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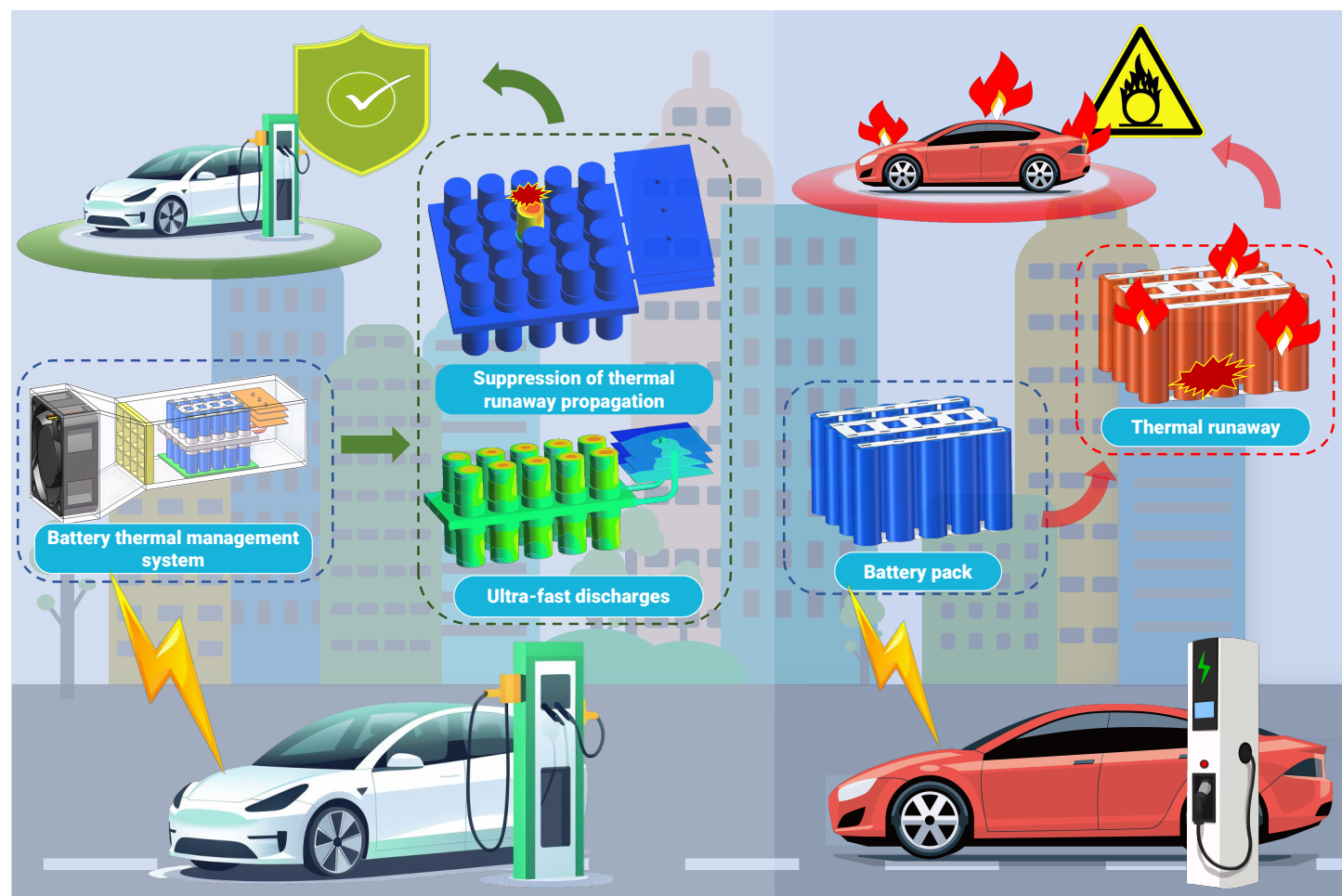
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GRAPHICAL ABSTRACT



PUBLIC SUMMARY

- Light-weight battery thermal management system with L-shaped heat-pipe thermal spreader (LHP-TSP) proposed.
- Numerical model with measured LHP thermal conductivity established and validated.
- Thermal performance with varying velocities and fins investigated.
- For 5C discharge battery temperature well maintained with high grouping efficiency.
- Propagation of thermal runaway successfully suppressed with LHP-TSP.

Role of heat-pipe thermal spreader on the thermal management of light-weight battery module and suppression of thermal runaway

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A novel air-cooling battery thermal management system integrated with L-shaped heat-pipes and thermal spreader is proposed for both ultra-high C-rate and thermal runaway suppression applications. Cylindrical batteries in 5×4 array are connected structurally with a thermal spreader embedded with finned L-shaped heat-pipes aligned to the air flow direction. The heat-pipes, which enable rapid temperature equalization of the upstream and downstream batteries, are characterized by the specially developed test jig. Both numerical simulation and experimental test are conducted to examine the thermal performances of battery module under varying discharge rates from 1C to 5C and air velocities from 1m/s to 5m/s. An experimental test for the battery module incorporating L-shaped heat-pipes was also conducted to verify the numerical simulation, with excellent agreement achieved. Under ultra-high discharge rates of 5C, the maximum temperature and the maximum temperature difference of the battery module at 5m/s vs 2m/s decrease by 15.92% and 28.79%, respectively, meeting the thermal requirements. Increasing the number of fins can further improve the thermal performance of the battery module. Besides, the battery module also demonstrates effective suppression of thermal runaway propagation. In the event of thermal runaway occurring in a single battery, the surrounding batteries reach 78.69°C maximum, well below the thermal runaway temperature, and thus successfully suppress the thermal propagation. Clearly the present study presents new ways to push the air-cooling limit for high performance light-weight battery packs.

INTRODUCTION

To reduce the carbon emission and mitigate energy shortages, electric vehicles have been developed in an unprecedented pace all over the world in the last decade. The development trend keeps increasing and the sales could reach around 17 million in 2024 according to International Energy Agency.¹ Lithium-ion batteries (LIBs) have become the primary energy source for electric vehicles due to their high energy density, long lifespan, and low self-discharge rate. A highest energy density of 360Wh/kg for commercialization has been reported.² In general, the heat generation increases with the energy density of the battery, which could cause the battery temperature to exceed the threshold temperature during extreme conditions such as fast discharge and high ambient temperature, leading to accelerated capacity degradation, shortened lifespan, and even thermal runaway accidents.³ A well-designed battery thermal management system (BTMS) is highly solicited to maintain the batteries within the appropriate temperature range for enhanced safety and prolonged lifespan,⁴⁻⁶ typically between 25°C and 40°C for the operation temperature with a temperature difference of less than 5°C.⁶

Various cooling media have been utilized in BTMS, including air,⁷⁻⁸ aqueous liquids,⁹ dielectric liquids,¹⁰⁻¹¹ and phase change materials.¹²⁻¹³ Air cooling is a method of dissipating heat from batteries by circulating air in between, which can be categorized into natural convection and forced convection. Due to the limited heat dissipation capability of natural convection, forced convection is typically implemented to cool the batteries.¹⁴ In general, air cooling is mostly favored due to its simple structure, small size, low cost, and ease of maintenance. Nonetheless, air cooling has been suffering from disadvantages such as low convective heat transfer coefficients and prominent heating effect along the flow path due to its low heat capacitance, which is yet to be enhanced in competing with the other cooling techniques such as liquid cooling technique.

