A Cooling design for an end cell in fuel-cell stack using forced air flow in metal foam: Modelling and Experiment

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Abstract. The effect of temperature on an end-cell plate in a proton exchange membrane (PEM) fuel cell stack is presented in this paper. Numerical analysis and experimental work investigated the impact of a heat exchanger filled with metal foam on the end-cell plate used for cooling a PEM stack. One side of the edge plate was subjected to constant heat flux (1.55 W/cm²) while the other side was insulated. A metal foam with a pore density of 40 and having a porosity of 60% was analysed. The model was simulated in ANSYS R3-CFD, and the thermal problem was solved using the local thermal non-equilibrium (LTNE) to investigate the heat transfer between the fluid and solid phases. The thermal solution studied the effect of metal foam heat exchangers at different airflow velocities between 0.2 and 1.5 m/s on end-cell plate’s temperature. The results showed heat transfer enhancement in plate temperature, proving that the use of the LTNE assumption illustrated the difference between the end-cell plate and fluid flow temperatures. The effect of inlet air velocity on the plate temperature was also examined. An experimental model of a commercial PEM stack was constructed out of aluminium. This model was tested at the same conditions as those employed in the numerical solution. The heat generated by a fuel cell was simulated by a surface heater providing the same heat flux as the actual PEM cell. The numerical and experimental results for the end-cell plate for both at flow direction and different inlet velocities were assessed. There was a good agreement between the numerical and experimental results.

1. Introduction
One of the most critical points in designing fuel cells is appropriate thermal management. Most fuel cells work in a consistently safe range of operating temperatures [1]. The PEMFC operating temperature is between 60-80°C. Keeping these temperatures in the reliable range and reducing the gradients of cell temperature is the aim of thermal management. The uniform distribution of cell temperature decreases the risk of stack flooding and dehydration of the membrane. The application and size of the PEMFC stacks are responsible when choosing suitable thermal management [2]. Some researchers investigated the thermal management of PEMFC stacks, but only a few studied the use of heat exchangers in PEMFC stack cooling applications. Significant studies have been conducted on the different strategies of thermal management for fuel cell stacks, and water-cooled methods were recently used in the thermal management of PEMFC stacks. This method requires an additional water-cooling loop that uses extra accessories, such as a radiator. So, to avoid increasing expenses, researchers were looking for something as an alternative. The heat exchanger was introduced to the field of the fuel cell stack industry to keep the stack at operating temperatures safe range and to remove the waste heat produced by the stack during...
operation [3]. The air-cooled systems in fuel cell stacks use air as flow coolant, where the air is forced to flow through the air cooling channels [4].

In some cooling systems, a cooling fan is placed at the bottom of the unit and blows the air across the cooling channels [5]. A few researchers have thermally modeled the PEMFC stack and analytically and numerically studied the thermal management of the fuel cell stacks. Chupin et al. [6] investigated a thermal design of water-cooling for the PEMFC stack. This was followed by Promislow and Wetton [7], who studied the thermal model to investigate temperature distribution in straight flow cooling channels for a fuel cell stack. Yu and Jung [8] numerically established a 2D thermal model of PEMFC; the water-cooled method was used. Their proposed model studied the conduction of heat transfer inside the membrane and the convection of waste heat removed by cooling water. A 2D model was numerically developed by Sasmito et al. [9], wherein the heat transfer by convective forced air was investigated. The study represented the effects of pressure drop in PEMFC stack performance. Previous research has been established by Sinha and Wang [10]. They examined the PEMFC performance when operated at high temperatures, where the applied temperature was constant at the external boundary conditions of the stack. A qualitative study by Ramousse et al. [11] studied the thermal management of the PEMFC and evaluated a thermal model of a fuel cell stack and studied the effects of the distribution of temperature on the stack. Several studies focused on using metal foam to enhance heat transfer in many applications [12]. Some parameters such as pore diameter and porosity play a major effect on the thermohydraulic behavior of metal foams. Due to the thermal properties of metal foam, heat exchangers are extensively used to improve heat transfer [13]. Odabae et al. [14] numerically and experimentally studied the difference between water and air-cooled heat exchangers for design optimization. In their study, the heat transfer and pressure drop across the heat exchanger in a PEMFC stack were investigated. The results presented the temperature distribution and airflow rate of varied models. Recently, Boyd et al. [15] numerically simulated a 3D model to study the thermal management of a metal foam heat exchanger that was used to remove waste heat from PEMFC. The results showed uniform distribution of the plate temperature at high inlet velocities. This paper presents the effect of high temperature on the edge plate of a PEMFC stack. The model will be simulated in ANSYS R3-CFD, and the thermal equations will be solved by LTNE to investigate the heat transfer between the fluid and solid phases. A metal foam with a pore density of 40 having a porosity of 60% will be used. This paper will study the effects of using metal foam heat exchangers at different air inlet velocities on end-cell plate temperature.

