Numerical simulation of heat and moisture transfer in co-current membrane heat exchangers

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Abstract. The paper presents the mathematical model, based on the narrow-channel approximation, for calculating the processes of heat and moisture transfer in membrane heat exchangers and taking into account possible condensation on heat exchange surfaces. The possibility of humidifying the supply air with moisture from the exhaust air in the cold period is considered. It is shown that the membrane co-current heat exchangers can operate without freezing up to the outdoor temperature equal to -20 °C.

1. Introduction
Regenerative and recuperative heat exchangers are increasingly being used to reduce energy consumption for heating and air conditioning of buildings. When outdoor temperatures are below 0 °C the ventilation may incur freezing of heat exchanger channels [1, 2, 3]. Such freezing may be prevented by the preheating of supply air, bypassing, reduction of the sensible heat transfer effectiveness of the heat exchanger, or preliminary dehumidification of the exhaust air.

Air may be dehumidified in the following ways: 1) by condensation on the heat exchange surface with subsequent removal of condensate from the channel [4]; 2) by absorption with various absorbents [5, 6]; 3) or by using a selectively permeable membrane that removes only water vapor [7, 8].

Heat exchangers with vapor-permeable heat exchange surfaces have recently gained wide popularity in the ventilation systems of residential buildings [9 – 19], since such devices allow controlling both air temperature and humidity. The data of [20 – 29] shows that heat exchangers of this type have a somewhat higher efficiency than those without permeable heat exchange surface. The efficiency of such heat exchangers depends on the season and the geographical latitude.

Membrane heat exchangers are mainly considered as dehumidifiers of supply air, incoming into room in summer. In winter, on the contrary, it is required to humidify the supply air up to comfortable values. This issue has not been practically covered in the literature.

To simulate heat and mass transfer processes occurring in the channels of the membrane heat exchanger, one may use the balance methods of heat and mass calculation [17] or the Navier-Stokes equations (or their simplified approximation in the form of a boundary layer or a narrow channel) [11, 23]. The latter way of modeling is more accurate and allows considering a large number of factors affecting heat and mass transfer: changes of concentration profiles of water vapor and temperature, reduction of the heat flow length, etc.

At vapor-air mixture cooling in heat exchangers, vapor may condense on heat and mass transfer surfaces even in the warm season, which changes the nature of mass transfer in the heat exchanger circuit. Such a change inside the membrane heat exchanger is virtually not considered in the literature.
The abovementioned determines the main purpose of this paper, namely, to study the capability of the membrane heat exchanger to humidify the supply air with moisture of the exhaust air in the cold season, to find the sensible and latent heat transfer effectiveness of the heat exchanger and to estimate the minimum air temperature at which no freezing of the heat exchanger channels occurs.

2. Problem statement and main features of heat and moisture transfer

The scheme of flows in question is shown on fig. 1. The heights of both channels are 2 mm, and the lengths are 300 mm. The supply air flows in the upper channel, and the exhaust air flows in the lower channel. The flow velocities of the vapor-gas mixture are identical in both channels, \( u = 1 \) m/s, except for the test case, when the flow velocity vary from 0.5 to 2.5 m/s.

The gas mixture flow inside the heat exchanger is considered in the narrow-channel approximation. The equations of motion in this case are similar to the boundary layer equations:

continuity equation:
\[
\frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} = 0 , \tag{1}
\]

equation of motion
\[
\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = \frac{\partial}{\partial y} \left( \frac{\rho u}{\partial y} \left( \frac{\partial u}{\partial y} \right) - \frac{dp}{dx} \right) , \tag{2}
\]
diffusion equation
\[
\rho u \frac{\partial K_i}{\partial x} + \rho v \frac{\partial K_i}{\partial y} = \frac{\partial}{\partial y} \left( -\rho D \frac{\partial K_i}{\partial y} \right) , \tag{3}
\]
and energy equation
\[
\rho u c_p \frac{\partial T}{\partial x} + \rho v c_p \frac{\partial T}{\partial y} = \frac{\partial}{\partial y} \left( -\lambda \frac{\partial T}{\partial y} \right) + \sum_{i=1}^{n} \left( c_{pi} - c_{pm} \right) \rho D \frac{\partial K_i}{\partial y} \frac{\partial T}{\partial y} + \mu \left( \frac{\partial u}{\partial y} \right)^2 + \frac{dp}{dx} . \tag{4}
\]