To push the performance limit of air cooling, its combinations with other cooling techniques such as heat-pipes can be used for thermal enhancement. Among the cooling combinations, heat-pipes can transfer heat from the battery to the ambient over long distances, thereby guaranteeing the performance and the flexibility of the cooling system layout.¹⁵ In comparison to natural convection, the maximum temperature and temperature difference of the cylindrical battery module can be decreased for the discharge rates ranging from 1C to 3C.¹⁶⁻¹⁷ Under a discharge rate of 3C and reciprocating spray cooling, the maximum temperature of the battery module measured 44°C, indicating a reduction of 32.1% in comparison with natural convection.¹⁸ For prismatic battery module, the experimental studies show that the discharge rate up to 2C can be marginally managed with flat heat-pipes in combination with air cooling or liquid cooling techniques.¹⁹⁻²⁰ Besides the single-loop heat-pipes, flat micro heat-pipe arrays and oscillating heat-pipes with multiple loops were also investigated for cooling prismatic and pouch battery packs.²¹⁻²² Flat micro heat-pipe arrays are composed of numerous individual and completely isolated micro heat-pipes, and each inner surface of the micro heat-pipes is equipped with microstructures such as micro-grooves to enhance the heat transfer.²³ Mostly the prismatic batteries were used with the discharge rates ranging from 1C to 2C.²⁴⁻²⁶

It is observed that the present studies on the heat-pipe based BTMS are mainly focused on the prismatic battery modules under moderate discharge rates. The studies on the cylindrical batteries are limited, partially due to the integration difficulty in cylindrical batteries with uneven surfaces. As a result, the maximum temperature of the cylindrical battery modules is hardly controlled below 40°C even at moderate discharge rates. The air-cooling BTMS systems for cylindrical batteries employing heat-pipes suffer from drawbacks such as structure complexity, and a large number of heat-pipes with excessive weight and low grouping efficiency. Moreover, the thermal conductivity of the heat-pipes is a critical parameter for BTMS heat dissipation, yet the experimental characterization of heat-pipe thermal conductivity in the field of battery thermal management is not well reported. A significant portion of the BTMSs with heat-pipes did not report the thermal conductivity data in the application scenarios,²⁷⁻²⁸ or just cited data from the existing literature.²⁹

In view of this, this paper focuses on the light-weight integration of the cylindrical battery module with L-shaped heat-pipe thermal spreader (LHP-TSP) for enhanced air cooling up to 5C discharge. A test jig is specially developed to measure the equivalent thermal conductivity of the heat-pipes. By incorporating the measured value, the numerical simulation model for the battery module is established and the thermal characteristics are examined numerically under different discharge rates and varying air velocities. The experimental test was also conducted to verify the numerical modeling. Based on the present configuration, the effects of the air velocity and fin number on the thermal performance of the battery module are investigated for the ultra-high discharge rates from 4C to 5C, which is usually not achievable based on the existing air-cooling BTMS. In addition, the thermal suppression effect of the present configuration is examined when a single battery undergoes thermal runaway. It is demonstrated that the present LHP-TSP design would remove the heat effectively to avoid the thermal propagation to the surrounding batteries. This study not only achieves light-weight air cooling of cylindrical batteries at ultra-high discharge rates, but also pioneers the use of heat-pipe thermal spreader for the suppression of thermal propagation in the air-cooling battery module, which would provide new insights in

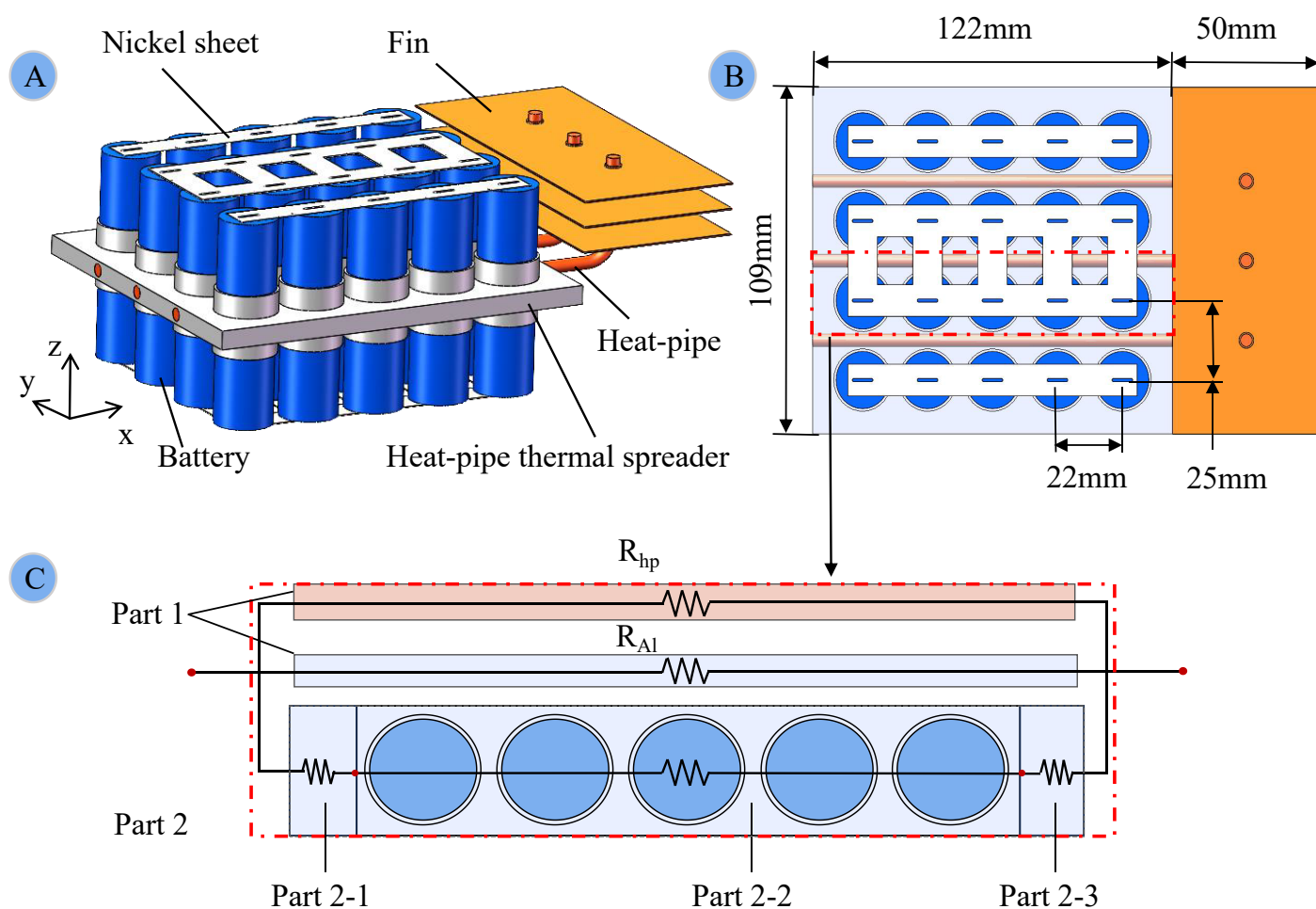


Figure 1. Schematic diagram of a battery module using embedded L-shaped heat-pipes coupled with air cooling (A) 5×4 aligned arrangement. (B) Top view. (C) Thermal resistance network.

the thermal management of battery systems in electric vehicles.

MATERIALS AND METHODS

Description of battery module LHP-TSP

Geometrical model. As depicted in Figure 1, the proposed battery module consists of 20 LG18650 batteries with 5 batteries in parallel in the x direction and 4 batteries in series in the y direction (5P4S), a TSP embedded with three LHPs, and nickel sheets as busbar connections. The heat-pipes with the diameter of 4mm were installed in the holes of the thermal spreader through brazing to ensure the air-tight contact with the TSP, and three copper fins were installed at equal intervals along the z direction on the downstream condensation sections of the heat-pipes to enhance the heat dissipation. Each fin has a length of 50mm, a width of 109mm, and a thickness of 0.5mm, with a gap of 7.5mm between adjacent fins. The battery terminals are connected by nickel sheets, with the spacing of 22mm in x direction and 25mm in y direction. The length, width, and thickness of the embedded TSP are 122mm, 109mm, and 6mm, and the height and thickness of the sleeve are 7mm and 0.8mm, respectively.

Heat-pipe characterization. The heat-pipe used in this study is an L-shaped copper-water heat-pipe which is a sealed copper container including the shell, wick, and vapor core. The schematic of the heat-pipe is shown in Figure 2A and the structural parameters are given in Table S1. When the evaporator section is heated, the liquid absorbs a large amount of heat and rapidly evaporates into vapor, which is then driven to the condenser section by the vapor pressure difference. In the condenser section, the vapor is cooled, condenses into the liquid, and then returns to the evaporator section

via capillary action, completing one cycle of operation.^{30–31}

Since the heat-pipe performance varies with both its design and the manufacturing process, here a test jig was developed to determine its thermal conductivity experimentally. As is shown in Figure 2A, the experimental setup consisted of a bracket, fan, duct, DC power supply, and thermocouples to the temperature acquisition system. A nickel-manganese alloy heating wire with a diameter of 0.2mm and a resistivity of $43.3\Omega\cdot\text{m}^{-1}$ was wound around the evaporator section of the heat-pipe for uniform heating. DC power (GPD-2303S) was connected to both ends of the heating wire, and the heating power of the wire was controlled by adjusting the voltage. Forced convective air cooling was employed in the condenser section of the heat-pipe, which had the welded fins to increase the heat dissipation area and thus enhanced cooling. Thermocouples were welded to the evaporator, adiabatic, and condenser sections of the heat-pipe, respectively, and a temperature data acquisition instrument (HIOKI-LR8410R) was used to record the temperatures of each part of the heat-pipe. It was worth noting that the thermocouple in the evaporator section should not contact the heating wire to avoid direct heating and thus large temperature errors. A layer of aerogel insulation material of 10mm thick was wrapped around the evaporator and adiabatic sections of the heat-pipe to minimize heat loss.

