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Modelling heat and mass transfer in a membrane-based air-to-air enthalpy exchanger

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Abstract The diffusion of total energy recovery systems could lead to a significant reduction in the energy demand for building air-conditioning. With these devices, sensible heat and humidity can be recovered in winter from the exhaust airstream, while, in summer, the incoming air stream can be cooled and dehumidified by transferring the excess heat and moisture to the exhaust air stream. Membrane based enthalpy exchangers are composed by different channels separated by semi-permeable membranes. The membrane allows moisture transfer under vapour pressure difference, or water concentration difference, between the two sides and, at the same time, it is ideally impermeable to air and other contaminants present in exhaust air. Heat transfer between the airstreams occurs through the membrane due to the temperature gradient. The aim of this work is to develop a detailed model of the coupled heat and mass transfer mechanisms through the membrane between the two airstreams. After a review of the most relevant models published in the scientific literature, the governing equations are presented and some simplifying assumptions are analysed and discussed. As a result, a steady-state, two-dimensional finite difference numerical model is setup. The developed model is able to predict temperature and humidity evolution inside the channels. Sensible and latent heat transfer rate, as well as moisture transfer rate, are determined. A sensitive analysis is conducted in order to determine the more influential parameters on the thermal and vapour transfer.

1. Introduction

In recent years, forced ventilation has been widely used in large buildings where it is required to provide a comfortable and healthy indoor environment for the occupants. Although exchange between indoor and outdoor air is necessary to maintain the indoor air quality, the energy demand associated with treating the outdoor air can be significant. A smart way to reduce this consumption is to adopt total energy recovery systems. These devices are able to transfer sensible heat and moisture between supply and exhaust airstreams. Since the latent heat of moisture can account for large part of the energy in the exhaust air, recovery of total energy is of great importance. A total energy recovery systems is mainly characterized by the type of air-to-air enthalpy exchanger in use. The two most common types of enthalpy exchangers are membrane based planar plate-type enthalpy exchangers and enthalpy wheels.

A membrane-based enthalpy exchanger is an air-to-air heat exchanger with a semi-permeable membrane used for heat recovery in modern building ventilation system. Supply and exhaust airstreams at different temperature and humidity transfer heat and moisture across the membrane by flowing through the exchanger. The thermal performance of the exchanger are mainly affected by its

geometry, and in particular by the flow configuration of the airstreams. Traditionally the airstreams were in crossflow, but in recent years the quasi-counterflow configuration has been widely used, due to its higher thermal performance. Figure 1 shows the membrane shape of enthalpy exchanger in crossflow and in quasi-counterflow configuration. The red lines highlight the computational discretization.

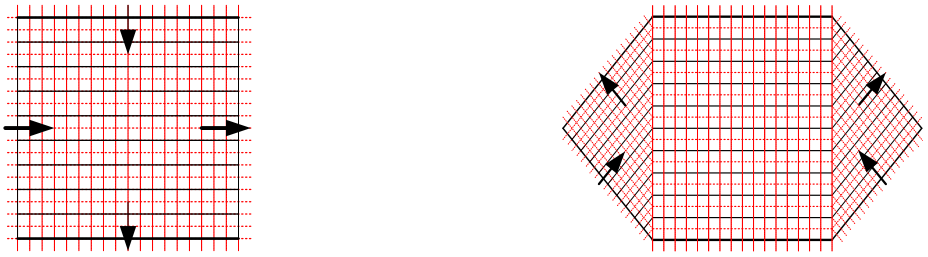

Figure 1. Plate of planar enthalpy exchanger in crossflow (left) and quasi-counterflow (right) arrangements.

In order to promote thermal and moisture transfer, the heat transfer and mass transfer resistance of the membrane should be as lower as possible. On the other hand, the membrane must be stiff enough to withstand the pressure difference and limit the surface warping under both dry and wet conditions. Since the moisture transfer is mainly affected by the moisture permeability of the membrane, it is essential to understand how its physical characteristics affect the exchange performance.

In the last decades, two research groups have performed experimental studies on the characterization of semi-permeable membranes for energy exchange purpose [1,2,3,4,5]. Table 1 summarizes the main results of their works.

