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Numerical investigation of liquid cooling cold plate for power control unit in fuel cell vehicle

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ABSTRACT

This paper proposed fifteen structure schemes of the liquid-cooled plate for thermal control of the power control unit (PCU) in fuel cell vehicle (FCV). At the given serpentine channel with inconstant width, pin fin arrays with various configurations were arranged to improve the performance of three heating zones with multiple heat sources. Based on the same setup and boundary conditions, numerical simulations were conducted for different schemes. The solutions were validated by grid independence check and comparison with previous researches. Effects of fin geometrical parameters (such as diameter, height, fin pitch and shape) on pressure drop and heat transfer characteristics were investigated. Furthermore, two dimensionless factors η_H and η_P were quantified to evaluate the heat transfer enhancement and pressure drop augmentation. The dimensionless performance evaluation factor P_{EF} was cited to assess overall performance of the cold plate. Based on three factors mentioned above, cooling performances of three heating zones and the whole plate were compared among all schemes. According to the performance comparison, scheme 12 employing circular fins with diameter of 4 mm was selected as the optimal solution for the cold plate.

1. Introduction

With increasing concerns on oil resources shortage and environmental pollution, fuel cell vehicle, one of the most potential automotive technologies, has been developed rapidly. In order to reduce the weight and size of FCV, some electronic devices in charging of power control, conversion, conditioning and high speed switching applications (such as DC–DC conversion and IGBTs and 12 V DC) are packaged into a power electronics module called power control unit. PCU is the brain of the FCV engine system, which manages the power flow between electric motor generator and fuel cell.

During last few years, heat dissipation requirement of PCU keeps going up along with size reduction and power density increase. Performance and reliability of the electronic devices packed in PCU are greatly affected by operating temperature. Therefore, high performance cold plate to maintain devices in the optimum temperature range [1–6] becomes an critical component.

According to the coolant adopted in the electronics cooling systems, the cooling systems can be divided into three types: air-cooling, single phase liquid cooling and liquid refrigerant cooling [7–9]. As for automobile application, cooling system using single phase liquid such as water/ethylene glycol is the most popular. In order to improve the performance of single phase liquid-cooled heat exchanger, micro scale-channels [10–16] and jet impingement [17,18] have been widely discussed. However, these technologies make the existing FCV cooling system complicated and expensive.

A commercially available cold plate, conventionally an aluminum plate attached to the electronic packaging, usually contains an internal serpentine cooling passage with inconstant width. Pin fins arrays mounted on the endwall of the passage play an important role for heat transfer enhancement (HTE). From technological and economical perspective, improving efficiency of the existing cold plate that contains serpentine channels attached by mini-fin arrays is a better solution for thermal management for PCU.

There have been experimental and numerical investigations on pin–fins channels. The effects of various geometrical parameters on the heat transfer and friction characteristics are topical subjects of the researches. The influences of the aspect ratio of streamwise and spanwise pitch on the thermal performance of inline and staggered pin fin arrays were reported in literatures [19–24]. And, the effect of pin-tip leakages was investigated in four rectangular channels with staggered pin–fin arrays [25]. Besides, shape is an important design factor of pin–fin array. The performance

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Nomenclature

A_f	cross-sectional area of fin
A_{fa}	surface area occupied by the fins
A_P	projected area of channel
A_T	total surface area of channel
C_c	clearance between top of fin and endwall of channel
D_c	equivalent hydraulic diameter of channel
D_f	hydraulic diameter of fin
f_f	friction factor across fin array
h	average heat transfer coefficient
H_f	height of fins
k	thermal conductivity of fluid
k_s	thermal conductivity of solid
N_f	total number of fins
N_2	fin number in streamwise direction
Nu	Nusselt number
Nu_f	Nusselt number across fin array
$\triangle P$	pressure drop
P_{EF}	performance evaluation factor
$\triangle P_{tot}$	total pressure drop of channel
P_f	perimeter of fin
Pr	Prandtl number of fluid
Pr_w	Prandtl number of fluid at wall temperature
Q	heat transfer rate
Re	Reynolds number
Re_f	Reynolds number of fin
S_1	spanwise pitch
S_2	streamwise pitch
T	temperature
T_{max}	maximum temperature
T_{in}	flow temperature at entry of the zone
u_{in}	flow velocity at entry of the zone
u_{max}	maximum velocity across fin array
V_f	volume of fins
V_t	Volume of smooth duct
W_c	width of channel

Greek symbols

μ	dynamic viscosity
ρ	fluid density
η_H	heat transfer enhancement effectiveness factor
η_P	pressure drop augmentation factor

comparisons between circular fins and non-circular fins such as hexagonal, diamond-shaped and elliptic fins were represented in Ref. [26–29]. The elliptic fins were found to achieve higher heat transfer coefficient and smaller pressure drop than the circular pin fins [28,29].

