Thermal Characterization of Battery Cold Plates

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Abstract

Proper thermal management of Lithium-ion batteries is a crucial design consideration in electric vehicles (EV). Liquid cooling is the preferred cooling technology for these batteries due to its high heat transfer coefficient and compactness. Cold plates utilized in electric vehicles need to maintain a battery temperature range of 20-40C and a temperature uniformity of less than 5C between the batteries. Design and optimization of cold plates require tradeoffs between conflicting requirements including thermal resistance, pressure drop, and manufacturing constraints. In the case of EV batteries, it is also very important to consider the surface temperature uniformity of the cold plate. The main goal of this paper is to thermally characterize EV battery cold plates using key design & performance parameters relevant to eMobility. This will help engineers develop and optimize solutions quickly that meet their performance targets.

A typical battery cold plate was chosen for this study with the dimensions of 250 x 500 x 10mm and a uniform heat load of 500W on both sides. The coolant used was a mixture of ethylene glycol and water. A simulation model was created using the commercially available CFD tool FloTHERM. Several design parameters were varied including fluid channel height, the number of flow turns, fin pitch, and type of coolant to determine their impact on the thermal performance. The thermal performance was characterized in terms of the effectiveness (ε) which is inversely proportional to the thermal resistance of the cold plate and the coolant mass flow rate. The temperature uniformity was calculated using the difference between the maximum and minimum surface temperature of the cold plate (ΔT_{MAX}). The power required for cooling (P) was also calculated as a product of the volumetric flow rate and pressure drop of the cold plate. It was found that increasing the number of flow turns and adding fins resulted in the most significant improvements in ε and ΔT_{MAX}.

INTRODUCTION

Lithium-ion (Li-ion) batteries are very popular in Electric Vehicles (EVs) because of their high energy density. However, they are highly sensitive to operating temperatures. Heat generation occurs due to a combination of joule heating due to resistive losses as well as due to electrochemical reactions [4]. The high amounts of heat make thermal management crucial to battery performance, safety, and reliability.

Typically, the batteries need to be maintained in a narrow temperature range of 20-40C to optimize the electrochemical performance, maximize power capability and minimize thermal aging. The temperature uniformity between the batteries is another very important consideration and needs to be maintained below a set threshold to avoid electrical imbalances. Also, if thermal management is not properly implemented, there is a safety risk of a thermal runaway problem, when elevated temperatures cause more exothermic chemical reactions which lead to even higher temperatures.

The heat dissipation of Li-ion batteries is proportional to the current flowing through them. The maximum current occurs during fast charging and depends on the charging rate or C- rate. The C-rate determines how quickly the battery is being charged.

For example, if a 5Ah battery needs to be charged in 1 hour, it is considered a 1C charging rate and the current required is 5A. To calculate the heat dissipation, multiply the square of the current with the impedance of the battery. So if the impedance is 10 milliohms, the heat dissipation will be 0.25W for a 1C charging. The heat dissipation goes up rapidly for higher C-rates. For a 5C charging rate (1/5 hrs), the heat dissipation will be 6.25W.

Traditionally, air cooling and liquid cooling are the most common cooling technologies for batteries. While air cooling has advantages in terms of its design simplicity and cost, the low heat transfer coefficient in air cooling becomes a big limitation to achieving the thermal resistance and temperature uniformity targets, especially when heat loads are high (fast charging). Also, when utilizing air cooling, large gaps need to be placed between the batteries to allow the air to flow through, increasing the overall volume of the battery pack.

Liquid cooling, however, provides the significantly higher heat transfer coefficient required to achieve the thermal performance targets. Since a liquid cooling system is already required for the HVAC system, the incremental cost of providing coolant for the batteries is mitigated. Another type of cooling method which is relatively uncommon is using phase change material (PCM). This has limitations in terms of the poor energy density of the PCM and low thermal conductivity, restricting its use to low power applications such as electric scooters.

Liquid cooling is growing in popularity in a wide variety of applications including servers, power electronics, and EVs due to its high heat transfer coefficient and smaller footprint. Liquid cold plates can be customized in a wide range of geometries, sizes, and manufacturing methods based on requirements and use. Electric Vehicle cold plates are typically thin (< 15 mm thickness) with a very large surface area to accommodate all the batteries. Due to cost and weight constraints, aluminum is the material of choice. Their construction can vary from simple tube type to more complex brazed fin assemblies. Figure 1 shows some of the commonly used types of battery cold plates. Maintaining surface flatness is one of the main manufacturing challenges with these designs.

