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# A parametric study of the performance of a planar membrane humidifier with a heat and mass exchanger model for design optimization

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ABSTRACT

The performance of a proton exchange membrane fuel cell (PEMFC) is seriously changed by the humidification capability available when equipped with a PTFE® membrane. Typically, the humidification of a fuel cell is carried out by means of an internal or external humidifier. A membrane humidifier is applied to the external humidification of residential power generation fuel cell due to its convenience and high performance. In this study, a static model is constructed to understand the physical phenomena of the membrane humidifier in terms of geometric parameters and operating parameters. The model utilizes the concept of planar type heat exchanger with mass transport through the membrane. The model is constructed with FORTRAN in a Simulink® environment for consistency with other components of the model we previously developed. The results show that the humidity of the wet gas and the channel length, the membrane thickness and wet gas inlet humidity are critical parameters affecting the performance of the humidifier.



## 1. Introduction

A proton exchange membrane fuel cell (PEMFC) has many advantages such as high efficiency, fast start-up, very low noise, compact and easy packaging, and no air pollution. The advantages make PEMFC be applied as alternative powertrain of automobiles. The PEMFC has also recently been applied to residential power generation because it can produce heat and power simultaneously. Because of its intrinsic aspect, the PEMFC should be well humidified for high performance with reliable operation.

A typical method of humidification is to moisten the entrained air and hydrogen with an external humidifier. Various types of external humidifiers are available such as bubbler type, injection type, vapor exchanging membrane type, and enthalpy wheel type and so on. A vapor exchanging membrane type is called a membrane humidifier which is an attractive candidate for a PEMFC because it is free from parasitic loss. The membrane humidifiers are classified as planar type and tubular type. Even though the performance of the humidifier is very significant to the overall performance of a PEMFC, a few studies for the tubular humidifier can be found in literatures [1–5]. The tubular type humidifier can maximize the heat and mass transfer performance with lots of tubes in very compact form but cost effectiveness mass production would be difficult. Compared with commercially viable tubular humidifiers, a planar humidifiers offer a cost efficient and mass producible alternative which motivates us to develop a planar type humidifier. Very detailed researches are found in planar membrane humidifier from the design methodology to $\varepsilon$-*NTU* modeling approach [6–9].

Nonetheless, simulation models in the literature are mostly focused on the tubular type humidifier [1–5] or assumes no condensation inside the humidifier [7–9]. However, when the water is condensed, the diffusion flux has to be higher than that of non-condensed case. Accordingly, when the simulation model is considering the thermodynamic formulation of liquid condensation, water transport mechanism seems to be more feasible. On the other hand, the counter flow configuration can possibly condense the water vapor at the inlet of wet inlet which can has very high convective heat transfer coefficient due to entrance effect. In this study, those two phenomena are captured with simulation model. After the simulation model is verified with experimental data, this simulation model is applied to the heat and mass transfer analysis of planar type humidifier.

## 2. Analysis methodology

### 2.1. Experimental setup for validation of simulation model

The planar humidifier consists of multiple layers of plates, each of which has flow channels on both sides. The Nafion® membrane

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**Nomenclature**

| | |
|---|---|
| $A$ | area ($m^2$) |
| $D_h$ | hydraulic diameter ($D_h$ = 4 × exposed area/perimeter) |
| $h$ | enthalpy (kJ/kg) or heat transfer coefficient ($W/m^2/K$) |
| $j$ | mass transport flux ($kg/cm^2/s$) |
| $k$ | heat conductivity (W/mK) |
| $L$ | length (m) |
| $\dot{m}$ | mass flowrate (kg/s) |
| $M_m$ | equivalent weight (g/mol) |
| $M_w$ | molecular weight of water (g/mol) |
| $N$ | number of membranes |
| $N_i$ | molar flux ($mole/cm^2$ s) |
| $p$ | pressure (kPa) |
| $q$ | heat flux ($W/m^2/s$) |
| $Q$ | heat transfer through membrane (W) |
| $S$ | exposed area ($m^2$) |
| $T$ | temperature (K) |
| $x_i$ | mole fraction |
| $\varepsilon$ | effectiveness factor |
| $\lambda$ | water content |
| $\omega$ | absolute humidity |
| $\mu$ | kinetic viscosity ($m^2/s$) |
| $\rho$ | density ($g/cm^2$) |

