Research on Electric Vehicle Cooling System Based on Active and Passive Liquid Cooling

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Abstract. Compared with internal combustion engine automobile, the battery capacity and motor conversion efficiency for electric vehicles (EV) are limited, which means it requires lower energy consumption. To this aim, EV needs more complex thermal management system and higher thermal management requirements. This paper proposes an active and passive liquid cooling-based system cooling scheme. Coolant circulation’s components model and refrigerant circulation’s components model are built. The performance parameters for the components are obtained by fitting the experimental data. To demonstrate the performance of the proposed method, simulation experiments are conducted. The results show that it is feasible and robust to cool and heat the battery using passive cooling circuits in low and medium temperature environments. During critical conditions where the ambient temperature changes from 28 °C to 32 °C, the active and passive liquid cooling-based scheme can not only guarantee the battery operating temperature, but also save energy.

1. Introduction

The high fuel consumption and exhaust emissions of internal combustion engine automobile have exacerbated the global energy crisis and environmental pollution, which causes the transportation industry faces severe energy and environmental problems [1]. With higher requirements for fuel economy and emission standards, auto companies are trying to update and improve traditional vehicles and developing new energy-saving and environmentally-friendly vehicles. Electric vehicles (EV) are currently the most likely to replace traditional vehicle [2]. However, the thermal management technology restricts the development of EV. Compared with internal combustion engine automobile, EV need to not only cool the motor, but also consider the different requirements of the battery at low and high temperatures, and the energy utilization efficiency under the battery capacity limited [3].

High temperature affects motor output torque, cogging torque, power, overload characteristics and motor life. Severe thermal runaway of the motor has even caused the essential transformation of the motor insulation material to lose its insulation ability, which has reduced the strength and hardness of the motor metal components [4]. There is no reasonable control of the battery thermal load, which easily
causes large problems such as excessive temperature, uneven heat production, and uneven temperature distribution. Temperature not only affects the performance of the battery, but even the risk of spontaneous combustion and explosion of electric vehicles when the thermal runaway occurs [5]. In the current research, the way for the driving motor cooling is mainly liquid cooling, however, there are many cooling methods for battery, including air, liquid, heat pipe and phase change material [6].

![Figure 1. Liquid cooling strategy](image1)

Compared with other battery cooling methods, liquid cooling is widely used due to its high efficiency, easy maintenance and technical maturity. Battery liquid cooling is divided into active and passive based on different control strategies [7]. The active liquid cooling is shown in Fig. 1(a). The coolant is forcedly cooled by the refrigerant through the internal heat exchanger. The passive liquid cooling is shown in Fig. 2(b). The coolant is cooled by air through the parallel flow heat exchanger.

A lot of research has been done on liquid cooling of batteries. But most of them focus on improving the system capacity by studying the coolant materials [8] and heat exchangers [9]. To improve the energy utilization efficiency of the thermal management system, this paper proposes a cooling system scheme of EV based on active and passive liquid cooling, as shown in Fig. 2.

![Figure 2. Schematic of cooling system](image2)

2. Numerical model
According to the different working medium, the cooling system is divided into a coolant circulation system using glycol and water (50/50), and a refrigerant circulation system using R134a.

2.1. Coolant circulation system

2.1.1. Pump model. Automobile water pumps generally use centrifugal pumps. The point where the derived system curve intersects the pump curve defines the operating point of the system. Therefore, the solution of the pump model is changed to solve the system head and pump head. Pump head is solved using a pump surface with the flow rate and speed. The system head is calculated Eq. (1) [10].
Where \( P, v, Z \) is coolant pressure, speed and height at the pump inlet and outlet, \( \rho \) is coolant density, \( g \) is gravitational acceleration.

2.1.2. Radiator model. Most front-end radiators for EV are parallel flow heat exchangers. It is difficult to model the heat exchanger structurally without the original three-dimensional mathematical model due to the complexity of the structure. But performance data is easily obtained through the heat exchanger experiment. Based on the empirical method to model radiator only needs the performance data under a certain working condition. The heat load exchanged is solved by defining the relationship between the heat transfer coefficient and the mass flow rate of the coolant and air. Eq. (2) is used to solve the heat load of exchange.

\[
Q = \alpha (m_{cool}, m_{air}) A \Delta T
\]  

Where \( Q \) is the heat load exchanged, \( \alpha \) is heat transfer coefficient, \( A \) is the area of heat exchanger, \( \Delta T \) is the temperature difference between the air and the coolant.

