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Design methodology for membrane-based plate-and-frame fuel cell humidifiers

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Abstract

Currently, polymer electrolyte membrane fuel cells require some method of humidification to operate effectively. External gas-to-gas membrane-based humidifiers can provide an efficient method to recycle exhaust heat and product water from the fuel cell stack. This work describes a design methodology involving a series of design equations for plate-and-frame membrane humidifiers. Humidifiers of different flow channel geometries were created with a rapid prototyping technique. These humidifier units were tested at different operating conditions in an attempt to validate the design equations. The ratio between the residence time of gas in the humidifier over the diffusion time of water from the surface of the membrane into the channel can be used as a design parameter. This ratio was shown to offer a good starting point for humidifier design, and a target range between 2.0 and 4.0 was identified (with a nominal desired value of 3.0). A humidifier design procedure and suggestions are presented based on this parameter and the packaging requirements of the humidifier in a fuel cell system. This algorithm was validated by creating a further prototype humidifier.

Keywords: Membrane humidifier; Gas-to-gas humidification; Humidifier design

1. Introduction

The polymer electrolyte membrane fuel cell (PEMFC) shows promise as an energy conversion device for mobile and stationary applications. PEMFCs can be used to extract the chemical energy in hydrogen and convert it into electrical energy to do work. However, many technical barriers must still be overcome before these devices can reach commercialization. One of these barriers is the lack of optimal water management subsystems in fuel cell power plants.

A key component of PEMFC systems is the humidifier, which adds sufficient water to the fuel cell reactant gases before they enter the fuel cell. This helps ensure that the fuel cell will operate effectively. In this work, a design methodology for a plate-and-frame humidifier involving a series of design equations was developed, tested and presented. Design methodologies for components such as heat exchangers are common, but as fuel cells are a newer technology, design methodologies for these sub-systems are limited. The development of the effectiveness parameter (R) via a series of experiments was critical to this design. Humidifier plates were created with the assistance of a rapid prototyping technique and assembled into humidifier stacks in order to validate the design.

1.1. Importance of fuel cell humidification

The presence of water within the polymer ionic conductor is indispensable for PEMFC operation: A dry membrane loses its ionic conductivity, thereby reducing its performance and durability. However, excess water present in other regions of the cell (e.g. the electrodes, gas diffusion layer, or the flow channels) can have a negative impact on cell performance as it may hinder

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Nomenclature

d	depth of flow channel (mm)
D	external humidifier depth (m)
$D_{\mathrm{WA}}(\mathrm{T})$	diffusivity of water in air, function of temperature ($\mathrm{m^2\,s^{-1}}$)
E	required space for sealing surfaces on edges of humidifier stack (mm)
F	Faraday's constant (96,485 $\mathrm{C\,mol^{-1}}$)
H	required space for the end plates of the humidifier stack (mm)
l	length of flow channel (mm)
L	external humidifier length (m)
m	number of plates in the humidifier
M	required space for manifolds on top and bottom of humidifier stack (mm)
MW_n	molecular weight ($\mathrm{kg\,mol^{-1}}$)
n	number of channels per plate
P	pressure (Pa)
P_{e}	fuel cell rated power (W)
$P_{\mathrm{H_2O}}$	saturated water vapour pressure (Pa)
Q	volumetric flow rate of gas ($\mathrm{m^3\,s^{-1}}$)
r	space rib thickness (mm)
R	dimensionless humidifier design factor, R-value
Re	Reynolds number
t	time (s)
T	temperature of air stream (K)
v	average gas velocity in channel ($\mathrm{m\,s^{-1}}$)
V_{c}	fuel cell rated individual cell voltage (V)
w	width of flow channel (mm)
W	external humidifier width (m)
WRR	water recover ratio (%)

Greek letters

δ	membrane thickness (mm)
λ	air stoichiometry for fuel cell reactant
τ	residence time (s)
τ_{D}	diffusion time (s)
ω	humidity ratio (kg water (kg dry air)$^{-1}$)

Subscripts

1	dry side inlet stream
2	dry side outlet stream
3	wet side inlet stream
4	wet side outlet stream
$\mathrm{H_2O}$	water
Air	air
min	minimum value for given packaging constraints
max	maximum value for given packaging constraints

mass transfer of reactants. Therefore, PEMFC operation requires a careful balance between the hydration levels in the membrane, and the presence and removal of excess water.

Water transport in a PEMFC can be described in terms of four major processes:

(i) Electro-osmotic drag from the anode to the cathode;
(ii) Back diffusion from the cathode to the anode;
(iii) Water production by the fuel cell reaction at the cathode; and
(iv) Drying and humidifying effects of the flowing reactants.

Exhaustive studies and reviews of microscopic and macroscopic water transport within PEMFCs are available [1]. Similar analysis of each mechanism is out of the scope of this work, but results indicate that drying conditions will prevail for nearly all desirable operating conditions above 60 °C [2]. One of the main conclusions from water balance studies is that optimal fuel cell performance requires some method of humidification. Accordingly, current fuel cell stack designs incorporate external or internal humidification schemes.

1.2. Methods of humidification

Fuel cells may be humidified by internal and external methods. Internal humidification refers to the addition of water directly into the fuel cell, or a method of keeping the product water of the fuel cell reaction within the fuel cell membrane. External humidification involves the use of a humidification unit to provide the fuel cell with humidified reactant gases, and it usually involves recycling the heat and humidity in the exhaust streams.

1.2.1. Internal humidification

One method of internal humidification relies on special fuel cell stack designs or flow field pathways [3,4]. These designs distribute the water within the fuel cell, but require greater stack complexity, potentially specialized porous flow field plate materials, and are often less effective than external humidification. Other internal humidification methods involve producing membranes with metal oxide additives to retain water produced in the fuel cell reaction within the membrane. Platinum additives can also be added to catalyze the reaction between permeating H_2 and O_2 gases, and generate water within the membrane itself [5–7]. Fuel cells with these types of membranes have demonstrated increased performance over other non-humidified fuel cells, but their performance can be still improved with additional humidification. Also, a careful balance and distribution of metal oxides and platinum is required to ensure that electrical short-circuiting does not occur. In addition, the effects of these additives on membrane lifetime and durability are not clear. Despite these limitations, these specialized membranes have been reported to assist fuel cells in cold-start conditions.

Wicks and sponges have also been integrated into fuel cells in order to provide hydration on demand and distribute water evenly over the electrochemically active area [8,9]. These methods humidify the fuel cell membrane, but they involve more advanced fuel cell design, may be more prone to gas leaks, and may not provide sufficient humidification in certain operating regimes.

