- 1) Performed 3Delectrochemical-thermal modeling of four battery cooling methods
- 2) Thermal performance of direct air cooling, direct liquid cooling, indirect (jacket) liquid and fin cooling are compared
- 3) Merits and limitations of each cooling method for occupying a fixed volume are summarized
- 4) Temperature rise for a fixed load is lower with direct or indirect liquid cooling lower than air and fin cooling.

Comparison of Different Cooling Methods for Lithium Ion Battery Cells

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Abstract

Choosing a proper cooling method for a lithium-ion (Li-ion) battery pack for electric drive vehicles (EDVs) and making an optimal cooling control strategy to keep the temperature at a optimal range of 15°C to 35°C is essential to increasing safety, extending the pack service life, and reducing costs. When choosing a cooling method and developing strategies, trade-offs need to be made among many facets such as costs, complexity, weight, cooling effects, temperature uniformity, and parasitic power. This paper considers four cell-cooling methods: air cooling, direct liquid cooling, indirect liquid cooling, and fin cooling. To evaluate their effectiveness, these methods are assessed using a typical large capacity Li-ion pouch cell designed for EDVs from the perspective of coolant parasitic power consumption, maximum temperature rise, temperature difference in a cell, and additional weight used for the cooling system. We use a state-of-the-art Li-ion battery electro-chemical thermal model. The results show that under our assumption an air-cooling system needs 2 to 3 more energy than other methods to keep the same average temperature; an indirect liquid cooling system has the lowest maximum temperature rise; and a fin cooling system adds about 40% extra weight of cell, which weighs most, when the four kinds cooling methods have the same volume. Indirect liquid cooling is more practical form than direct liquid cooling though it has slightly lower cooling performance.

Keywords: Li-ion battery, cooling method, cooling model, battery thermal management

1. Introduction

Energy-saving and environmentally friendly electric drive vehicle (EDV) adoption in the market is increasing and has more potential if batteries have more energy, travel longer, and are less expensive. The battery thermal management system to keep the temperature at a optimal range of 15°C to 35°C [1, 2] is essential for lithium-ion (Li-ion) battery packs in electrical vehicles (EVs) and hybrid electrical vehicles (HEVs) to extend lifetime and ensure operating safety. During vehicle operation, considerable heat is generated in the battery pack that needs to be rejected. How to remove the generated heat, and keep the temperature uniform has become a challenge because of the high requirement of gravimetric and volume energy in EDVs. Several cooling methods have been proposed and researched. Zolot et al. [3, 4, 5] evaluated the thermal

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behaviors of battery packs in the Honda Insight and Toyota Prius using air cooling. An air-cooling system worked very well in HEVs during standard drive cycles that could control the maximum temperature below the limit of 55°C and the temperature difference was no more than 5°C, but the maximum temperature was higher than the desired limit on an aggressive cycle. Choi and Kang [6] developed a thermal model to structure the flow system and determine the appropriate cooling capacity for an air-cooled HEV. Wang et al. [7, 8] Yang et al. [9] and Xu et al. [10] worked on the optimization of battery arrangement and airflow duct, respectively, to achieve better performance from air cooling. Zhao et al. [11] investigated parametric influence on a cylindrical battery module using air cooling. Metal foam was added to improve the performance of air cooling by Mohammadian et al. [12] Thus the temperature uniformity was improved. Pesaran and Kim et al. [13, 14] analyzed the merits and shortcomings of liquid cooling and air cooling. Chacko et al. [15] evaluated the performance of an indirect liquid cooling battery pack and concluded that active indirect liquid cooling/heating would be one of the most promising means to achieve battery thermal management. Kim et al. [16] studied the fin cooling system and discussed the advantages of using air to remove the heat from fin ends compared to using liquid. Wu et al. [17] designed a solution using a heat pipe to mitigate the temperature rise. Wang et al. [18] used the heat pipes to cool and heat the EV battery and found that cooling and heating via heat pipes is viable for EVs. A flat heat pipe was used by Tran et al. [19] to reduce the thermal resistance of a common heat sink and found it could handle more efficiently instant increases of the heat flux than the conventional heat sink. Kim et al. [20] researched the method of using phase change materials for cooling and found that they can provide some benefit to limiting peak temperature. Porous foam or carbon fiber was used by some researchers [21, 22, 23, 24] to enhance the performance of PCM cooling. Liu et al. [25] suggested thermoelectric cooling in battery thermal management systems and found it can keep a more uniform temperature distribution.