The effective thermal conductivity of a heat-pipe can be obtained from the temperature difference between the evaporator and condenser sections based on Fourier's law without detailed information on the vapor-liquid two-phase flow inside the heat-pipe. Namely,

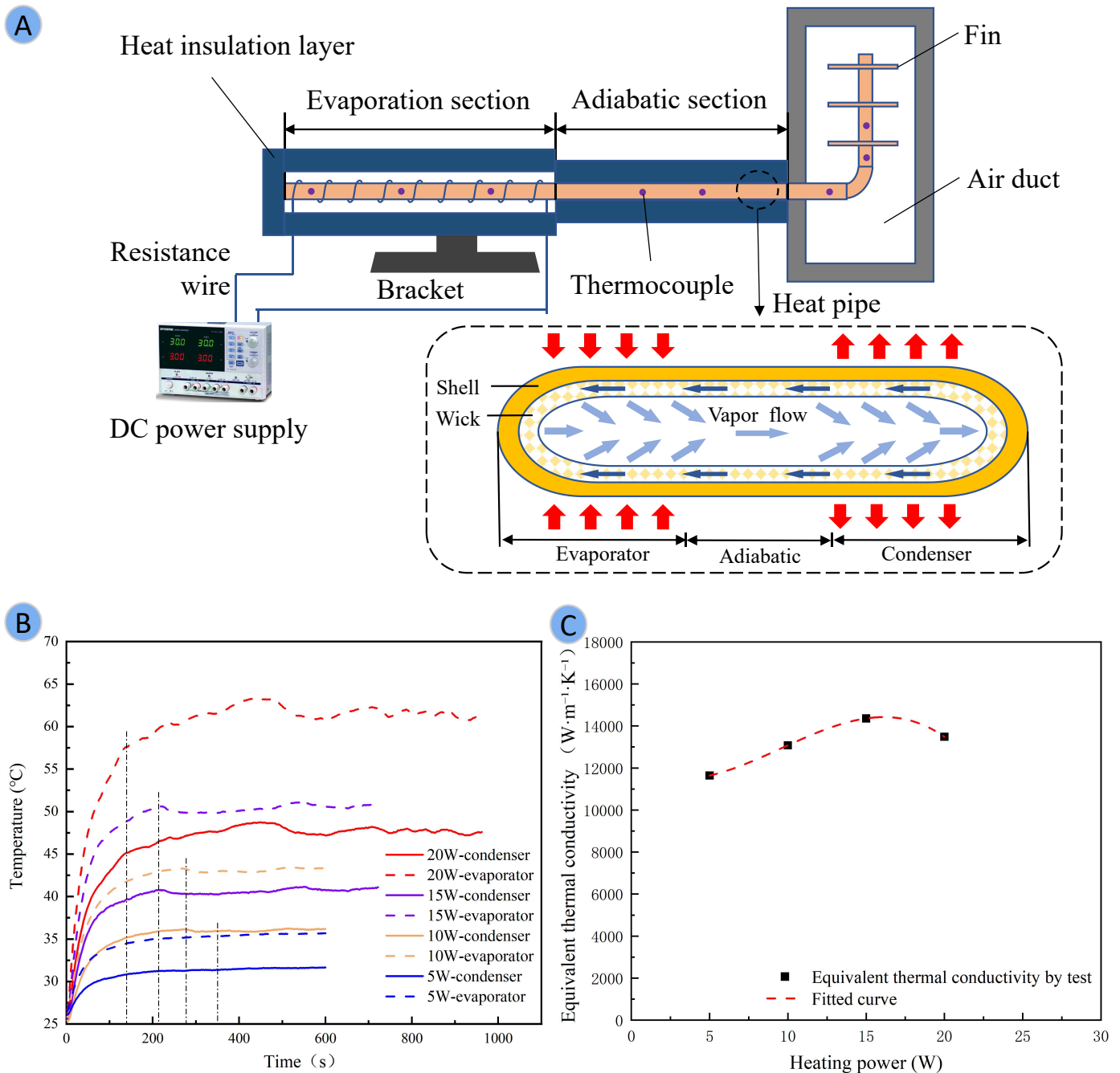


Figure 2. Schematic diagram of the heat-pipe (A) Operating principle and test jig for the equivalent thermal conductivity. (B) Temperature variation of the evaporator and condenser sections at different heating powers. (C) Variation of the equivalent thermal conductivity coefficient with heating power.

$$k_{hp} = \frac{l}{AR_{hp}} = \frac{lQ}{A(T_{we} - T_{wc})} \quad (1)$$

$$l = L_{wa} + \frac{L_{we} + L_{wc}}{2} \quad (2)$$

where k_{hp} , l , A , R_{hp} are the equivalent thermal conductivity, effective length, cross-sectional area, and thermal resistance of the heat-pipe. Q is the heat transfer power, T_{we} and T_{wc} are the temperature of the evaporator section and condenser section, respectively. L_{wa} , L_{we} , L_{wc} are the lengths of the adiabatic section, evaporator section and condenser section of the heat-pipe, respectively.¹⁶

The thermal test results for the heat-pipes are shown in Figure 2B. During the startup phase of the heat-pipes, the temperatures of both the evaporator and condenser sections rise rapidly, indicating the efficient thermal response

capability of the heat-pipes. As the heating time goes on, the temperatures of the evaporator and condenser sections stabilize, indicating that the steady state working condition is reached, at which the equivalent thermal conductivity of the heat-pipe can be obtained. When the heating power increases to 20W, noticeable oscillations and instability in the temperatures occur in both the evaporator and condenser sections mainly due to the boiling instability, after which the heat-pipe thermal conductivity is slightly lowered.

The effective thermal conductivity of the heat-pipes is calculated using Equation 1 and the results are shown in Figure 2C. With the increasing heating power, the effective thermal conductivity of the heat-pipes first increases and then decreases. In the subsequent numerical simulation, the average thermal conductivity $1.314 \times 10^4 W \cdot m^{-1} \cdot K^{-1}$ of the heat-pipe will be used to investigate the thermal performance at different C-rates. A theoretical calculation of the heat-pipe thermal conductivity at the saturation vapor pressure

at 40°C is also conducted (Appendix A), which is found to be $1.206 \times 10^4 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ fairly close to the experimental result with a deviation of -8.2%.³²⁻³⁴

According to the relative uncertainty analysis,³⁵ the uncertainty of the heat-pipe thermal conductivity can be determined using the root mean square technique for the relative uncertainties of the independent variables. The uncertainty for the thermal conductivity of the heat-pipe can be estimated using the following formula.

$$\frac{\delta k_{hp}}{k_{hp}} = \sqrt{\left(\frac{\delta l}{l}\right)^2 + \left(\frac{\delta A}{A}\right)^2 + \left(\frac{\delta Q}{Q}\right)^2 + \left(\frac{\delta(T_{we} - T_{wc})}{T_{we} - T_{wc}}\right)^2} \quad (3)$$

The effective length had a relative uncertainty of approximately 0.01% and the cross-sectional area of approximately 0.44%. The relative uncertainty of the heat transfer power supply at the heating powers from 5W to 20W varied from 2% to 0.5%. The relative uncertainty in the temperature difference between the condenser and evaporator sections at the heating powers from 5W to 20W was estimated to range from 4.95% to 1.6%. Based on the above analysis, at an ambient temperature of 25°C, the relative uncertainty in the thermal conductivity of the heat-pipe at heating powers of 5W to 20W varied from 5.4% to 1.6%.