In most previous studies [30–33], pin fin arrays were conducted in plain and straight rectangular ducts. However, the conclusions derived from those works cannot be applied to the pin–fin arrays mounted on the serpentine channel in PCU's cold plate. And due to varied device power densities and irregular geometry, an experimental or theoretical investigation on the cold plate would be too complicated, expensive and time-consuming. According to several similar researches [34–37], numerical simulation have been proved to be a simple, efficient and reliable approach to perform geometry optimization and thermal performance analysis on cold plate.

This study numerically investigated the thermal and hydrodynamic characteristics of the PCU's cold plate with various fin configurations. Impacts of geometrical parameters such as hydraulic diameter, inter-fin pitch ratios, clearance ratio and shape were discussed. Moreover, the performance evaluation factor (P_{EF}) was proposed to estimate overall performances of cold plate and to determine the optimal scheme under certain mass flow rate and heat fluxes.

2. Description of the cold plate

The power control unit consists of two DCF IGBTs, a DCF inductor, and a 12 V DC. DCF IGBT is a DC-to-DC converter that controls the output of the direct current for fuel cell, with heat emission of 350 W for each. DCF inductor is used for energy storage during DC-to-DC converting, producing heat load of 1500 W. The 12 V DC is used to convert high-voltage direct current to 12 V, producing heat load of 220 W. With consideration of electrical and manufacturing issues, the heat releasing devices are installed on the backside of the cold plate, as illustrated in Fig. 1. The outer dimensions of the aluminum cold plate are 700 mm $\times$ 430 mm $\times$ 25 mm. Inside the plate, a serpentine channel with depth of 9 mm, whose bottom endwall is 11 mm away from the heating surface of the plate, is fixed and adjusted with respect to the locations of multiple heat resources. Ethylene glycol–water mixture (50%:50%) is used as the coolant passing through the cold plate. More details about the cold plate are listed in Table 1.

As components' high heat flux dissipated into the cold plate, three heating zones, namely zone A, zone B and zone C, were separated from the cooling channel. Fin arrays were arranged on the internal wall of heating zones. It is well known that fin arrays could not only increase internal wetted surface area, but also lead to a high turbulence intensity, which would significantly enhance convective heat transfer performance. The proposed fifteen structure schemes with various fin array configurations for the given cooling channel are listed in Table 2. Circular fins with different diameters (3 mm and 4 mm) and with various heights (4.5–9 mm) were applied in schemes 1–13. Scheme 14 and 15 adopted 9 mm-height elliptic fins with equivalent diameter of 3 mm and 4 mm, respectively. All fins were positioned in staggered arrays, whose spanwise inter-fin pitch ratios were in range of 2.5–4. The streamwise pitches were half of the pitches in spanwise direction. The order of magnitude of fin dimensions was chosen for present investigation according to pin–fin channels applied in electronic industry. To evaluate the cooling performances of these schemes, a cold plate with smooth-walled cooling channel was introduced as the baseline geometry (defined as scheme 0). Altogether, performance comparisons of three heating zones and the whole plate were performed among sixteen schemes. The optimum geometry was selected for the PCU cold plate based on the requirements of low operating temperature, small pressure loss and high thermal heat transfer efficiency.

3. Numerical model

The finite volume method (FVM) based software FLUENT was employed for the numerical simulation. The model was established according to the following assumptions:

(1) Heating elements were simplified to four thermal resource surfaces on the substrate of the cold wall where uniform heat fluxes were applied.
(2) Considering the free convection of the air on the surfaces of the cold plate, 5% of the heat load was dissipated into the surrounding air instead of the coolant passing through the channel.