The vapor flow through the membrane is determined from the difference in partial pressures of water vapor in the channels:
\[
j_v = \Lambda \cdot \Delta p / \delta_m . \tag{5}
\]

In equations (1)–(5): \( \rho \) is the density, kg/m\(^3\); \( u, v \) are the flow velocities along and across the tube, m/s; \( x, y \) are the longitudinal and transverse coordinates, m; \( K \) is the mass concentration of the gas mixture component; \( T \) is the absolute temperature [K]; \( p \) is the pressure [Pa]; \( \mu \) is the dynamic viscosity, Pa·s; \( D \) is the diffusion coefficient, m\(^2\)/s; \( c_p \) is the specific heat of gas at constant pressure, J/kg·K; \( \lambda \) is the thermal conductivity, W/m·K; \( \Lambda \) is the membrane permeability coefficient for the mixture component \( i \), kg·m / Pa·s·m\(^2\); \( \delta_m \) is the membrane thickness, m; \( j \) is the mass flow through the membrane, kg/m\(^2\)·s; \( \Delta p = p_{11} - p_{12} \) is the difference in partial pressures of vapor in both channels, Pa.

With temperature decrease of the heat exchange surface, vapor condensation may occur. It changes the boundary condition for the diffusion equation: when the dew point temperature is reached, the concentration of water vapor on the wall is assumed to be equal to the saturated vapor concentration at the preset temperature (relative humidity on the wall is 100%). The flow of vapor penetrating through the membrane is, as earlier, determined by the relation (5), but in the cooled channel the vapor flow to the wall is determined from the mass balance:
\[ j_v = \frac{1}{1 - K_{sv}} \rho D \left( \frac{\partial K_v}{\partial y} \right)_w, \]  

(6)

where \( K_{sv} \) is the mass fraction of saturated vapor.

The vapor flow passing through the membrane is less than the vapor flow caused by condensation. As a result, a film of liquid is formed on the membrane. In the following calculation results, the influence of the formed film on the dynamics of flow and heat exchange is neglected.

The heat flow through the membrane is determined by the Newton-Richman law:

\[ q = \frac{\lambda_m}{\delta_m} (T_{w1} - T_{w2}), \]  

(7)

where \( \lambda_m = 0.1 \text{ } W/\text{m} \cdot K \) and \( \delta_m = 10^{-4} \text{ } m \) are the thermal conductivity and membrane thickness, and \( T_{w1} \) and \( T_{w2} \) are the temperatures of the membrane in the first and second channels, respectively. For each channel, the heat balance is determined as follows:

\[ q = -\lambda \left( \frac{\partial T}{\partial y} \right)_w - j_v \cdot r(T), \]  

(8)

where \( r(T) \) is the heat of phase transition, MJ/kg, approximated by the equation:

\[ r(T) = 0.282 (T_{vo} - T)^{0.371}, \]  

(9)

where \( T_{vo} = 647.3 \text{ } K \) is the critical temperature of water vapor. The properties of air, water vapor and the vapor-gas mixture, were calculated from the polynomial dependences presented in the works [30, 31]. The saturated vapor pressure was determined by the Goff–Gratch equation. The problem is solved by numerical integration of equations in physical coordinates for the implicit scheme on the non-uniform rectangular grid with compression near high gradients of velocity and temperature. The nonlinear differential equations are solved by the method of simple iterations at each step of integration with the accuracy of \( 10^{-3} \). The number of computation grid points in the transverse direction is 300 for each channel. The integration step in the longitudinal direction does not exceed \( 10^{-3} \) m.

