

## An analytic model of membrane humidifier for proton exchange membrane fuel cell

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### ABSTRACT

*An essential requirement for an operating PEM fuel cell is providing proper water content in the membrane. To avoid water flooding an appropriate water balance is required. Here, an analytic model of a planar membrane humidifier for PEM fuel cell is proposed where the effect of dimensional parameters includes membrane thickness, membrane area and channel hydraulic diameter are investigated. A Non-linear governing equations system is developed and solved. At each stage, the outlet temperatures, the water and heat transfer rates, relative humidity and the dew point at dry side outlet are presented and discussed. The humidifier is evaluated based on the decrease in difference between the dew point at wet side inlet and dry side outlet which leads to humidifier better performance. The results show that an increase in membrane thickness results in a decrease in dew point at dry side outlet which indicates a weak humidifier performance. Vaster membrane area can enhance humidifier performance. Here, big hydraulic diameters are not recommended.*

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### 1. Introduction

Water management is important for balancing PEM fuel cells operation. An essential requirement for operating PEM fuel cell especially at high conductivity is providing high water content in the membrane in order to ensure high ionic conductivity [1]. A low water content of the membrane could reduce the membrane durability and could leads to membrane electrode adhesion [2]; while high water content could lead to water accumulation in the cathode GDL. Too much water at the cathode can condense and close some of the GDL porositie sand thus prevents oxygen transfer to the cathode Catalyst layer. This phenomenon known as flooding leads to a lower chemical reaction rate in the cell ending with a voltage drop.

Flooding generally occurs at high current densities, especially at low flow rates and low temperatures [3]. Hence, for durability and efficiency of PEM fuel cell appropriate and optimum water content is essential [4]. This can be obtained by humidification of reactant gases before entering the cell and humidity control especially at anode side of the cell; though, air humidification is necessary to prevent the drying of the membrane at anode entrance as well. With no humidification of the reactant gases, a PEM fuel cell performs low about 20-40 percent than when operating on humidified state [5].

To date, a variety of methods have been developed with respect to humidification of reactant gases, mainly categorized into external and internal humidification. The internal humidification includes stack-integrated membrane humidifiers [6], steam or liquid water injection [7], use of membrane additives [8-10] and use of porous absorbent sponges to proper distribution of water [11, 12].

The external humidification includes bubble or dew point humidifiers [13], enthalpy wheel exchanger [14] and membrane humidification [15, 16].

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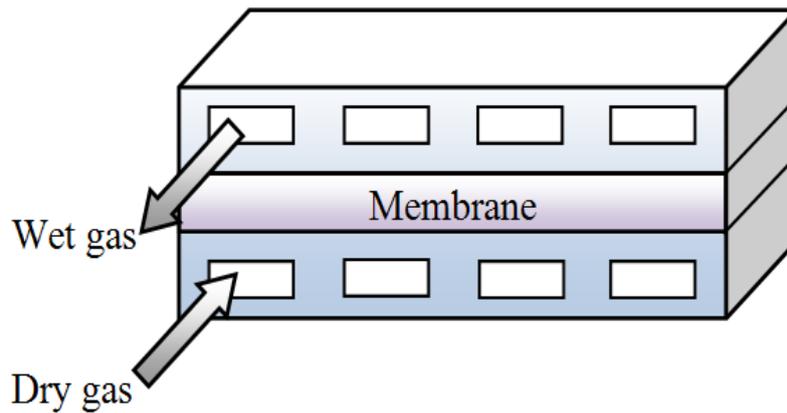


Fig. 1. A schematic of planar membrane humidifier

Among these methods, the membrane humidification is the simplest and most common method with the least energy consumption. It reduces complexity of fuel cell system and parasitic power. This method uses a semi-permeable membrane. In general, in terms of geometry, the membrane humidifiers could fall into two categories: planar and tubular. A schematic of these two kinds is illustrated in Fig. 1. In the membrane humidifier, wet gas (or liquid water) and fuel (or air) flows through the channels on either side of the membrane. Due to the difference in water concentration and temperature between the two sides, water and heat are transferred from wet side to the dry side through membrane with diffusion method which humidifies and heats the dry gas in the channel/membrane interface through vaporization. In humidifiers such as heat exchangers there are three main flow arrangements, counter-flow, parallel-flow, and cross-flow.

Where the internal and external bulk streams travel coaxially, in opposite directions. This type of setup is common in practice as it tends to provide excellent heat and water transfer characteristics relative to other arrangements. In most fuel cell systems, humidification system is based on cell exhaust gas recirculation. A schematic of the system is shown in Fig. 2. In this system, the dry side outlet of the humidifier is connected to the cathode side of the fuel cell stack; whereas, the outlet of the cathode is connected to the wet side inlet of the humidifier.

The gas at the outlet of the fuel cell stack has high

moisture content due to the production of water in the fuel cell stack. Thus, this stream is fed to the humidifier and is used to humidify the gases entering the cathode side. Cave and Merida [17] studied the effect of flow rates on the performance of a gas to gas membrane humidifier. Their experimental results showed that a reduction in flow rate at dry side results in an increase in the dew point at the dry side outlet. Huizing *et al.* [18] proposed a methodology for designing a planar membrane humidifier using a dimensionless parameter that communicates between the residence time and characteristic diffusion time in humidifier channels. Bhatia *et al.* [19] employed an analytical method to conduct sensitivity analysis. The results show that mass transfer affects the heat transfer extremely. An experimental study and thermodynamic transient modeling for a membrane humidifier for PEM fuel cell were conducted by Chen *et al.* [15, 20].

In their model each humidifier unit consists of three gas channels. The third channel here is to control heat transfer and can be adjusted by a sliding plate. Their objective was to identify the minimum number of units essential for a target fuel cell stack. Park and Oh [21] developed a 1-D analytical model of a water-to-gas membrane humidifier. The analytical expression was used to calculate the relative humidity along the gas flow channel as a function of gas flow rate, length and height.

In their model, the overall temperature of humidifier is assumed to be constant; whereas, the key role of temperature and heat in vapor transfer.