Different cooling methods have different limitations and merits. Air cooling is the simplest approach. Forced-air cooling can mitigate temperature rise, but during aggressive driving circles and at high operating temperatures it will inevitably cause a large nonuniform distribution of temperature in the battery [26, 27]. Nevertheless, in some cases, such as parallel HEVs, air cooling is adequate. A liquid cooling system is more effective but its complexity, cost, and potential leakage make manufacturers hesitate to use it. Adding phase change material to a battery pack could add unwanted mass and volume. Phase change materials also need to exhaust the heat they absorb during continuous cycling, but they could address maximum temperature situations. Heat pipe cooling for li-ion battery pack is limited by gravity, weight and passive control. [28]

Currently, air cooling, liquid cooling, and fin cooling are the most popular methods in EDV applications. Some HEV battery packs, such as those in the Toyota Prius and Honda Insight, still use air cooling. Indirect liquid cooling has been adopted by the Chevrolet Volt, and Tesla Model S. A123 used fins for heat removal and achieved temperature uniformity. A fierce debate is ongoing about which kind of cooling method should be applied to EDV battery packs. Because each cooling method has unique advantages and disadvantages, the purpose of this paper is to carry out cell-level thermal analysis in the battery pack design process to assess which cooling method should be adopted for different applications. That can help battery designers design packs according to properties such as cell chemistry, geometry, and working conditions. Developing a cooling strategy needs to be based on cell-level analysis results. In order to compare the advantages and disadvantages of different cooling methods and provide usable flow

rate range under a specific control target, this paper analyzes the effects of air cooling, direct liquid cooling, indirect liquid cooling, and fin cooling. The results are then applied to a large-capacity Li-ion pouch cell designed for EDVs. Compactness and high energy density (i.e. volume efficiency) of the battery pack draw great attention for integrating it a vehicle with limited space. In this paper, we compared the thermal performance of the each cooling method based on the same added volume for cooling. The one that had the minimum temperature rise and minimum temperature variation was considered better if the volume were the same. Cases with different coolant flow rates or heat transfer coefficients are simulated to compare the coolant flow power consumption, maximum temperature rise, and temperature difference in a cell. Additional cooling system weight is also compared, assuming the same volume among cooling methods. Optimum flow rate range is discussed in each case when all previously mentioned impacts are taken into account.

2. Battery modeling

A 35 Ah prismatic pouch Li-ion cell with dimensions of 169 mm width, 179 mm long, and 14 mm thick is modeled for all simulations. The picture of the battery selected for this analysis is shown in Fig. 1(b). Fig. 1(c) contains discharge curves of the cell at 25°C. The cell is designed for an EDV battery pack with 128.5Wh/kg energy density. Its anode material is graphite, and its cathode material is a combination of NMC and MnO. According to the geometry and structure of the battery, a three-dimensional model is built in ANSYS/Fluent, as shown in Fig. 1. Four parts comprise the model: the active volume, positive current tabs, negative current tabs, and skin. The active volume represents the stacked structure, including positive and negative active materials, separator layers, and aluminum and copper foils. The positive tab and negative tab are used to collect the current flow through the cell. The thin skin wrapping the active volume and part of the tabs representing the pouch is built to contain the internal component heat conduction through skin surfaces in the model.