Thermal enhancement effect with the heat-pipe thermal spreader. As shown in the partial schematic in Figure 1C, the effective thermal conductivity of the TSP and LHP-TSP along the flow direction can be analyzed through a thermal resistance network composed of Part 1 and Part 2 in parallel.

$$k_{TSP/LHP-TSP} = \beta_{Part2} \cdot k_{Part2} + (1 - \beta_{Part2}) \cdot k_{Part1} \quad (4)$$

where k_{Part1} , k_{Part2} , β are the thermal conductivities of Part 1, Part 2, and the volume fraction of Part 2, respectively.

$$k_{Part1} = \beta_{hp} \cdot k_{hp} + (1 - \beta_{hp}) \cdot k_{al} \quad (5)$$

where k_{al} is the thermal conductivity of aluminium.

$$k_{Part2} = \frac{l_{TSP}}{\frac{l_{Part2-1}}{k_{al}} + \frac{l_{Part2-2}}{k_{Part2-2}} + \frac{l_{Part2-3}}{k_{al}}} \quad (6)$$

Here $l_{Part2-1}$, $l_{Part2-2}$, $l_{Part2-3}$, $k_{Part2-2}$ represent the length of Part 2-1, the length and thermal conductivity of Part 2-2, and the length and thermal conductivity of Part 2-3, respectively. The thermal conductivity for Part 2-2 can be expressed by the Maxwell equation.³⁶

$$k_{Part2-2} = \left[\frac{k_x + 2k_{al} - 2\beta_x(k_{al} - k_x)}{k_x + 2k_{al} + \beta_x(k_{al} - k_x)} \right] \cdot k_{al} \quad (7)$$

with k_x represents the thermal conductivity in the x-direction of the battery.

The equivalent thermal conductivities of the TSP and LHP-TSP are obtained to be $89 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ and $1,177 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$, respectively. Compared to the TSP only case, the thermal conductivity of the LHP-TSP thermal spreader is increased by 12.2 times, demonstrating the superior heat dissipation performance of the present design configuration.

Numerical model and experimental test

Model description. Due to the symmetry of the battery module integrated with LHP-TSP, only half of the model was geometrically created using Solidworks to reduce simulation time, and it was imported into ANSYS Fluent 19 as a STEP format file, where the governing equations were solved using a double precision solver. In Fluent Meshing, boundary conditions such as velocity inlet, pressure outlet, adiabatic surfaces, and symmetry planes were set. The boundary layer was set using a smooth-transition method, consisting of 3 layers with a growth rate of 1.2 and a y^+ value ranging from 30 to 40. A computational mesh was generated using poly-hexcore finite elements, with a skewness of 0.54, as shown in Figure 3A. The battery model is established based on a 2.5Ah LG18650 cylindrical cell, considering the anisotropic thermal conductivity of the cylindrical cell. The physical properties of the materials are detailed in Table S2.

It is worth noting that the instantaneous heat generation rate at different

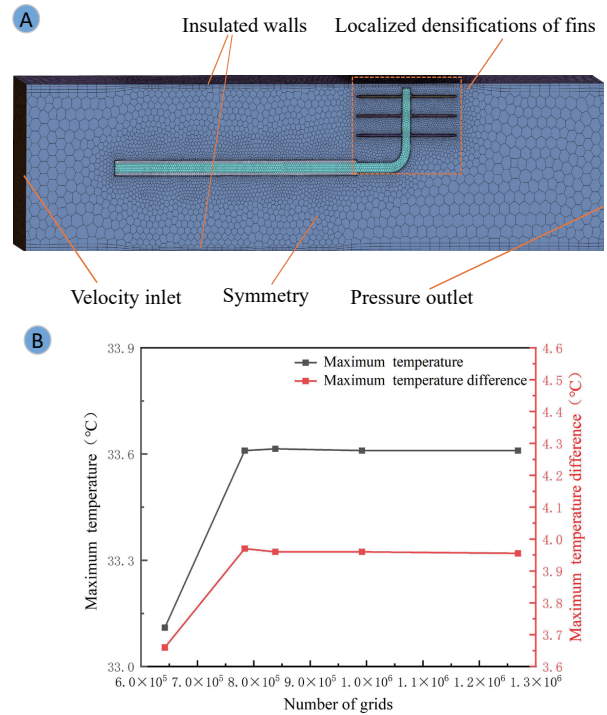


Figure 3. Grid Independence Verification (A) Computational domain for numerical modeling. (B) Results of grid independence analysis.

discharge rates is determined by the thermal protection method.³⁸ The obtained heat generation rate, as shown in Appendix B, is highly dependent on the discharge time or state of charge (SOC), which is input to the model as a User-Defined Function for computation in Fluent 19.0. It is found that, at a discharge rate of 3C and an air velocity of 2m/s, the maximum temperature and maximum temperature difference of the battery module decrease by 0.37°C and 0.11°C, respectively, which are 1.1% and 2.9% deviations from the case without busbar. Since the effect of the busbar on the thermal performance of the battery module is minimal, the busbar is neglected in the following numerical modeling to simplify the computation efforts. The other assumptions for the numerical model are as follows.

- (1) The contact thermal resistance between the battery and other components, as well as the effects of radiative heat transfer, are ignored.
- (2) The battery cell is considered as a cylinder composed of uniform but anisotropic battery material.
- (3) The cooling air is treated as an incompressible Newtonian fluid.
- (4) All the thermal properties of the materials are assumed to be constant and do not change over time.
- (5) The heat-pipe is modeled as a thermal conductor with measured thermal conductivity by neglecting the two-phase heat transfer within the heat-pipes.

In the air-cooling battery thermal management system proposed in this work, the Reynolds number is 4,094 at the inlet air velocity of 1m/s using Equation 8.³⁹ Thus, the air flow in the battery thermal management system is considered turbulent, and the standard wall function is used near the walls.

$$Re = \frac{\rho_a U_{max} D_b}{\mu_a} \quad (8)$$

$$U_{max} = \frac{v_0 \cdot S_b}{S_b - D_b} \quad (9)$$

where, Re , ρ_a , U_{max} , μ_a , D_b , S_b , v_0 are the Reynolds number, the density of air, the maximum velocity of air flowing between adjacent tube clusters, the dynamic viscosity of air, the diameter of the battery, the center-to-center spacing between adjacent batteries, and the inlet air velocity, respectively.

The K-ε turbulence model is employed to compute the turbulent flow field.