Table 1. Typical values of membrane properties

Material	λ_M W m^{-1} K^{-1}	$D_{w,M}$ kg m^{-1} s^{-1}	C_s -	θ_{max} kg$_w$ kg$_M^{-1}$	δ_M mm	*Reference*
Copolimer	-	$2{,}16 \cdot 10^{-8}$	2,5	0,23	0,02	[1]
Paper	0,44	$5{,}33 \cdot 10^{-9}$	6	0,92	0,055	[2]
Cellulose acetate	0,41	$7{,}98 \cdot 10^{-9}$	11.4	0,43	0,005+0,04[a]	[2]
Modified cellulose acetate	0,44	$2{,}5 \cdot 10^{-9}$	8.64	2,5	0,005+0,04[a]	[2]
PES	0,15[b]	$5{,}41 \cdot 10^{-7}$	2.49	0,07	0,095	[3]
PVDF	0,19[b]	$1{,}92 \cdot 10^{-6}$	10.26	0,03	0,082	[3]
Cellulose	0,4[b]	$7{,}10 \cdot 10^{-7}$	8.1	0,25	0,115	[3]

[a] thickness of the support PP net (polypropylene) [b] typical value reported in the literature

The literature contains also studies on modeling membrane-based enthalpy exchangers. Niu and Zhang [1] developed a theoretical model to evaluate the performance of a membrane-based enthalpy exchanger and to investigate the effect of the inlet air conditions. Zhang and Niu [6] developed performance correlations for quick estimation of the sensible, latent and enthalpy effectiveness of the exchanger. They based their further studies [2,7,8] on these works. Min et al. performed similar works by studying heat and mass transfer processes across the membrane [9,10,11]. They analyzed the effect of the heat of adsorption on the process of heat transfer in moisture exchange across a membrane [12,13,14].

In the current work, a detailed model of the coupled heat and mass transfer mechanisms through the membrane between the two airstreams has been developed. The model is able to predict temperature and humidity profile inside the channels. Using this model as a base, the performance of the enthalpy exchanger was evaluated in terms of sensible and latent heat. A sensitive analysis is conducted in order to determine the most influential parameters on the thermal and moisture transfer.

2. Theoretical model

1.1. Physical model

The model consists of supply stream and exhaust stream channels separated by a membrane. Figure 4 shows the schematic of a channel in a typical membrane-based planar plate heat exchanger. The membrane are separated by plastic spacer with the purpose to guide the stream across the axial direction.

In the quasi-counterflow arrangement, supply and exhaust streams in the core-section are in counterflow, whereas in the two head-section the streams are in crossflow (Figure 1, right). Heat and water vapour are transferred transversally to the flow direction through the membrane from the hot and humid stream to the cold and drier one. According to the Solution-Diffusion model [15], when moisture transfers through the membrane, it first adsorbs at high concentration side of the membrane releasing heat. Then, under the driving force of concentration gradients in the membrane, the adsorbed water diffuses through the membrane and finally desorbs from the other (low concentration) side of the membrane absorbing heat.

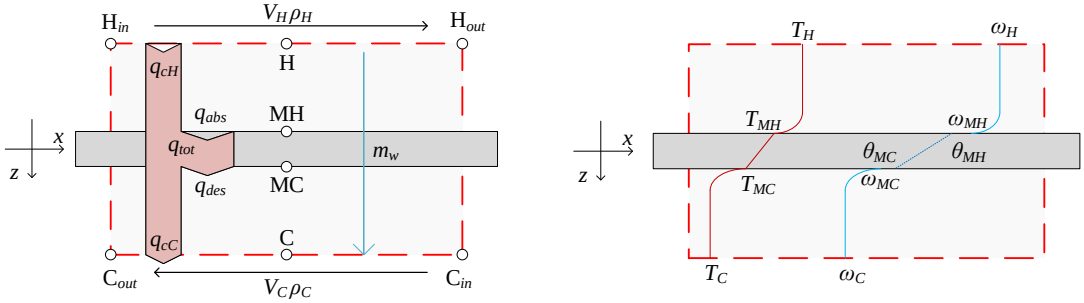

Figure 2. Left: one-dimensional schematic of the heat and mass transfer. Right: temperature T, humidity ω and water uptake θ distribution profiles across the membrane.

During the equilibrium between the membrane and the moisture at its surface, the sorption property of the membrane can be described using sorption isotherms. A popular sorption isotherm equation widely used in the gas separations permits to express the moisture uptake θ (kg_w kg_M^{-1}) of the membrane as a function of the relative humidity ϕ at the system temperature T [16]:

$$\theta = \theta_{\max}\left(1 - C_S + \frac{C_S}{\phi(T)}\right)^{-1} \tag{1}$$

where θ_{max} is the maximum moisture content of the membrane corresponding to the saturation condition (ϕ=100%) and C_S is the membrane sorption coefficient.

