Effectiveness Extrapolation of Rotary Energy Exchangers and Its Impact on Energy Simulation

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Abstract
Air-to-air energy exchangers are used in buildings to transfer energy between the exhaust air and the entering outdoor air. Building simulation models are usually used to evaluate the benefits of applying air-to-air energy recovery in HVAC systems. The current practice is to simulate air-to-air energy recovery based on the rated performance while the impact of real operating conditions on effectiveness is either simply ignored or not fully considered. No research has been performed to justify the current practice. In this paper, the impact of effectiveness extrapolation on energy simulation is investigated using the whole building simulation program — EnergyPlus. A standalone retail building model is used to estimate and compare the system energy consumption by considering the impact of air face velocity, the flow rate ratio between the two air streams, and inlet air psychrometric conditions on energy recovery performance. It is found that ignoring the impact of inlet air psychrometric conditions on exchanger effectiveness has negligible impact (<0.5%) on both heating and cooling energy consumption. However, the air face velocity and especially the airflow ratio between the two airstreams have much higher impact on the heating energy consumption.

Highlights
• Laboratory tests are used to evaluate the wheel effectiveness correlations selected from the literature
• The correlation for wheel sensible effectiveness is improved
• The EnergyPlus model of a standalone retail building model is used
• A user-defined object in EnergyPlus is used to model the effectiveness of energy wheels

Introduction
Air-to-air energy recovery is an energy efficient technology that is increasingly used in many building HVAC systems. During building operation, fresh outdoor air needs to be supplied to the building to meet the minimum ventilation requirements as prescribed in ASHRAE Standards 62 (ASHRAE 2019a). The outdoor air is usually conditioned (i.e., heated/humidified in winter and cooled/dehumidified in summer) before it is sent to the occupied spaces. Conditioning outdoor air becomes part of the load of heating and cooling equipment (e.g., furnace, compressor, heating and cooling coils) and consumes a lot of energy, especially when a large amount of outdoor air is required. Meanwhile, air in the conditioned space is exhausted to the outdoor at a flow rate close to that of the outdoor air intake. Because the exhaust air has been conditioned, it can be used to preheat or precool the outdoor air, which is the underlying concept of air-to-air energy recovery. By transferring energy between the exhaust air and the outdoor air, air-to-air energy recovery has the potential to significantly reduce the energy consumed to condition the outdoor air. Using air-to-air energy recovery can also downsize heating and cooling equipment, resulting in capital cost reduction. Using energy recovery is required in many situations according to the prevalent energy efficiency standards for commercial buildings such as ASHRAE Standard 90.1 (ASHRAE 2019b).

Air-to-air energy exchangers come in different types and sizes, and they can be classified according to geometry, construction type, heat transfer, number of fluids, etc. Common types include fixed-plate exchangers, regenerative exchangers, heat pipes, and run-around loops (ASHRAE 2020a).

As a widely used energy efficiency technology, air-to-air energy recovery modeling is supported by many energy simulation programs. Therefore, many studies are available in literature using simulation programs to investigate the benefits of energy recovery in building design. Liu et al. (2010) used EnergyPlus to investigate the energy savings of using energy recovery ventilator (ERV) in a residential building with different climatic conditions, enthalpy efficiency, fresh air change rate, and fan power consumption of ERV. Zhou et al. (2007) used EnergyPlus to study the energy performance of ERV with different indoor temperature setpoints in Shanghai and Beijing, China. They found that ERV operation in cold climate (Beijing) was uneconomical when the cooling setpoint was above 24°C. Jiru (2014) used prototype commercial building models developed in EnergyPlus to estimate the energy savings from the combination of energy conservation measures, including air-to-air energy recovery. In the series of Advanced Energy Design Guides, air-to-air energy recovery was included in the recommended design package for several different commercial building types. The recommendation was made based on EnergyPlus simulations (e.g., Thornton et al. 2009).

Almost all previous work that used building simulations to evaluate the benefits of air-to-air energy recovery relied
on the product performance data at rated conditions. Because of the constraints of the energy recovery models and data availability, the impact of operating conditions (e.g., inlet air temperature, humidity, air face velocity, and airflow ratio between the two air streams) on energy recovery effectiveness is either simply ignored or not fully considered. No research has been performed to justify the current practice. This paper aims to investigate the impact of effectiveness extrapolation for rotary energy wheels on energy simulation using the whole building simulation program — EnergyPlus.