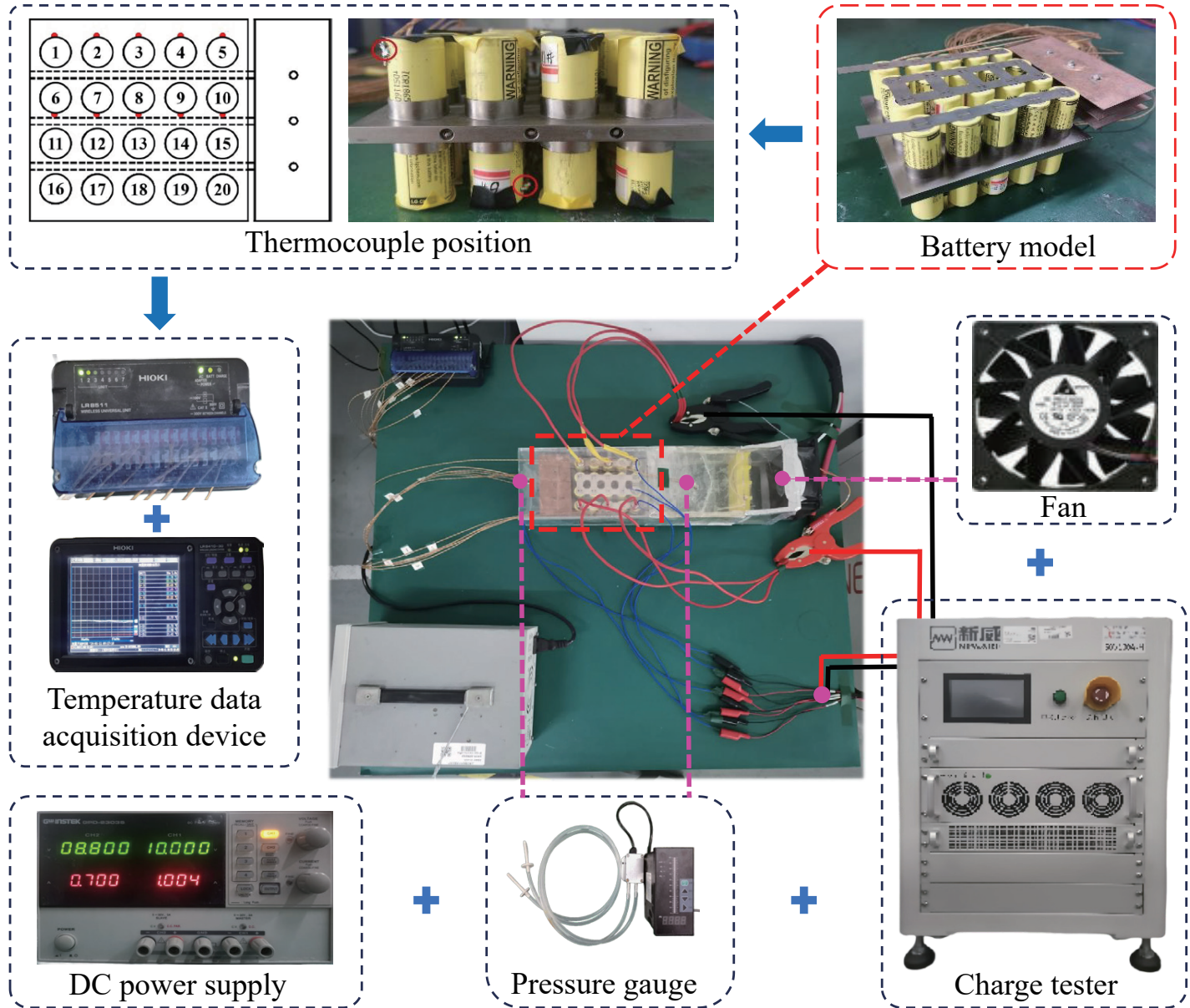


Figure 4. Test setup diagram of the air-cooling battery module in combination with LHP-TSP

By solving the following equations, the flow field in the battery thermal management system can be obtained. ³⁹⁻⁴¹

Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (10)$$

Momentum conservation equation:

$$\rho_a(u_i \frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial t}) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_t) \frac{\partial u_i}{\partial x_j} \right] \quad (11)$$

Turbulent kinetic energy equation:

$$\rho_a(u_i \frac{\partial K}{\partial x_j} + \frac{\partial K}{\partial t}) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial K}{\partial x_j} \right] + \frac{\mu_t}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 - \rho_a \epsilon \quad (12)$$

Turbulent kinetic energy dissipation equation:

$$\rho_a(u_i \frac{\partial \epsilon}{\partial x_j} + \frac{\partial \epsilon}{\partial t}) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} \right] + C_1 \frac{\mu_t}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 \frac{\epsilon}{K} - C_2 \rho_a \frac{\epsilon^2}{K} \quad (13)$$

$$\mu_t = \rho_a C_\mu \frac{K^2}{\epsilon} \quad (14)$$

Energy equation for air cooling:

$$c_a \rho_a \left(\frac{\partial T_a}{\partial t} + u \frac{\partial T_a}{\partial x} + v \frac{\partial T_a}{\partial y} + w \frac{\partial T_a}{\partial z} \right) = k_a \left(\frac{\partial^2 T_a}{\partial x^2} + \frac{\partial^2 T_a}{\partial y^2} + \frac{\partial^2 T_a}{\partial z^2} \right) \quad (15)$$

Energy equation for the battery:

$$\rho_b c_b \frac{\partial T_b}{\partial t} = k_x \frac{\partial^2 T_b}{\partial x^2} + k_y \frac{\partial^2 T_b}{\partial y^2} + k_z \frac{\partial^2 T_b}{\partial z^2} + q_b \quad (16)$$

where u , v and w are the velocity components in the x , y , and z directions, p is pressure. K , ϵ represent turbulent kinetic energy and turbulent kinetic energy dissipation rate, respectively. μ , μ_t represent the molecular dynamic viscosity coefficient and the turbulent viscosity, respectively. C_1 , C_2 , C_μ , σ_k and σ_ϵ are parameters of the K- ϵ turbulence model, their values are 1.44, 1.92, 0.09, 1.3, and 1.3, respectively. q_b is the volumetric heat generation rate in the battery. Additionally, ρ_a , ρ_b , c_a , c_b , k_a , k_x , k_y , k_z , T_a , and T_b represent the density, specific heat, thermal conductivity, and temperature of the air and the battery, respectively.

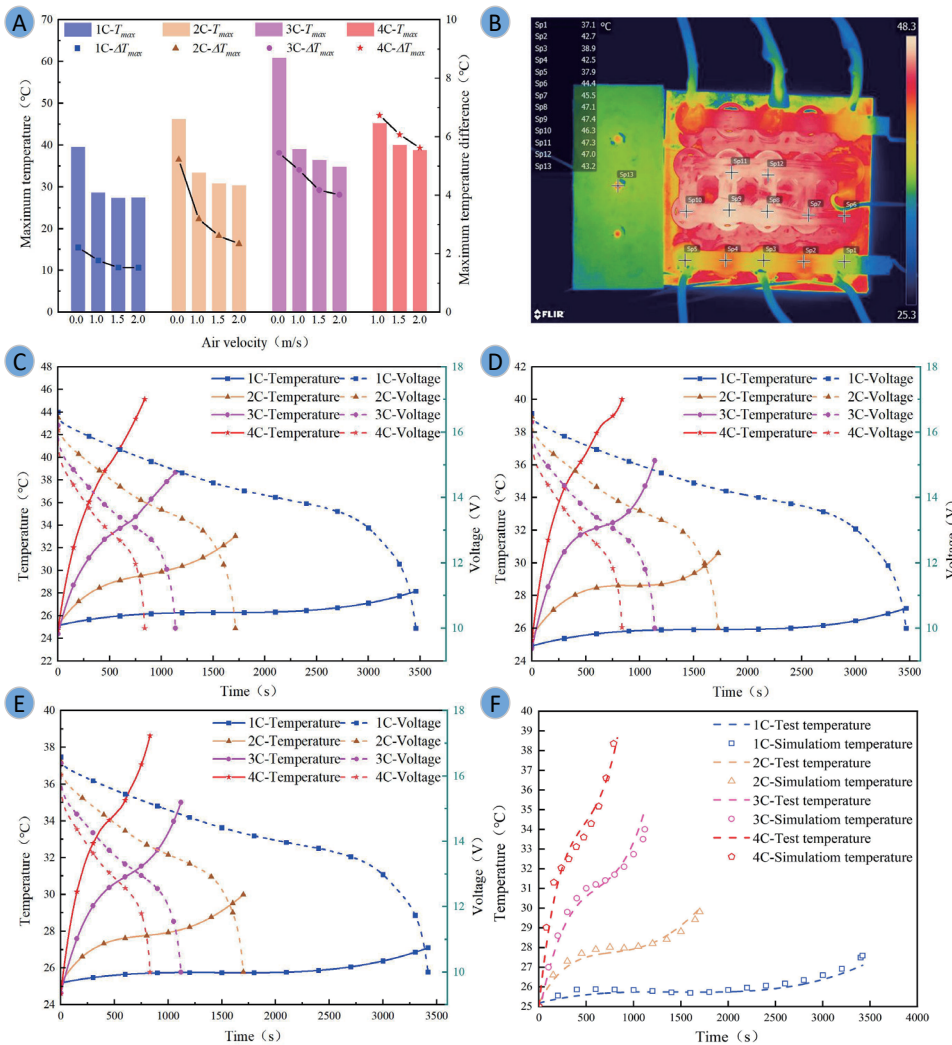


Figure 5. Measured thermal characteristics (A) Maximum temperature and temperature difference at different air velocities. (B) Thermal image under natural convection at the end of 2C discharge. Evolutions of maximum temperature (No. 10 battery) and voltage at the air velocity of (C) 1.0 m/s, (D) 1.5 m/s, (E) 2.0 m/s. (F) Comparison of the simulation results vs experimental measurements for the maximum temperature.

ature detection points in the simulation are consistent with the thermocouple positions in the experimental test). Finally, the customized busbar (made of nickel plates of 0.16 mm thick and 10 mm wide) was welded to the positive and negative terminals of the battery using a spot-welding machine to complete the electrical connections of the battery module.

Figure 4 depicts the experimental test platform of the air-cooling battery module in combination with LHP-TSP, including fan (DELTA-PEC1212DE), DC power supply (GPD-2303S), charge tester (NEWRARE-60V100A-H), temperature data acquisition device (HIOKI-LR8410R), pressure gauge and so on. The battery module was connected to the charge tester for constant current-constant voltage charging at 1C and constant current discharging from 1C to 4C. The DC power supply was connected to the fan to control the air velocity. The pressure gauges were placed at the inlet and outlet to measure the pressure drop. The precision of the experimental equipment is listed in Table S4.

