Simulation and Experimental investigation of Battery Thermal Management System for a Hybrid Vehicle

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Abstract. As a essential component in battery pack, thermal management system plays a vital role in controlling battery temperature, prolonging battery lifetime, enhancing battery safety. A battery liquid cooling system for hybrid vehicle is described in this paper. Based on vehicle thermal requirement, the battery thermal system design is optimized by 3D CFD simulation and thermal analysis in STAR-CCM+. Simulation result shows that the targets of battery thermal system can be fulfilled. Two use cases are analyzed in battery bench test, same as the simulation use cases. Comparing simulation result with bench test, it shows that the deviation of battery maximum temperature in high-temperature conditions is 1.8% and in low-temperature condition is 11%. A complete development process of battery TMS is illustrated in this paper and provide research direction for cooling plate optimization and bench test method.

1. Introduction
At present, lithium-ion battery is dominant in the power battery of new energy vehicles. The ternary lithium battery has the advantages of stable electron-chemical performance, wide discharge voltage range, high specific energy and good cycle performance, which is suitable for electric bicycle and electric vehicle power system [1, 2]. The performance of lithium-ion battery mainly depends on the working temperature and voltage. Over normal voltage and temperature range, lithium-ion battery will produce irreversible damage [3, 4]. There is a direct linear relationship between electron-chemical reaction rate and temperature. At low temperature, the reaction rate and charge discharge capacity of the battery are reduced, which makes it difficult for lithium-ion to embed into the lattice, and finally the battery capacity decreases. At high temperature, the reaction rate accelerates, which leads to more heat release, and the heat accumulation may lead to the risk of thermal runaway [5]. In addition, the uneven temperature distribution in the battery pack is also one of the problems in the wide application of lithium-ion battery, which is mainly caused by high ambient temperature, internal current change of cell and uneven heat conduction of cell structure [6]. In order to ensure the excellent performance and cycle times of the battery, the working temperature of the battery should be controlled at 20°C - 40°C, and the temperature difference of the battery should be below 5K [7].

Therefore, the thermal management design of power battery should take into account both cooling and heating functions to ensure that the battery works in the appropriate temperature range and avoid battery life degradation due to over temperature or quite low temperature. Liquid cooling system is widely used in EV cars and hybrid vehicle, which can meet the requirement of fast charging and high-power output condition. According to the thermal management requirements of a hybrid vehicle battery,
a TMS architecture and cooling plate structure are proposed. The flow field of the cold plate is obtained by solving the solution in STAR-CCM+. Also 3D model of the battery pack is established to obtain module temperature field through numerical analysis. Afterwards the battery bench test is carried out, to compare with simulation results.

2. Numerical model development for module and pack

2.1. Governing equations

2.1.1. The Navier-Stokes equations. The governing conservation equations in the module zone along with thermal system needs to be defined, which is discreted to be used by numerical methods. The Navier-Stokes equations are used to describe each phenomena which comprises a transient term, diffusion term, convection term and a source term. They are formulations of mass, momentum and energy conservation laws for fluid flows, which in the Cartesian coordinate system rotating with angular velocity $\Omega$ about an axis passing through the coordinate system’s origin can be written in the conservation form as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0$$

(1)

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} + \frac{\partial p}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \tau_{ij} + \tau^R_{ij} \right) + S_i \quad i = 1, 2, 3$$

(2)

$$\frac{\partial \rho H}{\partial t} + \frac{\partial (\rho u_i H)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ u_j \left( \tau_{ij} + \tau^R_{ij} \right) + q_i \right] + \frac{\partial p}{\partial x_i} - \tau^R_{ij} \frac{\partial u_i}{\partial x_j} + \rho \varepsilon + S_i u_i + Q_{nt} \quad H = h + \frac{u^2}{2}$$

(3)

Where $\rho$ is the fluid density, $u$ is the fluid velocity, $\tau_{ij}$ is the viscous shear stress tensor, $S_i$ is a mass-distributed external force per unit mass, $H$ is the total enthalpy content, $q_i$ is the diffusive heat flux. $\varepsilon$ is the turbulent dissipation, $h$ is the thermal enthalpy.

2.1.2. Transport equations. Most fluid flows in battery cooling plate are turbulent. To simulate and predict turbulent flows, time-averaged effects of the flow turbulence on the flow parameters are considered. To close the equations, it mostly employs transport equations for the turbulent kinetic energy and its dissipation rate, the so-called k-ε model. The Reynolds-stress tensor has the following form as Boussinesq assumption:

$$\tau^R_{ij} = \mu_t \left[ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right] - \frac{2}{3} \mu \frac{\partial k}{\partial x_j}$$

(4)

Here $\mu$ is the dynamic viscosity coefficient, $\mu_t$ is the turbulent eddy viscosity coefficient and $k$ is the turbulent kinetic energy. Two transport equations ater used to describe the turbulent kinetic energy and dissipation:

$$\frac{\partial \rho k}{\partial t} + \frac{\partial (\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \mu_t \left[ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] \frac{\partial u_i}{\partial x_j} - \rho \varepsilon + S_k$$

(5)

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_i \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} \mu_t \left[ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] \frac{\partial u_i}{\partial x_j} - C_{2\varepsilon} \frac{\varepsilon^2}{k} + S_\varepsilon$$

(6)
Where $C_{1\epsilon}$, $C_{2\epsilon}$, $\sigma_k$, $\sigma_\epsilon$ are model constants, $C_{1\epsilon} = 1.44$, $C_{2\epsilon} = 1.92$, $\sigma_k = 1.0$, $\sigma_\epsilon = 1.3$; $S_k$ and $S_\epsilon$ are user defined source terms.