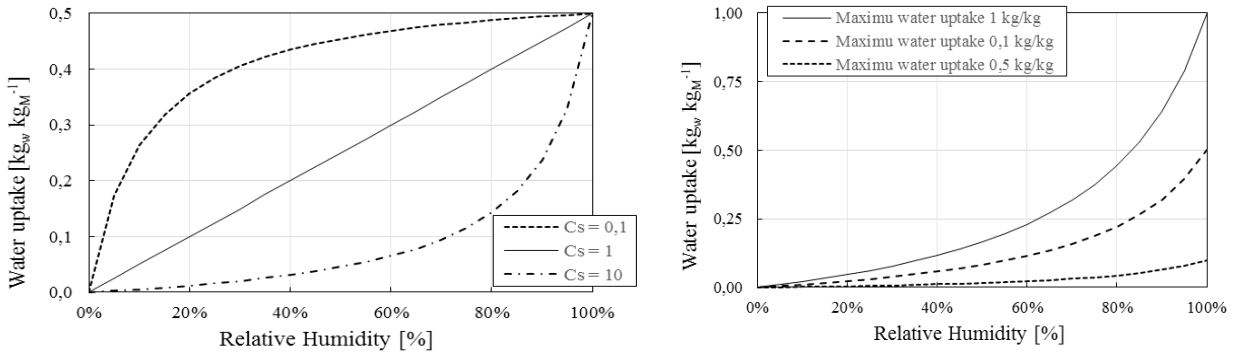

Figure 3. Typical sorption curves when varying the the sorption coefficient C_S (left) and the moisture uptake θ_{max} (right) .

C_S is a variable that represents the shape of the sorption curve and the type of sorption. For C_S =1 (silica gel), the sorption curve is linear with the relative humidity and corresponds to Herny sorption.

C_S <1 (molecule sieve) indicates that the sorption curve is an upward convex curve, which corresponds to Langmiur sorption. C_S >1 (polymers) yields a downward convex curve, which applies to most of the membrane used for enthalpy exchanger as can be seen in Table 1. Figure 3 shows sorption curves for different values of the sorption coefficient C_S (left) and of the maximum moisture uptake θ_{max} (right).

1.2. Modeling assumptions

The mathematical model of coupled heat and mass transfer in a membrane based heat exchanger is based on the following assumptions:

1. Both the heat and mass transfer processes are at steady state.
2. Heat conduction and vapour diffusion in parallel direction to the channel (x, y) are negligible compared to the bulk convection.
3. Heat conduction through the plastic spacer is negligible as compared to the bulk convection.
4. Moisture diffusion in the membrane only occurs in the transvers direction (z) along the thickness
5. The physical properties of the membrane are constant with temperature and the water uptake (profiles of temperature and moisture uptake in the membrane are linear in the transversal direction).
6. Adsorption and desorption of moisture are in equilibrium adsorption state.
7. The heat of adsorption-desorption is assumed constant and equal to the latent heat of condensation-vaporization.
8. Sorption hysteresis is neglected.

Niu and Zhang [1] reported that the effect of axial heat conduction and vapour transfer can be totally neglected for Peclet number Pe ($Pe = Re\ Pr$) greater than *100*.

The effect of the heat of adsorption-desorption on the process of heat and moisture exchanges across a membrane has been studied by Min and Wang [12,13]. They showed that the heat of adsorption-desorption is not a constant but depends on the membrane surface adsorption capacity and temperature, which are affected by the heat and mass transfer characteristics. The heat of adsorption can be considered composed by two parts: the heat of condensation, caused by the interaction among the adsorptive molecules, and the surface energy, caused by the interaction between the adsorptive and adsorbent molecules affected by the system temperature. In the work [12] of Min and Wang has been shown that for small moisture mass flux, the effect of the surface energy can be neglected and the heat of adsorption can be treated as a constant. Furthermore, as shown in their sequent work [13], a variable heat of adsorption-desorption requires the estimation of several membrane physical properties, usually difficult to find in literature. Hu et al. [14] made a similar study on the effect of adsorption heat on the heat transfer. They treated the adsorption heat as a constant and assumed it to be equal to latent heat of condensation. Therefore, in this work the heat of adsorption-desorption is assumed constant and equal to the latent heat of condensation-vaporization, too.

It has been assumed that the temperature along the membrane thickness exhibits a linear distribution. This condition corresponds to ignore the effect of the enthalpy carried by the mass transfer in the membrane on the temperature distribution. This assumption is reasonable because the membrane is usually really thin (less than *0,1* mm) and the mass flux is quite small compared to the bulk flow.