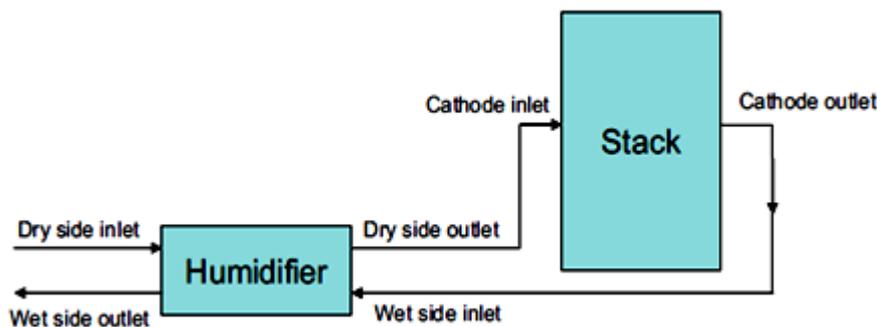


Fig. 2. A schematic of humidifier in an exhaust gas recirculation system [16]

This assumption especially in the calculation of output relative humidity creates a serious error. In their equation, the effect of wet side inlet flow rate and pressure are not presented. Sabharwal *et al.* [16] studied a 2-D modeling of a planar membrane humidifier, but their results do not contain the outlet temperatures and heat transfer rates. They did not investigate the effect of dimensional parameters. Yu *et al.* [22] proposed a static model for a planar membrane humidifier. They investigated the effect of the humidity of the wet gas, membrane thickness and channel length on humidifier performance. In their calculation the amount of liquid water generated during humidifier work is evident, while the relative humidity and temperature at outlet are not presented. In this study, an analytic model of a planar membrane humidifier is developed in order to investigate the effect of dimensional parameters. For this purpose, the thermodynamic equations and vapor mass transfer equation are solved. At each stage, the outlet temperature of dry side and wet side, water vapor transfer rate through membrane, the relative humidity and the dew point of the dry side outlet are presented and discussed.

**2. Mathematical modeling**

The model domain consists of wet side and dry side channels with the membrane in the middle (see Fig. 3). The air flow entering the dry side of the humidifier is moisture free. The wet side inlet contains a mixture of air and water vapor, which comes from the fuel cell cathode. Water and heat are transferred by the membrane from wet side to the dry side.

**2.1. Assumptions**

The following assumptions are made in implementing this model:

- i) steady-state flow,
- ii) ideal gas mixtures,
- iii) laminar and incompressible flow with low Reynolds numbers and pressure gradients,
- iv) omitted diffusion of air across the membrane,

- v) isotropic and homogeneous membrane characterized by effective porosities and permeabilities,
- vi) well insulated from its surroundings,
- vii) neglected the kinetic and potential energies of the gas molecules,
- viii) no external work,
- ix) constant specific heats,
- x) constant total convection heat transfer coefficient, and
- xi) the fuel cell cathode exhaust gas saturated at 80°C

**2.2. Modeling equations**

Humidifier is a thermodynamic system. Based on the first law of thermodynamics, as indicated in Fig. 3, the governing equation for control volumes 1 and 2 are given as

$$\dot{m}_{1,air,out}h_{1,air,out} + \dot{m}_{1,v,out}h_{1,v,out} = \dot{q} + \dot{m}_{1,air,in}h_{1,air,in} + \dot{m}_{1,v,in}h_{1,v,in} + \dot{m}_{v,mem}h_{mem}, \tag{1}$$

$$\dot{m}_{2,air,out}h_{2,air,out} + \dot{m}_{2,v,out}h_{2,v,out} = -\dot{q} + \dot{m}_{2,air,in}h_{2,air,in} + \dot{m}_{2,v,in}h_{2,v,in} - \dot{m}_{v,mem}h_{mem}. \tag{2}$$

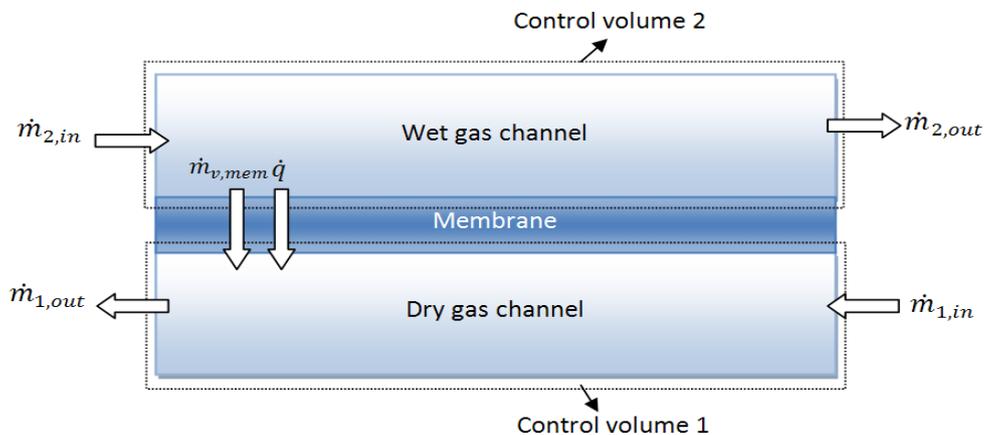
where  $\dot{m}_{1,air,out}$ ,  $\dot{m}_{1,v,out}$ ,  $\dot{m}_{1,air,in}$ ,  $\dot{m}_{1,v,in}$  are the air and vapor mass flow rate entering and leaving control volume 1 and  $\dot{m}_{2,air,out}$ ,  $\dot{m}_{2,v,out}$ ,  $\dot{m}_{2,air,in}$ ,  $\dot{m}_{2,v,in}$  are the air and vapor mass flow rate entering and leaving control volume 2.  $\dot{m}_{v,mem}$  is the vapor mass flow rate transferred from CV2 to CV1 through membrane.  $\dot{q}$  is the heat transfer rate from CV2 to CV1.