The relative uncertainty of temperature primarily arises from the uncertainty of the measuring equipment and random errors associated with temperature. A K-type thermocouple (ETA-G-K-30) has an accuracy of

$\pm 0.2^{\circ}\text{C}$ within a temperature rise of 40°C , corresponding to a relative uncertainty of 0.5%. The temperature data logger system also has a relative uncertainty of 1.0%, which leads to a total relative uncertainty in temperature of 1.13% by using the root mean square technique.

RESULTS AND DISCUSSION

Battery module: forced convection vs natural convection

The measured temperature characteristics of the battery module varying with the discharge rate under different air velocities are shown in Figure 5A. As the discharge rate increases, both the maximum temperature and the maximum temperature difference in the module increases for the forced convection and the natural convection. As has been shown in Figure 5A, the maximum temperature can be maintained below 40°C even under the ultra-high discharge rate of 4C at 2 m/s, which amount to 27.43°C , 30.32°C , 34.84°C and 38.83°C for discharge rates of 1C to 4C, in comparison with the natural convection case of 39.63°C , 46.62°C , 60.89°C and above. Nonetheless, the battery temperatures are significantly higher than the appropriate operation temperature range ($15^{\circ}\text{C} \sim 40^{\circ}\text{C}$) for the LIBs under natural convection at 2C discharge rate and above.

To visualize the temperature distribution of the battery module under natural convection cooling conditions, an infrared camera was used to capture the thermal images of the battery module at a 2C discharge rate, as shown in Figure 5B. Among them, Sp1 ~ Sp10 are the temperatures of the busbars atop batteries No.1 to No.10, which are close to the battery temperatures measured with thermocouples within 1.5°C deviation. e.g., the maximum temperature of 46.2°C appears in Battery No.9, with the busbar temperature of 47.4°C . The busbar, made of nickel, was blackened to facilitate the thermal imaging. It is expected that the temperature could exceed 50°C if the

Mesh independence verification. The number of grid cells in the model significantly affects both the accuracy and speed of computations. Given the highly nonlinear nature of the governing equations in the research model and the presence of various geometric scales, it is essential to assess the grid independence. The mesh independence is examined by benchmarking the maximum temperature and maximum temperature difference with different grid cells during the battery's 3C discharge process at an air velocity of 2 m/s. As shown in Figure 3B, increasing the number of grids from 838,053 to 1,281,648 results in only a 0.01% difference in the maximum temperature of the module and a 0.1% difference in the maximum temperature difference of the module. To effectively save computational time, the intermediate mesh with 838,053 grid cells is used in the simulation study.

Experimental test method. In this study, experimental validation is conducted on a 5×4 array of 18650 battery model under an ambient temperature of 25°C . The battery cathode is made of $\text{LiNi}_{0.5}\text{Co}_{0.2}\text{Mn}_{0.3}$ whereas the anode is made of the graphite. The parameters of the battery cell are listed in Table S3. To ensure the consistency of the battery module performance, 20 batteries with consistent individual parameters such as internal resistance, voltage, and capacity were selected from a batch of 36 batteries of the same production batch. The average internal resistance of the 20 screened batteries is $13.98\text{m}\Omega$, with a standard deviation of 1.45%. The discharge capacity is 2500 mAh with a standard deviation of 0.64%, and the average voltage at 100% state of charge is 4.176 V with a standard deviation of 0.02%. To record the actual temperature of the battery module during the experiment, Type K thermocouples (ETA-G-K-30) with an error less than 0.2°C were used to monitor battery temperature changes. To avoid interfering electrical connections to the battery module, the thermocouples were soldered on the lateral surface 2 mm away from the top surface (the temper-

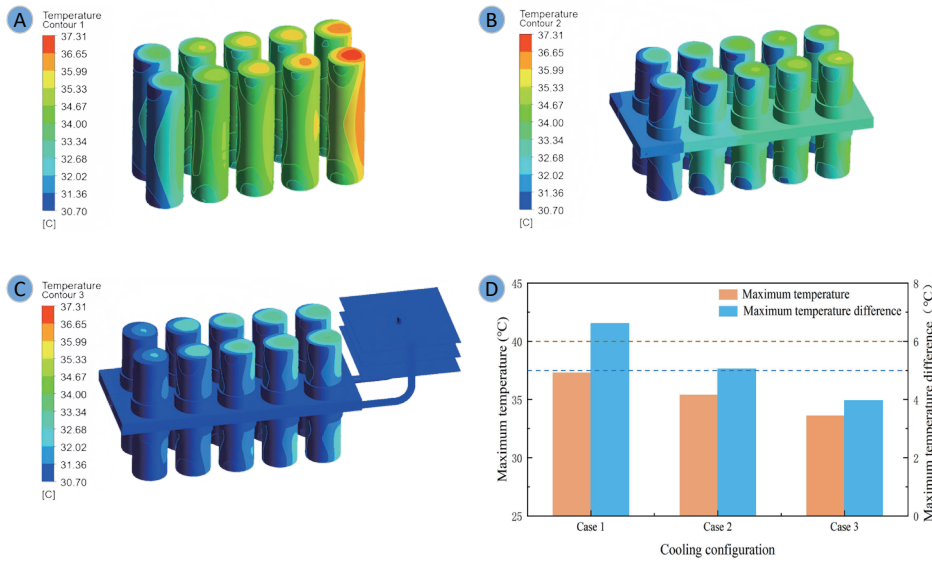


Figure 6. Battery module temperature result under a discharge rate of 3C and an air velocity of 2m/s (A) Case 1- bare battery module. (B) Case 2- battery module with TSP. (C) Case 3- the present module with LHP-TSP. (D) Maximum temperature and temperature difference of the battery modules for different cases.

discharge rate further increases. In other words, natural convection may not be sufficient for cooling the battery module at the large C-rates 2C and above. As such, the forced convection cooling strategy is adopted in the following to control the temperature rise of the battery module to meet the safety requirements for the battery.

battery temperature rises rapidly. As the discharge goes on, the temperature rise rate in the battery is slowed down, corresponding to the moderate heat generation rate at the middle portion of SOC. At the end of the discharge process, the internal resistance of lithium-ion intercalation into the cathode increases, resulting in a significant drop in voltage and rise in the internal