2. Modelling and numerical approach

In this model, the edge cooling plate was machined, where all shoulders between the channels were removed and became one large air-cooling channel. The large air channel was filled with metal foam, as shown in Figure 1(a). The resulting convection heat transfer problem and its boundary conditions are given schematically in Figure 1(b).

![Figure 1](image_url)

*Figure 1. (a) End plate filled with metal foam, (b) Air-cooled end cell: channel filled with metal foam*

The morphological properties are the metal foam used are given in Table 1.
Table 1. The sample specifications and metal foam properties.

| Metal foam (PPI) | Dimension (mm) | Porosity ε (%) | Heated plate | Heat flux (W/m²) |
|------------------|----------------|----------------|--------------|-----------------|
| 40               | 3.56 x100      | 60             | Aluminium   | 1.55            |

The following governing equations were solved using ANSYS- CFD:

Continuity:
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]  

(1)

Momentum:
\[
\frac{\partial p}{\partial x} + \frac{\mu}{\varepsilon K} \frac{\partial^2 u}{\partial x^2} - \frac{\rho_f u^2}{\sqrt{K}} = 0
\]  

(2)

where \( \varepsilon \) and \( K \) are the porosity and permeability of the foam, respectively.

Fluid energy equation:
\[
k_f \left( \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial x^2} \right) + h_s f \sigma (T_s - T_f) = \rho c_p \left( \frac{\partial (u T_f)}{\partial x} + \frac{\partial (v T_f)}{\partial y} \right)
\]  

(3)

Solid energy equation:
\[
k_s \left( \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial x^2} \right) + h_s f \sigma (T_f - T_s) = 0
\]  

(4)

where \( u \) and \( v \) are the velocity components in the flow direction (x) and perpendicular to the flow direction (y), and \( p \) is the pressure. \( \rho \) and \( \mu \) are the density and viscosity of the fluid, respectively. \( f \) is the form drag coefficient for the foam. \( k_f \) and \( k_s \) are the effective thermal conductivities of the fluid and the solid, respectively. \( \varepsilon \) and \( K \) are the porosity and permeability of the foam, respectively. \( h_s f \) is the interfacial heat transfer coefficient, and \( \sigma \) is the interfacial surface area density (m²/m³) inside the foam. \( T_f \) and \( T_s \) are the fluid and solid temperatures, respectively.

The following boundary conditions were imposed:

At the channel inlet:
\[
u = u_{in}, \, v = 0, \text{ and } T_f = T_{in}
\]  

(5)

At the heated walls:
\[
y = 0, \, q^* = -k_f \frac{\partial T_f}{\partial y} = -k_s \frac{\partial T_s}{\partial y}, \, u = v = 0
\]  

(6)

At isolated walls:
\[
q^* = 0, \, u = v = 0
\]  

(7)

At the channel outlet:
\[
p = 0 \text{ and } \frac{\partial T_f}{\partial x} = \frac{\partial T_s}{\partial x} = 0
\]  

(8)

3. Experimental work

An identical sample to the commercial stack, EFC-100-03-6-ST, was built precisely in the lab from aluminium to study heat transfer on the end-cell plate and to employ this application on the fuel cell stacks air-cooled system. In this model, all shoulders between air channels in the bipolar plate were removed, which made the air channels one large channel filled with metal foam as a heat exchanger. The stack produces 500 Watts for an active area of 100 cm². The stack has three cells, and each cell provides 166.7 watts. The heat flux applied to each cell was 1.55 W/cm². Figure 2(a) shows the new plate design of the stack with large, air-cooled channels. The bipolar plate has large channels filled with 40 PPI having a porosity of 60% of aluminium foam, and the air was forced to flow inside the air-cooled channels. Figure 2(b) shows the back side of the plate with a surface heater to apply constant heat flux.
Figure 2. (a) The new plate design of the PEMFC stack with large, air-cooled channels, (b) The installation of the thermofoil heater on the back of the end-cell plate.

Super Flow: SF600 was used in this experiment, as shown in Figure 3. The high rates of airflow produced by this machine make it efficient to force the air through the sample’s thin channels. Using a flow direction controller, one must simply control the intake and exhaust of the airflow direction. This machine also has a flow controller to maintain the desired air velocity in the end-open tunnel. A square open-end tunnel has a length of 60 cm, and identical cross-section size of 12.50 cm was tightly connected to the machine inlet section. The fabricated test section made of clear Plexiglas has a square cross-section size similar to the air tunnel outlet that was connected to the end of the tunnel by latches. For pressure drop measurements, a heavy-duty pressure transducer made by Honeywell was installed, and a Nanometer was used for inlet temperature and velocity measurements.