The model is solved in Fluent using the Multi-Scale Multi-Dimensional (MSMD) battery module [29]. The electrical and thermal fields are solved using equation 1 and 2, specifically [30].

$$\begin{cases} \nabla \cdot (\sigma_+ \nabla \phi_+) = -j \\ \nabla \cdot (\sigma_- \nabla \phi_-) = -j \\ j = I/Vol \end{cases} \quad (1)$$

Where σ is the effective electric conductivities for the electrode, ϕ is the phase potential for the electrode, + and - present the positive and negative respectively, j is the volumetric transfer current density computed by the current I divided by volume Vol of the sub model.

$$\begin{cases} \frac{\partial \rho C_p T}{\partial t} - \nabla \cdot (k \nabla T) = \dot{q} \\ \dot{q} = \sigma_+ \nabla^2 \phi_+ + \sigma_- \nabla^2 \phi_- + j[U - (\phi_+ - \phi_-) - T \frac{dU}{dT}] \end{cases} \quad (2)$$

Where T is temperature, $\dot{q}$ is the heat generation rate during the battery operation, k is the thermal conductive coefficient, C_p is the heat capacity, and U is the open circuit voltage of the battery.

A 1RC equivalent circuit model is used in this paper. To acquire the parameters of the

equivalent circuit model and validate model performance, two sets of experiments were conducted. The equivalent circuit model parameters are calculated from multi-pulse discharge and charge data using the simulation software Matlab [31]. A 95A discharge process is used to validate the voltage and temperature predicted by the model. Nine T-type thermocouples are pasted on the surface of the battery to monitor the temperature at each area. During the discharge process, the battery is made to be adiabatic by placing it in an adiabatic battery testing calorimeter, Phi-TEC, produced by HEL [32, 33]. An Arbin BT-2000 is used as the discharge equipment and to monitor voltage. An Agilent 34901A is used to acquire the temperature and other data. The experimental and simulated results are shown in Fig. 2. The experimental temperature rise data used here is the average value of nine thermocouples, and the simulation temperature rise is the average temperature of the whole battery. Both the voltage drop and temperature rise of the simulation compare well to the experimental data, indicating that the model can simulate the real battery.

3. Cooling methods configuration and simulations

Fig. 3 shows the schematic of each cooling method. For better visualization, the cooling part is shown with increased thickness. All four methods use the two largest side surfaces of the cell to reject heat. Parallel coolant passing by each cell in a battery pack is assumed, so the nonuniformity in the pack is not considered. Fig. 3(a) shows the configuration of air cooling or direct liquid cooling. In air cooling and direct liquid cooling, the coolant flows through the gap between two cells and contacts cell side surfaces directly. Although the two methods have the same configuration at the cell level, direct liquid cooling is much more complicated than air cooling from the perspective of the pack level and flow distribution. Dielectric mineral oil is used in direct liquid cooling, which adds extra weight to the battery pack and needs a more complex circulation system, whereas air won't gain significant weight and is easy to circulate using fans in direct air cooling. Further, the liquid leakage problem is worth careful consideration in direct liquid cooling. Fig. 3(b) shows the configuration of fin cooling. Heat generated during discharge/charge is conducted through the largest side surfaces to the fin, which is sandwiched between two cells. Then the heat is dissipated from the edge of the fin, which is usually sitting on a cold plate or cooled by air directly. The cold plate can be cooled by liquid or other methods. Only one side of the fin edge is cooled in this paper. The fins, usually aluminum, add extra weight to the battery pack. The dimension of each fin end is $169\text{mm} \times 197\text{mm} \times 1\text{mm}$. Fig. 3(c) shows the indirect liquid cooling configuration. A "jacket" is used to contain the cooling liquid, conduct the heat from the cell to the coolant, and restrict the coolant in specific cooling channels. The coolant is usually water/glycol. The jacket and coolant add some weight to the battery pack, and the leakage problem should be considered during operation. Fig. 3(d) shows the direct liquid cooling configuration; the coolant is a dielectric mineral oil. The properties of each material used in each cooling system are listed in Table 1.