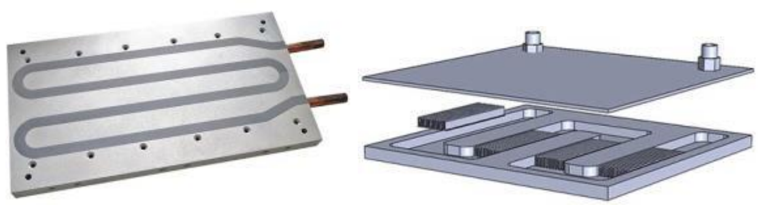

Fig. 1.
Common types of cold plates with tube type (left) and brazed fin assembly type (right)

Another important constraint is the surface temperature uniformity which is required for the electrical balance between the batteries. Since battery cold plates have a large surface area, maintaining temperature uniformity becomes very challenging. Optimized cold plate designs must be able to meet these thermal requirements, maintain safety and reliability standards, and be cost-effective and easy to manufacture. In this paper, we explore ways to effectively characterize the thermal performance using design and performance parameters that are suitable for battery cold plates.

PROBLEM DESCRIPTION

For this study, a typical battery aluminum cold plate design was selected. The overall dimensions are 250 x 500 x 10mm for the baseline with small inlet and outlet ports as shown in figure 2. It has 2mm top and bottom covers with three flow baffles in the fluid region. A heat load of 500W is applied on both sides of the cold plate. This heat load is approximately equivalent to a pack of 100 cylindrical batteries at a 5C charging rate. The coolant used is a 30-70 mixture of ethylene glycol and water with a flow rate of 5 liters per minute (LPM) at an inlet temperature of 20C. These chosen values are typical of battery cold plates in practical applications.

Fig. 2. Schematic of baseline battery cold plate showing top view (above) and side view (below) with flow baffles

Computational fluid dynamics (CFD) simulations are carried out using FloTHERM to determine the thermal performance of the battery cold plates for different design configurations. The simulation model was kept as simple as possible using a fixed flow boundary condition at the inlet and a free pressure condition at the outlet. A structured hexahedral mesh was used. The heat load from the batteries was modeled as a uniform heat source applied over the entire top and bottom surface of the cold plate. All simulations were carried out under steady-state conditions.

The simulation model was first run for a baseline case which consisted of a fluid height of 6mm with 3 flow baffles (or 3 flow turns) using a 30-70 mixture of ethylene glycol and water at 5 LPM. No internal fins were added yet. Using the baseline case as a reference, several design parameters were varied to evaluate their effect on thermal performance including thermal resistance, surface temperature uniformity, and pressure drop. The following design parameters were varied, all relative to the baseline case.

1. The height of the fluid channel varied from 4 to 8mm.
2. The number of flow baffles (or flow turns) was varied from 1 to 7
3. The coolant flow rate was varied from 1 to 7 LPM
4. Adding fins and varying the fin pitch from 3mm to 7mm

THERMAL PERFORMANCE PARAMETERS

A set of suitable thermal performance parameters were defined to compare the thermal characteristics of different types of cold plates. These performance parameters were carefully chosen to be generally applicable to all types of battery cold plates and include not only the thermal performance but also the cooling power required.

The effectiveness of the cold plate, ε, is a dimensionless parameter defined as the ratio of the actual heat load removed by the coolant (Q) to the maximum possible heat load that can be removed (Q_{max}) at the same coolant flow rate [6].

If T_{fi} and T_{fo} are the inlet and outlet temperature of the coolant and T_{max} is the maximum surface temperature of the cold plate, then Q and Q_{max} can be written as

$$Q = m\, C_p\, (T_{fo} - T_{fi}) \qquad (1)$$
$$Q_{max} = m\, C_p\, (T_{max} - T_{fi}) \qquad (2)$$

Where m is the mass flow rate and C_p is the specific heat of the coolant.
Therefore, the effectiveness of the cold plate ε can be written as

$$\varepsilon = Q/Q_{max}$$
$$= (T_{fo} - T_{fi}) / (T_{max} - T_{fi}) \qquad (3)$$

Equation (3) shows that ε has a theoretical maximum value of 1 when T_{max} equals T_{fo}. This is an ideal condition that occurs when a given coolant flow rate is utilized to extract the maximum possible heat from the cold plate.

