Development of numerical heat and mass transfer model for predicting total heat exchange performance in Energy Recovery Ventilator

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Abstract. An energy recovery ventilator (ERV) is a widely used equipment that can recover sensible and latent heat. To improve its performance, it is essential to understand the heat and moisture transfer mechanism in the ERV. Against this background, overarching objectives of this study are to develop mathematical/numerical models for predicting and clarifying the hygro-thermal (i.e. heat and moisture) transfer mechanism in the heat exchange element of the ERV in terms of (i) simplified model for sensitivity analysis, and (ii) comprehensive model to integrate hygro-thermal transfer equations with computational fluid dynamics (CFD) analysis. Toward this end, we conducted fundamental experiments to measure temperature, humidity, and enthalpy exchange efficiencies in the scale-down ERV unit model, and then, numerical analyses were conducted according to the experimental scenario. As results of this study, we confirmed the reasonable prediction accuracy of our proposed numerical models for predicting total heat exchange performances.

1. Introduction

In terms of sustainable design for residential houses, the optimization of energy consumption of heating, ventilating and air-conditioning (HVAC) systems is recognized as essential and then the use of an energy recovery ventilator (ERV), which can recover sensible and latent heat, is considered as one of the most effective way to reduce ventilation load in buildings. ERV has been widely researched for several decades, however, novel approaches and findings are still being reported by recent studies. In the previous study, we conducted and reported field measurements for performance evaluation of an ERV installed in an office space. We also quantitatively investigated the temperature and enthalpy exchange efficiency under various indoor and outdoor environmental conditions. In addition, we developed an ERV with carbon dioxide (CO2) demand-controlled ventilation system, which could adjust the amount of outside air introduced through the total heat exchanger after sensing the CO2 concentration indoors. As for the numerical analysis, we developed an analytical method integrating CFD and building energy simulation (BES). In order to achieve further performance improvement of ERV systems, it is still essential to understand their heat and moisture transfer mechanism and to optimize the geometry of the flow channel and the constituent materials.

Against this background, the overarching objectives of this study were to develop simplified mathematical/numerical models to predict and clarify hygro-thermal (i.e., heat and moisture) transfer mechanisms in the heat exchange element of an ERV and for sensitivity analysis, and comprehensive model to integrate the simultaneous heat and mass transfer equations with CFD analysis. Towards this end, firstly we conducted fundamental experiments to measure temperature, humidity, and enthalpy exchange efficiencies as functions of air flow rate, in the scaled-down ERV unit model. Secondly, numerical analyses based on a simplified hygro-thermal transfer model were executed according to the...
experimental scenario. Thirdly, comprehensive heat and mass transfer analyses integrated with CFD were also performed and prediction accuracies of simplified model and comprehensive model were discussed while comparing with the experimental results.

2. Numerical Models and Methods

2.1. Simplified model

To investigate simply and rapidly the total energy exchange mechanism of an ERV system, a simplified numerical simulation model for steady-state analysis has been developed. A trapezoidal flow channel in the total heat exchange element was assumed as shown in Figure 1.

![Figure 1. Flow channel in the total heat exchange element](image)

In this model, the sensible heat loss in inflow channel 1 (in Figure 2) is defined by Equation (1).

\[ q_1 = \rho_1 C_{p1} Q_1 (T_{1,\text{in}} - T_{1,\text{out}}) \]  

where, \( \rho_1 \) [kg/m\(^3\)] represents air density, \( C_{p1} \) [J/(kg K)] is specific heat of the air, \( Q_1 \) [m\(^3\)/s] is volumetric flow rate, \( T_{1,\text{in}} \) [K] is inflow air temperature, and \( T_{1,\text{out}} \) [K] is outflow air temperature.

The convective heat transfer on the surface of the heat exchange panel, \( S_{up} \), is expressed by equation (2) according to the Newton’s law of cooling.

\[ q_2 = h_1 S_t (T_{w1} - T_{in}) \]  

where, \( h_1 \) [W/(m\(^2\) K)] is convective heat transfer coefficient, \( S_t \) [m\(^2\)] is equivalent heat transmission area, \( T_{w1} \) [K] is the surface temperature on the heat exchange panel, and \( T_{in} \) represents air temperature calculated as the arithmetic mean value of \( T_{1,\text{in}} \) and \( T_{1,\text{out}} \).

Heat conduction from the upper surface \( S_{up} \) of the heat exchange panel to the lower surface \( S_{down} \) is expressed by the Fourier’s first law.

\[ q_3 = -\lambda_2 S_{min} \frac{dT}{dx} = -\lambda_2 S_{min} \frac{(T_{w2} - T_{u2})}{l_s} \]  

where, \( \lambda_2 \) [W/(m K)] is heat conductivity in the heat exchange panel, \( T_{w2} \) [K] is the surface temperature at the lower part of the panel, \( S_{min} \) [m\(^2\)] represents the heat transmission area estimated by \( S_t \) and \( S_2 \), and \( l_s \) [m] is the thickness of the heat exchange panel.