Generally, as shown in Fig. 4, the following comprise a battery pack cooling loop: a battery pack, a fan/pump, a heat exchanger, and coolant pipes [34]. In this paper, the volume for different cooling methods is assumed to be the same—that is, the gap between two cells used for cooling in different cooling methods is the same. Decreasing the hydraulic diameter has positive effects on battery cooling, whereas the power consumption of the cooling system will increase [14]. We use a 1mm gap between two cells for cooling, as adopted by some manufactures and researchers for prismatic pouch cells in air cooling [35] and fin cooling [16]. The gap used for cooling between batteries in all methods discussed in this paper is also set to 1mm. The extra weight of different cooling methods is calculated and shown in Table 2. The mass percentage is

the ratio of cooling system weight to cell weight.

The extra weight of air cooling is negligible compared to the battery weight (1.01 kg). Fin cooling adds maximum extra weight, approximately 39%, to the battery when all cooling methods have the same volume. Direct liquid cooling and indirect liquid cooling add approximately 2.95% and 7.16% weight to the battery, respectively, which is acceptable in EDV applications. In conclusion, considering the structure and extra weight added to a battery, air cooling is the simplest and lightest method, fin cooling adds the most extra weight, the weight added in indirect liquid cooling and direct cooling is moderate, and direct liquid cooling add less extra weight than indirect liquid cooling because the density of the aluminum “jacket” is almost three times that of mineral oil.

4. Simulation results and discussion

A series of simulations were conducted to estimate the effects of cooling by changing the flow velocity of coolant in air cooling and liquid cooling. We let the average temperature rise at the end of discharge reach a minimum of 4°C so that all cooling methods are comparable. The velocity range is 0-20m/s for air, 0-0.01m/s for mineral oil and 0-0.05m/s for water/glycol. Because the method of conducting heat from the cooling surface of the fin cooling can be variable, we assume that the thermal resistance of the cold plate is negligible and specify only the heat transfer coefficient directly on the cooling surface. The heat transfer coefficient can be calculated from the flow in the cold plate. The typical value is 5-25W/m²/k using air cooling. A value of 390W/m²/k was used for indirect liquid cooling [34]. All the simulations have the same initial and inlet temperature, the battery discharge current is set to 2.71C and the operating state of charge range is from 1 to 0.2. The average heat generation rate during the process calculated from experimental data is approximately 15.7W. The simulation results—the average temperature of the battery in each of the different cooling methods—are shown in Fig. 5. The cases in which air flow speed is greater than 4m/s in air cooling and when the heat transfer coefficient is greater than 400W/m²/k are higher than what is currently practical in an EV cooling system, so the temperature rise curves are plotted in dashed lines.

Fin cooling with a reasonable heat transfer coefficient can hardly control the average temperature rise below 8°C, even when the maximum heat transfer coefficient is 500W/m²/k. The minimum average cell temperature rise at the end of discharge using air cooling is approximately 8°C when the flow speed is during the realizable range. The flow speeds or heat transfer coefficients used in those cases are in equal intervals, whereas the average temperature rise is not decreased proportionally as the flow speed or heat transfer coefficient increases. The average temperature decrease rate at the end of discharge is reduced with the increasing flow speed or heat transfer coefficient. The average temperature of the battery reflects the portion of heat generated during discharge that was not conducted by the cooling system. This means that at high flow speeds, the added cooling effect becomes less significant with the increase in coolant flow rate, whereas the power used to blow or pump the coolant increases dramatically. The ideal power consumption used for the coolant driving force can be calculated using equation 3:

$$power = \sum_{i=1}^n \Delta P_i \dot{V}_i \quad (3)$$

where ΔP_i is the pressure drop from the coolant inlet to the coolant outlet of one coolant channel, $\dot{V}_i$ is the volume flow rate in the coolant channel, and i represents the coolant channel number. The same inlet velocity is applied to the boundary conditions, so all $\dot{V}_i$, in certain cases, are equal. Thus, the power consumption can be expressed as equation 4:

$$power = \frac{1}{n} \sum_{i=1}^n \Delta P_i \sum_{i=1}^n \dot{V}_i \quad (4)$$

The $\sum_{i=1}^n \Delta P_i$ and $\sum_{i=1}^n \dot{V}_i$ are obtained from the simulation results of each cooling method, as shown in Tables 3, 4 and 5.

Fig. 6(a) shows the simulation results of discharge voltage from the cell state of charge 1 to 0.2. The average temperature when flow rate of air cooling, direct liquid cooling, and jacket cooling is around 5g/s and when the heat transfer coefficient is 50W/m²/k are shown in Fig. 6(b). The average temperature of jacket cooling is 5°C lower than air cooling and 2°C lower than direct liquid cooling when the mass flow rate of the coolant are nearly 5g/s.

The average temperature rise at the end of a 2.71C discharge compared to the power consumption of air cooling and liquid cooling are plotted in Fig. 7(a). Because various cooling methods can be used to cool the cold plate in fin cooling, only a heat transfer coefficient is specified on the cooling surface. The power consumption of fin cooling is not discussed here.

Fig. 7(a) shows the power consumption of different cooling methods when we need to control the maximum temperature at the end of a 2.71C discharge to a certain value. Much more power is needed to control the maximum temperature within a certain range when using air cooling compared to liquid cooling. Jacket cooling consumes less power than oil cooling.

Fig. 7(b) shows that the average temperature at the end of a 2.71C discharge decreases as the mass flow rate increases, whereas the decreasing rate varies considerably. Jacket cooling has the highest rate of decline, because of the relatively large heat capacity. Air cooling has the smallest decline rate because of the small heat capacity and poor thermal conductivity of air.

As previously mentioned, the average temperature of the battery reflects the portion of heat generated during discharge that is not rejected by the cooling system. Four cases in which average temperature rise at the end of discharge is closest to 8°C in each cooling method, are selected to analyze the temperature distribution of different cooling methods when all cooling methods rejected the same heat from the cell. The mass flow rate, pressure drop, and power consumption of each case using air cooling, oil cooling, and jacket cooling are shown in Table 6. Note that the general trend of jacket cooling being better than direct liquid cooling is for the assumptions of this study and should not be extrapolated as a general case.

The temperature distribution at the end of discharge of each selected case is shown in Fig. 8. Fig. 8(a) - 8(d) shows the results of air cooling, direct liquid cooling, indirect liquid cooling, and fin cooling, respectively. The heat transfer coefficient at the cold plate of fin cooling in the selected case is 300W/(m²K).

In air and direct liquid cooling, temperature increases gradually from inlet to outlet. The positive tab (top) is hotter than the negative tab due to more heat is generated in the positive tab and heat conduction is slower than the negative tab. Because the positive tab, which is made of aluminum, has relatively smaller thermal and electrical conductivity than the negative tab, which is made of copper, as shown in table 7. Temperature difference between the cell surface and the coolant in air cooling is larger than that of liquid cooling as a result the relatively small thermal conductive coefficient of air, which is indicated by the temperature at the edge of inlet side. The temperature distribution of direct liquid cooling is similar to air cooling, but it has a larger temperature effect on the battery. So the temperature gradient is larger than the air cooling in this case. The temperature difference of the jacket cooling is largest when the average temperature at the end of the discharge is controlled to 8°C. The temperature distribution of jacket cooling is more complicated than other cooling methods because of the roundabout flow routing. The

lowest temperature is around the inlet, whereas the highest temperature is located at the top of the battery on the outlet side. Although the temperature of coolant becomes higher with the flow direction, which means the coolant at outlets is the hottest. Heat generated by the cell around outlets is conducted to the inlets part of the cell because of the relatively bigger temperature gradient at that direction and low temperature around inlets. So the cell temperature around outlets is lower than the right-top. In fin cooling, heat at the opposite side of the cold plate needs to be conducted through the fins to the cooler side, therefore resulting in a relatively high temperature difference in the battery.