Using (1) and the commonly used parameter called thermal resistance of the cold plate ε which is defined as

$$\theta = (T_{max} - T_{fi})/Q \qquad (4)$$

Using (3) and (4) ε can be rewritten in terms of θ as,

$$\varepsilon = 1 / (m\, C_p\, \theta) \qquad (5)$$

Equation 5 shows the relationship between effectiveness and thermal resistance of the cold plate. For a given fluid (C_p) and mass flow rate (m), ε is inversely proportional to θ. Maximizing ε, therefore, involves minimizing the thermal resistance for a given coolant flow rate. Conversely, for a given thermal resistance specification, maximizing ε involves minimizing the coolant flow rate (energy efficiency). The cooling energy efficiency is an important criterion for battery cold plates because it is linked to the pump sizing and the overall energy efficiency of the electric vehicle.

Temperature uniformity between the batteries is a critical requirement for EV batteries to maintain the electrical balance. For liquid-cooled batteries, the temperature variation between the batteries is closely tied to the temperature variation of the cold plate surface. There are several different ways to measure the surface temperature uniformity of the cold plate. Since the specification is usually defined as the difference between the minimum and maximum surface temperature of the cold plate (ΔT_{max}), this was chosen as the measure of temperature uniformity.

Since the energy efficiency of the cooling system is directly linked to the overall EV energy efficiency, the cooling power (P) is another important performance parameter. P is defined here as the product of pressure drop, Δp, and volumetric flow rate of the coolant, V. In SI units, P is measured in Watts.

$$P = V * \Delta p \qquad (6)$$

SIMULATION RESULTS

The baseline case was first run and analyzed to evaluate the thermal performance using a fluid height of 6mm, 3 flow baffles (or turns), and no fins. Several mesh sizes were evaluated until the solution was found to be mesh independent.

Figure 3 shows the surface temperature and speed distribution inside the cold plate at a coolant flow rate of 5 LPM at an inlet temperature of 20C. As expected, the cold plate surface temperature rises from the inlet to exit due to the rise in the fluid temperature. Recirculation regions are seen near the inlet and at the end of each turn causing a local rise in temperature above those regions. The inlet fluid channel is significantly cooler than the other channels due to the high heat transfer coefficient near the inlet jet. The maximum temperature of the cold plate was around 27C and the surface temperature variation ΔT_{max} was around 5.4C, both within the typical performance requirements of battery cold plates. The fluid temperature rise from inlet to outlet is around 3.1C. The thermal resistance and pressure drop were calculated to be 0.07 C/W and 2276 Pa respectively. The pressure drop is due to the combined effects of friction, flow turns, and contractions at the inlet and outlet. Using equations (5) and (6) the values of ε and P were calculated to be 0.44 and 0.19W respectively.