3. Validation of mathematical model

Air-to-air heat exchangers are characterized by sensible and latent heat transfer effectiveness, which are determined by the following equations:

\[ \varepsilon_s = \frac{G \left( T_{si} - T_{so} \right) + G_i (T_{vo} - T_{si})}{2G_{\min} (T_{si} - T_{ei})}, \]  

(10)

\[ \varepsilon_L = \frac{G \left( K_{si} - K_{so} \right) + G_i (K_{vo} - K_{si})}{2G_{\min} (K_{si} - K_{ei})}, \]  

(11)

where \( G \) and \( G_i \) are the mass flows in the first and second channels, respectively, kg/s; \( G_{\min} \) is the minimum of the two vapor-gas flows; \( T_{si} \) and \( K_{si} \) are the temperature and mass fraction of water vapor at the supply inlet; \( T_{so} \) and \( K_{so} \) are those at the supply outlet; \( T_{ei} \) and \( K_{ei} \) are the ones at the exhaust inlet; and \( T_{vo} \) and \( K_{vo} \) are those at the exhaust outlet.

The mathematical model was tested by comparing the simulation results with the data of [23]. The geometry and dimensions of the heat exchanger from [23] correspond to the abovementioned ones. In
this case, the supply air temperature in winter was taken as $t_s = -10^\circ C$, and in summer $t_s = 30^\circ C$; and the relative humidity was $\varphi = 50\%$ and $61\%$ in winter and in summer, respectively. The exhaust air temperature for both periods was assumed to be $t_e = 25^\circ C$, and the relative humidity was $45\%$. The comparison was made on the value of sensible heat transfer effectiveness (fig. 2). Its maximum value in the case of the co-current flow is about 50%. This value is achieved for flow velocities of 0.5 m/s and lower. With flow velocity increase, the $\varepsilon_s$ decreases since complete heat exchange needs more time and length. The data of numerical modeling (filled points) is in satisfactory agreement with the data of [23], which proves the adequacy of the used mathematical model.

4. Results and discussion
As it has been already mentioned in the introduction, the problem of freezing of the circuit with exhaust air is quite acute for air-to-air heat exchangers, operating in the cold seasons. When using heat exchangers with countercurrent or cross-current flows in winter, their sensible heat transfer effectiveness is specifically reduced to avoid frosting. Co-current flow heat exchangers have lower effectiveness compared to counter- and cross-flow ones ($\varepsilon_s \leq 50\%$), therefore temperatures above zero may be set at exhaust channel outlet up to supply air temperatures $t_s = -20^\circ C$. Figure 3 shows changes in the temperature profile in different cross sections of the co-current heat exchanger without permeable membrane. Relative humidity of supply air is 90%, temperature is $t_s = -20^\circ C$ (upper channel), and for exhaust air (lower channel) $\varphi = 45\%$ and $t_e = 25^\circ C$. At the lower channel outlet, the average temperature is approximately equal to $t_e = 1^\circ C$. The temperature profiles for the case with permeable and impermeable membrane for water vapor almost coincide, so the figure presents the results only for the case of impermeable membrane.
For the case with impermeable membrane the temperature increase in the upper channel leads to a decrease in relative humidity. If there is no additional humidification dry air is supplied in the room (fig. 4 solid lines). In the lower channel, relative humidity increases due to temperature decrease. In this case, the mass fraction of water vapor in the upper channel remains unchanged, and in the lower channel decreases due to condensation (fig. 5 solid lines). The use of membrane with high permeability coefficient for water vapor as a heat transfer surface facilitates the vapor transfer from the lower to the upper channel. In the latter one, the relative humidity decreases in the initial sections since the heat transfer process through the membrane is much faster than the mass transfer. Then it increases due to temperature leveling in the channels and vapor supply through the membrane (fig. 4 dotted lines). In this case, over the heat exchanger length, the mass fraction of water vapor \( (K) \) in the lower channel decreases due to water vapor removal and condensation on the membrane, and in the upper channel \( K \) increases due to vapor penetration through the membrane (fig. 5). In the studied length of the membrane heat exchanger, the mass fraction of water vapor in the two channels changes approximately two times, and at the output of the heat exchanger channels the humidity ratio of air is almost the same. The determining factor in the change of humidity ratio in the lower channel is not the value of vapor flow through the membrane, but the flow due to condensation.