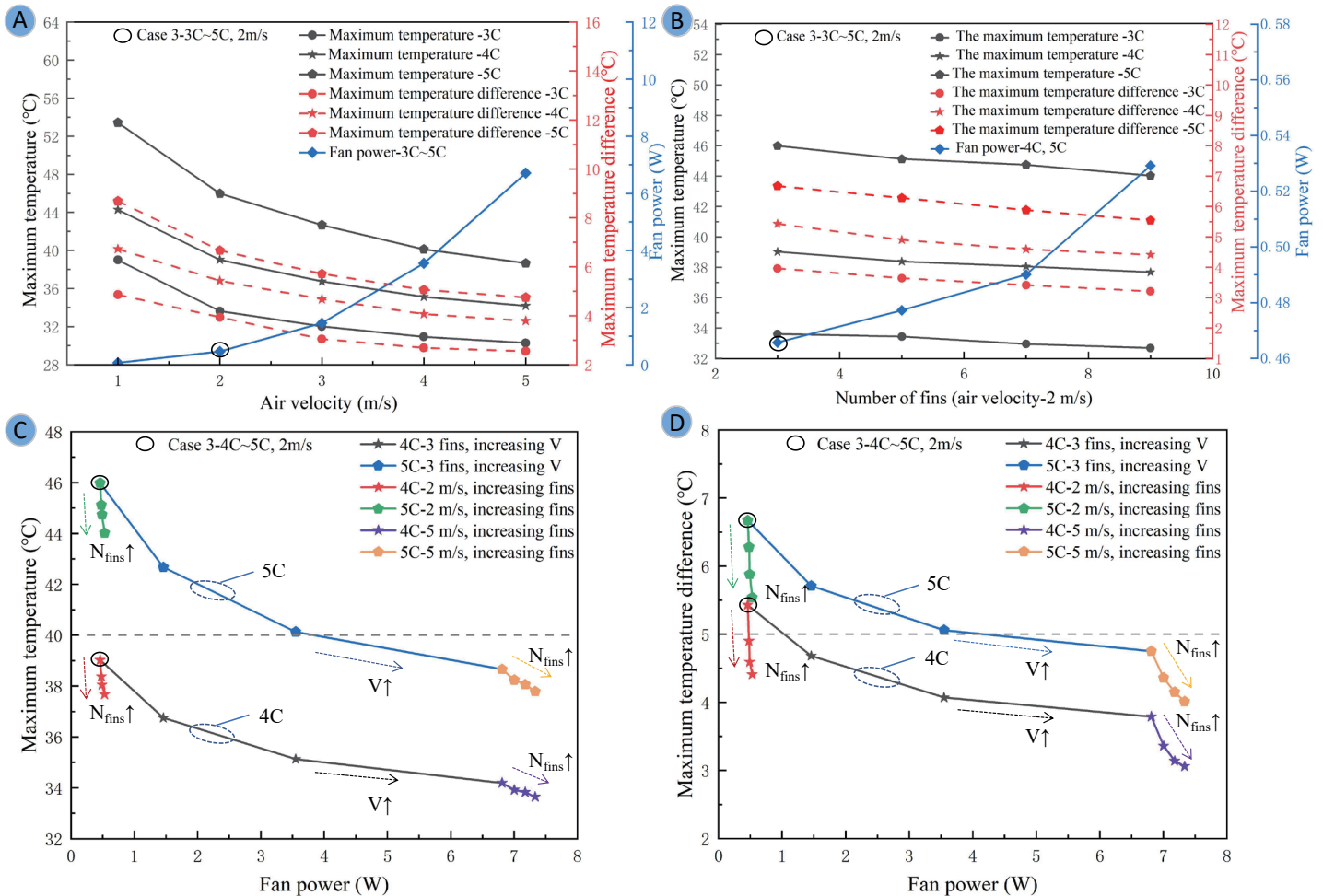


Figure 7. Effects of air velocity and fin number variation on the thermal performance and fan power of the battery module for high C-rates (A) Maximum temperature, maximum temperature difference and fan power for 3C, 4C and 5C discharges of the battery module. (B) At an air velocity of 2m/s and discharge rates from 4C to 5C. (C) The effect of fin number and air velocity on the battery module maximum temperature at different discharge rates of 4C and 5C. (D) the maximum temperature difference

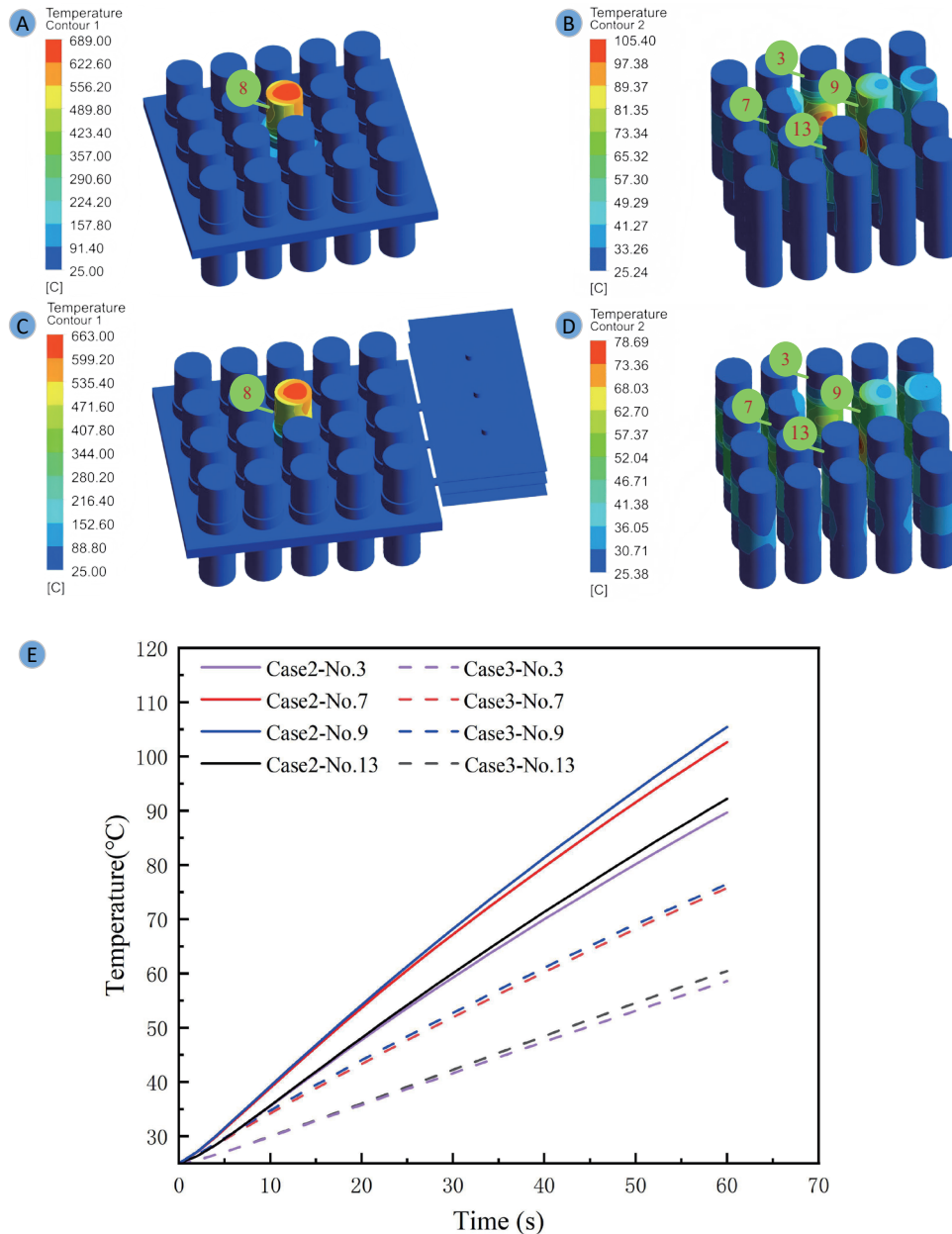


Figure 8. The temperature profiles under thermal runaway of No.8 battery at 2m/s (A) Case 2-module temperature. (B) Case 2-temperature of the other batteries exclusive of No.8 battery. (C) Case 3-module temperature. (D) Case 3-temperature of the other batteries exclusive of No.8 battery. (E) Temperature evolutions for the batteries No.3, No.7, No.9 and No.13 adjacent to the thermal runaway battery in Case 2 and Case 3.

which indicates the good accuracy of the simulation model. The small errors could be attributed to the uncertainty in the measured battery thermal parameters and the equivalent thermal conductivity of the heat-pipes.

Effect of cooling configurations

To highlight the effectiveness of the battery module with LHP-TSP, the thermal performances are compared for the three scenarios: Case 1-bare battery module without TSP, Case 2-battery module with TSP, and Case 3-the present module with LHP-TSP. The temperature profiles for all the three cases under the ambient temperature of 25°C, an air velocity of 2m/s, and a discharge rate of 3C are shown in Figure 6. The temperature results as indicated in Figure 6D show that Case 1 and Case 2 fail to meet the appropriate temperature difference of 5°C. However, due to the significant increase in the thermal conductivity of the module by the LHP-TSP as discussed in Section 2.3, Case 3 can easily meet the thermal management requirements. Compared to Case 1, Case 3 exhibits the reductions of 9.76% and 38.43% in maximum temperature and maximum temperature difference, respectively. Such a comparison demonstrates that the addition of LHP-TSP greatly reduces the temperature and the temperature non-uniformity across the battery module with potential increase in the cycle life.

It is interesting to compare the present experimental work with recently published experimental studies on the BTMSs with heat-pipes, as shown in Table S5. It is seen that the C-rate and heat generation in this work is the highest

among the compared literature. Under the same C-rate level, both the maximum temperature and temperature difference of the present battery module are lower than those in the other works including the cylindrical and prismatic battery modules. In addition, the air-cooling BTMS proposed in this work features light-weight characteristics, with a mass grouping efficiency significantly higher than those involving cylindrical batteries, close to the prismatic battery modules yet with better thermal performance for high C-rates. In fact, the present battery module even outperforms the prismatic battery module if taking into account the high discharge rate of 3C and above in the present study. As such, the present air-cooling BTMS with LHP-TSP exhibits advantages of excellent thermal performance and high grouping efficiency besides the structural advantages such as simplified structure and easy integration with cylindrical batteries.