1.2.2. External humidification

The bubbler is the most common method of external humidification. In a bubbler humidifier, the reactant stream is passed through a sparger into a heated column of water. The gas is dispersed into a large number of small bubbles that ensure a large contact area for humidification. The amount of water transferred to the gas stream is a function of water temperature, the contact area of the water–air interface, and the residence time of the air bubbles in the column [10]. This type of humidifier coupled with subsequent heating provides good humidity control. However, this method is impractical beyond the stationary laboratory scale due to the significant parasitic losses associated with heating the columns, carrying the water, and the required scaling of the column length for larger systems.

Direct water or steam injection can also provide effective and precise humidity control [11]. However, this method involves a more complex system which may cause parasitic power losses due to the pumping and heating of the injected water or steam.

Water and heat are produced in the fuel cell reaction, and if the reactant streams have been heated and humidified prior to entering the fuel cell, there will be sufficient excess water and heat in the fuel cell exhaust streams to adequately humidify the inlet gases [2]. Hence, the heat and moisture in the exhaust gases can be recycled to increase overall system efficiency. With this approach, gas-to-gas humidifiers can be implemented as coupled heat and moisture exchangers. There are two basic types of these exchangers, the enthalpy wheel and the membrane-based humidifier. These systems provide limited control over the humidification and heating of the inlet streams, but they can be scaled to suit the requirements of applications with varying power outputs. They usually take the form of open structures that cause small or negligible parasitic power losses.

Another type of external humidifier is the enthalpy wheel. Humidifiers of this type are produced by the company Emprise Corporation [12]. The humidifier functions by utilizing a hygroscopic core material which slowly rotates from the hot and wet fuel cell exhaust stream to the dry the inlet streams. This rotating material carries hot moisture from the fuel cell exhaust stream to the inlet stream, heating the inlet stream prior to entering the fuel cell. The core rotation causes some parasitic power losses in the system.

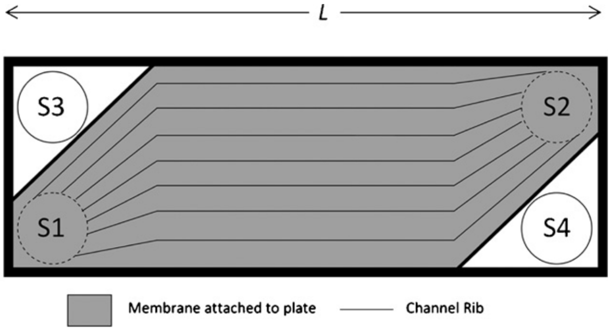

Fig. 1. Sample drawing of a planar membrane humidifier plate.

Membrane humidifiers fall into two basic categories: planar and tubular. These devices are similar to plate-and-frame and shell-and-tube heat exchangers, respectively. The membranes in these humidifiers are usually hydrophilic polymers that act as the media for transporting moisture and heat from the exhaust streams to the inlet reactants. This paper will focus on the design and performance characterisation of planar gas-to-gas membrane humidifiers.

1.3. Planar gas to gas humidification

Although membrane humidifiers have been developed, conventional designs are modeled after other well-known devices such as plate-and-frame heat exchangers. In the simplest implementation, a series of rigid plates similar to those shown in Fig. 1 are separated by a membrane and aligned one on top of another to form a stack. The membrane is not permeable to gases, but it allows heat and water transport via micro- and macro-pores, hydrophilic additives, or by virtue of hydrophilic properties of the membrane material. This topology defines flow channels with two membrane sides (wet and dry) and two sides delimited by the walls of the separating channel ribs as shown in Fig. 2. The channel cross-sections (e.g. trapezoidal or square) are usually constant. The humidifier is comprised of plates similar to the one shown in Fig. 1 which have flow inlet and outlet ports, and flow channels placed on top of the membranes. Many of these plates are combined together to produce a humidifier stack which has two set of inlets and outlets, one set for the

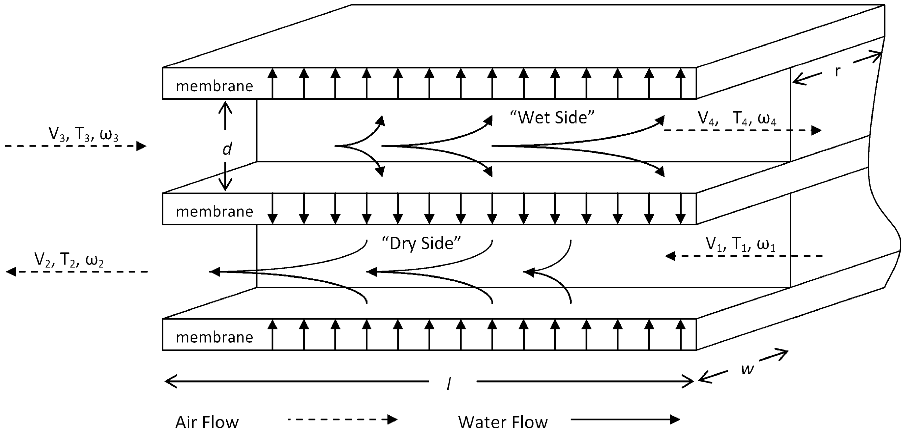

Fig. 2. Schematic of transport in the channel of the heat and moisture exchanger.

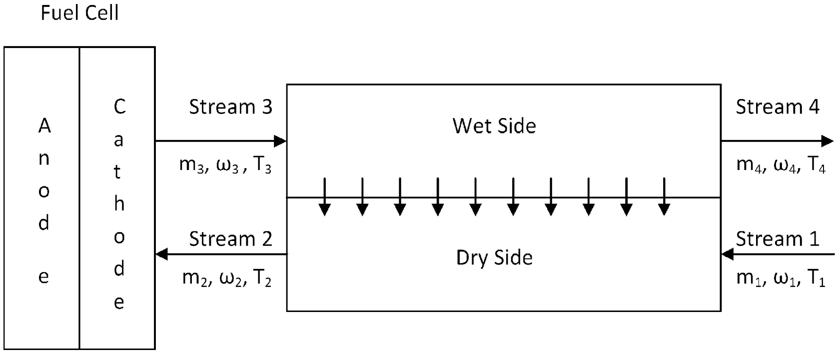

Fig. 3. General schematic for operation of humidifier.

cool, dry stream, which is humidified in the unit before entering the fuel cell, and one set for the hot, wet stream which is taken from a stream exiting the fuel cell stack. The overall flows to the humidifier are shown in Fig. 3. The hot, wet stream entering the humidifier from the outlet of the cathode of the fuel cell will not have exactly the same volumetric flow rate as that which enters the humidifier on the cool dry side, since oxygen is consumed and water is generated within the fuel cell. In the humidifier, hot and humidified excess gases exiting the fuel cell flow along one side of the membrane while cooler, dry reactants flow into the fuel cell along the other side of the membrane. The vapour pressure and temperature differences across the membrane act as the driving forces for water and heat flux. Some related patents describe this type of humidifier [13–15], and Cave and Merida [16] provides some experimental results for a single channel test rig.