1.3. Governing equations and boundary conditions

The governing equations for studying the coupled heat and mass transfer in a control-volume are based on the energy and mass balances. Referring to Figure 2, the subscript H and C are associated to the hot and cold air stream, while MH and MC refer to the hot and membrane and cold side of the membrane, respectively:

$$\begin{array}{ll} \textit{Hot air stream} & \dot{V}_H \rho_H c_{p,H} \dfrac{\partial T_H}{\partial x} + 2\alpha_H \left(T_H - T_{MH}\right) = 0 \qquad \dot{V}_H \dfrac{\partial \omega_H}{\partial x} + 2\alpha_{m,H} \left(\omega_H - \omega_{MH}\right) = 0 \\ \textit{Cold air stream} & \dot{V}_C \rho_C c_{p,C} \dfrac{\partial T_C}{\partial x} + 2\alpha_C \left(T_C - T_{MC}\right) = 0 \qquad \dot{V}_C \dfrac{\partial \omega_C}{\partial x} + 2\alpha_{m,C} \left(\omega_{MC} - \omega_C\right) = 0 \end{array} \tag{2}$$

where V, ρ an c_p are the volumetric flow rate ($m^3\ s^{-1}$), the density ($kg\ m^{-3}$) and the specific heat ($J\ kg^{-1}\ K^{-1}$) of the airstreams, The symbols α ($W\ m^{-2}\ K^{-1}$) and α_m ($kg\ m^{-2}\ s^{-1}$) are used to identify the heat and mass transfer coefficients, respectively. T (K) and ω ($kg_v\ kg_{da}^{-1}$) are the temperature and the absolute humidity of the moist air. For the membrane,

$$\dot{m}_w c_{p,w} \frac{\partial T_M}{\partial z} - \lambda_M \frac{\partial^2 T_M}{\partial x^2} - \lambda_M \frac{\partial^2 T_M}{\partial y^2} - \lambda_M \frac{\partial^2 T_M}{\partial z^2} = 0 \qquad \dot{m}_w = -D_{wM}\left(\frac{\partial \theta}{\partial z} + \frac{\partial \theta}{\partial x} + \frac{\partial \theta}{\partial y}\right) \tag{3}$$

where the moisture mass transfer flux m_w ($kg\ m^{-2}\ s^{-1}$) is dependent on the moisture diffusion coefficient in the membrane D_{wM} ($kg\ m^{-2}\ s^{-1}$) and the moisture uptake θ ($kg_w\ kg_M^{-1}$) gradient of the membrane.

Based on the assumptions of the previous paragraph, the boundary conditions for the coupled heat and mass transfer can be expressed as follows:

Hot air stream		*Cold air stream*		*Membrane*	
$T_H\big\|_{x=0} = T_{H,in}$	$\frac{\partial T_H}{\partial y} = 0$	$T_C\big\|_{x=0} = T_{C,in}$	$\frac{\partial T_C}{\partial y} = 0$	$\frac{\partial T_M}{\partial x} = 0$	$\frac{\partial T_M}{\partial y} = 0$
$\omega_H\big\|_{x=0} = \omega_{H,in}$	$\frac{\partial \omega_H}{\partial y} = 0$	$\omega_C\big\|_{x=0} = \omega_{C,in}$	$\frac{\partial \omega_C}{\partial y} = 0$	$\frac{\partial \theta}{\partial x} = 0$	$\frac{\partial \theta}{\partial y} = 0$

where $T_{H,in}$, $\omega_{H,in}$ and $T_{C,in}$, $\omega_{C,in}$ are the temperature and absolute humidity of the two airstreams at the inlet condition

Hot air – membrane interface *Cold air – membrane interface*

$$-\lambda_M \left.\frac{\partial T_M}{\partial z}\right|_{\delta=0} = 2\alpha_H\left(T_H - T_{MH}\right) + \dot{m}_w \Delta h_{abs} \qquad -\lambda_M \left.\frac{\partial T_C}{\partial z}\right|_{z=\delta_M} = -2\alpha_C\left(T_C - T_{MC}\right) + \dot{m}_w \Delta h_{des} \tag{4}$$

where Δh_{abs} and Δh_{des} ($J\ kg^{-1}$) represent the isosteric heat of adsorption and desorption, respectively

1.4. Coupled heat and mass transfer

Figure 2 shows a schematic representation of the energy balance in a control-volume membrane. Based on this diagram, the heat transfer described in the previous governing equations can be discretized as follows:

$$2\alpha_H\left(T_H - T_{MH}\right) + \dot{m}_w \Delta h_{abs} = \frac{\lambda_M}{\delta_M}\left(T_{MH} - T_{MC}\right) = 2\alpha_C\left(T_{MC} - T_C\right) + \dot{m}_w \Delta h_{des} \tag{5}$$

where δ_M (m) is the thickness of the membrane.