The mass conservation equations gives

$$\dot{m}_{2,v,in} - \dot{m}_{2,v,out} = \dot{m}_{v,mem} = \dot{m}_{1,v,out} - \dot{m}_{1,v,in}. \tag{3}$$

The parameter h shows the enthalpy and  $h_{mem}$  is the membrane enthalpy and it can be expressed as

$$h_{mem} = C_{p,v} \cdot T_{mem}. \tag{4}$$



**Fig. 3.** A schematic of a membrane humidifier performance

where  $C_{p,v}$  is vapor specific heat and  $T_{mem}$  is membrane temperature and approximated as

$$T_{mem} = \frac{T_{1,out} + T_{2,out}}{2} \quad (5)$$

The assumption here is that the amount of transferred water mass is assumed to be to the membrane vapor diffusion rate; thereby, the amount of transferred water mass can be determined as [20]

$$\dot{m}_{v,mem} = D_w \frac{C_2 - C_1}{t_m} M_v A \quad (6)$$

where  $M_v$  is the vapor molar mass,  $A$  is the membrane area and,  $C_1$  and  $C_2$  represent water concentrations in CV1 and CV2, respectively.  $D_w$  is determined by the following empirical equation:

$$D_w = D_\lambda e^{2416 \left( \frac{1}{303} - \frac{1}{T_{mem}} \right)} \quad (7)$$

where the coefficient  $D_\lambda$  is determined as using [23]

$$D_\lambda = \begin{cases} 10^{-6} \lambda_m < 2 \\ 10^{-6} (1 + 2(\lambda_m - 2)) & 2 \leq \lambda_m \leq 3 \\ 10^{-6} (3 - 1.67(\lambda_m - 3)) & 3 < \lambda_m < 4.5 \\ 1.25 \times 10^{-6} \lambda_m \geq 4 \end{cases} \quad (8)$$

and  $\lambda_m$  is the membrane water content, obtained using

$$\lambda_m = 0.043 + 17.81 a_m - 39.85 a_m^2 + 36.0 a_m^3 \quad (9)$$

Moreover,  $a_m$  is the membrane relative humidity (RH) and assumed to be given by

$$a_m = \frac{\Phi_1 + \Phi_2}{2} \quad (10)$$

where  $\Phi_1$  and  $\Phi_2$  are the RH of CV1 and CV2, respectively and are determined as

$$\Phi_1 = \Phi_{1,out} = \frac{P_{1,out} \omega_{1,out}}{P_{1,sat} (\omega_{1,out} + \beta)} \quad (11)$$

$$\Phi_2 = \Phi_{2,out} = \frac{P_{2,out} \omega_{2,out}}{P_{2,sat} (\omega_{2,out} + \beta)} \quad (12)$$

and

$$\beta = \frac{M_v}{M_{air}} \quad (13)$$

The water concentrations CV1 and CV2 are obtained using

$$C_1 = \frac{\rho_{m,dry}}{W_{m,dry}} \lambda_1 \quad (14)$$

$$C_2 = \frac{\rho_{m,dry}}{W_{m,dry}} \lambda_2 \quad (15)$$

where  $\rho_{m,dry}$  is the membrane dry density,  $W_{m,dry}$  is the membrane dry equivalent weight. The variable  $\lambda_1$  and

$\lambda_2$  represent water content of CV1 and CV2, respectively are given as

$$\lambda_1 = 0.043 + 17.81 \phi_1 - 39.85 \phi_1^2 + 36.0 \phi_1^3 \quad (16)$$

$$\lambda_2 = 0.043 + 17.81 \phi_2 - 39.85 \phi_2^2 + 36.0 \phi_2^3 \quad (17)$$

The heat transfer rate is determined using

$$\dot{q} = UA \Delta T \quad (18)$$

where  $\Delta T$  is the log mean temperature difference between the two control volumes in the cross flow humidifier and is given by

$$\Delta T = \frac{(T_{2,in} - T_{1,out}) - (T_{2,out} - T_{1,in})}{\ln((T_{2,in} - T_{1,out}) / (T_{2,out} - T_{1,in}))} \quad (19)$$

and  $U$  is the total heat transfer coefficient defined as

$$U = \frac{kNu}{2D_h} \quad (20)$$

Also,  $k$  is the thermal conductivity,  $Nu$  is the Nusselt number, and  $D_h$  is the channel hydraulic diameter.

### 2.3. Numerical procedures

A set of non-linear equations are solved by using repeat method in FORTRAN programming software in order to obtain the following three main unknown parameters, temperature at dry side and wet side outlet and water transfer rate. With these parameters and other input parameters, the relative humidity at dry side outlet, heat transfer rate and the dry side outlet dew point are calculated and reported. The equation set was solved for various geometric parameters.

## 3. Results and discussion

### 3.1. Model validation

The geometric and physical parameters of the humidifier system are listed in Table 1. The relative humidity is one of the most important outputs of the humidifier performance. For model validation, the variation of relative humidity at dry side outlet with the velocity dry side inlet in this study is compared with that of Park and Oh results [21] results where the same operating and input conditions are applied (see Fig. 4). The findings of this study are in good agreement with Park and Oh results with the exception that in their model, the effect of temperature variation is not considered.

A good performance of humidifier is justified when dry side outlet condition approaches the fuel cell condition.

The dry side outlet gas in addition to the high temperature should have a high relative humidity. In studies conducted by Hwang and Yu [24], Cave and Merida [17] and Yu *et al.* [22]; the dew point is used as a criterion for an appropriate performance of humidifier.