### Effectiveness Correlations

The work by Simonson and Besant (1999a, 1999b), as referred to in Informative Appendix F of ASHRAE Standard 84 is important for modelling rotary energy exchangers. They derived and developed a group of dimensionless numbers for energy wheels to reduce the dimensions of input variables. These dimensionless numbers were then used to establish a numerical model for coupled heat and moisture transfer between the two air streams of a rotary energy exchanger. The numerical model was validated with laboratory experiments covering a range of mass flow rates, temperature, and humidity. Based on the validated numerical model, they developed correlations to calculate the effectiveness of rotary energy exchangers.

The correlations proposed by Simonson and Besant (1999b) are significant to investigate the impact of different input variables on exchanger effectiveness at full-load conditions. Based on the numerical model of heat and mass transfer in energy wheels (Simonson and Besant 1999a), Simonson et al. (2000a. 2000b) further investigated the potential of using wheel speed control and bypass control to module the energy and moisture transfer rates at part-load conditions. Simonson et al. (2000b) modified the effectiveness correlations for full-load conditions to model the performance of energy wheels with unbalanced airflow rate. For both balanced and unbalanced airflow rates, the effectiveness correlations need a number of technical parameters about the energy wheel, such as the matrix mass, mass fraction of desiccant, volume fraction of desiccant, and surface area density of wheel. Unfortunately, most of these parameter values are regarded as proprietary information and thereby difficult to obtain.

Recognizing the challenge of applying the correlations that need proprietary information, Jeong and Mumma (2005) used the same numerical model from Simonson and Besant (1999a, 1999b) as the basis and developed “practical” correlations by modelling the wheel effectiveness based on the air face velocity, the flow ratio between the two airstreams, the dry-bulb temperature and the relative humidity of the two inlet airstreams, all values of which are supposed to be easily obtained. They developed two sets of correlations for different desiccant materials, one set for wheels with silicone gel and the other set for wheels with molecular sieve. Both sets of correlations are based on typical wheel characteristics including the wheel rotational speed of 20 rpm and the wheel depth of 200 mm. For example, the correlations for energy wheels with molecular sieve are expressed as:

\[
\eta_s = a_0 + a_1 V_i + a_2 T_i + a_3 \Phi_i + a_4 Q_R + a_5 \eta T_i + a_6 V_i \Phi_i + a_7 V_i Q_R + a_8 T_i \Phi_i + a_9 T_i Q_R + a_{10} \Phi_i Q_R + a_{11} T_i \Phi_i Q_R + a_{12} \Phi_i Q_R + a_{13} T_i \Phi_i Q_R (1)
\]

where, \( \eta \) and \( \eta_s \) are the sensible effectiveness and latent effectiveness, respectively; \( V, T, \Phi \) represent the air face velocity (m/s), air temperature (°C), and air relative humidity, respectively; \( Q_R \) is the airflow ratio between the exhaust air and the supply air; the subscripts \( i \) and \( e \) represent inlet supply air and inlet exhaust air, respectively; \( \alpha \) and \( \beta \) are correlation coefficients.