2. Background for design process

In contrast to flows in circular tubes and parallel plates, the analysis of fluid flow in rectangular, square, and trapezoidal ducts requires two-dimensional analysis. There are no generalised solutions for the velocity, temperature, and concentration boundary layers along the channels. Moreover, if heat and mass transfer are included in the problem formulations, the approximations to these profiles require sophisticated numerical methods that are computationally demanding. Often flows in the humidifier unit will contain saturated reactant streams as well as condensed or condensing water, this two-phase flow further complicates computations. As a result, the optimisation of new humidifier topologies has relied on a balance between empirical design experiments, and simplified, predictive models that incorporate these results.

The design of planar gas-to-gas humidifiers involves optimizing the channel geometry and selecting the appropriate membrane materials. These factors must maximize water and heat transfer while simultaneously complying with practical design constraints such as minimal pressure drop, minimal volume, and low cost. The performance of the humidifier can be described in terms of the water transported from the wet stream to the dry stream, or similarly the output dew point temperature of the dry stream. Desirable performance occurs when the water transport rate is maximized or, conversely, when the dew point of the exiting dry stream approaches that of the incoming wet stream. In this work the performance was described by a water recovery ratio expressed as a percentage:

$$\mathrm{WRR} = \frac{\omega_2 - \omega_1}{\omega_3} \times 100 \tag{1}$$

In which ω is the humidity ratio in the gas:

$$\omega = \frac{\mathrm{MW_{H_2O}}}{\mathrm{MW_{air}}} \times \frac{P_{\mathrm{H_2O}}}{P - P_{\mathrm{H_2O}}} \tag{2}$$

These parameters are used to provide a simple comparison metric of the performance of various humidifiers.

The general schematic for transport in the exchanger can be found in Fig. 3. Where ω_2 refers to the humidity ratio at the dry stream outlet (stream 2), ω_1, the humidity ratio at the dry stream inlet (stream 1), and ω_3, the humidity ratio at the wet stream inlet (stream 3). The water recovery ratio compares the amount of water that has been transferred to the dry stream in the humidifier to the amount of water that was provided to the humidifier in the wet stream. The gas entering at stream 1 is generally ambient and has little water content, and in the experiments completed for this work the air supply was pre-dried, so ω_1 was zero. In this work, wet gas entering at stream 3 was saturated with water to 55, 65, or 75 °C. Consequently, the range of experimental values for ω_3 was between 0.11 and 0.39. In fuel cell systems, stream 3 will be exiting the fuel cell as exhaust. This gas stream will likely be saturated with water at the fuel cell operating temperature and will often contain some condensed water droplets. As the dry, low temperature air in stream 1 passes through the humidifier to point 2, it will gain moisture content and heat; ω_2 will indicate how much moisture the humidifier has transferred.

It is desirable to have a design equation by which proper humidifier design can be ensured. Full humidifier membrane hydration (at steady state) is assumed since the fuel cell exhaust generally contains some liquid water and further condensation will occur in the humidifier, also the membrane used is hydrophilic. Laminar flow is assumed since the Reynold's number (*Re*) for all experimental flows presented here was smaller than 500. Using these assumptions, the performance of a heat and moisture exchanger can be described by defining a ratio of residence to diffusion time within the humidifier channels [13]. As described in US Patent 6,416,895, the optimal performance regime of the exchanger can be achieved by ensuring that the residence time of a parcel of gas passing through the humidifier channels is greater than the diffusion time of water from the membrane surface into the channels. The patent considers the residence time of water molecules in the flow channel, and the time required for the molecules to diffuse (through air) over the depth of the channel. A channel is defined by dimensions w, l, and d as shown in Fig. 2. The residence time in the channel can be calculated from knowledge of this geometry and the flow rate of gas to the humidifier.

$$\tau = \frac{mnLwd}{2Q} \tag{3}$$

In Eq. (3), n is the number of channels per plate and Q is the volumetric flow rate to the humidifier. The number of plates in the humidifier, m, is used since the plates must be stacked to create the humidifier, as the plates are stacked, one plate will create a set of 'dry side' channels and the next will create a set of 'wet side' channels. For a humidifier of eight plates, there will be four sets of 'dry channels' and four sets of 'wet channels'. The full flow, Q will pass through only half of the total number of plates per pass, so the equation must be divided by two. The diffusion time for a hypothetical water molecule in the chamber has been calculated in the aforementioned patents as a ratio involving the channel depth and D_{WA}, the diffusion coefficient of water in air at the average temperature across the humidifier channel [13,14].

$$\tau_D = \frac{d^2}{D_{\mathrm{WA}}} \tag{4}$$

These authors have described empirical correlations between the maximum water flux across and the humidifier geometry as a dimensionless parameter, R, defined as the ratio of residence time to channel diffusion time:

$$R = \frac{\tau}{\tau_D} = \frac{mnlwD_{\mathrm{WA}}}{2Qd} \tag{5}$$

According to the previous studies, the best humidifier performance was obtained for R-values between 0.75 and 3. Despite the serious limitations in this simplified model, this parameter (R) can be an effective indicator of overall humidifier efficiency. In this study, the usefulness of this parameter was examined for humidifier plates of different channel depths, and at varying operating temperatures and flow rates. Recommendations for a humidifier design procedure will then be presented utilizing the R-value.

3. Experimental

3.1. Prototype humidifiers

A rapid prototyping machine was used to create three sets of eight humidifier plates with varying channel geometries. These plates are similar to the sample drawing of a plate shown in Fig. 1. Half of the eight plates are mirrored, so when stacked, four plates will comprise the 'wet side' of the humidifier, and four plates will comprise the 'dry side' of the humidifier. The parameter changed between each of these sets of plates was the channel depth, while length and width were held constant. Three humidifiers were assembled from these plate sets, the plate geometries are listed as humidifiers A–C in Table 1.

The humidifier plate sets were created using a Stratasys fused deposition modelling (FDM) Titan rapid prototyping machine. The humidifier plates were designed in using Solidworks 2005 computer aided design (CAD) software. The CAD files were then pre-processed using the Insight software provided by Stratasys. This ensured that the extrusion paths for rapid prototyping were optimized. The Insight files were the loaded to the Stratasys FDM Titan rapid prototyping machine for prototyping. Each plate set was created with acrylonitrile butadiene styrene (ABS) plastic. FDM created the humidifier plate sets by depositing ABS in layers following the extrusion path created using the Insight software.