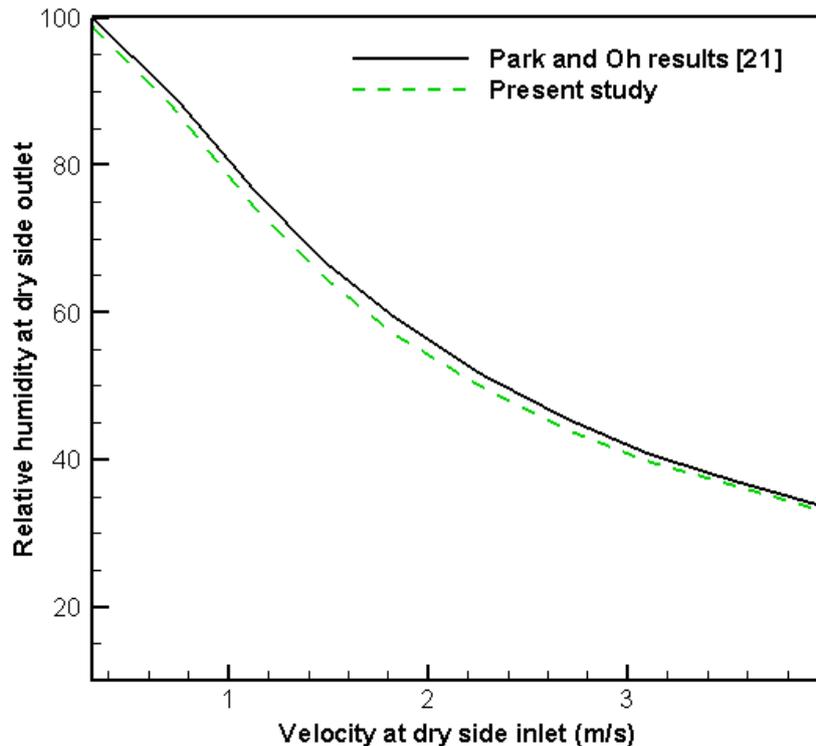
**Table 1.** Dimensional parameters and transport properties

Description	Unit	Value
<b>Dimensional parameters</b>		
Channel length	mm	30
Flow channel height	mm	1
Flow channel width	mm	1
Membrane thickness	μm	25
<b>Operating conditions</b>		
Dry side inlet temperature, T	K	305
Wet side inlet temperature, T	K	345
Pressure, P	kPa	101.5
Relative humidity at dry side inlet	%	0
Relative humidity at wet side inlet	%	100.
Mass flow rate at dry side and wet side	mg s <sup>-1</sup>	4.0
<b>Transport parameters</b>		
Air specific heat, C <sub>p</sub>	kJ kg <sup>-1</sup> K <sup>-1</sup>	1.007
Vapor specific heat, C <sub>v</sub>	kJ kg <sup>-1</sup> K <sup>-1</sup>	1.88
Air thermal conductivity, K	W m <sup>-1</sup> K <sup>-1</sup>	0.028
Nusselt number	-	2.9
<b>Material properties</b>		
Membrane dry density, ρ <sub>m,dry</sub>	g cm <sup>-3</sup>	1.0
Membrane dry equivalent weight, W <sub>m,dry</sub>	kg mol <sup>-1</sup>	1.0

The closer the dry side outlet dew point to the wet side inlet dew point, leads to the more ideal state of humidifier performance. This criterion is subjected to the effect of both the temperature and relative humidity. The dew point at dry side outlet is calculated as follow

$$p_{sat,(T=DewPoint)} = p_{sat,(T=T_{L,out})} \times \phi_{L,out} \quad (21)$$

In this study it is assumed that the humidifier is a part of an exhaust gas recirculation system (see Fig. 2) where the wet side inlet temperature is the cell performance temperature (353K). In this system the relative humidity of gas at wet side inlet is assumed as saturated, thus the wet side inlet dew point is the inlet temperature that is 353K. The closer the dew point at dry side outlet at 353K, leads to the better humidifier performance.

**Fig.4** Comparison presented study with analytic model presented by Park and Oh [21].

3.2. Effect of mass flow rate

In the gas to gas membrane humidifier systems, especially at high stoichiometric ratios, the exhaust gas recirculation system is often used for humidification. At these systems, the inlet mass flow rates at dry side and wet side are almost equal, while due to oxygen consumption in the cathode, the flow rate at wet side is a bit lower than that of the dry side.

Figures 5 and 6 show the variation of outlet temperatures, water vapor transfer rate and relative humidity at dry side outlet with mass flow rate. To study the effect of mass flow rate on the humidifier performance, the mass flow rate is varied between 0.5 and 8 mg.s<sup>-1</sup>. An increase in the inlet mass flow rate leads to a decrease in dry side outlet temperature and an increase in wet side outlet temperature.

This phenomenon introduces a significant increase in the heat flux which lies between 88 and 862w/m<sup>2</sup>. It is observed from Fig. 6, that the water transfer rate decreases with an increase in mass flow rate. A decrease in the temperature at dry side outlet results in a decrease in the saturation pressure and according to Eq. (11), a decreased saturation pressure can lead to an increased relative humidity.

A decrease in the water transfer rate leads to a decrease in the relative humidity. Here, it could be deduced that, these two opposing factors affect the relative humidity at dry side outlet which contribute to determine the dominant factor. In studying the mass flow rate, the effect of water transfer rate on relative humidity is the dominant factor. As shown in Fig.6 the relative humidity at dry side outlet decreases with an increase in mass flow rate. This decline occurs more in rates higher than 5 mg/s.