For energy wheels having molecular sieve for the desiccant material, wheel depth of 200 mm and rotational speed of 20 rpm, the coefficients take the following values: 1.05319 (\( a_0 \)), -0.022312 (\( a_1 \)), 1.24609E-3 (\( a_2 \)), 5E-3 (\( a_3 \)), -0.032 (\( a_4 \)), 1.17969E-4 (\( a_5 \)), 2.5E-3 (\( a_6 \)), -0.0325 (\( a_7 \)), -1.32813E-3 (\( a_8 \)), -2.32188E-3 (\( a_9 \)), -0.01 (\( a_{10} \)), -1.26562E-3 (\( a_{11} \)), 4.53125E-4 (\( a_{12} \)), 3.625E-3 (\( a_{13} \)), 1.18598 (\( a_\Phi \)), -0.026498 (\( a_\Phi \)), -0.02742 (\( a_Q \)), -9.8253E-3 (\( a_\delta \)), -0.117875 (\( a_\beta \)), -0.18408 (\( a_\beta \)), -0.030625 (\( a_\beta \)), -3.82031E-3 (\( a_- \)), 1.95833E-3 (\( a_- \)), -0.17775 (\( a_- \)), -0.012417 (\( a_\beta \)), -0.031875 (\( a_\delta \)), 1.19604E-3 (\( a_\beta \)), 0.047533 (\( a_- \)), 0.059808 (\( a_- \)), -5.96875E-3 (\( a_- \)), 0.014 (\( a_- \)), 0.017417 (\( a_\delta \)), 0.081667 (\( a_\delta \)), 1.34375E-4 (\( a_- \)), 8.125E-3 (\( a_- \)), 2.34375E-3 (\( a_- \)), -2.916667E-3 (\( a_- \)), 0.215 (\( a_- \)), -2.316667E-3 (\( a_- \)), -2.80417E-3 (\( a_- \)), -0.1153 (\( a_- \)), -0.023333 (\( a_- \)), -5.625E-4 (\( a_- \)), 5.666667E-3 (\( a_- \)).

The laboratory test results from ASHRAE RP-1799 (Wang et al. 2023) are used to validate the correlation Equations 1-2. In ASHRAE RP-1799, two sets of energy wheels, with each set consisting three wheels of different sizes (i.e., small, medium, and large) were tested. Only the large wheel from the second set is used here because this wheel satisfies the conditions (i.e., molecular sieve, wheel depth= 200 mm, rotational speed = 20 rpm) of using the correlations. A total of 7 effectiveness tests were performed at Intertek’s energy recovery ventilation equipment testing facility in Cortland, NY. Table 1 shows the inlet air conditions for all effectiveness tests.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>18°C</td>
</tr>
<tr>
<td>Humidity</td>
<td>40%</td>
</tr>
<tr>
<td>Airflow</td>
<td>2000 m³/h</td>
</tr>
</tbody>
</table>

The results of the laboratory tests show that the correlations match the test results with an accuracy of ±10%.

Based on the test conditions in Table 1, Equations 1-2 are applied. Figure 1 compares the laboratory test results and the correlation results and it shows the following:

- For the sensible effectiveness, the test results and the correlation results have quite large differences. The correlation results are approximately 10%-12% higher than the test results. On average, the difference is 11.87%.
For the latent effectiveness, the test results and the correlation results match very well except for Test 4, which shows an almost 5% difference.

Table 1: Laboratory test conditions

<table>
<thead>
<tr>
<th>Test no.</th>
<th>$T_{sl}$ (℃)</th>
<th>$\vartheta_{sl}$ (%)</th>
<th>$T_{st}$ (℃)</th>
<th>$\vartheta_{st}$ (%)</th>
<th>$V_{st}$ (m/s)</th>
<th>$Q_R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>37.8</td>
<td>42.4</td>
<td>21.1</td>
<td>56.1</td>
<td>3.30</td>
<td>1.0</td>
</tr>
<tr>
<td>2</td>
<td>32.2</td>
<td>50.3</td>
<td>21.1</td>
<td>56.1</td>
<td>3.30</td>
<td>1.0</td>
</tr>
<tr>
<td>3</td>
<td>37.8</td>
<td>42.4</td>
<td>21.1</td>
<td>56.1</td>
<td>2.31</td>
<td>1.0</td>
</tr>
<tr>
<td>4</td>
<td>37.8</td>
<td>49.5</td>
<td>23.9</td>
<td>58.8</td>
<td>3.30</td>
<td>1.0</td>
</tr>
<tr>
<td>5</td>
<td>37.8</td>
<td>29.4</td>
<td>21.1</td>
<td>48.3</td>
<td>3.30</td>
<td>1.0</td>
</tr>
<tr>
<td>6</td>
<td>26.7</td>
<td>54.4</td>
<td>20.0</td>
<td>54.8</td>
<td>3.30</td>
<td>1.0</td>
</tr>
<tr>
<td>7</td>
<td>23.9</td>
<td>78.3</td>
<td>18.3</td>
<td>66.0</td>
<td>3.30</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Because this is the first third-party effort to validate the correlations from Jeong and Mumma, it is not possible to find sources from the literature that directly explain why there is a large difference between the test results and the correlation results. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial model of energy wheels to develop the correlations. After some research, we think one possible reason could be the neglect of axial heat conduction when applying the numerical heat and transfer model of energy wheels to develop the correlations. Simonson and Besant (1999b) indicated that axial

