool down to the initial temperature. For this study one module with 5 MOSFETs die is applied. The experiment is conducted under

1. 0.5 - 2% (250-1000W) power loss of 50kW power converter and
2. Flow rate ranges between 1 - 5LPM.

The power loss, P is produced by an electric current, I passing through the power device with a resistance, R . Since, power loss, P is the product of voltage and current. A multimeter is used to monitor the voltage drop across drain-source terminal of the power device when the current source is turn-on. Table 3 summarised the voltage and current combination that produce various power loss, P . Temperatures are recorded at steady state. The experiment conducted with the aim to verify the simulation model.

Power loss (%)	Total power loss (W)	Power loss per module (5 MOSFETs) (W)	U (V)	I (A)
0.5	250	42	0.84	49.9
1	500	84	1.33	63
1.5	750	125	1.53	83.1
2	1000	167	1.76	93.8

Table 3 Heat source generation parameters

(b)

Fig. 3. (a) Schematic of experimental setup (b) graphic picture of experimental setup

To verify the simulation model, we compared the simulation results with experiments under different flow rate with inlet temperature of 32 °C. The temperature is measured at the inlet (T_{in}), outlet (T_{out}) and two different point of each power module (T_2 and T_3).

The geometries is built by using Gambit and while the hybrid mesh is generated by ANSYS Workbench. ANSYS Fluent R17.0 is employed to solve the 3D conjugate heat transfer problem. Two turbulent models such as $k-\omega$ SST and $k-\varepsilon$ with enhanced wall treatment are used and validated in this study. When the $k-\varepsilon$ with enhanced wall treatment model is employed, the distance between the near-wall cell centroid and the wall is $y^+ \leq 3$. In fact, the y^+ value is in the order of unity. The number of the prismatic layer is 20. The grid for $k-\omega$ SST model is the same as for the enhanced wall treatment since both turbulent models have similar near-wall requirements. The same turbulent models comparison has been done in [19].

Fig. 4 shows the comparison of the selected turbulent models with experiment in term of temperature and heat transfer coefficient at the base plate where the temperature is monitored during the experiment. The max differences between experiment and simulations results for both turbulent models is not exceed 3%. However the results of $k-\varepsilon$ model for T_2 and T_3 values are closer to experimental one. While in term of the T_{out} both models predicts well. Based on better accuracy, the $k-\varepsilon$ with enhanced wall treatment model is selected in this study. Same turbulent model is selected in [20].

Fig. 4. Experimental and numerical data for (a) temperature and (b) heat transfer coefficient at the base plate of MOSFET

A good agreement between experiment and simulation results is clarifying the accuracy of the analysis through FVM simulation. For further analysis the simulation tools are used to observe the effect of the liquid coolant variations (type, flow rate, power loss) against weight of the cooling system.

4. Results and discussions

4.1 Types of coolant

In this study four types of coolant are considered (air, water, oil and fuel). Some of these coolants are readily available in aircraft at 70 °C. Their thermo-physical properties are shown in Table 4 at this temperature.

Air cooling is the preferred options to cool the converter due to ease of use. However the limitations in properties such as low specific heat, low mass density as well as a low heat transfer coefficient make it impractical for high power density converters [21].

The alternative way to cool converter down is to use liquid as a coolant. Water is one of the widely-used types of cooling owing to its good thermo-physical properties. But it carries a significant weight penalty at the aircraft level as it requires additional heat exchange system. Oil can be selected as a coolant option due to its availability in aircraft system. Moreover oil properties such as heat capacity and heat transfer capability provide low temperature difference between coolant and heat source which makes it valuable for aircraft application. In this study SAE 10W/40 engine oil is selected [22].

The final option of coolant is fuel. Besides significant weight reduction since there is no necessity to install additional heat exchanger system, fuel thermo-physical properties are quite comparable with those of oil. Furthermore, while oil requires a high pump power due to high viscosity, fuel viscosity value is close to that of water.