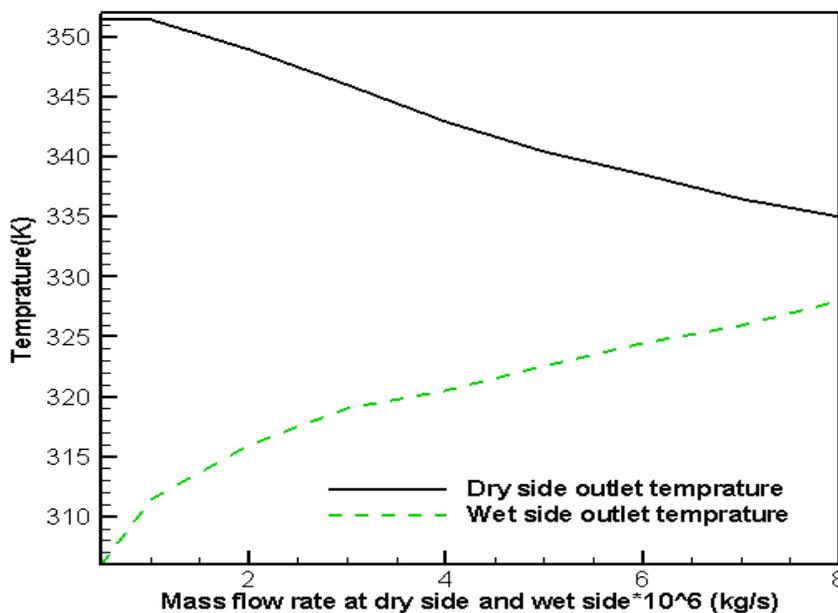


Fig. 5. Effect of mass flow rate on the temperature at dry and wet side

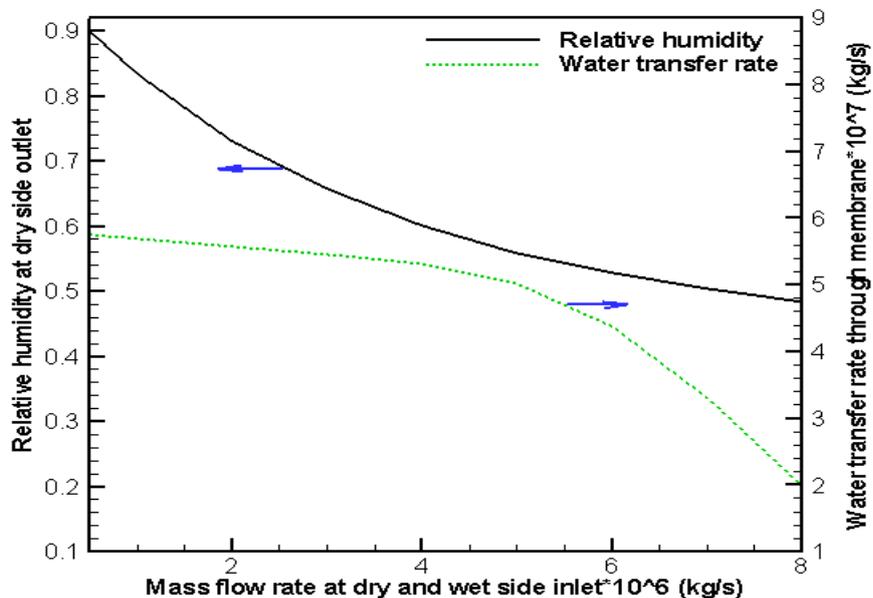


Fig. 6. Effect of mass flow rate on the water transfer rate and relative humidity at dry side outlet

Decreasing temperature and relative humidity at dry side outlet are two agreeing factors that cause a decrease in dew point. The dry side outlet dew point reduces from 348.95 to 319.9K with an increase in mass flow rates (see Fig. 7), indicating a decline in humidifier performance. This finding is in good agreement with the experimental trends observed by Hwang *et al.* [24].

### 3.3. Effect of membrane thickness

The membrane thickness effect is studied by varying it between 0.1 and 100  $\mu\text{m}$ . The variation of outlet temperatures with membrane thickness is shown in Fig. 8. It is observed here that the dry side outlet temperature is almost constant.

The variation of water transfer rate and relative humidity at dry side outlet in relation to membrane thickness is presented in Fig. 9. Water transfer rate decreases significantly with respect to an increase in membrane thickness. Due to consistent outlet temperature at dry side, a decrease in water transfer rate results in a decrease in relative humidity at dry side outlet. The variation of relative humidity decreases with the membrane thickness to more than 75  $\mu\text{m}$ . In this state heat flux is almost constant with a value of 567  $\text{W}/\text{m}^2$ . Due to the low gradient of the increase in the dry side outlet temperature, a decrease in relative humidity at dry side outlet results in a decrease in the dew point at dry side outlet. Applying thinner membrane is preferable, though the thinner membrane is accompanied by the leakage and repute risk.