$$\alpha_1 V_{st} \vartheta_{st} + \alpha_2 V_{st} T_{si} + \alpha_3 Q_R + \alpha_4 V_{st} T_{si} + \alpha_5 V_{st} \vartheta_{st} + \alpha_6 V_{sl} \vartheta_{sl} + \alpha_7 V_{st} Q_R + \alpha_8 T_{st} \vartheta_{st} + \alpha_9 T_{st} Q_R + \alpha_{10} \vartheta_{sl} Q_R + \alpha_{11} V_{st} T_{si} \vartheta_{sl} + \alpha_{12} V_{st} T_{si} Q_R + \alpha_{13} T_{st} \vartheta_{st} Q_R = 0.1187$$

Based on the modified correlation equation, the sensible effectiveness is updated as shown in Figure 2. This figure shows that the modified correlation results are very close with the test results. Equations 2 and 3 will be used in a whole-building simulation model to investigate the impact of operating conditions on energy simulation results.

Figure 2: Comparison of the sensible effectiveness between the test results and the results of modified sensible correlation after accounting for the axial conduction.

**Building Model**

A building model is needed to investigate the impact of effectiveness extrapolation on simulation results. Prototype building models have been developed by the Building Codes Program of the U.S. Department of Energy (DOE 2022). There are 16 prototype building models for different commercial building types such as office building, primary school, outpatient healthcare, and standalone retail. For each building type, a total of 19 EnergyPlus models exist for different climate zones, as defined in ASHRAE Standard 169. The standalone retail building model is selected to be used in this work.

The standalone retail building has a rectangular footprint (54.2 m × 42.4 m), one floor, a total floor area of 2298 m² and a floor-to-ceiling height of 6.1 m. The building has windows on its front side, which accounts for 25% of the front façade area. Regarding the construction, the building has a slab-on-grade floor, concrete block walls and a built-up roof on metal deck. The thermal performance of building envelope meets the minimum requirement of ASHRAE Standard 90.1-2019 (ASHRAE 2019b). Figure 3 shows an axonometric projection of the building.

Figure 3: The axonometric view of the modelled standalone retail building (DOE 2022).

The building is divided into five thermal zones (Figure 4), including back space, core retail, point of sale, front retail and front entry. These five thermal zones respectively account for 16.5%, 70%, 6.5%, 6.5% and 0.5% of the total
floor area. Internal loads (i.e., lighting, occupants, and plug loads) are modeled in these zones based on the relevant standards (e.g., ASHRAE Standard 90.1 and ASHRAE Standard 62.1) and typical practices.

![Diagram](https://example.com/diagram.png)

**Figure 4: Thermal zoning of the modelled standalone retail building (DOE 2022).**

Except for the front entry that is served by a unit heater to address the heating loads, all other thermal zones are served by single-zone packaged rooftop air-conditioning units. Each roof unit is equipped with a two-stage cooling, a multi-speed supply fan, and a gas furnace.

The original prototype model for standalone retail buildings has been developed for different climates. Each climate is represented by one representative location in the simulation model. Only the warm and humid climate (i.e., climate zone 3A in ASHRAE Standard 169 and 90.1), with Atlanta, GA being its representative location, is selected in this work. Major reasons behind this selection include the following:

- The warm and humid climate is appropriate for the use of energy recovery. This climate also covers many areas in the U.S. and therefore, the findings from this work can be widely applicable.
- The use of air-to-air energy recovery in warm and humid climate usually does not require frost controls. As to be discussed later in Section 4, customized programs need to be developed to capture the impact of operating conditions on the effectiveness of energy exchangers. Because developing customized programs in EnergyPlus is error prone and time consuming, it is wise to minimize the scope of customization by avoiding frost controls, which must be considered in cold climates.