Fig. 9 shows the temperature difference (T_{diff}) and average temperature rise (T_{rise}) of each cooling method at the end of discharge as a function of mass flow rate. Although the average temperature rise decreases with the increase in mass flow rate, the temperature difference becomes larger with the increase in mass flow rate at the low mass flow rate. After a certain mass flow rate, the temperature difference begins to decline with the increase in flow speed. Indirect liquid cooling has the biggest maximum temperature difference, because it has the longest flow channel, and it has the biggest decrease rate after the maximum temperature difference. Air cooling has the smallest maximum temperature difference because of its poor thermal conductivity. Therefore, the operating mass flow rate range of each cooling method should be carefully considered to control both the temperature rise and temperature difference under set values and at the same time using minimum fan or pump power. For example, the control target of the temperature rise and temperature difference are both specified to be less than 4°C (below the blue dashed line in Fig. 9). For air cooling, the temperature difference will never exceed the target value, so the temperature rise takes the dominant design point. For jacket cooling, to constrain the temperature rise to 4°C and the temperature difference to 4°C, the mass flow rate is 0.65g/s and 0.70g/s, respectively. Thus, the temperature difference becomes the dominant design point, which means that the mass flow rate should be larger than 0.70g/s.

5. Conclusions

In this paper, an electrochemical-thermal battery model for a prismatic cell was built using ANSYS/Fluent, and its performance was validated. Four cooling structures were analyzed based on the model: air cooling, direct liquid cooling, indirect liquid cooling, and fin cooling. The extra weight of the cooling systems is calculated and compared. The cooling effect of each cooling method is accessed using a series of simulations that change flow speed or heat transfer coefficient from the perspective of coolant flow power consumption and average temperature rise. The simulation results are useful to battery pack designers of electrical vehicle to assess and choose a proper cooling method under the volumetric constrain. They can also get a usable flow rate of different cooling methods for a specific control target.

From this study, the following conclusions can be drawn:

1. Fin cooling adds the most extra weight when all cooling methods have the same volume.
2. Air cooling consumes the most parasitic power.
3. Both fin cooling and air cooling are constrained by minimum average temperature rise considering the realistic heat transfer coefficient range and fan power range in an electric car.
4. A maximum temperature difference exists at the low mass flow rate area when the mass flow rate increases from zero.
5. Indirect liquid cooling has the highest maximum temperature difference point because of the longest coolant channel, but the maximum temperature difference decreases more

quickly after the highest value than in other methods because of the higher heat capacity of water/glycol coolant.

6. Using the low mass flow rates of indirect liquid cooling to control the temperature rise and temperature difference within a battery should be avoided.

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Table 1: Material Properties

Item	Air[36]	Mineral Oil[36]	Water/Glycol[36] (50/50)	Aluminum[37]
Density (kg/m^3)	1.225	924.1	1069	2719
Specific heat capacity ($J/kg/k$)	1006	1900	3323	871
Thermal conductivity ($W/m/k$)	0.0242	0.13	0.3892	202.4
Kinematic viscosity (m^2/s)	1.46e-5	5.6e-5	2.58e-6	—

Table 2: Extra Weight Added By Cooling System for Each Cell

Item	Air	Mineral Oil	Water/Glycol	Fin
Extra mass (kg)	Negligible	0.0298	0.0723	0.394
Mass percentage (%)	0	2.95	7.16	39.0