For further comparison, two commercial gas-to-gas membrane humidification units from DPoint Technologies were obtained and tested. As well, a single membrane testing module was used for comparison. The dimensions of these units are listed as humidifiers D–F in Table 1.

The membrane utilized in all humidifiers consisted of precipitated colloidal silica bound by ultra high molecular weight polyethylene. This membrane had a thickness of 180 μm. The same membrane was used in all humidifier units. The plate material for the three rapid prototype plate sets was ABS. The plate material for the single membrane testing module was machined polymethyl methacrylate. The plate material for all other humidifiers was glass filled polypropylene. Much of the heat transported in the exchanger is due to the latent heat carried in the water vapour, so the plates contribute little to the overall heat transport in the unit. Additionally, the thermal conductivity values for these plastics are relatively similar so the effect of the plate material on heat transfer should not be significant when comparing the effectiveness of these humidifiers.

Table 1
Geometries for various humidifiers tested

Humidifier	Number of plates, m	Number of wet and dry sides	Number of channels per plate, n	Channel width, w (mm)	Channel depth, d (mm)	Channel length, l (mm)
A	8	4	20	3.0	1.0	255
B	8	4	20	3.0	1.4	255
C	8	4	20	3.0	2.0	255
D	2	1	7	3.0	1.0	160
E	50	25	20	3.0	1.6	255
F	50	25	7	3.0	1.1	125

3.2. Testing procedures

The humidifier stacks were assembled, and sealed to ensure no external or crossover leaks were present. A test stand with the appropriate control of air stream temperature, humidity, and flow was utilized to test the performance of the humidifier stacks. Air was supplied on one side of the plates completely dry and at room temperature, at a given set of flow rates, at position 1 in Fig. 3. Simultaneously, air that had been heated and saturated to 100% relative humidity at 55, 65, and 75 °C was supplied on the side of the plates at the same set of flow rates at position 3 in Fig. 3. At position 2 in Fig. 3, the dry bulb and wet bulb temperatures were measured using thermocouples. From these values, the mass balance over the humidifier was used to determine the overall water and heat transport performance of the humidifiers.

The test station provided air to the humidifier at position 1 pre-dried at a controlled flow rate, and ambient temperature. Air supplied at position 3 was passed through a sparger to heat and saturate the gas stream to the desired temperature. A heated line from the sparger to the prototype humidifier maintained the gas stream at the desired temperature. The test station employed data acquisition software which tracked the flow rates and temperatures of the gas streams entering and exiting the prototype humidifiers. Humidifiers were run at given flows and temperatures until steady state operation was achieved, at this point data was recorded in 15 s intervals for 10 min at steady state. Temperature data under steady state operation was averaged over the operating time for data analysis.

4. Results and discussion

4.1. Residence time

From Eq. (3) it can be seen that decreasing the gas flow to the humidifier will increase the residence time of the gas in the humidifier, this decreases the velocity of gas passing over the membrane in the channels. Increasing the residence time of gas in the humidifier will lead to greater time for moisture and heat to transfer to the dry stream. This increased transport time will lead to increased performance as indicated by the water recovery ratio experimentally. The effect of residence time on the water recovery ratio is shown in Fig. 4. It can be seen that increasing the residence time by decreasing the supply gas flow has a great effect on the water recovery ratio at low residence times. However, as the flow is further decreased, the positive effect of increasing residence time diminishes and little increase in water recovery is gained by greater increases in residence time. In designing the humidifier it will be desirable to have a higher residence time, so the velocity in the humidifier channels should be low, and the channel length should be long. However increasing the residence time in the humidifier demonstrates diminishing returns past a certain point. This means that it is important to design the humidifier so that the channels are long enough that a sufficiently large residence time will be achieved at the rated flow, but not so long that optimal overall design, size and material usage is compromised.

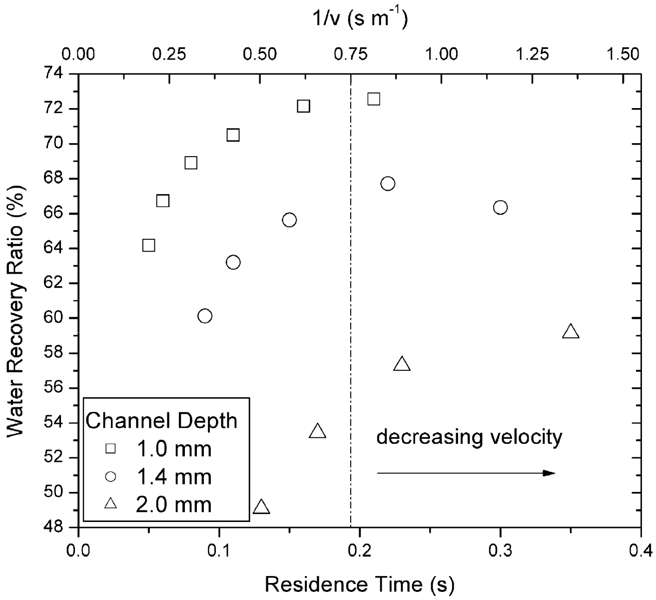

Fig. 4. Humidifier performance as a function of residence time, values for three plate sets with different channel depths (1.0, 1.4 and 2.0 mm), at stream 3 temperature and dewpoint of 75 °C, flow rates from 10 to 60 L min^{-1}.

4.2. Diffusion time

From Eq. (4) it is predicted that increasing the channel depth in the humidifier will increase the diffusion time of water from the membrane surface into the center of the humidifier channel. Three different humidifiers were made with plates having channel depths, *d* of 1.0, 1.4, and 2.0 mm to determine if the diffusion time does in fact affect humidifier performance. Increases in diffusion time will lead to a decrease in humidifier performance; this is shown in Fig. 5. The velocities and residence times were calculated for each of the three sets of plates at different flow rates. If the channel depth had no effect on the humidifier performance then it would be expected that the three humidifiers

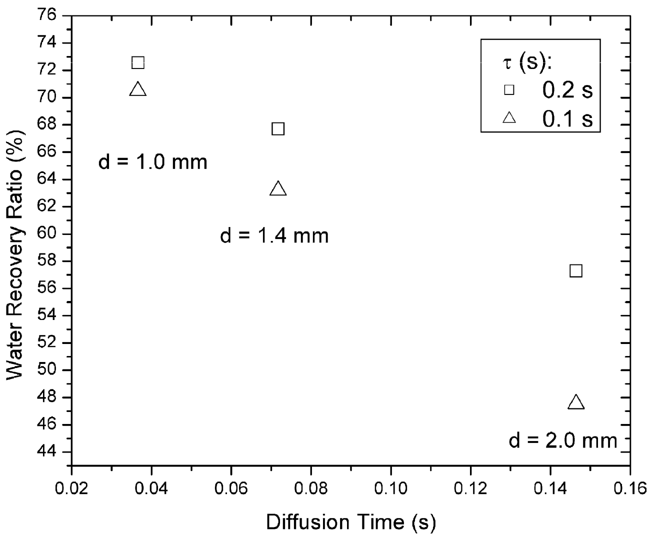

Fig. 5. Humidifier performance as a function of diffusion time, values for three plate sets of differing channel depths (1.0, 1.4 and 2.0 mm) at residence times of 0.1 and 0.2 s, at stream 3 temperature and dewpoint of 75 °C.