Property	Air	water	oil	fuel ^[23]
Density (kg/m ³)	1	977.63	860	770
Heat capacity (J/kgK)	1010	4190	2080	2450
Viscosity (kg/ms)	2.1e-05	0.000404	0.053	0.000308
Thermal conductivity (W/mK)	0.03	0.67	0.139	0.105

Table 4 Thermo-physical properties of coolants

4.2 Effect of power loss percentage

Fig. 5 shows simulation results of the maximum temperature on the MOSFET and thermal resistance under power loss in three phase inverter six switch ranges from 1 to 3% of 50kW power converter by applying different types of coolant at the flow rate of 2LPM. According to the datasheet maximum operating junction temperature of CAS100H12AM1 1.2kV, 100A silicon carbide half-bridge power module is 125 °C as it is indicated by red dash line. Overall temperature on the MOSFETs increases linearly as power loss percentages increase for all types of coolants in this study. Water as a coolant provides the lowest temperature followed by fuel and oil accordingly. Moreover the difference in maximum temperature between coolants increases with power loss. However only water and fuel are able to provide a temperature less than maximum operating temperature at 1% of power loss. While water can meet the requirements of $T_{max} < 125$ °C up to 1.5% of power loss. Table 5 illustrates pressure drop and pump power of all types of coolants. Fuel yields the lowest pressure drop as well as pump power followed by water while oil results in double the fuel due to high kinematic viscosity.

Fig. 5. (a) Maximum temperature and (b) thermal resistance versus power loss at flow rate of 2LPM

Type of coolant	Pressure drop (kPa)	Pump power (W)
Water	122.693	4.08
Oil	239.211	7.96
Fuel	118.015	3.93

Table 5 Pressure drop and pump power under 1-3% of power loss

Fig. 6 shows the temperature of the coolant at the outlet of the cold plate. The temperature difference between inlet and outlet depends on the thermo-physical properties of the coolant. As higher the values of properties such as density, heat capacity and thermal capacity as lower the temperature difference between inlet and outlet. While the viscosity has the reverse effect. That is the reason why water results in the lowest temperature rise followed by fuel and oil. For a lower outlet temperature lower pump power is needed. Fig. 7 shows the temperature distribution and pressure drop under 1% of power loss at a flow rate of 2LPM. Its noted that in the cases of water and fuel the temperature is uniformy distrubited along the three power modules of power converter while oil is unable to remove the heat and the MOSFETs are overheated. Overall pressure drop is higher in the entrance regions of the pipe due to flow acceleration and increased wall shear stress and at the end of the pipe it is equal to the atmospheric pressure. Fuel results in the lowest pressure drop followed by water and oil respectively.

Fig. 6. Outlet temperature under 1% of power loss and flow rate of 2LPM

Fig. 7. (a-c) Temperature distribution and (d-f) pressure drop across the 6 pass cold plate of water, fuel and oil accordingly

4.3 Effect of flow rate

Fig. 8 presents the temperature, pressure drop and pump power on flow rate ranges from 2 to 8LPM under 1%, 2% and 3% of power losses. It is apparent from the Fig.8 that temperature decreases with increasing the flow rate. This is because larger amount of working fluid is capable of carrying out the same portion of heat flux. However the temperature saturates at the flow rate 8LPM. Further increase in flow rate does not improve the cooling performance of cold plate. As discussed before, both water and fuel are able to maintain the temperature on the MOSFETs at less than maximum operating junction temperature (125 °C) even at 2LPM for 1% power loss. Nevertheless in the presence of oil the thermal resistance is decreasing more noticeably with flow rate however it becomes constant by reaching flow rate of 8LPM as well. In a case of 2% power loss none of the coolants are able to ensure temperature lower than operating junction temperature at a low flow rate. Although once the flow rate

is more than 2LPM and 8LPM water and fuel respectively can be used as the coolants, oil must be eliminated. On the condition that 3% of power loss is applied, neither of the coolants can meet the requirements even at high flow rate. Generally pressure drop increases with flow rate. Pressure drop of water and fuel are close enough at low flow rate and differ with flow rate. Fuel results in the lowest pressure drop whereas oil in the highest. As the pumping power is a function of pressure drop, it shows a similar tendency as pressure drop.