The convective heat fluxes q_c ($W\ m^{-2}$) between the bulk flows and the membrane surfaces are defined as,

$$q_{c,H} = \alpha_H\left(T_H - T_{MH}\right) \qquad q_{c,C} = \alpha_C\left(T_{MC} - T_C\right) \tag{6}$$

Substitution of Equation 6 in Equation 5 yield to the expression:

$$q_{c,C} = q_{c,\mathrm{H}} = \left(\left(T_H - T_C\right) - \dot{m}_w \Delta h_{lat}\left(\frac{\delta_M}{\lambda_M}\right)\right) \Big/ \left(\frac{1}{2\alpha_M} + \frac{\delta_M}{\lambda_M} + \frac{1}{2\alpha_C}\right) \tag{7}$$

Besides the bulk temperatures (T_H and T_C) the convective heat transfer coefficients (α_C and α_H) and the conductivity of the membrane (λ_M), the convective heat is dependent from the moisture flux across the membrane m_w. The moisture mass flux is given by,

$$\dot{m}_w = 2\alpha_{mH}\left(\omega_H - \omega_{MH}\right) = \frac{D_{wM}}{\delta_M}\left(\theta_{MH} - \theta_{MC}\right) = 2\alpha_{mC}\left(\omega_{MC} - \omega_C\right) \tag{8}$$

According to the sorption isotherm of Equation 1, the moisture uptake, θ_{MH} and θ_{MC}, on the hot and cold side of the membrane can be related to the relative humidity at the membrane surface, hence Equation 2 becomes:

$$\theta_{MH} = \theta_{\max}\left(1 - C_S + \frac{C_S}{\phi(T_{MH})}\right)^{-1} \qquad \theta_{MC} = \theta_{\max}\left(1 - C_S + \frac{C_S}{\phi(T_{MC})}\right)^{-1} \tag{9}$$

In order to obtain an expression of the mass flux similar to that for the convective heat flux, it is useful to relate relative and absolute humidity. Using the well-known Clapeyron equation and assuming atmospheric pressure (*101325* Pa), the relationship between relative to absolute humidity can be approximated as:

$$\frac{\phi}{\omega} = 10^{-6} e^{5294/T} - 1{,}61\phi \tag{10}$$

As reported by [15] neglecting the second term on the right of Equation 10 leads to an error less than 5%. Thus, the mass flux across the membrane assume the following expression:

$$\dot{m}_w = \frac{D_{wM}}{\delta_M}\theta_{\max}\left(\left(1 - C_S + \frac{C_S}{\omega_{MH} 10^{-6} e^{5294/T_{MH}}}\right)^{-1} - \left(1 - C_S + \frac{C_S}{\omega_{MC} 10^{-6} e^{5294/T_{MC}}}\right)^{-1}\right) \tag{11}$$

This results is similar to that obtained from Min and Su [9,10]. It has to be note that the previous equation differs from that given by Zhang and Liang [7]: in the present formulation the moisture gradient across the membrane has been taken in account instead of evaluating the sorption characteristic based only on the humid side of the membrane. Since humidity can change greatly across the membrane the two formulations may differ substantially.

From the above observations, it results that heat transfer and mass transfer in membrane-based enthalpy exchangers are coupled processes that should not be traded separately.

3. Numerical model

A 2-dimensional, finite-difference model has been developed to calculate the heat and moisture exchange in the membrane based planar plate enthalpy exchanger in quasi-counterflow arrangement.

The computational domain has been chosen as a three-dimensional elementary cell taken between two consecutive channels as represented in Figure 4.

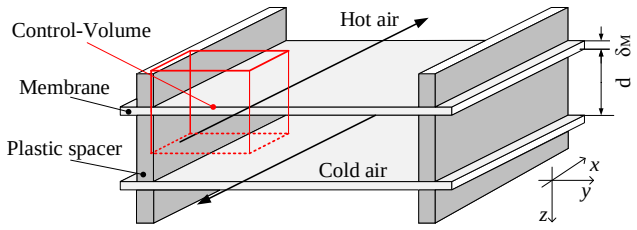

Figure 4. Schematic of a channel in membrane based planar plate heat exchanger and control-volume identification for a counterflow arrangement.

As shown in Figure 1, the geometry of the single control volume depends on its position in the exchanger. The number of elements in the head sections is determined by the number of divisions for each channel, while in the core section the number of element depends on the divisions along the axial direction and the number of division for each channel, as well.