of different channel depths would perform similarly at the same residence times. It can be seen in Fig. 5 that this is not the case. At the same residence times, the humidifiers made with plates with smaller channel depths performed much better than those with larger channel depths. These results are similar to those reported for heat exchange in fully developed laminar flow under forced convection in rectangular ducts. Such as the case for a heat exchanger in which the channel side walls are considered adiabatic while heat is transferred from the top and bottom channel walls. This case presented by Shah and London found that decreasing the aspect ratio of duct width over duct height led to a decreased Nusselt number, this would indicate decreased heat transfer for deeper channels [17]. Results show the diffusion time value proposed in Eq. (4) may be a good metric for performance in a planar gas-to-gas humidifier. Increasing the diffusion time by increasing the channel depth leads to a decrease in overall humidifier performance. Accordingly, the channel depth in the humidifier design should be minimized. However, a smaller channel depth will lead to increased pressure drop across the humidifier unit; this must be considered in an appropriate design.

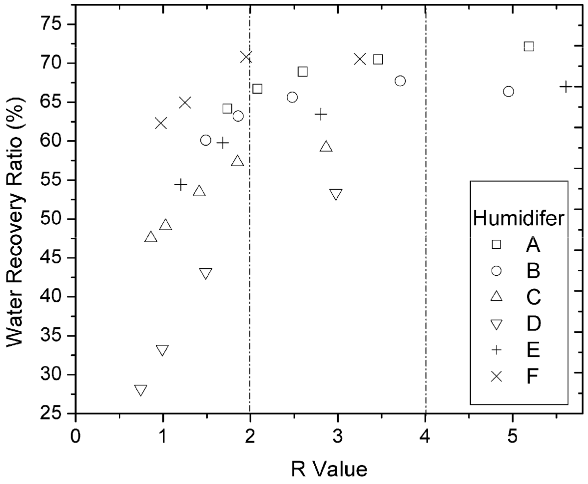

Fig. 6. Stream 2, dry outlet humidity ratios for experiments at different R-values with three plate sets (1.0, 1.4 and 2.0 mm) at three wet inlet stream 3 dew point temperatures (55, 65 and 75 °C) and various flow rates (10–60 L min^{-1}).

4.3. The R-value parameter

The residence time and the diffusion time have differing effects on humidifier performance. Increased residence times will have a positive effect on the humidifier performance within a limited operation range. Whereas increased diffusion times in the humidifier channel will have a negative effect on the humidifier performance. Individually, the residence time and diffusion time do not offer a single parameter by which to quantify humidifier performance. The R-value parameter presented in Eq. (5) attempts to combine both of these effects into a dimensionless number that can be used for humidifier design. The water recovery ratio for each experiment was plotted against the calculated dimensionless R number in Fig. 6. It can be seen that the performance of the humidifier indicated by ω_2 drops rapidly for R-values below 2.0 for any design or operating conditions. However, humidifier performance tends not to increase significantly at R-values greater than 4.0. It can be noted from Fig. 6 that increasing the wet inlet dew point temperature at stream 3 from 55 to 75 °C increases the water content in the dry outlet, stream 2. This is due to the increased heat and moisture provided to the humidifier by increasing the stream 3 temperature and dewpoint, which increases the driving force for water transport across the membrane. It can be seen that a well-designed humidifier should have an R-value in the range of 2–4 to ensure that good humidifier performance will be achieved. Optimally a value of 3 should be targeted for good performance. Various humidifiers produced by DPoint Technologies with different geometries have been tested and found to show optimal performance in this range. The geometries of all the humidifiers created are summarized in Table 1 and their performance plotted against their R-values can be found in Fig. 7.

4.4. Humidifier design procedure

For a plate-and-frame humidifier with the basic volume dimensions as shown in Fig. 8, and channel dimensions in Fig. 2, the overall humidifier channel design procedure follows a series of design equations as summarized in Fig. 9 and described below. The humidifier should be designed according to the requirements and constraints defined by the fuel cell system. Requirements will be based on the nominal flow rate to the fuel cell, the fuel cell operating temperature, and the required input water con-

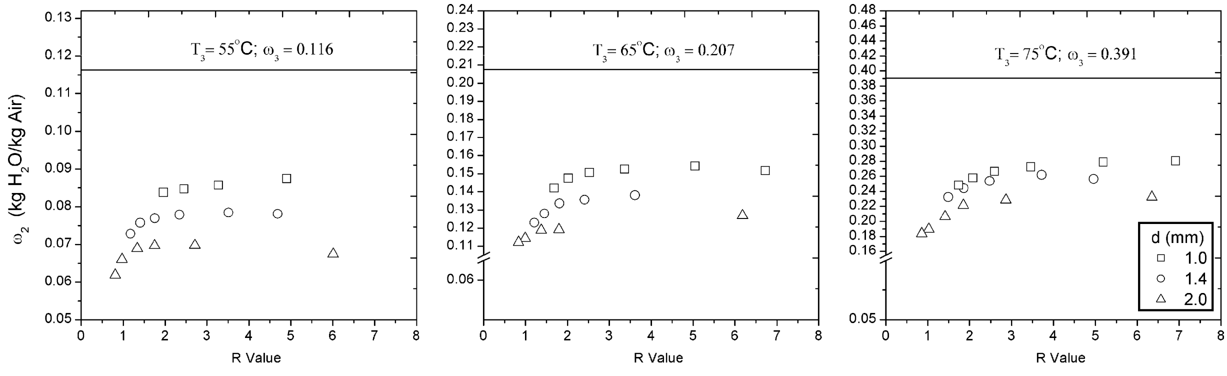

Fig. 7. Water recovery ratios for various humidifiers (see Table 1) as a function of the R-value at stream 3 dew point temperature of 65 °C.