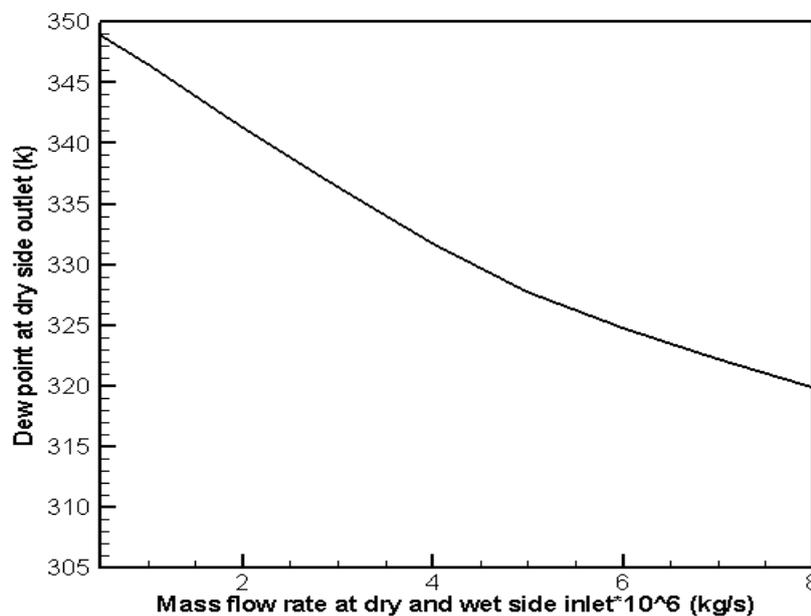


Fig. 7. Effect of mass flow rate on the dew point at dry side outlet

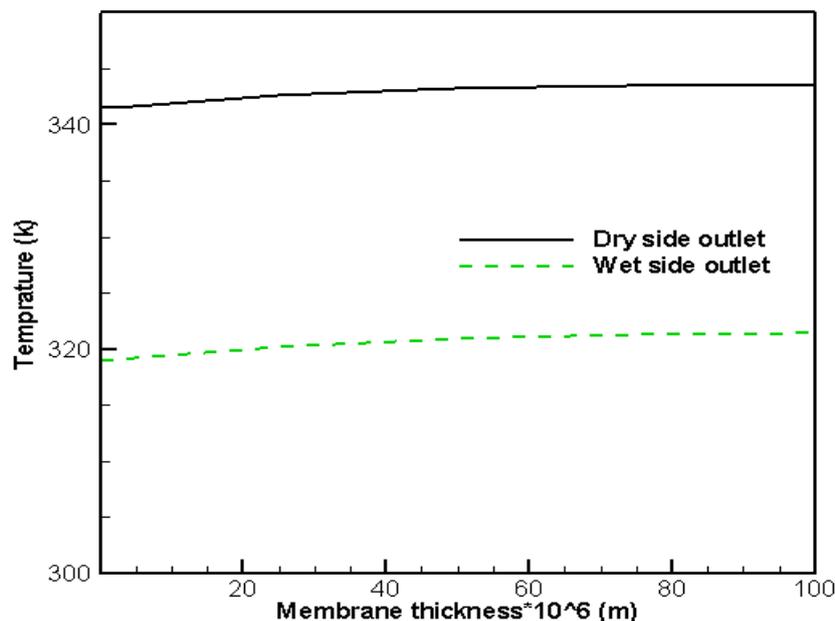


Fig. 8. Effect of membrane thickness on the temperature at dry and wet side

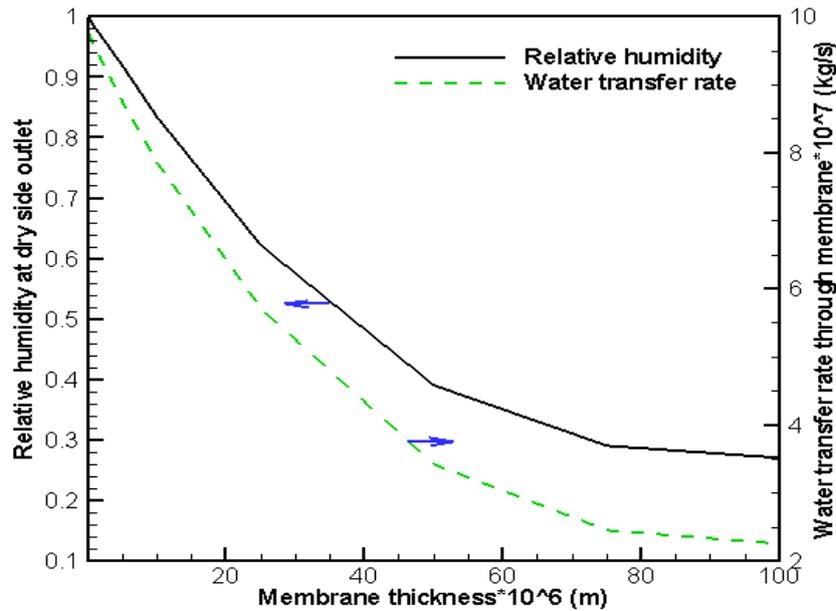


Fig. 9. Effect of membrane thickness on the water transfer rate and relative humidity at dry side outlet

3.4. Effect of Membrane area

Membrane area is one of the most important parameters that affect the humidifier performance. As indicated in Eqs. (6) and (18), this parameter directly affects the water and heat transfer rate. The membrane area can be changed by changing its length (length of the channel) and its width (depth of the channel). Park and Oh [21] have proposed a correlation that is achieved through the effect of channel length and some other parameters on the relative humidity. Their results indicate that increasing the length of the channel to a specified amount, the output relative humidity increases dramatically. In their model, the overall temperature of humidifier is assumed to be constant leading to an error.

Moreover, the main parameter of performance evaluation, i.e. dew point is not presented in their analytic model. The effect of the membrane area on the outlet temperatures, and the effect of the membrane area on water transfer rate and relative humidity at dry side outlet are presented in Figs. 10 and 11. Water transfer rate increases significantly with an increase in the membrane area, so that despite the increase in the temperature, the relative humidity increases. Increasing the temperature and relative humidity at dry side outlet are two agreeing factors respecting in increasing the dew point. It is observed in Fig. 12 that an increase in the membrane area up to 5 cm<sup>2</sup> leads to remarkable increase in dew point which is appropriate. In this state, heat flux increases from 479 to 737 W/m<sup>2</sup>.