Figure 3. Schematic diagram of the experimental setup.

Two holes were drilled to install the speedometer probe and pressure drop device sensor. The sample was inserted into the test section and isolated on all sides by the Plexiglas and sealed by thermal silicon. No air leakages were allowed in this experiment. So, the open-end wind tunnel must tightly connect to the test section by latches. A foam was inserted into the test section entrance to ensure a fully developed...
flow state. Both the differential pressure transmitter device and twelve thermocouples were connected to the data acquisition. The measurement of the plate's temperature and pressure drop was measured according to the desired inlet velocities. As recommended, the measurements were taken three times and the average was considered.

The root-sum-squares Eqn. (9) was used to calculate the uncertainty in the pressure drop measurement [16]. Fixed and reading errors were declared by the instrument manufacturer. A fixed error of 2% and a reading error of 0.25% were stated by the heavy-duty pressure transducer’s manufacturer.

\[
\delta p = \pm \sqrt{e_f^2 + e_r^2}
\]

where \(\delta p\) is the pressure uncertainty, \(e_f\) is the fixed error, and \(e_r\) is the reading error. The inlet velocity was measured by a digital manometer. The uncertainty of 3% as a fixed error and 0.015 m/s as a reading error were reported by the device manufacturer. The pressure uncertainty is 2.06%, and air velocity uncertainty is 3%.

4. Results and discussions
The numerical and experimental results are shown in Figure 4 (a) for end-cell plate temperatures at 2.5 cm near the entrance in the flow direction. Increasing inlet velocity decreased both temperatures and brought them to the safe operating temperature range. The numerical and experimental temperature results in the middle of the plate at different inlet velocities for the end-cell plate. In Figure 4(b), both temperature trends had the same behaviour, decreasing with the increase of the inlet velocity until they reached the lowest temperatures at 1.5 m/s. Shown in Figure 4(c) is the result of the end-cell plate temperature at the end of the plate, which was exactly at 7.5 cm. The result showed a good agreement between the numerical and experimental temperatures measurement. The effect of the inlet velocity was observed, where increasing the velocity decreased the total plate temperatures.
Figure 4. Numerical and experimental temperatures for end-cell at (a) 2.5 cm, (b) 5 cm, (c) 7.5 cm.

The numerical and experimental results for the end-cell plate at 0.2 m/s of inlet velocity at flow direction are shown in Figure 5 (a). This figure shows the effect of inlet airflow velocity on the plate temperature at different locations. At 2.5 cm from the channel entrance, the temperature was about 380 for both temperature readings. The plate temperature slightly increased with the increase of the velocity and that was because of the heat produced from heaters on the plate base. Increasing the inlet velocity from 0.2 to 0.6 m/s decreased the plate temperature by about 10 degrees at 2.5 cm in the flow direction as shown in Figure 5(b). The numerical temperature results were higher than the experimental temperature results. There was a slight increase in the plate temperature, but the total plate temperature decreased when the inlet velocity was increased to 0.6 m/s. Figure 5(c) shows the comparison between the numerical and experimental temperature measurements for different plate locations at the same inlet velocity. There was a big drop in the plate
temperature; increasing the inlet velocity to 1 m/s decreased the plate temperatures by about 20 degrees total plate temperature. The temperature measurements for the end-cell plate at 1.5 m/s of inlet velocity in the flow direction are shown in Figure 5(d). Increasing airflow velocity to 1.5 m/s decreased the numerical and experimental temperature measurements. Both temperature measurements had the same behaviour. The plate temperature slightly increased, but generally decreased in the total plate temperature. The numerical and experimental results had a good agreement.
Figure 5. Numerical and experimental for end cell at (a) 0.2 m/s, (b) 0.6 m/s, (c) 1 m/s, (d) 1.5 m/s

5. Conclusions
The impact of maximum temperature on an edge plate filled with metal foam and used for cooling the PEMFC stack was investigated in this paper. The thermal equation of both fluid and solid phases was solved by LTNE. The study was performed for metal foam with 40 PPI and having porosities of 60% at different airflow field velocities. The results showed the heat transfer enhancement at plate temperature and proved that the use of the LTNE assumption illustrated the difference between the edge
plate temperature and fluid flow temperature. Increasing inlet air velocity decreases the plate temperature and there was a good agreement between numerical and experimental results. The difference in temperature between numerical and experimental results for the end-cell plate was investigated. The lowest difference was about 0.3% in 7.5 cm at 1.5 m/s of inlet velocity, while the highest difference was about 0.8 % at 1 m/s.

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