*Subscripts and superscripts*

| | |
|---|---|
| $a$ | air |
| $cond$ | condense |
| $conv$ | convective heat transfer |
| $diff$ | diffusion |
| $d$ | dry membrane |
| $fg$ | phase change from steam to water |
| $g$ | gas |
| $i$ | inlet |
| $m$ | membrane |
| $l$ | liquid |
| $p$ | perimeter |
| $o$ | outlet |
| $sat$ | saturation |
| $tr$ | transport through membrane |
| $v$ | vapor |

of each layer separates the hot humid gas from the cold dry gas. A schematic of the internal structure and an actual image of planar membrane humidifier (called *p*-small) are shown in Fig. 1 [10]. To maximize the mass transport efficiency, the flow direction of the hot humid gas is opposite to the flow direction of the cold dry gas. The channels of *p*-small are square and the channel ribs are thick enough to sustain the compression force for bonding. Geometrical and operating parameters are shown in Table 1.

Experimental apparatus is shown in Fig. 2. The mass flow rates, temperatures, and humidity of the inlet gases were independently controlled while the pressure, temperature, and humidity were monitored at the inlet and outlet of the humidifier [10]. The amount of heat and mass transfer were calculated from the measured experimental data such as exit temperature, pressure, and relative humidity. The airflow of each channel was carefully balanced using the mass flow controller to reduce the experimental disturbance by the difference of flow rate and pressure. Therefore, the only difference in the gas flow rates at the channel inlets was the amount of vapor present.

### 2.2. Modeling approach to understand the performance variation of prototype planar membrane humidifier

#### 2.2.1. Conservation equations

A simulation model is developed to understand the heat and mass transfer trends over various parametric changes. The gas in the simulation is assumed to be ideal gas and it is also assumed that no liquid water is induced into the channel. The control volume of the model is shown in Fig. 3. The humidity and temperature of the dry air in control volume 1 (c.v.1) increase simultaneously by mass transport and the gas flow direction of c.v.1 is opposite to the gas flow direction of control volume 2 (c.v.2). Those two features mean that the hot humid gas at the exit is rarely condensed. Therefore, the species conservation of the cold dry gas is as follows:

$$\frac{d}{dx}(\dot{m}_{g1}x_{v1}) = j_{m,1}L_p. \tag{1}$$

The amount of hot humid gas of c.v.2 at the exit decreases the vapor fraction due to the mass exchanging process with cool dry gas in c.v.1. Furthermore, the condensation of humid gas can take place, depending on the heat exchanging characteristics. The mass flow is decreased by those two consumptions:

$$\frac{1}{L_p}\frac{d}{dx}(\dot{m}_{g2}x_{v2}) = -j_{m,1} - j_{cond}, \tag{2}$$

Additionally, mass conservations of c.v.1 and c.v.2 are:

$$\frac{1}{L_p}\frac{d}{dx}(\dot{m}_{g1}) = j_{m,1}, \tag{3}$$

$$\frac{1}{L_p}\frac{d}{dx}(\dot{m}_{g2}) = -j_{m,1} - j_{cond}, \tag{4}$$

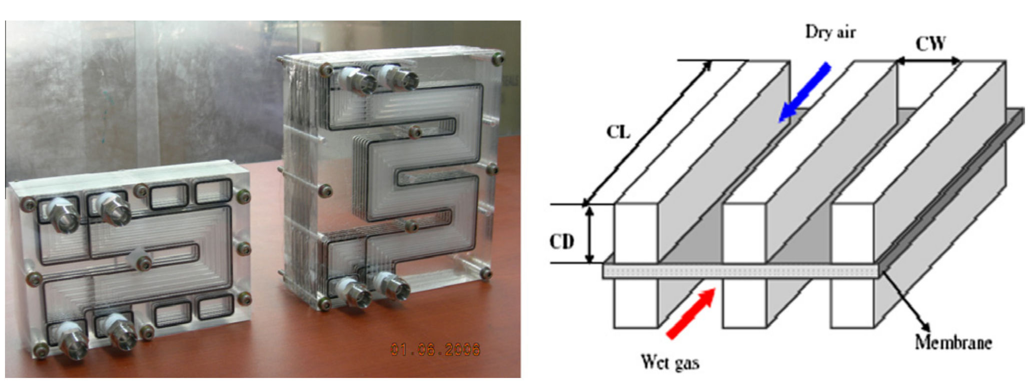


**Fig. 1.** Actual image (prototype I) and schematic of planar membrane humidifier prototype and the information of geometry/flow direction.