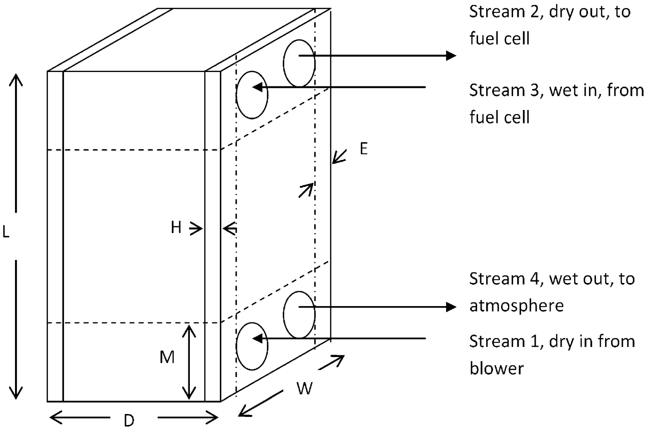

Fig. 8. External humidifier geometry.

tent. The nominal flow rate to the humidifier is calculated from P_e, the rated power of the fuel cell, V_c, the rated cell voltage of individual cells in the fuel cell stack at the given operating conditions, and λ, the required air stoichiometry provided to the fuel cell [2].

$$m_{\text{air}} = \lambda \frac{P_e}{4FV_c}\frac{\text{MW}_{\text{air}}}{x_{O_2}} \tag{6}$$

This value is converted to volumetric flow, Q, which is used in Eq. (5) to design the humidifier. If less than saturated reactant conditions are required at fuel cell inlet (stream 2), then the rated flow can be decreased proportionally to the relative humidity required.

As shown in Fig. 5, it is desirable to minimize the channel depth in the humidifier. However, manufacturing constraints may prevent the creation of a humidifier with very shallow channels. Also, a decrease in channel depth will lead to an increase in the pressure drop across the humidifier. Increased pressure drops will lead to increased parasitic power losses from the air supply blowers in the fuel cell system. Further, exhaust air exiting the fuel cell and entering the wet side of the humidifier often contains condensed or condensing water, so it is important to ensure that the channel depth is not so small that channel blockage may occur due to water droplets which adhere to the channel surface. For this analysis, the minimal channel depth will be set at 1 mm.

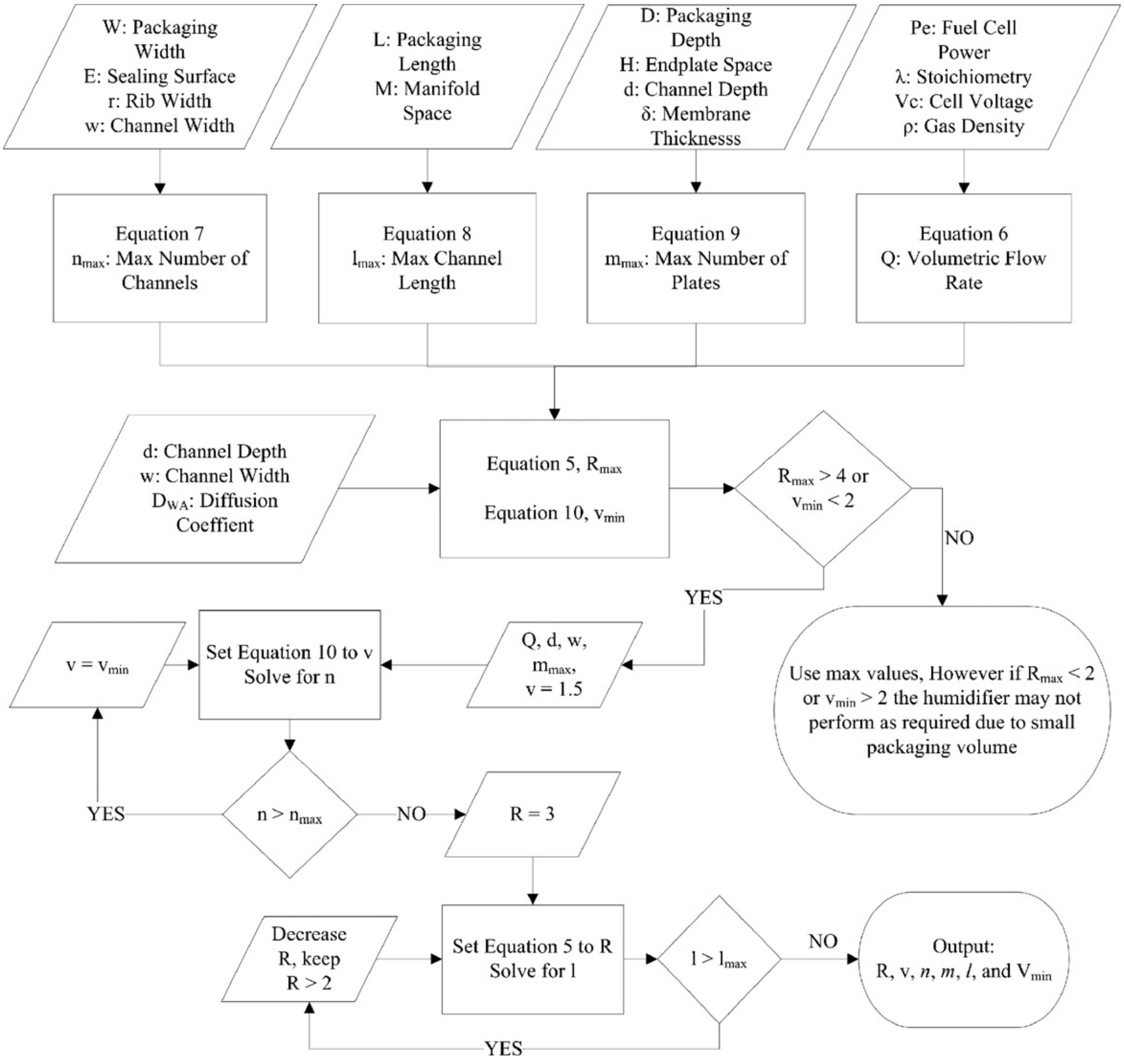

Fig. 9. Flowchart for humidifier design procedure.

The channel width for the humidifier is constrained by the differential pressure across the humidifier membrane, the membrane tensile strength, and creep resistance of the membrane. Increasing the channel width will increase the residence time, the membrane area, and overall humidifier performance, and decrease the overall humidifier size. This means larger channel widths are desirable. However, in the presence of increased or constant differential pressures the membrane may stretch or creep into the humidifier channel if the channel width is too large. This means that it is important to understand the membrane material properties when choosing the humidifier channel width. For this sample analysis, the maximum channel width will be set to 3 mm. The remaining variables, l, the humidifier length; n, the number of channels per plate; and m, the number plates in the humidifier, will depend on the packaging requirements for the humidifier in the fuel cell system. These values must be solved by an iterative approach in which packaging restraints, L, W, and D are met while the R-value is kept within the desired range of 2.0–4.0. The humidifier external geometry is outlined in Fig. 8.