**Modeling Air-to-Air Energy Recovery in EnergyPlus**

EnergyPlus has two models for air-to-air energy recovery (both sensible and latent heat exchange) between two air streams. These two models are HeatExchanger:AirToAir:SensibleAndLatent and HeatExchanger:Desiccant:BalancedFlow, which are called Objects in EnergyPlus models.

For the first model (Figure 5), the energy exchanger’s thermal performance can be specified by providing the sensible and latent effectiveness at 75% and 100% of the rated supply airflow rate at two standard operating conditions. These two operating conditions refer to the winter heating condition and the summer cooling condition, which were used by AHRI before 2018 to rate the thermal performance of energy exchangers.

To obtain the operating effectiveness of the energy exchanger, the model calculates the average volumetric airflow rates through the exchanger at every simulation time step and then applies linear interpolation or extrapolation to determine the actual operating effectiveness of the energy exchanger based on the effectiveness values at 100% flow and 75% flow, which are specified in the input, and the ratio between the actual and nominal airflow rates through the wheel. For example, the operating sensible effectiveness is calculated as (DOE 2021):

$$
\varepsilon_{s, operatingFlow} = \varepsilon_{s, 75\% Flow} \cdot \left( \frac{\epsilon_{s, 100\% Flow}}{\epsilon_{s, 75\% Flow}} \right) \cdot \frac{FR}{0.75} \cdot \frac{1}{1-0.75}
$$

where, $\varepsilon_s$ is the sensible effectiveness; $FR$ is the ratio of the average operating volumetric airflow rate to the nominal airflow rate; the subscripts $operatingFlow$, $75\% Flow$, and $100\% Flow$, respectively represents the operating flow condition, 75% of the nominal flow rate condition, and 100% of the nominal flow rate condition.

The above description indicates that the first model considers the impact of airflow rate on effectiveness, but it does not consider the impact of entering air properties (i.e., dry-bulb temperature and wet-bulb temperature) and unbalanced airflow on effectiveness. Because it is not possible to change the effectiveness according to entering air properties, the first model cannot be used in this work.

The second model, HeatExchanger:Desiccant:BalancedFlow, considers the exchanger performance through a performance data object type called HeatExchanger:Desiccant:BalancedFlow:PerformanceDataType1 (Figure 6). This performance data object type calculates the regeneration air outlet temperature and humidity ratio based on the predefined correlation equations while the equation coefficients can be user-specified. For example, the equation for calculating the dry-bulb temperature of regeneration outlet air is defined as (DOE 2021):

$$
RTO = b_1 + b_2 \cdot RWI + b_3 \cdot RTI + b_4 \cdot \frac{RWI}{RTI} + b_5 \cdot PWI + b_6 \cdot PTI + b_7 \cdot \frac{PWI}{PTI} + b_8 \cdot V
$$

where, $RTO$, $RTI$, and $PTI$ represent the dry-bulb temperature (°C) of the regeneration outlet air, the regeneration inlet air and the process inlet air, respectively; $RWI$, and $PW1$ represent the humidity ratio of the regeneration inlet air and the process inlet air, respectively.
Another equation with the same format as Eq. 5 but with different coefficients is used to calculate the regeneration outlet air humidity ratio. Even though the second model captures the impact of inlet air properties on exchanger performance, it cannot be used in this work for the following reasons. First, the roles of the two air streams may switch during annual simulation. For example, the exhaust air is regarded as the regeneration air in summer but as the process air in winter. Second, Eq. 5 is totally different from Eq. 2 and Eq. 3, which are the performance correlation equations to be studied.

Because both of the two authentic models for air-to-air energy recovery in EnergyPlus cannot be used to support this research, a new strategy must be developed. Fortunately, EnergyPlus provides a method for users to include customized models for HVAC components. The object Coil:UserDefined is used to model air-to-air energy recovery by using the two air connections: one connection for supply air and the other one for exhaust. Figure 7 presents the object Coil:UserDefined for the air-to-air energy recovery in the packaged system serving the core retail.