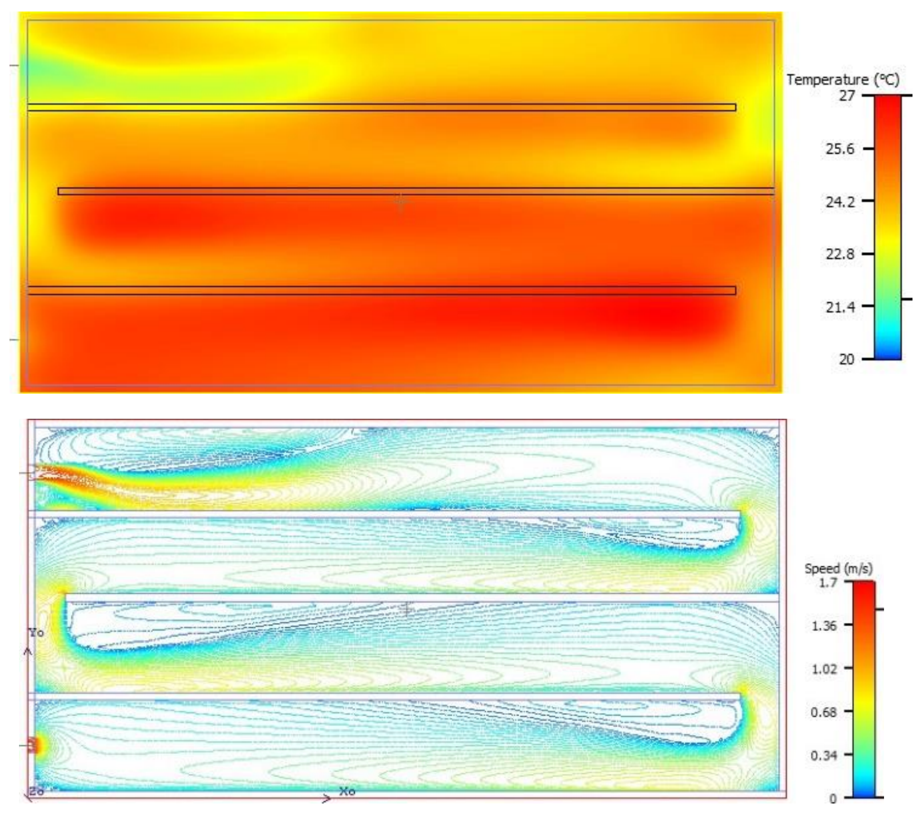

Fig. 3. Simulation results for baseline case showing surface temperature distribution on the cold plate (top) and speed distribution (bottom) using three flow baffles (turns)

The first set of runs was carried out by varying the design parameters relative to the baseline configuration. Fins were not included in these runs which helped to keep the mesh size small while still providing valuable inputs on the effect of various design parameters. Figures 4 to 6 show the effect of varying the various design parameters on the effectiveness (ε), temperature uniformity (ΔT_{max}), and cooling power (P). As shown, design parameters exert more influence on the performance than others.

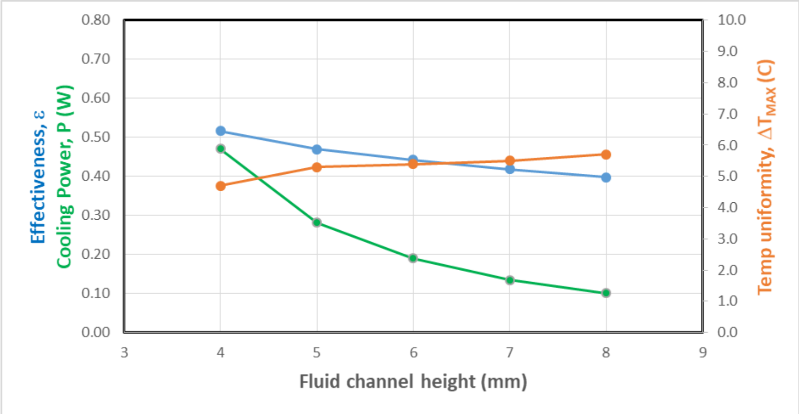

Fig. 4. Simulation results showing the values of ε, ΔT_{max} and P versus fluid channel height.

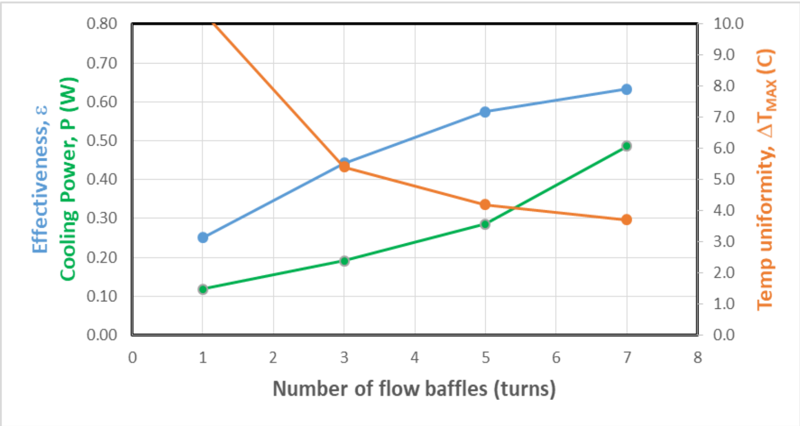

Fig. 5. Simulation results showing the values of ε, ΔT_{max} and P versus number of flow baffles (turns).

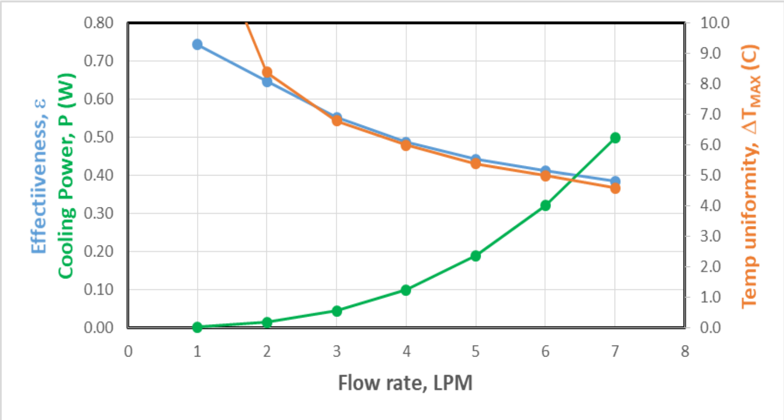

Fig. 6. Simulation results showing the values of ε, ΔT_{max} and P versus coolant flow rate.