2.2. Battery temperature distribution

In order to calculate how fast TMS cools the battery, the battery heat generation mechanism needs to be determined. As we all know, cell heat generates from phase change inside the cell, electron-chemical reactions, mixing effects and Joule heating. The following calculation developed by Bernardi [8] is commonly utilized to determine the heat generation in the cell:

$$ q = I \left[ (E_0 - E) + T \frac{dE_0}{dT} \right] $$

(7)

With the utilization of a battery TMS and the effect of natural convection, the battery temperature can be calculated by the following equation below:

$$ m_b C_{P,b} \frac{\partial T}{\partial t} = q - \bar{h} A (T_b - T_0) - m_c C_{P,c} (T_{c,out} - T_{c,in}) $$

(8)

Here $E_0$ and $E$ are open circuit and cell potentials, $C_{P,b}$ is the weight-average value of cell heat capacity, $C_{P,c}$ is the cooling liquid heat capacity, $m_b$ is the cell mass, $m_c$ is the cooling liquid mass per unit time, $\bar{h}$ is effective heat transfer coefficient, $A$ is battery surface area.

3. Simulation and experimental results

3.1. TMS architecture and requirement

The research object of this paper is the battery thermal management system of a hybrid passenger car, and the system requirements and multi-objective planning are evaluated at the initial stage of design. Based on the performance requirements of the whole vehicle for the battery system, combined with the basic parameters of the battery cell and the module temperature sensor layout, the performance objectives of the battery TMS are as follows: the battery working temperature range is 5 ~ 50 °C, the temperature difference of whole package is $\leq 10$ °C. Through one-dimensional simulation analysis and decomposition of the electric system target, the battery continuous charge and discharge power is obtained, and the battery heat generation is roughly estimated. Considering the weight requirements of the battery system, the cost of battery system and the space boundary of battery system, the battery TMS is determined as the liquid cooling method. The structure of battery thermal management system is mainly composed of battery chiller, electronic expansion valve, temperature and pressure sensor, air conditioning system, expansion tank, water pump, PTC heater and cooling plate, as shown in Figure 1.

![Figure 1. Vehicle TMS architecture and battery cooling plate.](image)
3.2. CFD and thermal simulation

By using Hyper Mesh software, the geometric model of liquid cooling structure is established. In STAR-CCM+ software, polyhedron and prismatic boundary layer grid are used to divide the fluid domain. The inlet flow rate is determined quantitatively according to the design of vehicle TMS, which is generally 10L / min. The hydraulic simulation results with flow rate distribution of each inlet are illustrated in Figure 2. The pressure drop of this battery TMS is 15.3kPa, lower than 30kPa the maximum pressure drop limit of coolant loop. Also the coolant runs a comparatively uniform distribution for each inlet, which is good for battery temperature uniformity.

![Hydraulic simulation result of battery cooling plate.](image)

In this paper battery heat generate calculation is resulted from coupling Amesim and STAR-CCM+ method, simultaneously updated according to battery present temperature and SOC. The thermal simulation results are analyzed by STAR-CCM+ post-processing tools, as shown in Table 1 and the temperature field screen-shot in module Z direction are shown in Figure 3. Here two typical use cases are utilized to analysis the performance of battery TMS. High velocity represents extreme condition in high temperature environment and heating in low temperature environment, in which the initial battery temperature is 40°C and -30°C.

| Use case     | Ambient temperature (°C) | Max. cell temperature (°C) | Min. cell temperature (°C) | Time (s) |
|--------------|--------------------------|----------------------------|----------------------------|----------|
| High velocity| 45                       | 50                         | 47.6                       | 165.5    |
| Heating      | -30                      | 15.3                       | 5                          | 2340     |

![Temperature field screen-shot in module Z direction.](image)

**Figure 2.** Hydraulic simulation result of battery cooling plate.

**Figure 3.** Battery temperature field screen-shot in module Z direction.
The experimental results of battery bench test are illustrated in Fig.4. Two same use cases are adopted to evaluate battery temperature range and difference. Here we can see the battery temperature change meets the objective of battery temperature range, and the maximum difference is 8.5°C lower than 10°C.

![Figure 4. Results of battery bench test.](image)

(a) High velocity use case               (b) Battery heating use case

4. Conclusion
This paper provides a development process of liquid cooling system as battery TMS for a hybrid vehicle. According to 3D CFD simulation, the cooling plate is optimized which develops into better uniform heat transfer. The battery temperature distribution shows that this battery TMS design has high heat exchange efficiency and keeps the battery working at suitable temperature range. Compared with experimental results, the deviation of battery maximum temperature in high-temperature conditions is 1.8% and in low-temperature is 11%. The experimental results are in good agreement with simulation analysis.

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