The external humidifier width, W, will be related to n the number of channels per humidifier plate, w the channel width, and r the spacer rib thickness. The spacer rib thickness will have to be sufficient to ensure that the ribs can easily line up when the humidifier plates are stacked during assembly, for this analysis the *R-value* will be set to 2.0 mm. The channel width (w) and the rib thickness (r) values are set by the aforementioned mechanical requirements, this means the external humidifier width, W will be governed by the number of channels chosen for each humidifier plate. If an external width is imposed on the humidifier design by the packaging requirements, then the maximum number of channels per plate will be set by

$$n_{\max} = \frac{W_{\max} - 2E + r}{w + r} \tag{7}$$

In which E is the required space for the outer sealing edges of the humidifier, as shown in Fig. 8. It is important to note that the choice of overall humidifier thickness and number of channels per plate may also depend on the manifolding and inlet and outlet header design for the humidifier.

The external humidifier length L will be a function of l, the humidifier channel length, and may also be constrained by the packaging requirements for the humidifier. It will also be important to ensure that the humidifier length is sufficiently large enough to ensure a good residence time for the gases in the humidifier channel. The maximum humidifier channel length will be set by

$$l_{\max} = L_{\max} - 2M \tag{8}$$

In which M is the required space for the humidifier inlet and outlet manifolds.

Finally the external humidifier depth, D will be a function of d the channel depth, and δ the membrane thickness. The humidifier depth, D may be constrained by packaging requirements. The maximum number of humidifier plates, $m_{\max}$ will be set by

$$m_{\max} = \frac{D_{\max} - 2H}{d + \delta} \tag{9}$$

In which H is the required space on either end of the humidifier stack for the endplate as well as the inlet and outlet ports.

Another thing to consider in the humidifier design is that increasing the velocity of the gas in the channel to higher values, as seen in Fig. 4, will rapidly decrease the performance of the humidifier. Keeping the velocity in the channel below 2 m s^{-1} ensures that performance will be sufficient for most designs. Velocity in the channel can be calculated by

$$v = \frac{2Q}{mnwd} \tag{10}$$

Using the values for $m_{\max}$, and $n_{\max}$ in Eq. (10) the minimum channel velocity, $v_{\min}$ can be calculated. If the minimum velocity is less than 1 m s^{-1} then the humidifier design can likely undergo further optimization, and if the minimum velocity is greater than 2 m s^{-1} then the flow may be too large and the packaging volume may be too small for the humidifier requirements. A target velocity of 1.5 m s^{-1} will be used in this design procedure.

Eqs. (5)–(10) are used together to solve for the humidifier geometry based on the required flow and packaging requirements for the humidifier. Often the packaging requirement for the humidifier in the fuel cell system will cause the humidifier to be constrained in at least one of the variables W, L, or D. With or without packaging requirements imposed on the humidifier design, it will be desirable to minimize the overall humidifier volume. The volume will be a product of the final external depth, width, and length of the humidifier (D, W, and L) which can be solved by rearranging Eqs. (7)–(9):

$$V = DWL \tag{11}$$

Since the combined channel width, w and rib thickness, r are generally larger than the combined channel depth, d and membrane thickness, δ the overall volume, V will be kept to a minimum by using the maximum number of plates and the minimum number of channels in order to achieve the R-value and velocity values imposed on Eqs. (5) and (10). This means that in Eq. (11), D will be maximized and W and L will be minimized. However, increased values of m, means more humidifier plates are used in the stack, meaning longer assembly times, and an increased likelihood of assembly error, which may lead to leaks and humidifier failure. It may be prudent to consider this in the design and set an upper limit on the number of plates per stack in a feasible range. Also when designing the humidifier plates, it would be beneficial to make plates that can be used for a large number of different humidifiers so that manufacturing costs can be minimized. With one type of plate design set, humidifiers for fuel cells of many different rated powers can be created by changing the number of plates in the humidifier, m.

The humidifier design procedure is demonstrated in the following paragraphs, the selected variables are summarized in Table 2 and the overall procedure is summarized in Fig. 9. A humidifier based on this design was created and rated performance results are presented. For this analysis, P_e, the rated

Table 2
Parameters used in the sample humidifier design presented

Parameter	Unit	Design note	Value for example
P_e	W	Nominal power for fuel cell	500
V_c	V	Nominal cell voltage for fuel cell	0.7
λ	–	Stoichiometry coefficient for fuel cell	2.0
Q	$m^3 s^{-1}$	Calculated required flow to humidifier	4.0×10^{-4}
W	m	Packaging width for humidifier, constraint	0.160
L	m	Packaging length for humidifier, constraint	0.200
D	m	Packaging depth for humidifier, constraint	0.125
d	m	Channel depth, want to minimum allowable	0.080
w	m	Channel width, want maximum allowable	0.003
R	m	Channel rib width, want minimum allowable	0.002
E	m	Required space for sealing surface on edges of humidifier	0.005
M	m	Required space for manifolds on top and bottom of humidifier stack	0.020
H	m	Required space for endplates on humidifier stack	0.025
δ	m	Membrane thickness	2×10^{-4}
n_{max}	–	Maximum number of humidifier channels per plate, output from Eq. (7)	10
l_{max}	m	Maximum length of humidifier channel, output from Eq. (8)	0.160
m_{max}	m	Maximum number of humidifier plates, output from Eq. (9)	25
R_{max}	–	Output from Eq. (5), with max values	4.9
v_{min}	$m s^{-1}$	Output from Eq. (10), with max values	1.01
V_{max}	L	Maximum packaging volume	0.96
R		Set as target value	3.0
v	$m s^{-1}$	Set as target value	1.5
m	–	Value for volume minimizing procedure	25
n	–	Value output from Eq. (10) with target v	7
l	m	Value output from Eq. (5) with target R	0.145
V	L	Volume, minimized while keeping v and R in desired ranges	0.64

power for the fuel cell will be 500 W, λ the air stoichiometry will be 2.0, and V_c, the individual cell voltage will be 0.7 V. Using Eq. (6) the required mass flow to the humidifier is calculated to be 5.1×10^{-4} kg s^{-1} and Q, the standard volumetric flow, is 4.0×10^{-4} m^3 s^{-1}. The channel depth is set to an assumed minimum of 1.0 mm, and the channel width at its maximum for the selected membrane mechanical properties at 3.0 mm, and the channel rib thickness was set to 2.0 mm. The required edge thickness of the plates, E, is set to 5 mm. The required space for the endplate on the humidifier stack, H, is set to 25 mm and the required space for the inlet and outlet manifolds, M, is set to 20 mm. The membrane thickness, δ, is 0.2 mm. The external humidifier length, width, and depth constraints for packaging will be assumed to be $W_{max} = 60$ mm, $L_{max} = 160$ mm, $D_{max} = 80$ mm, giving a maximum packaging volume, $V_{max} = 0.96$ L.