Increasing the number of flow baffles (or turns) and coolant flow rate causes a significant reduction in ΔT_{max} due to the increase in heat transfer coefficient (higher fluid velocities). Increasing the number of flow baffles, in particular, not only improves the temperature uniformity but also improves ε which implies that the available coolant flow rate is being utilized more efficiently for cooling. Figure 7 shows the fluid velocity and surface temperature distribution for the case of 7 flow baffles.

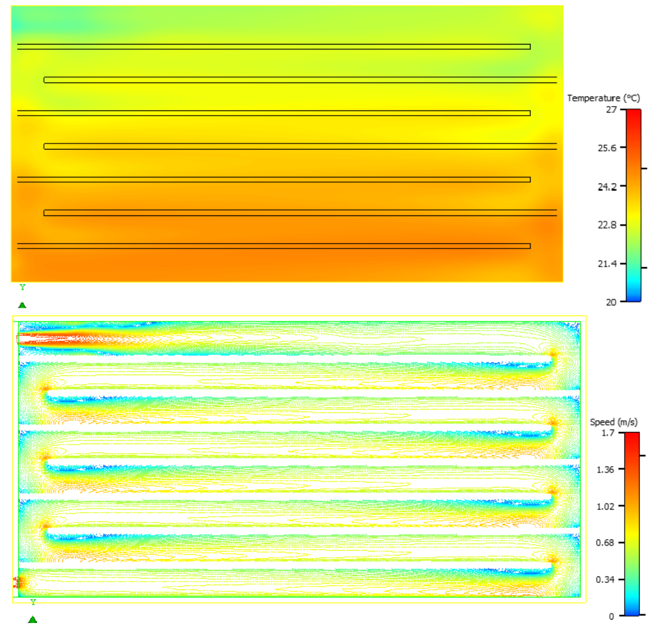

Fig. 7. Simulation results showing surface temperature distribution on the cold plate (top) and speed distribution (bottom) using seven flow baffles

The flow distribution and surface temperature distribution are more uniform relative to the baseline case (3 flow baffles) with a significant reduction in the flow recirculation regions. Increasing the fluid channel height lowers the fluid speed leading to a significant reduction in P (lower pressure drop). There is a relatively smaller reduction in ε due to the increase in thermal resistance. ΔT_{max} increases slightly due to the reduction in heat transfer coefficient.

Finally, fins were added to the cold plate and a simulation was run with a fin pitch of 5mm added to the baseline configuration. Figure 8 shows the temperature distribution on the cold plate and speed distribution with 5mm fin pitch. Flow distribution is nonuniform in the inlet channel but becomes more uniform moving towards the exit channel. The surface temperature distribution also shows a similar pattern.

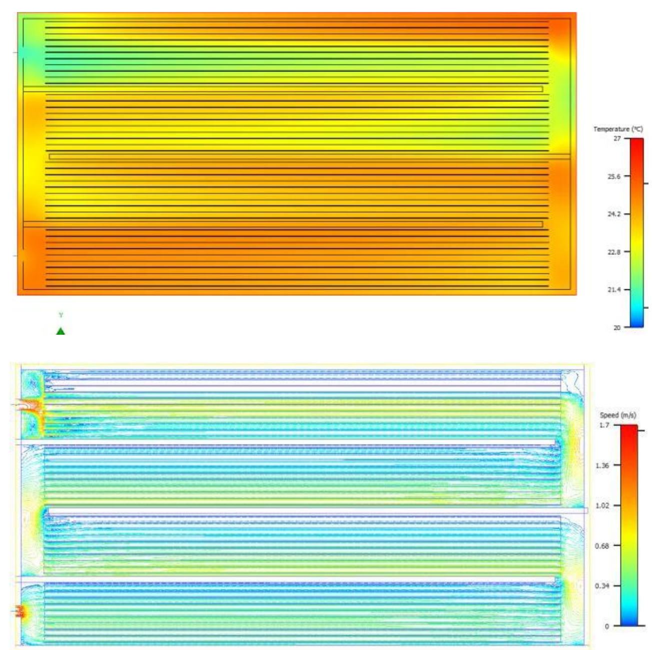

Fig. 8. Simulation results showing the surface temperature distribution (top) and speed distribution (bottom) with a fin pitch of 5mm