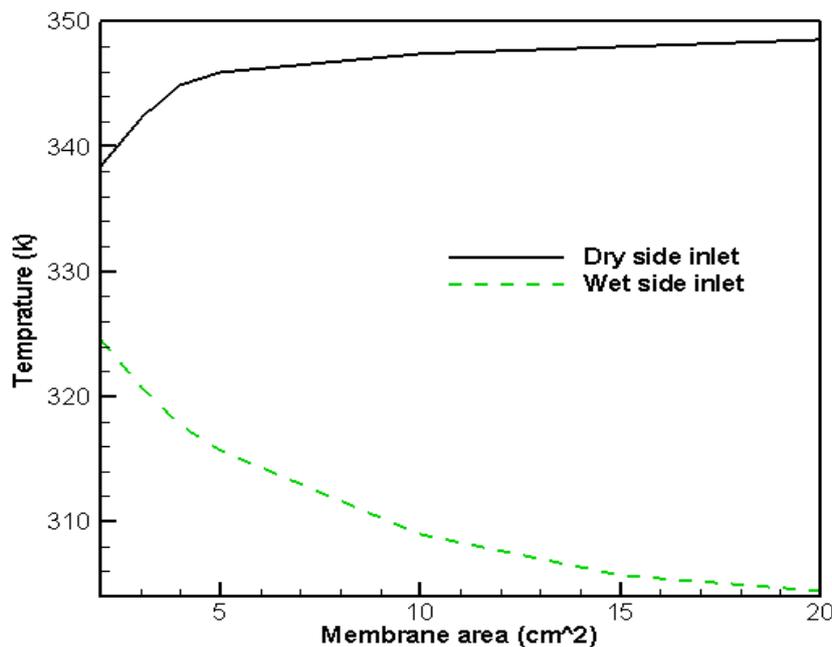


Fig. 10. Effect of membrane area on the temperature at dry and wet side outlet

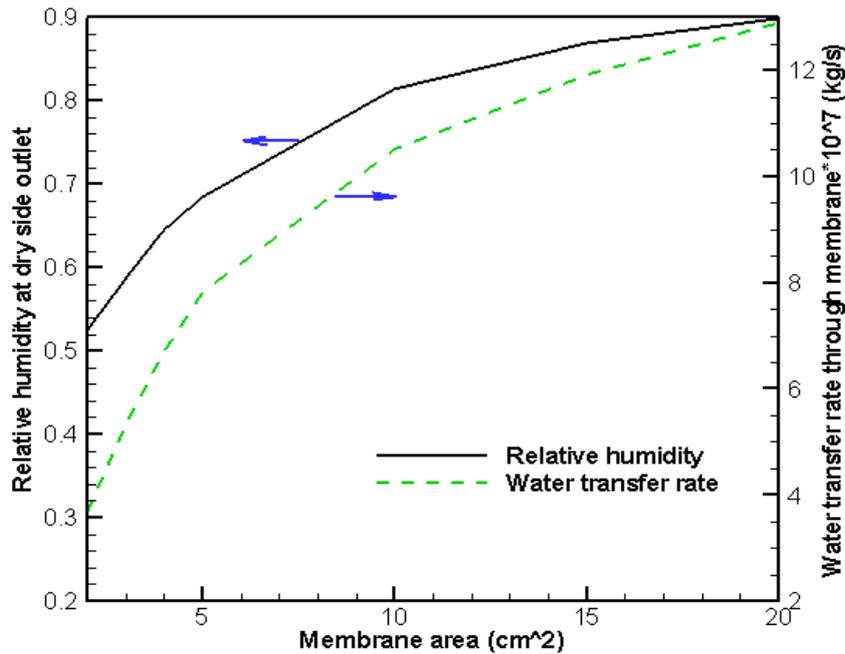


Fig. 11. Effect of membrane area on the water transfer rate and relative humidity at dry side outlet

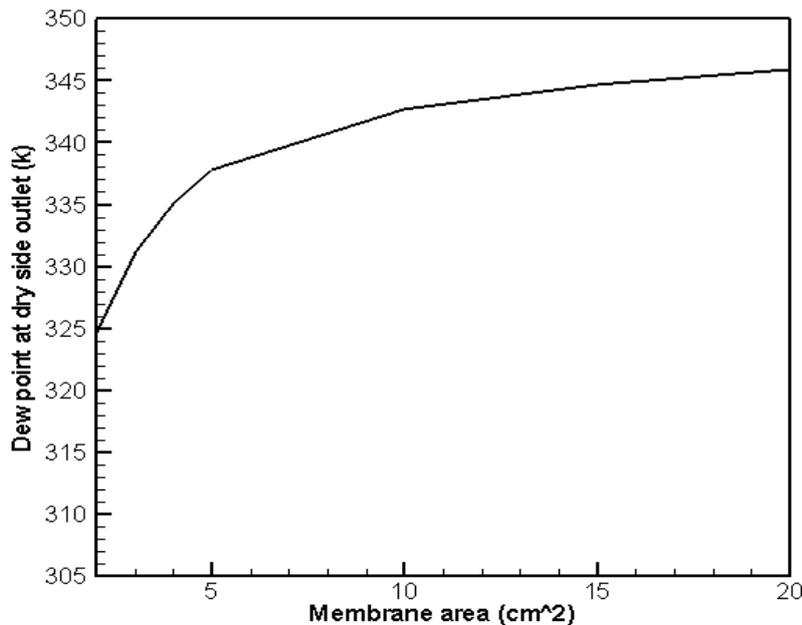


Fig. 12. Effect of membrane area on the dew point at dry side outlet

### 3.5. Effect of hydraulic diameter

The hydraulic diameter is varied between 0.5 and 4 mm, keeping the inlet mass flow rate constant with the same value for hydraulic diameter of both the channels. Hydraulic diameter can be changed in several ways. If the hydraulic diameter is changed through a change in height, it is mandatory to note that the heights less than 1 mm causes manufacturing difficulties and dramatically increase the production cost; while heights more than 4 mm might deform the thin membrane or break it due to the lack of support and pressure difference across the membrane. If the hydraulic diameter is changed through a change in the geometry of the cross section, the change of Nusselt

number also should be considered. The hydraulic diameter has an inverse relation with the heat transfer coefficient (see Eq. 20). As shown in Fig. 13, an increase in hydraulic diameter leads to an increase in the temperature at dry side outlet. It is observed from Fig. 14 that despite a decrease in water transfer rate, the relative humidity boosts.

According to Eq. 21 a decrease in temperature can leads to a decrease in dew point. The relative humidity can leads to an increase in dew point. The combined effect of these two opposing factors should be investigated.

The variation of dew point at dry side outlet with hydraulic diameter is shown in Fig. 15. The dew point decreases with an increase in hydraulic diameter. This

is due to dominating effect of a decrease in temperature on an increase in relative humidity. Hence, an increase in hydraulic diameter results in a

decline in humidifier performance. The heat flux decreases with an increase in hydraulic diameter varying between 617 and 300 W/m<sup>2</sup>.

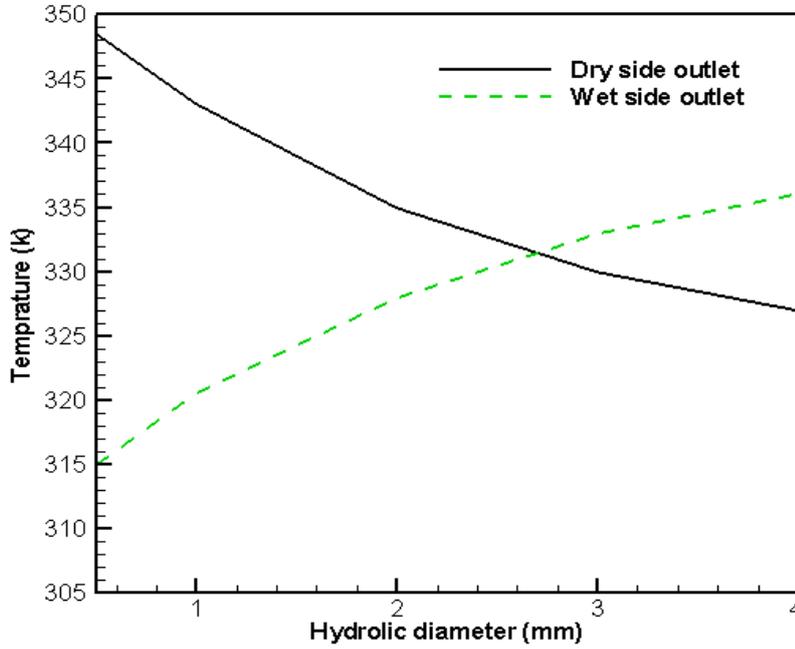


Fig. 13. Effect of hydraulic diameter on the temperature at dry and wet side outlet