Each control-volume requires iterative calculations to simultaneously solve Equations 5 and 8. The bulk air conditions are taken as average value between inlet and outlet conditions for each element. A main loop updates the inlet conditions for each control-volume until convergence is reached.

3.1. Heat and mass transfer coefficient in the boundary layer

To evaluate the convective heat transfer coefficient, the recent work of Gendebien et al. [17] has been taken as reference. The heat transfer coefficient in the boundary layer can be described by Nusselt correlation.

In the model, laminar flow has been considered for *Re < 2000*. The fully developed laminar flow heat transfer coefficient α (W m^{-2} K^{-1}) in rectangular duct can be calculated as function of the aspect ratio of the cross section only. The Nusselt number correlation developed by Shah and London [18] has been used. The resulting Nusselt number is then corrected to takes account of the Prandtl number and the thermal entry length effects according to Kakaç et al. [19] and to Wibulswas [20], respectively. Generally for turbulent flow, the Nusselt number correlations developed for circular tube can be use also for different cross sectional shape [21]. In this model, the Gnielinski correlation [21] is used to calculate the Nusselt number for *Re > 4000*. The Gnielinski correlation requires to estimate the friction factor, which is calculated with the correlation developed by Petukhov for smooth surfaces [21]. For transitional flow, between *Re = 2000* and *Re = 4000*, a linear interpolation has been used.

Once the heat transfer coefficient α is known, the mass transfer coefficient α_m (kg m^{-2} s^{-1}) can be obtained from the Chilton-Colburn analogy [21].

4. Results and discussion

Simulations were first conducted to investigate the effects of the membrane characteristics on the coupled heat and mass transfer through the membrane. Calculations were done for one geometrical configuration and one set of inlet air conditions for a variety of membrane parameters.

The parameters of the simulation are summarized in Table 2.

Table 2. Model parameters.

Total length	l_{total}	400 mm	Inlet temperature (H)	$T_{H,in}$	35 °C
Total width	w_{total}	250 mm	Inlet relative humidity (H)	$\phi_{H,in}$	75 %
Total height	h_{total}	150 mm	Inlet mass flow rate (H)	$V_{H,in}$	50 m^3 h^{-1}
Core length	l_{core}	200 mm	Inlet temperature (C)	$T_{C,in}$	20 °C
Spacer thickness	t_{spacer}	1 mm	Inlet relative humidity (C)	$\phi_{C,in}$	50 %
Number of layers	n_{layer}	30	Inlet mass flow rate (C)	$V_{C,in}$	50 m^3 h^{-1}
Number of channels	$n_{channel}$	9			

The results of the different simulations are represented in terms of sensible, latent and total effectiveness, respectively defined as:

$$\varepsilon_s = \frac{Q_s}{\min(\dot{V}\cdot\rho\cdot c_p)_{in}\left(T_{H,in}-T_{C,in}\right)} \qquad \varepsilon_l = \frac{Q_l}{\min(\dot{V}\cdot\rho)_{in}\left(\omega_{H,in}-\omega_{C,in}\right)\Delta h_{lat}}$$
$$\varepsilon_t = \frac{Q_s+Q_l}{\min(\dot{V}\cdot\rho\cdot c_p)_{in}\left(T_{H,in}-T_{C,in}\right)+\min(\dot{V}\cdot\rho)_{in}\left(\omega_{H,in}-\omega_{C,in}\right)\Delta h_{lat}} \tag{12}$$

Calculations were then conducted to estimate the difference between the coupled and the decoupled heat and mass transfer. Decoupling heat and mass transfer can be realized by ignoring the latent heat terms in Equation 9. The results were compared in terms of sensible and latent heat transfer rate and by visualization of the temperature and humidity distributions.

4.1. Parametric analysis

The parametric analysis has been conducted to investigate the performance of the enthalpy exchanger with different membrane characteristics. Diffusion coefficient $D_{w,M}$, sorption coefficient C_s, maximum water uptake θ_{max}, membrane thickness δ_M and membrane thermal conductivity λ_M have been varied in their typical ranges.