Fig. 8. Simulation results of MOSFET temperature at (a) 1% power loss, (b) 2% power loss, (c) 3% power loss, (d) pressure drop and (e) flow rate versus flow rate

Fig. 9 presents the heat transfer coefficient with respect to flow rate at power loss ranges from 1 to 3%. Heat transfer coefficient is determined by $h=q/A(T_w-T_m)$. Overall water has the highest heat transfer coefficient followed by fuel and oil respectively. The heat transfer coefficient increases with flow rate. However nearly at 8LPM it saturates and further increase in flow rate doesn't show significant improvement. The value of heat transfer coefficient is compared with J. Li et al. [24] where h is a bit lower in comparison with this study due to low Re .

Fig. 9. Simulation results of heat transfer coefficient versus flow rate at (a) 1% of power loss (b) 2% of power loss (c) 3% of power loss

4.4 Effect of 2 pass cold plate

Fig. 10 shows the temperature distribution and pressure drop under 1% of power loss at flow rate of 2LPM. The cooling performance is the same as for 6 pass cold plate where water yields the lowest temperature among all types of coolants while oil the highest. However as compare to 6 pass cold plate, even in a case of water, 2 pass cold plate is unable to cool the temperature on the MOSFETs down to less than the operating junction temperature under conditions of 2% power loss, as Fig. 11(b) shows. Even though it has longer effective pipe length than 6 pass cold plate. This can be explained as follows, since the pipe of the 6 pass cold plate is designed in that way to make sure that the MOSFETs are placed exactly under it, eventually it leads to a better conduction heat transfer rather than longer effective pipe length. On the other side, 2 pass cold plate provides lower pressure drop as well as pumping power as compared with 6 pass cold plate. This is due to less number of bends which contribute to the additional penalty in pressure drop and pump power. Temperature is saturated at 8LPM, which means further increase in flow rate does not improve the cooling performance while it will result in an additional penalty in pressure drop along with pump power.

(c)

(d)

(e)

(f)

Fig. 10. Simulation results of (a-c) temperature distribution and (d-f) pressure drop across the 2 pass cold plate of water, fuel and oil accordingly

(a)

(b)

Fig. 11. Simulation results of (a) temperature at 1% power loss, (b) temperature at 2% power loss, (c) pressure drop and (d) flow rate versus flow rate

Fig. 12 presents heat transfer coefficient versus flow rate. The overall trend is the same as 6 pass cold plate. However heat transfer coefficient is lower for 2 pass cold plate as compared to 6 pass cold plate.

Fig. 12. Simulation results of heat transfer coefficient versus flow rate at (a) 1% of power loss (b) 2% of power loss

For the reason to compromise higher heat transfer coefficient and associated higher pressure losses, it's suitable to express the results in term of aerothermal efficiency which conceive how much pressure is needed to get a particular level of heat transfer enhancement [19] as shown in Fig. 13.

$$\eta = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{1/3}} \quad (12)$$

Where Nu and f are Nusselt number and friction factor ($Nu = hD_{hi}/k$, $f = \Delta PD_{hi}/(0.5\rho U_i^2 L)$), Nu_0 and f_0 are Nusselt number and friction factor in a smooth channel from Blasius and Dittus–Boelter relations for turbulent flow

$(Nu_0=0.023Re^{0.8}Pr^{0.4}, f_0=0.184Re^{1/5})$ and from Sieder and Tate relations [25] for laminar flow $(Nu_0=1.86(RePrD/L)^{1/3}(\mu_b/\mu_s)^{0.14}, f_0=64/Re)$.