The convective heat transfer from the lower surface \( S_{down} \) to the air inflow in channel 2 is described as follows.

\[ q_4 = h_2 S_2 (T_{w2} - \bar{T}_L) \]  

\[ q_5 = \rho_2 C_{p2} Q_2 (T_{2,\text{out}} - T_{2,\text{in}}) \]  

The spacer panel inserted between the heat exchange panels works as a heat radiation fin. This fin effect is modelled as the increase of effective heat transmission area.
where, $S_i$ [m$^2$] is the effective heat transmission area based on the surface area of the heat exchange panel, $\alpha$ [-] represents fin efficiency ($0 \leq \alpha \leq 1$), $L$ [m] is the effective fin length of the spacer panel ($L$ is defined as half of the hypotenuse length ($= L_{PP}/2$)), $h$ is the convective heat transfer coefficient around the fin, $l_p$ [m] is the thickness of the spacer panel as fin, and $\lambda_p$ is heat conductivity.

In terms of heat balance in the heat exchange unit under steady-state conditions (Equation (8)), each temperature, $T_{1,\text{out}}$, $T_{2,\text{out}}$, $T_{\text{w1}}$, and $T_{\text{w2}}$, can be identified under the fixed inflow boundary condition of $T_{1,\text{in}}$ and $T_{2,\text{in}}$.

Concerning the latent heat transfer (moisture transfer), we considered the analogy between heat and mass transfer, and the Lewis relation ($T \rightarrow X$, $\lambda \rightarrow \rho D$, $h/C_p \rightarrow h_D$). Here, the complicated hygrothermal transfer mechanisms are simplified and the humidity transfer model is developed based on absolute humidity as transfer potential.

$$M_1 = \rho D (X_{1,\text{in}} - X_{1,\text{out}})$$

$$M_2 = h_D S (\tilde{X}_H - X_{\text{w1}})$$

$$M_3 = -\rho DS_{\text{min}} \frac{dX}{dx} = -KS_{\text{min}} \frac{(X_{\text{n2}} - X_{\text{w2}})}{l_p}$$

where, $X$ [kg/kg'-DA] is absolute humidity, $D$ [m$^2$/s] is diffusion coefficient of humidity in the heat exchange panel, $h_0$ [kg/(kg/kg'-DA)·m²·s] is convective mass transfer coefficient of absolute humidity, and $K$ [kg/((kg/kg'-DA)·m·s)] represents the coefficient of moisture permeability in the heat exchange panel. For the numerical analysis, the total heat exchange element is modelled as shown in Figure 3. It is discretized as a structured equidistant mesh (1 mm × 1 mm).

![Figure 3. Overview diagram of the numerical analysis](image_url)

2.2. Comprehensive model that integrates hygro-thermal transfer equation with CFD
For further understanding the transient heat and mass transfer phenomenon in an ERV, we developed comprehensive numerical model that integrated hygro-thermal transfer equations with CFD analysis. In the simplified model, sensible and latent heat transfer on the partition plates were modelled by using the convective heat transfer coefficient and convective moisture transfer coefficient. In order to improve the prediction accuracy in this boundary layer region at the vicinity wall surfaces, the CFD analysis with
low Reynold number type $k-\varepsilon$ model which can resolve particularly the viscous sub layer near the wall surfaces. And then the heat and moisture transfer rate can be quantitatively calculated by directly applying the Fourier’s law and Fick’s law for sensible and latent heat respectively. In this analysis, ensemble-averaged equation systems were applied, and the governing equations are as follows:

\[
\frac{\partial U_i}{\partial x_i} = 0 \quad (12)
\]

\[
\frac{\partial U_i}{\partial t} + \frac{\partial U_i U_j}{\partial x_j} = -\frac{1}{\rho} \frac{\partial P}{\partial x_i} + \nu \frac{\partial^2 U_i}{\partial x_j \partial x_j} - g_i \beta_T (T - T_r) - g_i \beta_X (X - X_r) \quad (13)
\]

\[
\frac{\partial T}{\partial t} + \frac{\partial U_i T}{\partial x_i} = \frac{1}{\rho_c} \frac{\partial}{\partial x_i} \left( D_T \frac{\partial T}{\partial x_i} - \dot{u}_t \right) \quad (14)
\]

\[
\frac{\partial X}{\partial t} + \frac{\partial U_i X}{\partial x_i} = \frac{1}{\rho_c} \frac{\partial}{\partial x_i} \left( D_X \frac{\partial X}{\partial x_i} - \dot{u}_X \right) \quad (15)
\]

where, (12) is the continuous equation, (13) is the Navier-Stokes equation, (14) is the energy (temperature) transport equation, and (15) is the moisture transport equation. The eddy viscosity and the correlation terms are defined as follows:

\[
-u_{ij} = v_i \left( \frac{\partial U_j}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} k \delta_{ij}, \quad v_i = C_{\mu} \mu \frac{k^2}{\varepsilon}, \quad -u_{it} = \frac{v_i}{\varepsilon} \frac{\partial T}{\partial x_i}, \quad -u_{ix} = \frac{v_i}{\varepsilon} \frac{\partial X}{\partial x_i} \quad (16)
\]

Here we used Abe-Kondoh Nagano model as low Re type $k-\varepsilon$ model. In addition to these governing equations, transport equations of turbulent energy $k$ and its dissipation rate $\varepsilon$ are also analyzed.