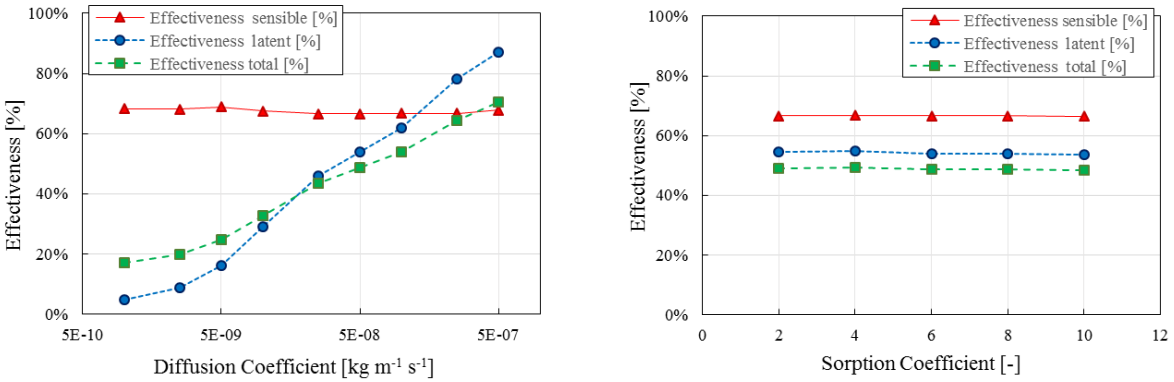

Figure 5. Effectiveness for different values of the diffusion coefficient $D_{w,M}$ of water in the membrane (left) ($C_s = 6$) and for different values of the sorption coefficient C_s (right) ($D_{w,M} = 5 \cdot 10^{-8}$ *kg m*$^{-1}$ *s*$^{-1}$). Results are obtained at $\theta_{max}=0,3$ kg kg^{-1}, $\delta_M = 0,05$ mm, $\lambda_M = 0,3$ W m^{-1} K^{-1}.

Figure 5 (left) shows the variations of the sensible, latent and total heat performance with the moisture diffusion coefficient in the membrane. When the moisture diffusion coefficient increases, the moisture flow rate increases as well, causing an enhancement in the latent heat flux. Since a larger moisture diffusion coefficient will cause a larger water uptake gradient in the membrane, it will increase the moisture mass flux between the two sides and the latent heat as a consequence. The total effectiveness, is the results of combining the sensible and the latent heat transfer. Since the sensible effectiveness does not present wide variation, the total effectiveness has a similar variation to that of the latent effectiveness.

The sorption coefficient affects only marginally the heat and the moisture transfers, as plotted in Figure 5 (right) It has to be mentioned that in the considered range, typical for enthalpy exchanging membranes, the variation of the sorption causes only slightly change on the sorption curve. As a consequence, the sensible, latent and total effectiveness remain almost constant with the sorption coefficient.

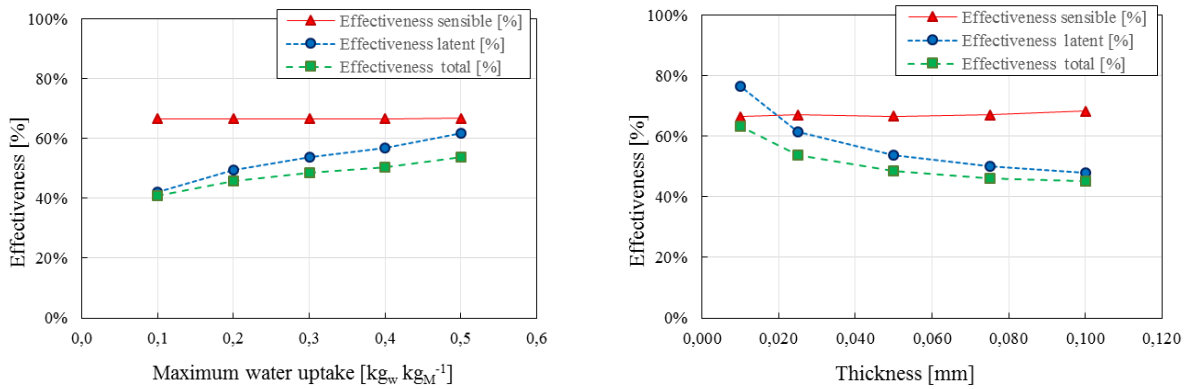

Figure 6. Effectiveness for different values of the maximum water uptake in the membrane θ_{max} (left) ($\delta_M = 0,05$ mm) and for different values of the membrane thickness δ_M (right) ($\theta_{max}=0,3$ kg kg^{-1}). Results are obtained at $D_{w,M} = 5 \cdot 10^{-8}$ kg m^{-1} s^{-1}, $C_s = 6$, $\lambda_M = 0,3$ W m^{-1} K^{-1}.