Customized programs using the Energy Management System (EMS) feature of EnergyPlus are then developed. Figure 8 shows the pseudocode of the implemented EMS program.
Energy Simulation and Results

The customized programs have been implemented for all four packaged single-zone systems in the standalone retail model. To investigate the impact of inlet air psychrometric conditions on energy simulation, two simulation models are developed. One simulation model uses the performance correlation equation (Eq. 2 and Eq. 3) to dynamically assign the exchanger’s effectiveness values based on the inlet air temperature and relative humidity at each simulation timestep. The second simulation model uses fixed effectiveness values which are calculated from Eq. 2 and Eq. 3 but based on the fixed standard operating conditions (Table 2). The winter condition is used when the wheel operates to recover heating while the summer conditions is used when the wheel operates to recover cooling. Both simulation models assume 1) the supply air and the exhaust air have balanced airflow, that is $Q_R = 1$; and 2) the inlet air face velocity is at the manufacturer-recommended value of 3.3 m/s (the same value used in the laboratory tests as discussed in Section 2), that is $V_{si} = 3.3$ m/s.

Table 2: The inlet air conditions used to calculate fixed effectiveness values.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Entering Supply Air</th>
<th>Entering Exhaust Air</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dry-bulb temperature (°C)</td>
<td>Wet-bulb temperature (°C)</td>
</tr>
<tr>
<td>Winter</td>
<td>1.7</td>
<td>0.6</td>
</tr>
<tr>
<td>Summer</td>
<td>35</td>
<td>25.6</td>
</tr>
</tbody>
</table>

Table 3: Comparison of the simulation results between the cases with fixed and dynamic effectiveness values. All cases in this table have $Q_R = 1$ and $V_{si} = 3.3$ m/s.

<table>
<thead>
<tr>
<th>System</th>
<th>Case</th>
<th>Energy consumption (MJ)</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Natural gas</td>
<td>Electricity</td>
</tr>
<tr>
<td>System 1</td>
<td>Dynamic effectiveness</td>
<td>16753</td>
<td>41142</td>
</tr>
<tr>
<td>(back space)</td>
<td>Fixed effectiveness</td>
<td>16740</td>
<td>41183</td>
</tr>
<tr>
<td>System 2</td>
<td>Dynamic effectiveness</td>
<td>9278</td>
<td>216111</td>
</tr>
<tr>
<td>(core retail)</td>
<td>Fixed effectiveness</td>
<td>9243</td>
<td>216177</td>
</tr>
<tr>
<td>System 3</td>
<td>Dynamic effectiveness</td>
<td>2597</td>
<td>30339</td>
</tr>
<tr>
<td>(point of sale)</td>
<td>Fixed effectiveness</td>
<td>2595</td>
<td>30410</td>
</tr>
<tr>
<td>System 4</td>
<td>Dynamic effectiveness</td>
<td>4649</td>
<td>32172</td>
</tr>
<tr>
<td>(front retail)</td>
<td>Fixed effectiveness</td>
<td>4639</td>
<td>32207</td>
</tr>
</tbody>
</table>

Table 3 summarizes the EnergyPlus simulation results for all four packaged single-zone systems. The results include the natural gas energy consumption for heating, the electricity consumption for cooling and fan, and the relative difference between the two cases (with and without the consideration of inlet air psychrometric conditions on exchanger effectiveness). This table shows that ignoring the impact of inlet air psychrometric conditions on exchanger effectiveness has minor impact (<0.5%) on both heating and cooling energy consumption. Because of the software constraint when performing the simulation study and the space limitation, we select System 4 only to investigate the impact of air face velocity ($V_{si}$) and airflow ratio of the two airstreams ($Q_R$) on energy simulation. The model that uses effectiveness correlations is perturbed sequentially in the following manner:

- The air face velocity is changed by -10% ($V_{si} = 2.97$ m/s), -5% ($V_{si} = 3.14$ m/s), 5% ($V_{si} = 3.47$ m/s), and 10% ($V_{si} = 3.63$ m/s) and, which
reflect the scenario that the energy wheel operates at an air face velocity different from the rated value. This scenario is commonly observed in the field because the system design supply airflow rate is likely different the nominal airflow rate specified by the wheel manufacturer. It is worth noting that for all cases of different air face velocities, the supply airflow rate is kept the same. The change of air face velocity is due to the selection of different wheel sizes.

- For each case of air face velocity, the airflow ratio is decreased by 5% ($Q_r = 0.95$) and 10% ($Q_r = 0.9$), which reflect the typical scenario that the exhaust airflow rate is less than the supply airflow rate due to air leakage in return ducts and local exhaust fans.