Substituting the given values into Eqs. (7)–(9); n_{max}, the maximum number of channels, l_{max}, the maximum channel length, and m_{max}, the maximum number of plates are calculated. For this example these values are 10 channels, a 200 mm channel length, and 25 plates respectively. These values are then entered into Eqs. (5) and (10) to determine R_{max} and v_{min} for the humidifier at the maximum allowable humidifier size. If R_{max} is below 2 or v_{min} is greater than 2 m s^{-1} then the size constraints are too small to design an optimized humidifier for the rated flow under the given packaging requirements. If R_{max} is greater than 4 then the humidifier can be made smaller and can be optimized. In this case the R_{max} value is 4.9 and v_{min} is 1.01 m s^{-1} so the humidifier can be made smaller than the maximum volume. The humidifier number of channels (n), the length of the channels (l), and the number of plates (m) can be decreased to find a combination of the three that outputs an R-value between 2 and 4, with a target of 3 in this design procedure. There will be multiple combinations of these values at which R will be 3.0. So as an added constraint, the channel gas velocity will be kept between 1.0 and 2.0 m s^{-1} and the gas velocity target for this procedure will be 1.5 m s^{-1}. This leads to a volume minimizing procedure, which will also ensure good humidifier performance.

Generally, the increasing the number of plates allows for the greatest increase in performance with the least increase in volume. Accordingly, the volume minimizing procedure uses the maximum number of plates, m_{max}, to minimize the stack volume, and a target value of 1.5 m s^{-1} is set for the channel velocity, v, to ensure good performance. This value is input into Eq. (10) and the required number of channels, n is solved. If n is greater than n_{max} at this velocity, then the velocity is increased to v_{min} and n_{max} is used in the design. The values are then input into Eq. (5), which is set to the target value of 3.0, and the channel length is solved. If the channel length required is greater than l_{max} then the R-value may be adjusted between 2.0 and 4.0 to achieve the necessary length. At this point all the humidifier dimensions are solved for the given packaging constraints and the overall volume should be minimized while sufficient humidifier performance is maintained.

The final design in the example had a channel length, l of 145 mm, each humidifier plate had seven channels (n), and there were 25 plates in the humidifier (m). At these set points R was 3.0, and v was 1.5 m s^{-1}, so it will be known that the designed

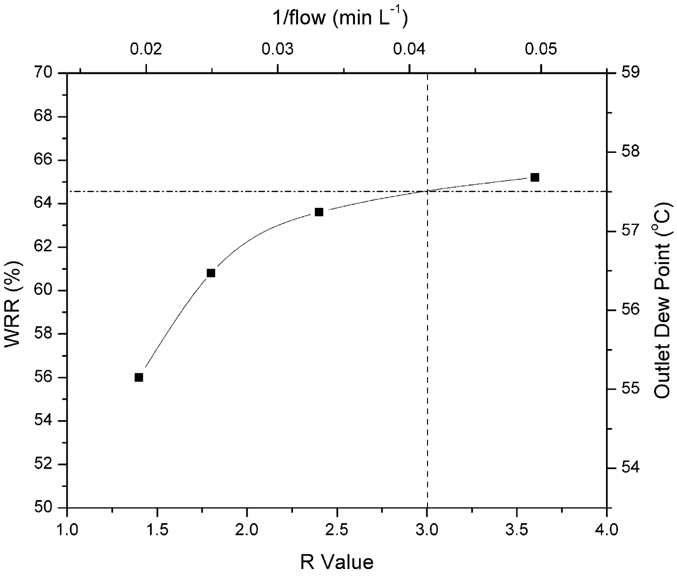

Fig. 10. Performance of humidifier created using the presented design procedure at a stream 3 dew point of 65 °C.

humidifier will perform well. The final humidifier volume was 0.96 L. This humidifier was manufactured at DPoint Technologies and was tested at various flow rates. The performance results for this humidifier can be found in Fig. 10. The rated flow was 24 L min^{-1} (0.042 min L^{-1}) for a 500 W fuel cell at 65 °C and a stoichiometry of 2.0. Fig. 10 shows that the humidifier is approaching the peak performance at this flow rate. This indicates that the humidifier has been designed for sufficient performance while minimizing the humidifier volume, subject to the packaging constraints.

It is important to point out that there are limitations to this approach. The overall design equations used for the *R*-values calculated are not rigorous, and do not account the actual amount of water transported in the humidifier unit. As well, the procedure depends on empirical observations of the optimal velocity and *R*-value performance ranges for humidifiers based on relatively straight channels. A recent publication by Cave and Merida [16] may provide a correction factor for the diffusion within a channel which may be used to further refine the results in a future work. Nonetheless, the presented *R*-value approach has shown to provide a good basis for humidifier design as shown in Figs. 7 and 10. The assumptions made may not be valid for a broad range of pressure and relative humidity conditions, and may be dependent on the membrane material used for the humidifier. Appropriate membrane selection will also be of great importance to the humidifier design (and is the subject of ongoing work); the *R*-value approach presented here only considers the geometrical considerations involved in the humidifier design. Unusual humidifier channel geometry and manifolding may limit the accuracy of this approach. Finally, the presented approach assumes that the humidifier is being designed for maximum performance with a minimum volume, particularly within specific special constraints. More efficient humidifiers may be designed given unlimited packing spaces; however a similar analysis would be beneficial in this case.

5. Conclusions

Prototype humidifiers were created to determine the validity of the *R*-value as a dimensionless parameter for humidifier design. The *R*-value is a combination of the residence time and the diffusion time of water molecules in the humidifier channels. Increasing the residence time of water molecules in the humidifier channels was found to increase the humidifier performance. However, greater increases in the residence time past a certain point tend to demonstrate diminishing returns in humidifier performance. Increasing the diffusion time of water from the surface of the membrane into the channel leads to a decrease in the overall humidifier performance. Combining these effects, the optimal *R*-value range for good planar gas-to-gas membrane humidifier design was found to be between 2.0 and 4.0, in which the optimal value is near 3.0 Future works should focus on developing parameters for humidifier design based on a more fundamental analysis. Design approaches using various types of membrane materials should be also be considered. The ideal *R*-value volume minimizing design algorithm was presented demonstrating a humidifier design procedure based on geometrical constraints for packaging the humidifier in the fuel cell system. This procedure can be used as a starting point for humidifier design.

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