Effects of the inlet air velocity and fin number

In this section, the effects of the inlet air velocity ranging from 1~5m/s and fin number from 3~9 fins on the thermal performance are investigated by the numerical simulation. The thermal performance of Case 3 is studied at the high C-rates of 3C to 5C and the air velocities are increased from 1m/s to 5m/s are shown in Figure 7A. With the increase in air velocity, the maximum temperature and the maximum temperature difference decrease asymptotically.

polarization resistance of the battery. This rapid increase in the internal resistance contributes to the sharp temperature elevation of the battery.

The maximum temperature difference of the battery module also decreases with the increase of air velocity, and the decrease is more obvious at high discharge rate. Under discharge rates of 1C to 3C, the maximum temperature difference of the battery module under the air velocity of 2m/s are 1.52°C, 2.34°C, and 4.01°C, which are reduced by 31.22%, 55.17% and 26.29% compared with natural convection. Even under the ultra-high discharge of 4C, the maximum temperature difference is 5.62°C, which is slightly higher than the 5°C temperature difference limit. As limited by the fan power, the maximum air velocity of 2m/s was reached in the experimental test. The air velocity and fin number can be further increased to reduce the maximum temperature difference of the battery module at the high discharge rate of 4C, which will be investigated by numerical simulation in the later sections.

To verify the accuracy of the simulation model, the discharge processes of 1C, 2C, 3C and 4C were simulated at an ambient temperature of 25°C and an air velocity of 2m/s in comparison with the maximum temperature of the battery modules in the test. The results are shown in Figure 5F. The average relative errors between the simulation and test results during 1C, 2C, 3C and 4C discharging processes are 0.60%, 1.10%, 3.53% and 3.71%, respectively,

cally, while the fan power increases at a quartic rate with the velocity. The fan power consumption can be calculated using the following equation.³⁹

$$W = VF \cdot \Delta p \quad (17)$$

where VF represents the volumetric flow rate, and Δp represents the pressure drop.

Under the discharge rate of 3C, the maximum temperature and the maximum temperature difference of the battery module are reduced from 33.61°C and 3.96°C to 30.29°C and 2.55°C, respectively, representing a reduction of 9.88% and 35.61% for the air velocity increasing from 2m/s to 5m/s. Meanwhile the fan power increases from 0.46W to 6.71W. Clearly increasing the air velocity significantly improves the thermal performance of the battery module, but the fan power also increases substantially. As such, it is necessary to consider the design factors such as temperature, temperature uniformity, and fan power in the design and optimization of thermal management systems.

Similarly, for the higher discharge rate of 4C and 5C, increasing the air velocity from 2m/s to 5m/s leads to the drop of the maximum temperature and maximum temperature difference. Especially for the ultra-high discharge rate of 5C, the maximum temperature and maximum temperature difference of the battery module under the air velocity of 5m/s can be controlled to 38.66°C and 4.75°C, which are 15.92% and 28.79% lower than the 2m/s case. It is demonstrated that, even under an ultra-high discharge rate of 5C, this BTMS can still meet the performance requirements of the module.

Figure 7B shows the effect of fin number on the temperature characteristics and fan power at the air velocities of 2m/s and the high discharge rates from 3C to 5C. The maximum temperature and the maximum temperature difference of the battery module decrease gradually as the number of fins increases, while the fan power increases slightly. Compared with 9 fins, Case 3 with 3 fins (Case 3-3C, 2m/s as encircled) shows a reduction in the maximum temperature and maximum temperature difference for 3C to 5C discharge by 3.26%, 3.46%, 4.26% and 20.35%, 18.78%, 16.94%, respectively, while the fan power increases by 24.21%.

Figures 7C-D summarizes the effects of increasing air velocities and fins on the thermal characteristics and fan power of the battery module at the ultra-high discharge rates of 4C and 5C. As shown in Figures 7C-D, the increase in the fin number has more pronounced effect on the thermal improvement with little increase in the fan power, which is more favorable in practice. For the ultra-high discharge rate of 5C, the thermal performance requirements of the module can be met either with 3 fins at an air velocity of 4.5m/s and above (Case 3) or with 3 fins and above at a velocity of 5m/s. As such, this figure can serve as a guide for the selection of optimized cooling configurations for the present battery module with LHP-TSPs.

Suppression of thermal runaway propagation

To further verify the thermal safety performance of the battery module, the heat generation of triggered thermal runaway is set at No.8 battery. It is noted that catastrophic thermal runaway could be propagated to the other surrounding batteries if effective suppression measures are absent. The simulation analysis based on Case 2 with aluminum thermal spreader and Case 3 with the LHP-TSP configurations is carried out to investigate the thermal suppression effect when a single battery undergoes thermal runaway. The inlet air velocity is set to 2m/s and the heat generation rate of the thermal runaway battery is set to 600W.⁴¹ The simulation results of thermal runaway propagation are shown in Figure 8. It is seen that the thermal runaway temperature of the No.8 battery reaches a peak value of 689°C for Case 2 and the adjacent No. 9 battery at the downstream reaches the maximum temperature of 105.4°C, which is close to the thermal runaway temperature around 120°C,⁴¹ risking the cascaded thermal runaway reactions. In comparison, Case 3 with the LHP-TSP configuration can reduce the maximum temperature of the adjacent No.9 battery to 78.69°C, equivalent to the reduction of 33.2% in temperature rise, which significantly mitigates the thermal propagation hazard.

It is noted that the viscous heat transfer limit and capacity of the heat-pipe is inversely correlated to its effective length.⁴² In the present case, No. 8 battery with triggered thermal runaway acts as the evaporator and all the

other parts of the heat-pipes and the adjacent batteries can be regarded as the condenser, which greatly shortens the heat dissipation path. Thus the heat-pipe limit and capacity could be further increased to enhance the suppression of the thermal runaway propagation. As such, the present battery module with LHP-TSP structure facilitates the heat transfer from the thermal runaway battery to the surrounding spreader without triggering the thermal runaway of the adjacent batteries, which exhibits excellent thermal suppression effect in unleashing the heat-pipe super conduction and inhibiting the propagation of thermal runaway.

CONCLUSIONS

(1) The heat-pipe thermal conductivity has been experimentally characterized to guarantee accurate numerical modeling with the deviations less than 3.71%. The equivalent thermal conductivity of the heat-pipe measured $1.314 \times 10^4 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ and the equivalent thermal conductivity of the LHP-TSP in the flow direction is $1,177 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$, 12.22 times higher than that of the TSP without heat-pipes.

(2) For the air-cooling BTMS integrated with LHP-TSP, both experimental test and numerical simulation show that the maximum temperature and maximum temperature difference of the battery module can be maintained within 35°C and 5°C at an air velocity of 2m/s for the discharge rates up to 3C, leading to a high-performance light-weight design with enhanced cooling capability at a high grouping efficiency of 78.39%.

(3) The impact of different BTMS configurations on the temperature characteristics of the battery modules is numerically investigated. Compared to Case 1 without TSP, the present battery module with LHP-TSP (Case 3) showed the reductions of 9.76% in the maximum temperature and 38.43% in the maximum temperature difference.

(4) For the air velocity increasing from 2m/s to 5m/s, the maximum temperature and maximum temperature difference under the discharge rate of 5C at an air velocity of 5m/s can be reduced to 38.66°C and 4.75°C. For the fin number increasing to 9 fins, the maximum temperature and maximum temperature difference can be controlled to 37.79°C and 4.01°C. As such, the ultra-high discharge rate of 5C can be satisfied either by increased velocity or fin number.

(5) For No. 8 battery undergoing thermal runaway, all the surrounding batteries are controlled below 78.69°C, which demonstrates the successful suppression of thermal runaway for the present battery module with LHP-TSP structure.

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AUTHOR CONTRIBUTIONS

Jiansheng Zhang: Writing – original draft, Methodology, Investigation, Conceptualization. Hengyun Zhang: Writing – original draft, Funding acquisition, Conceptualization, Formal analysis, Investigation. Shen Xu: Writing – review & editing. Wenlin Yuan: Writing – review & editing. HongFei Zheng: Writing – review & editing.

DECLARATION OF INTERESTS

The authors declare no competing interests.

DATA AND CODE AVAILABILITY

The data that support the findings of this study are available from the corresponding author upon reasonable request.

SUPPLEMENTAL INFORMATION

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