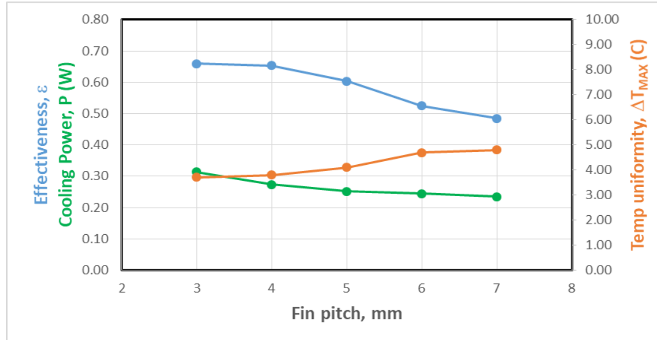

Fig. 9. Simulation results showing the values of ε, ΔT_{max} and P versus fin pitch.

The maximum surface temperature is lower compared to the baseline case (25.3C vs 27C). The effectiveness improved significantly (ε of 0.60 vs 0.44) due to the increase in finned surface area. The required cooling power increased marginally (P of 0.25W vs 0.19W) due to the increase in frictional losses in the finned regions as well as the increase in flow velocities. As expected, the temperature uniformity improved relative to the baseline case (ΔT_{max} of 4.1C vs 5.4C) due to the increase in heat transfer coefficient. Simulations were run by varying the fin pitch from 3 to 7mm (figure 9). Although adding more fins (reducing fin pitch) helps to improve ε (i.e. lower thermal resistance), it has a relatively smaller impact on ΔT_{max} and P. Figure 10 shows the surface temperature and speed distribution with a fin pitch of 3mm. The surface temperature uniformity and flow distribution are marginally better relative to the 5mm pitch.

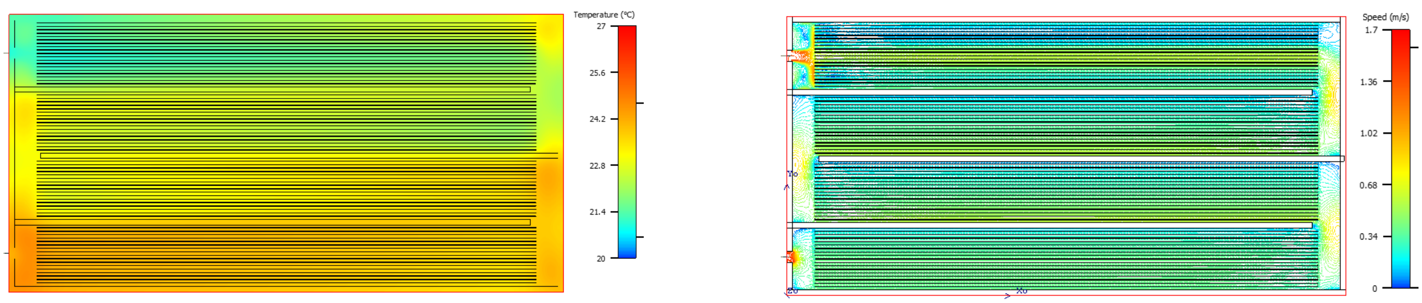

Fig. 10. Simulation results showing surface temperature distribution (left) and speed distribution (right) using a fin pitch of 3mm

CONCLUSIONS

Here are the main conclusions that can be drawn from this study.

1. The performance parameters ε, ΔT_{max} and P provide a good indicator of the overall thermal performance and cooling efficiency of the cold plates.
2. Surface temperature uniformity of the cold plate is closely tied to the flow distribution and minimizing the flow recirculation regions.
3. Increasing the number of flow baffles (or turns) is an effective way to improve both ε and ΔT_{max}. However, this comes at a cost of higher cooling power P (higher pressure drop).

4. Although increasing the coolant flow rate reduces ΔT_{max}, it also lowers ε and increases P. In other words, it is not an energy-efficient way to improve the temperature uniformity of the cold plate.
5. Increasing the fluid channel height reduces the fluid speed thereby reducing P significantly while not affecting ε and ΔT_{max} as much.
6. Adding fins is an effective way to improve both ε and ΔT_{max} without increasing P significantly.
7. Increasing the fin density (reducing fin pitch) improves ε (lower thermal resistance) but does not have a significant impact on ΔT_{max} or P.

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