As for the analysis inside the partition materials of the total heat exchange element, following transient hygro-thermal transfer equations are solved.

\[
(C_P + L_v) \frac{\partial T}{\partial t} = \frac{\partial}{\partial x_i} \left( \lambda_T \frac{\partial T}{\partial x_i} \right) + \frac{L_k}{\partial x_i} \frac{\partial X}{\partial t} \quad (17)
\]

\[
(k \rho_{\text{air}} + \kappa) \frac{\partial X}{\partial t} = \frac{\partial}{\partial x_i} \left( \lambda_X \frac{\partial X}{\partial x_i} \right) + \nu \frac{\partial T}{\partial t} \quad (18)
\]

At the surface of material and air interface, the flux conservation is imposed as the boundary condition, and the coupled analysis between the gas phase and the solid phase is carried out.

In order to verify and validate the prediction accuracy of our proposed models, numerical analyses are carried out in accordance with the boundary conditions of fundamental experimental.

3. Experimental Setup

Fundamental experiments were conducted to investigate the total heat exchange efficiency of the heat exchange element. In this study, we prepared a relatively small sized total heat exchange element with a trapezoidal shaped flow channel for minimizing the measurement uncertainties. The outline of the total heat exchange element for the experiment is shown in Figure 4. The size of the element is W 30 mm × H 450 mm (184 laminated layers in total). A running section was set at the leading edge of the measuring section. Temperature exchange efficiency and enthalpy exchange efficiency were measured using two-climate chambers. The experimental procedure specified in ISO 16494 was adopted as the standard test method. The outline of the experimental setup is shown in Figure 5.
4. Results

The fundamental experiments using scale-down ERV unit model were carefully executed and the measurements results were calculated as time and ensemble averaged value for multiple trials. Figure 6 shows the results of the fundamental experiments for temperature, humidity, and enthalpy exchange efficiencies $\eta$ as functions of air flow rates, and the numerical simulation results under the same boundary conditions are also shown as overlapping the experimental results. In the experiment results, the significant flow-rate dependence on temperature, humidity, and enthalpy exchange efficiencies were confirmed. Because of the scale-down effect of the target element model, temperature, humidity, and enthalpy exchange efficiencies in this experiment were evaluated relatively low compared with those in general-size conventional ERV unit.

As shown in Figure 6, the simplified numerical simulation results were reasonably consistent with the results from experiments. The statistical differences of these were confirmed to be less than 1.1 pt. Figure 6 also shows the results of each exchange efficiencies estimated by the comprehensive model to integrate heat and mass transfer analyses with CFD. Under the condition of inflow velocity $U_{in}=9.36$ m/s, significant deviation of 27% was observed in the temperature exchange efficiency between the simulation and the experimental results. The temperature exchange efficiency under the low inflow velocity conditions, and humidity and enthalpy exchange efficiency under all inflow velocity conditions were in good agreement with the experimental results. It is inferred that the decrease in prediction accuracy in the high inflow velocity region might be due to the insufficient prediction accuracy of CFD analysis which is a basis for calculating heat and mass transfer rate. The precise quality control of CFD simulation and the appropriate selection of turbulent model will have significant impact on the flow and heat and mass transfer characteristics in the heat exchange element. The further discussion about the improvement of the prediction accuracy of CFD part is in future issue.
5. Discussions and Conclusions
The development of numerical simulation methods for clarifying the performance of ERV element are important to improve the total energy exchange efficiency and the quality of HVAC design. As shown in Figure 6, the results of each heat exchange efficiency of the small-size ERV element could be precisely measured and compared with the results of the numerical analyses. The simplified numerical analysis and the comprehensive numerical analysis with CFD were performed in accordance with the experimental boundary conditions and then a good prediction accuracies of the proposed numerical simulation models were confirmed quantitatively.

The simplified model could be expected to apply for performance optimization with parametric studies because of the light computational load, and the comprehensive model is able to be utilized to grasp the local and detail characteristics, e.g., impact of the change in flow path design. As a consequence, these numerical simulation models will result in such significant value to the zero-energy buildings (ZEB) design in terms of the optimization of ventilation efficiency.

The findings obtained in this study can be summarized as follows.

(1) Fundamental experiments for targeting small-scale total heat exchange element model were performed. Exchanges efficiencies of temperature, humidity, and enthalpy as functions of air flow rates were precisely measured and reported.

(2) A simplified numerical model to predict heat and mass transfer in the heat exchange element model was developed. Its prediction accuracy was confirmed to be consistent with the experimental data.

(3) A comprehensive numerical model that integrate simultaneous heat and mass transfer equation with CFD analysis was also developed. Its prediction accuracy showed good consistency with experimental data.

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