In the considered case, in both channels of the heat exchanger, the pressure is equal to atmospheric one, and the moisture transfer across the membrane occurs due to a difference in initial concentrations of water vapor. The water vapor flow caused by the difference in partial pressures of the vapor above and below the membrane is much less than the flow resulting from the condensation (fig. 6). The difference is three orders of magnitude. The vapor flow rate at condensation decreases from \( \sim 10^{-4} \) to \( \sim 10^{-6} \) kg/m\( ^2 \)s. The vapor flow caused by the difference in partial pressures is \( \sim 10^{-7} \) kg/m\( ^2 \)s. The membrane cannot transmit all condensed moisture, and in the lower channel, even with rather high permeability of the membrane (\( \Lambda = 10^{-11} \)), the water vapor reaches the saturation state. So, a liquid film may form on the membrane surface.
Figure 6. Distribution of vapor flows caused by condensation (1) and by the difference of partial vapor pressures (2) along the heat exchanger length.

The membrane permeability increase in winter (fig. 7a) has almost no effect on the sensible heat transfer effectiveness for different relative humidity of the supply air, since at temperatures below zero the change in relative humidity hardly affects the change in humidity ratio. The reverse trend is observed in summer.

The latent heat transfer effectiveness (fig. 7b) increases with an increase in membrane permeability during the winter and summer periods. With a small permeability of the membrane, the $\varepsilon_L$ in winter is higher than in summer, due to vapor condensation on the membrane.

Figure 7. Sensible heat transfer effectiveness (a) and latent heat transfer effectiveness (b) vs. membrane permeability in summer (triangle) and winter (square).

The use of membrane heat exchangers for indoor ventilation during cold seasons allows humidifying the supply dry air to comfortable values. However, there are a number of difficulties to take into account. So, in the above-described mode, the supply air at the heat exchanger outlet has a temperature approximately equal to 0°C. At further air heating (no vapor supply) the relative humidity of air decreases, and when reaching the room temperature $\varphi$ is approximately 20%. This suggests that to increase the supply air humidity up to comfortable values ($\varphi=45\%$) an additional stage of moisture transfer is required, or it is necessary to increase the vapor flow through the membrane at the first stage. There are two ways to increase the vapor flow: the first is to use a more permeable membrane; and second is to increase pressure in the lower channel or to decrease it in the upper one.

According to equation (5) this will lead to a proportional increase in the flow of water vapor through the membrane. The selected membrane with high permeability coefficient and the set pressure difference in the channels must ensure the equality of water vapor flows caused by the difference in partial...
pressures and the flow resulting from condensation. Otherwise, the condensate forms on the heat transfer surface, necessitating its removal from the heat exchanger.

Conclusion
The paper presents the mathematical model for describing the heat and moisture transfer in direct-flow membrane heat exchangers with possible condensation of water vapor on the heat transfer surface. The results obtained using the proposed model have been verified on the results of other works. The results of calculations have showed the possibility of humidifying the air, supplied indoors in winter, by moisture from the exhaust air. It has been also proved that membrane heat exchangers can operate without freezing up to the outdoor temperature equal to -20 °C.

Humidification of the supply air by applying the membrane heat exchangers has a number of limitations. These include the formation of condensation on the heat transfer surface and insufficient humidification of the supply air. These limitations may be eliminated by using more vapor-permeable membranes or additional step of heat and mass transfer.

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