2.2. Refrigerant circulation system
All components in refrigerant circulation system is considered as a control volume with inlet and outlet mass flow. It is assumed that there is no mass accumulation after flowing through the component, that is, the outlet mass flow rate is equal to the inlet mass flow rate, as Eq. (3). The energy balance equation is defined by the enthalpy difference between the inlet and outlet of each component. The enthalpy is a function of the pressure and temperature at component’s inlet and outlet. Eq. (4) is energy balance equation of single component.

\[
\sum m_{out} = \sum m_{in}
\]

\[
\sum m_{in} h_{in} + \sum Q_{in} + W = \sum m_{out} h_{out} + \sum Q_{out}
\]

Where \( m \) is the mass flow rate, \( h \) is the enthalpy, \( Q \) is the heat exchanged, \( W \) is effective power, subscripts \( in \) and \( out \) indicate the inlet and outlet.

2.2.1. Compressor model. Compressor model is processed using a multivariable process model. compression process is regarded as an isentropic process, so the outlet entropy of compressor is equal to the inlet entropy. The mass flow of the compressor is solved by Eq. (5). The multivariable coefficient is derived from Eq. (6).

\[
m = \rho_{in} V_D \left[ 1 + C - C \left( \frac{P_{out}}{P_{in}} \right)^{1/n_p} \right] \omega
\]

\[
n_p = \frac{\ln P_{out} - \ln P_{in}}{\ln \rho_{out} - \ln \rho_{in}}
\]
Where \( V_v \) is volumetric displacement, which is the inherent property of the compressor, \( C \) is clearance factor fitted by experimental data, \( \omega \) is the compressor speed, \( \rho_{in}, \rho_{out}, P_{out} \) and \( P_{in} \) are density and pressure at the compressor inlet and outlet.

Actual compression process is not a complete isentropic compression. The isentropic coefficient is introduced, defined by the ratio of the enthalpy difference between the inlet and outlet of the isentropic compression and the inlet and outlet enthalpy difference of the actual compression process, as shown in Eq. (7). Compressor outlet pressure can be calculated through the isentropic compression process. Power of the compressor is used to perform work on the working medium. The compressor power consumption is solved by Eq. (8), the mechanical efficiency of the compressor is introduced.

\[
\eta_{isen} = \frac{h_{isen} - h_{in}}{h_{out} - h_{in}}
\]

\[
P = \frac{m(h_{out} - h_{in})}{\eta_m}
\]

Where \( h_{isen} \) is enthalpy at the isentropic compression outlet, \( h_{in} \) is enthalpy at compressor inlet, \( h_{out} \) is outlet’s enthalpy of actual compression, \( m \) is the mass flow rate, \( P \) is the compressor power.

2.2.2. **Electronic expansion valve model.** is modeled as an adiabatic component. According to the energy balance equation, the outlet enthalpy of EXV is equal to the inlet enthalpy. The mass flow of the electronic expansion valve is calculated by Eq. (9).

\[
m = \theta A \sqrt{2 \rho_{in} \left( P_{in} - P_{out} \right)}
\]

where \( m \) is the mass flow rate, \( \theta \) is the ratio of the valve opening area, \( A \) is flow area, \( \rho_{in} \) is refrigerant density at EXV inlet, \( P_{in}, P_{out} \) are refrigerant pressures at inlet and outlet. In order to increase the adjustment range, divide the flow area \( A \) by a coefficient \( C \), and multiply the value of \( \theta \) by \( C \).

2.2.3. **Heat exchangers model.** The structural diversity and complexity of heat exchangers, including the cross-sectional geometry of internal refrigerant flow channels, the flow arrangement, and the type and arrangement of external fins, determine the precise model of heat exchangers. A lot of geometrical structures data and complete description of refrigerant internal flow patterns is required in modeling heat exchanger. And that is difficult to be gained. But the performance data under a certain working condition are easy obtained through the heat exchanger experiment. Theoretical heat exchanger model can be established by fitting shape factor with performance data. The air-conditioning heat exchanger is modeled as a finless tube with the same heat transfer area and the same internal volume as the actual model. The heat exchanger is calculated by Eq. (10).

\[
(SF_{xin}) \alpha_{xin} A_{in} (T_{wall} - T_{in}) = (SF_{xout}) \alpha_{xout} A_{out} (T_{out} - T_{wall})
\]

Where \( \alpha \) is heat transfer coefficient, \( SF \) is the shape factor of \( \alpha \), \( A \) is the area of heat exchanger, and \( T \) are the temperature of working medium and wall. The subscript \( in \) and \( out \) is the inside and outside of the heat exchanger, The wall is inside and outside contact surface.
3. Parameter configuration

3.1. Pump Surface

The determination of the pump operating point is to find out whether the system head and pump head are equal at a specific volumetric flow rate. The system head solving method has been introduced. Pump head is related to speed and volumetric flow rate, which is inherent performance data of the pump. According to a pump experimental data, the relationship between flow rate and head and the relationship between flow rate and efficiency are obtained at different duty cycles. Adopting linear interpolation method to obtain the pump head surface as shown Fig. 3(a), and the pump efficiency surface as shown Fig. 3(b).