Fig. 13. Effect of flow rate on aerothermal efficiency for pass (a) 6 and (b) 2 pass cold plate at 1% power loss

4.5 Weight and size consideration

In aircraft the cooling system having stringent space and weight limitations. This leads to the necessity of investigation on weight consideration of liquid coolant system for power electronics converter. Weight of the pump is the main contributor to the existing cooling system (fuel, oil) while in a term of non-existing coolants (water) on the aircraft heat exchanger and all pipe connections must be taken into account as well. However some authors while calculating the power converter weight distribution limited only to heat sink weight consideration and didn't take the cooling system pump weight into account [26]. There should always be a trade-off between cooling performance and weight of the cooling system. It is obvious that the weight and size of the pump is increasing with pump power. Total cold plate weight is a summation of the cold plate weight and weight of the coolant. The 6 pass cold plate weight is 0.55kg and 2 pass is 0.5kg. As far as the weight of the coolant is considered, the usage of liquid coolant contributes more to the total weight of the heat sink compared to air coolant. Since the weight of the coolant is directly proportional to the density of the coolant, water results in the heaviest heat sink followed by oil, fuel and air respectively. However the maximum coolant weight contribution to the total weight of heat sink in this study is 10.5%, 9.2%, 8.2% and for water, oil and fuel accordingly.

$$Weight_{coolant} = \rho \cdot \pi \cdot r^2 \cdot l \quad (13)$$

Where, l is a pipe length of the cold plate ($l_{6pass}=0.098m$ and $l_{2pass}=0.071m$)

In order to investigate the cooling system pump of Guangdong Lingxiao pump industry Co., LTD (CMF series) company is chosen [27]. Though pump series CMF is not exactly a specific pump for the power converter for aircraft application. Since selected power converter has not been installed in aircraft yet the selected pump is considered at the moment in this study. Same pump is used in an experimental work of this study. CMF series is a casing-oriented multistage stainless steel pump. Under the processing of advanced welding equipment, heat treatment, the stress caused by end-cut plate in tension and press forming is eliminated to make it of high strength, with no deformation, long life, for a safe and reliable usage. It is under low-noise and less-vibration and durability. CMF series pump is used in the experimental part of this study when water is applied as a coolant. Also it was used in simulations when all three coolants (water, fuel, oil) are considered.

Fig. 14 represents the pump head, pressure drop and pump power dependence on flow rate. Pump head is defined as the useful mechanical energy transferred to the low per weight of the fluid handled and impact on the size of the pump. Pump head increases with pressure drop and decreases with flow rate. It means pressure drop decreases as flow rate increases under constant pump power. Fig. 14(b) as well as Table 6 show that pump power dependence on flow rate and corresponding pump weight.

Since the 6 pass cold plate results in higher pressure drop compared to 2 pass cold plate, further discussions will be based on the worse case consideration. In case of 1% power loss both water and fuel can be applied at 2LPM. Under these circumstances CMF2-20 can be chosen with pump head diameter of 14cm and weight of 8kg. On the condition that oil is selected even the most powerful pump in this range cannot provide the temperature less than it is needed. If power converter dissipates 2% power loss water can be adopted with CMF2-30 under pump head diameter of 21cm and weight of 8.5kg while CMF2-40 with 32cm and 9.5kg is required in the presence of fuel. All results are summarised in Table 7. It means the fuel usage as compared to water will increase the pump head size and pump weight by 66% and 12% accordingly at 2% of power loss. There is no weight and size contribution at the 1% power loss using fuel as a cooling agent.