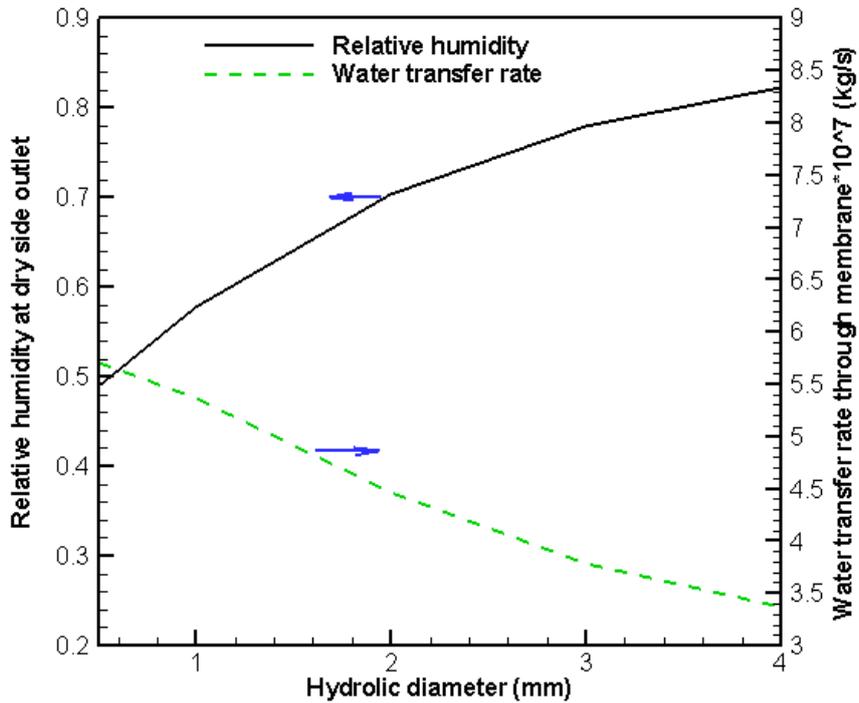


Fig. 14. Effect of hydraulic diameter on the water transfer rate and relative humidity at dry side outlet

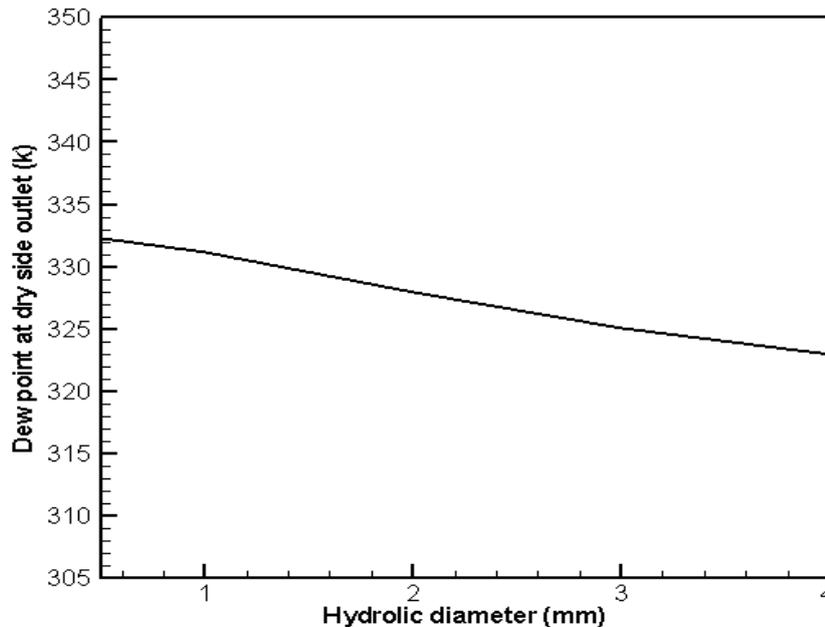


Fig. 15. Effect of hydraulic diameter on the dew point at dry side outlet

#### 4. Conclusion

A steady-state analytic model of a planar membrane humidifier with cross-flow configuration is presented where the effect of dimensional parameters are investigated. At studying each parameter, the outlet temperatures, the water and heat transfer rates, the relative humidity and the dew point at dry side outlet are discussed. The closer the dry side outlet dew point to the wet side inlet dew point leads to the more ideal state of humidifier performance.

The major findings of this study are concluded below:

1. In a membrane humidifier with the equal mass flow rates at the dry side and wet side channels, an increase in the flow rate leads to a decrease in dew point at dry side outlet exhibiting a decline in the humidifier performance.
2. A thinner membrane leads to a higher dew point at dry side outlet; consequently, humidifier illustrates better performance. It is important to note that too thin membrane causes leakage and rupture.
3. The dew point at dry side outlet increases with an increase in the membrane area up to 5 cm<sup>2</sup> that enhances the humidifier performance. An increase in the dew point with an increase the area more than 5 cm<sup>2</sup> is not of significance; therefore, probably using areas more than 5 cm<sup>2</sup> is not economical.
4. Higher values of hydraulic diameter result in a decrease in dew point, despite the increase in relative humidity. Thus using higher values of hydraulic diameter is not recommended.

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