![Pump Surface](image1)

(a) Pump surface

![Efficiency Surface](image2)

(b) Efficiency surface

**Figure 3.** Pump performance surface
According to Fig. 3 (b), the pump efficiency can be obtained, then the actual power consumption of the pump can be calculated from Eq. (11). \( W \) is the effective work of pump, derived from Eq. (12).

\[
P = \frac{W}{\eta} \tag{11}
\]

\[
W = \rho \, g \, H \, V \tag{12}
\]

Where \( \eta \) is pump efficiency, \( \rho \) is coolant density, \( g \) is gravitational acceleration, \( H \) is head, and \( V \) is coolant volumetric flow rate.

3.2. Radiator Surface
Setting the temperature to 50 °C, relative humidity to 50% at air side inlet of radiator and the temperature to 75 °C at coolant side inlet of radiator, the temperature at air side and coolant side outlet of radiator were measured in different air velocity and coolant volumetric flow rate. Calculate the actual heat exchange based on the temperature difference between the radiator inlet and outlet. The performance curve of the heat exchanger is obtained by linear interpolation, as shown in Fig. 4.

![Figure 4. Heat transfer coefficient of radiator](image)

3.3. Clearance factor
For a given compressor, a fixed clearance factor can be used to simulate the behavior of the compressor under various operating conditions. The determination of the clearance factor requires the combination of performance experimental data to establish a single compressor test model. The related parameters of a compressor for calculating the clearance factor are shown in the Table 1. These variables in the table are evaporation pressure, condensation pressure, suction superheat, condensation subcooling, cooling capacity, heating capacity, compressor power, compressor mechanical efficiency, and compressor isentropic efficiency.
Table 1. Parameters of compressor.

| $P_e$ | $P_c$ | $\Delta T_{op}$ | $\Delta T_{sc}$ | Speed | $Q_c$ | $Q_h$ | $Q_m$ | $\eta_m$ | $\eta_{isen}$ |
|-------|-------|-----------------|-----------------|-------|-------|-------|-------|----------|-------------|
| 0.3   | 1.5   | 10              | 5               | 3000  | 2.86  | 3.71  | 1.22  | 0.714    | 0.834       |
| MPa   | MPa   | K               | K               | Rpm   | kW    | kW    | kW    | /        | /           |

3.4. Shape factor

The shape factor of the air-conditioning heat exchanger is fitted according to the experimental data. According to the mass flow rate, humidity, temperature, pressure at the inside and outside inlet and the pressure, temperature at the inside and outside outlet, the heat exchange and so on. Constantly iteratively obtain the correction factor of the heat exchange coefficient at the heat exchanger inside and outside through the basic heat exchange formula.

4. Simulation and results analysis

The thermal load model of the motor, motor controller unit (MCU), and battery is modeled as the function of speed based on the test data using polynomial fitting [11]. Eq. (13) is a general formula for a simple thermal load model established by polynomial fitting based on experimental data. The order of Eq. (13) is determined by the accuracy constraint of Eq. (14) and Eq. (15). DCDC thermal load is related to the low-voltage power consumption, is set to 100w in this study.

$$Q = a_0 + a_1v + a_2v^2 + a_3v^3 + a_4v^4 + ... + a_nv^n$$  \hfill (13)

$$\sum_{i=1}^{n} (Q_{fit}(v_i) - Q_{test}(v_i))^2 < 0.005$$  \hfill (14)

$$|Q(0)| \leq 0.001$$  \hfill (15)

Where $Q$ is the thermal load, $a_i$ is fitting coefficient, $n$ is order of formula, $v$ is vehicle speed, $Q_{fit}$ is the thermal load obtained by fitting, and $Q_{test}$ is the heat load obtained by experiment.

4.1. Motor cooling analysis

The wind velocity through front-end radiator is set to the maximum amount of air provided by the fan, assuming nothing to do with vehicle speed. The speed of the water pump is set to the maximum value to establish the maximum heat exchange under different working conditions. Test whether the cooling system meets the heat dissipation requirements of the motor at different vehicle speeds and ambient temperatures. The stable temperature of the motor after 3000s is shown in Fig. 5(a), and the stable temperature at the outlet of front-end radiator is shown in Fig. 5(b).