Fig.14. (a) Pump head diameter, pressure drop and (b) pump power versus flow rate

Model	Driving motor (kW)	Q (LPM)	16.7	24.9	33.3	41.6	49.9	58.2	Weight (kg)
CMF2-20	0.37	H (cm)	18	16	14	13	11	10	8
CMF2-30	0.37		27	24	21	20	17	14	8.5
CMF2-40	0.55		35	32	28	26	23	17	9.5
CMF2-50	0.55		43	40	35	33	28	22	11
CMF2-60	0.75		50	48	42	38	32	25	12

Table 6 Pump performance table

1%			2%		
Size (cm)	Weight (kg)	Flow rate (LPM)	Size (cm)	Weight (kg)	Flow rate (LPM)
Fuel	14	8	32	9.5	8
Water	14	8	21	8.5	3

Table 7 Pump weight and size

4.6 Comparison of power devices

One of the possibility to decrease the size and weight of the cooling system of the converter in aircraft is to use the new generation devices. This is due to that the new generation devices have lower thermal resistance and higher operating junction temperature. Table 8 compared pump weight and size of the CAS100H12AM1 and CAS120M12BM2 power devices. Since the thermal resistance Junction-to-Case for MOSFET of CAS120M12BM2 power device is lower than for CAS100H12AM1, it results in lower weight and smaller size of the pump [28]. The usage of CAS120H12BM2 gives further benefits: The weight is lighter by 16% and 6% in term of fuel and water accordingly at 2% power loss. The size is smaller by 56% and 33% in term of fuel and water accordingly at 2% power loss. At 3% power loss water still able to maintain the temperature lower than junction temperature.

1%				2%				3%			
Size (cm)		Weight (kg)		Size (cm)		Weight (kg)		Size (cm)		Weight (kg)	
CAS10	CAS12	CAS10	CAS12	CAS10	CAS12	CAS10	CAS12	CAS10	CAS12	CAS10	CAS12
0H12A	0M12B	0H12A	0M12B	0H12A	0M12B	0H12A	0M12B	0H12A	0M12B	0H12A	0M12B
M1	M2	M1	M2	M1	M2	M1	M2	M1	M2	M1	M2
F	14	14	8	8	32	21	9.5	8.5	-	-	-
ue											
l											
W	14	14	8	8	21	14	8.5	8	-	14	-
at											
er											

Table 8 Pump weight and size comparison of CAS100H12AM1 and CAS120M12BM2 power devices

5. Conclusions

In this work, weight consideration in liquid coolant system for power electronics converter is studied. Cooling systems with aircraft existent liquid (oil and fuel) as well as non-existent liquid (water) is investigated. 6 and 2 pass cold plates for power converter have been numerically designed using ANSYS and then experimentally verified. The experiment and simulation results are well matched. Further numerical research is conducted under conditions where the coolant inlet temperature is at 70 °C. The attractiveness of coolants is evaluated based on the main aircraft issues such as weight and size.

6 pass cold plate is shown to be better in comparison with 2 pass cold plate. Even though the effective pipe length of 2 pass cold plate is longer than 6 pass, the arrangement of copper pipe exactly underneath the MOSFETs plays more significant role in heat removal.

In a case of 1% power loss, both water and fuel are able to maintain the temperature on the MOSFETs less than operating junction temperature of 125 °C with flow rate of 2LPM. In those cases, there is no pump weight and size penalty for both coolant. If power loss is increased to 2%, although both water and fuel are able to keep the maximum operating junction temperature below 125 °C, there is a penalty of 12% pump weight and 66% pump head size increase in fuel when comparing with water.

Water exhibits the best performance among other coolants; however the non-existing cooling system results in an additional weight and size penalty. Despite the desirable electrical insulating property of oil studied in this case, it cannot be applied in aircraft due to incapacity to meet the requirements and high pressure drop unless high viscous oil is changed to low viscous oil. Fuel is the most promising coolant since it satisfies the necessary cooling conditions as well it provides the least pressure drop therefore leads to the lighter pump weight and size.

Further improvement in cooling system in aircraft as well device is considered. The advanced cooling system will further decrease thermal resistance while the new generation devices increase the maximum operating temperature which make oil as the potential coolant. The usage of low viscosity oil could also make it as applicable coolant agent on aircraft.

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