Table 3: Pressure Drop and Coolant Flow Rate of Air Cooling

Flow speed (m/s)	2	4	6	8	10
$\sum_{i=1}^2 \Delta P_i$ (Pa)	144	296	454	618	788
$\sum_{i=1}^2 \dot{V}_i$ (mL/s)	390	780	1171	1561	19524
Flow speed (m/s)	12	14	16	18	20

$\sum_{i=1}^2 \Delta P_i (\text{Pa})$	966	1152	1342	1540	1745
$\sum_{i=1}^2 \dot{V}_i (\text{mL/s})$	2342	2732	3123	3513	3904

Table 4: Pressure Drop and Coolant Flow Rate of Oil Cooling

Flow speed (m/s)	0.001	0.002	0.003	0.004	0.005
$\sum_{i=1}^2 \Delta P_i (\text{Pa})$	204	406	610	814	1016
$\sum_{i=1}^2 \dot{V}_i (\text{mL/s})$	0.20	0.39	0.59	0.78	0.98
Flow speed (m/s)	0.006	0.007	0.008	0.009	0.01
$\sum_{i=1}^2 \Delta P_i (\text{Pa})$	1220	1422	1626	1830	2040
$\sum_{i=1}^2 \dot{V}_i (\text{mL/s})$	1.17	1.37	1.56	1.76	1.95

Table 5: Pressure Drop and Coolant Flow Rate of Jacket Cooling

Flow speed (m/s)	0.005	0.01	0.015	0.02	0.025
$\sum_{i=1}^6 \Delta P_i (\text{Pa})$	378	756	1135	1515	1895
$\sum_{i=1}^6 \dot{V}_i (\text{mL/s})$	0.07	0.14	0.22	0.29	0.36
Flow speed (m/s)	0.03	0.035	0.04	0.045	0.05
$\sum_{i=1}^6 \Delta P_i (\text{Pa})$	2275	2656	3038	3421	3804
$\sum_{i=1}^6 \dot{V}_i (\text{mL/s})$	0.43	0.50	0.58	0.65	0.72

Table 6: Design Parameters of Different Cooling Methods with Same Temperature Rise

Item	Air	Mineral Oil	Water/Glycol(50/50)
Average temperature rise (°C)	8.4	8.2	8.3
Flow velocity V (m/s)	4	0.00275	0.015
Pressure drop $\frac{1}{n} \sum_{i=1}^n \Delta P_i$ (Pa)	148	279	189
Mass flow rate (g/s)	0.96	0.49	0.29
Ideal power consumption (mW)	116	0.15	0.051

Table 7: thermal and electrical conductivity of positive and negative tab [38]

Tap	Material	Thermal Conductivity	Electrical Conductivity
		$(W \cdot m^{-1} \cdot K^{-1})$	$(S \cdot m^{-1})$
Positive	Aluminium	249	3.50×10^7
Negative	Copper	353	5.96×10^7

Fig. 1: Battery characters and model

Fig. 2: Voltage and temperature comparison of the simulation and experimental results

Fig. 3: Cooling configuration with fix and the same gap between cells

Fig. 4: General schematic of battery pack cooling loop

Fig. 5: Average battery temperature at different flow speeds or h during 2.71C discharge

Fig. 6: Voltage and Temperature simulation results

Fig. 7: Simulation results of different cooling methods

Fig. 8: Temperature distribution of different cooling methods

Fig. 9: Temperature rise and difference at different mass flow rates

fig 1a

fig 1b

fig 1c

fig 2a

fig 2b

fig 3a

fig 3b

fig 3c

fig 3d

fig 5a

fig 5b

fig 5c

fig 5d

fig 6a

fig 6b

fig 7a

fig 7b

Temperature
Contour 1

ANSYS
R15.0

fig 8a

Temperature
Contour 1

ANSYS
R15.0

fig 8b

Temperature
Contour 1

ANSYS
R15.0

[C]

fig 8c

Temperature
Contour 1

ANSYS
R15.0

[C]

Code
Plate

fig 8d

fig 9