After running all cases with perturbed air face velocity, the results are shown in Figure 9 for electricity consumption of System 4 and Figure 10 for natural gas energy consumption. In these two figures, the base case has applied $V_s = 3.3$ m/s and all cases have $Q_r = 1.0$ in effectiveness correlations. These two figures indicate that for cases of the same $Q_r$, the energy consumption increases (decreases) as the air face velocity increases (decreases). This means that the recovered energy decreases with increasing air face velocity, which is expected. Increasing the air face velocity by 5% and 10% leads to an increase of natural gas energy consumption by 1.4% and 2.8% but negligible increase of electricity consumption (<0.2%). Decreasing the air face velocity by 5% and 10% leads to a reduction of natural gas energy by 1.3% and 2.6% and only a minor reduction of electricity consumption (0.2%~0.3%).

After running all cases with perturbed airflow ratio, the results are shown in Figure 11 for electricity consumption of System 4 and Figure 12 for natural gas energy consumption. These two figures indicate that for a given air face velocity, both electricity and natural gas energy consumption increase as the airflow ratio $Q_r$ decreases. This means that the recovered energy decreases as the airflow ratio between the exhaust air and the supply air decreases, which is expected. Decreasing $Q_r$ by 5% and 10% leads to a minor decrease of electricity consumption by 0.2% and 0.4% and an increase of natural gas energy consumption by ~5% and nearly 10%.

The reason of minor impact of perturbing airflow ratio and air face velocity on electricity consumption lies in the impact of the inlet air properties on the DX cooling efficiency. Typically, the efficiency decreases with the decreasing inlet air dry-bulb temperature and wet-bulb temperature. Therefore, if the energy recovery effectiveness is overestimated in the cooling mode, the entering air to the cooling coil will have lower temperature and humidity, which then lowers the cooling coil efficiency. This means that the overestimated energy recovery reduces the cooling load but it also reduces the DX cooling efficiency. Similarly, underestimated energy recovery increases the cooling load but it also increases the DX cooling efficiency. Thus, the overall impact on cooling electricity is small. In contrast, for heating, the inlet air properties have no impact on the furnace efficiency, which explains the bigger impact of energy recovery effectiveness on natural gas energy consumption.

![Figure 9: The impact of air velocity faced by the energy wheel on the packaged system electricity consumption. All cases have balanced airflow rates and account for the impact of inlet air temperature and relative humidity on wheel effectiveness.](image)

![Figure 10: The impact of air velocity faced by the energy wheel on the packaged system natural gas consumption. All cases have balanced airflow rates and account for the impact of inlet air temperature and relative humidity on wheel effectiveness.](image)

![Figure 11: The impact of airflow ratio on the packaged system electricity consumption. All cases account for the impact of inlet air temperature and relative humidity on wheel effectiveness.](image)
Figure 12: The impact of airflow ratio on the packaged system natural gas consumption. All cases account for the impact of inlet air temperature and relative humidity on wheel effectiveness.

Conclusions

Based on the laboratory test results for an energy wheel, a set of correlation equations from the literature are verified or modified for use in the simulation. The EnergyPlus model for a standalone retail building is used to investigate the impact of operating conditions on simulation results. Major conclusions include the following:

- Ignoring the impact of inlet air temperature and humidity on exchanger effectiveness had minor impact (<0.5%) on both heating and cooling energy consumption.
- The deviation of operating air face velocity from its nominal value has a noticeable but small impact on both heating and cooling energy consumption. A 10% deviation of the air face velocity could deviate the heating energy consumption by 2.8% and the cooling energy consumption by 0.3%.
- Relative to the assumption of balanced flow rates between the two air streams, decreasing the airflow ratio \( Q_r \) by 10% led to an increase of electricity consumption by 0.4% and an increase of natural gas energy consumption by nearly 10%.

This work could be extended in future by 1) expanding the investigation of airflow ratio and air face velocity to all four systems of the standalone retail model, 2) exploring the impact of effectiveness extrapolation in other building types and climates, and 3) developing effectiveness correlations and studying their impact on other energy exchanger types such as plates.

References