A more influential parameter on the sorption curve is the maximum water uptake in the membrane, as can be seen from Figure 3 (right). The effects of its variation are shown in Figure 6 (left). The increase in the maximum water uptake causes an increase in the maximum potential gradient for the water uptake across the membrane. It results in an enhancement of the moisture transfer flux and of the latent heat transfer as well. As the maximum water uptake varies, the sensible effectiveness remains almost unchanged.

As mentioned in the introduction, in order to minimize the membrane thermal and mass resistances itis fundamental to reduce the membrane thickness. Figure 6 (right) shows the variation of the effectiveness with the variation of the membrane thickness. The sensible effectiveness present only slightly variation as the thickness changes. The reason is that the dominant thermal process is the convective heat transfer in the boundary layer of the airstreams. In the considered conditions, the ratio of conductive (membrane) to convective resistance results less than 1/100. The latent and total effectiveness increases by decreasing the membrane thickness. The trend is super-linear with the thickness decrease. It confirms that with constant sorption characteristics of the membrane, an effective way to enhance the moisture and latent heat transfer is to reduce the thickness of the membrane.

As seen above, the sensible effectiveness is only slightly affected by the variation of the membrane parameters. Even the variation in the membrane thermal conductivity have small influences on the surface temperatures of the membrane, which affect sensible and latent heat transfer. In fact, the performance of the enthalpy exchanger exhibits almost no change with the membrane thermal conductivity varying from *0,1* to *100* W m^{-1} K^{-1} (not shown in the graphs). The reason is that the membrane thickness is really small, leading to a negligible thermal resistance compared to the thermal resistance of the convective processes. For a membrane-based enthalpy exchanger operating under typical conditions, the membrane thermal resistance accounts only for a small fraction of the total thermal membrane resistance. The heat transfer is dominated by the convection from the bulk flow to the membrane surfaces, while the membrane mass resistance dominates the humidity exchange. These present results confirm the observations of Min and Su [8,9] .

4.2. Coupled and decoupled heat and mass transfer comparison

Decoupling the heat and mass transfer consists in ignoring the latent term in Equation 2 and following consequences. The result is a temperature profile independent from the moisture transfer. The dependence of the moisture transfer by the surface temperatures remains in the expression of the relative humidity using Clapeyron equation.

The main effect of decoupling the heat and mass transfer is a reduction in the computational effort to solve the heat and mass transfer problem, which allows to reduce the calculation time.

Coupled and decoupled heat and mass transfer models are compared under the same conditions reported in Table 2. For these simulations, the membrane is characterized by a diffusion coefficient of $5 \cdot 10^{-8}$ kg $m^{-1}s^{-1}$, a sorption coefficient equal to *6*, a maximum water uptake of *0,3* kg kg^{-1}, a membrane thickness equal to *0,05* mm and a membrane thermal conductivity set to *0,3* W m^{-1} K^{-1}.

Minor differences have been registered in the temperature distribution. The calculated sensible heat transfer rate results *160* W and *162* W for the coupled and decoupled model, respectively, with a sensible effectiveness of *65* %. As for the temperature, the absolute humidity presents only slightly difference between the coupled and the decoupled models. The latent heat transfer rate results *409* W for the coupled solution and *410* W for the decoupled model, with a latent effectiveness of *65* %, which leads to a total effectiveness of *47* %.

5. Conclusions

A numerical study was performed to analyze the combined heat and moisture transfer across a membrane. A first set of calculations were performed for a representative air inlet conditions by changing the membrane parameters. A further calculation has been performed to compare the difference between coupled and decoupled heat and mass transfer models.

From the calculations it can be deduced that the sensible heat transfer is practically not affected by the membrane properties. In fact, the heat transfer is dominated by the convection from the bulk flows to the membrane surfaces. The membrane thermal resistance accounts only for a small fraction of the total thermal resistance. The only way to enhance the sensible heat flow rate is to enlarge the exchange surface or increase the convective coefficient. On the contrary, the membrane properties have a strong influence on the latent heat transfer because the membrane mass resistance dominates the humidity exchange. The most influential membrane characteristics result to be the moisture diffusion coefficient and the maximum water uptake in the membrane. When a membrane material has been selected, an

effective way to maximize the latent heat transfer is to reduce the membrane thickness as much as possible. At the same time the thickness of the membrane must guarantee enough stiffness to withstand the pressure differential and limit the surface warping under both dry and wet conditions.

Within the assumptions of this study, no evident difference has been registered in the predictions by the coupled and the decoupled heat and mass transfer models. Only slightly differences can be seen in the temperature and absolute humidity profiles. The adoption of the decoupled mass transfer model will lead to spare computational time and simultaneously to predict the performance of a membrane-based enthalpy exchanger with reasonable accuracy.

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