According to the analysis of Figure 5, the motor circuit of this system has good heat dissipation performance. When the vehicle speed is 120km/h and the ambient temperature is 50 °C, the highest stable temperature of the motor and water at the front-end heat exchanger outlet are 69.5 °C and 60.7 °C. In addition, it can be seen from the figure that when the ambient temperature is 10 °C, 20 °C, the temperature at the outlet of front-end heat exchanger is less than 30 °C at various speeds. This shows that in the passive cooling relying on front-end heat exchanger, the motor, MCU and DCDC heat dissipation capability are still provided to the battery.
Temperature severely affects battery performance, so the battery operating temperature is often controlled within the range of 20 °C to 30 °C. PTC is used to heat the battery in a low temperature environment. However, the battery has a large heat capacity, and it is obviously unrealistic to heat the battery temperature from -15 °C to 20 °C. Taking the battery in this study as an example, regardless of the heat load of the ambient temperature, the energy required to heat the battery from -15 °C to 20 °C is about 14021.75 kJ.

**Figure 5.** Motor cooling characteristics

**4.2. Waste heat utilization and battery passive cooling**

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In this study, an adjustable intake grill is considered. When the ambient temperature is low, the intake grill is closed, the heat exchange of front-end heat exchanger is weakened, and the thermal load of the motor circuit is used to heat the battery. Starting from the temperature of the battery that has been heated to 0°C by PTC, the residual heat utilization of the motor and the temperature rise of the battery based solely on its own thermal load are studied under the UDDS cycle. The simulation results are shown in Figure 6, which (a) is heating process and (b) is cooling process. After 3000s, the temperature rise of the battery is only 1.9°C only based on the battery’s thermal load. The temperature increase of the battery combined with the motor circuit’s waste heat about 7.8°C.

Due to battery is always accompanied by the generation of thermal load during driving, the heat accumulation must cause the temperature to rise. To ensure battery performance and service life, the battery needs to be cooled when the temperature is greater than 30°C. In order to study the passive cooling capacity of the battery, the battery initial temperature was set at 30°C, the ambient temperature was 25°C and 28°C, and the battery temperature change under passive liquid-cooled and non-radiated 3000sUDDS cycle conditions was obtained in Figure 6 (b). The results show that in the medium and low temperature environment, passive liquid cooling is feasible in battery cooling.
4.3. Active-passive switch under critical condition

Adopting active liquid cooling or passive liquid cooling should be determined according to the cooling capacity that can be provided. Furthermore, the constraint condition of Eq. (16) is obtained. When Eq. 16 is satisfied, passive liquid cooling is used, otherwise active liquid cooling is used.

\[
Q_b + \frac{T_b + (T_{in} - T_b) e^{-\alpha A (m_c C_{pc})^{-1}} - T_{in}}{m_c C_{pc}} < 0
\]  

(16)

Where \( Q_b \) is the battery thermal load, \( T_b \) is the battery temperature, \( T_{in} \) is the battery inlet temperature, assuming equal to the external heat exchanger outlet temperature, \( m_c \) is the cooling fluid mass flow, \( C_{pc} \) is coolant heat specific heat capacity, and \( A \) is the battery and coolant contact area.
Under UDDS cycle conditions, setting ambient temperature to be a triangular wave with a period of 1000s, a maximum temperature 32 °C and a minimum temperature 28 °C, the cooling system performance is obtained in different cooling ways, which include no cooling, active cooling, passive cooling, combining active and passive cooling. Battery temperature change curve as shown in Fig. 7.

When the battery is no cooling, the battery temperature is about 31.9 °C after 3000s. Battery temperature was reduced to 29 °C and turned off, when active liquid cooling is used. The active cooling needs to be turned on twice to control the temperature within 30 °C, and the air conditioning system consumes about 233.11kJ. Only passive cooling is used. The maximum battery temperature is 30.46 °C, which exceeds the target value of 0.46 °C. Adopts combining active and passive liquid cooling, active liquid cooling is turned on once within 3000s, power consumption is about 141.43kJ, and the maximum battery temperature 30 °C.
5. Conclusion
This paper proposes an active and passive liquid cooling-based EV cooling system scheme, which is used for the motor, MCU, DCDC and battery cooling. The mathematical models for each component of the system are established by combining experimental data and component mechanism. The performance data for each component is obtained by experimental data fitting using linear interpolation. To demonstrate the proposed method, the simulation tests are conducted based on the model built by Flowmaster. The experiment results show that the proposed method can satisfy the system cooling requirement for different kinds of device. The combination of active and passive liquid cooling schemes can not only guarantee the battery operating temperature, but also save energy.

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