Energy conservation potential of an indirect and direct evaporative cooling assisted 100% outdoor air system
MH Kim, JH Kim, OH Kwon, AS Choi and JW Jeong
BUILDING SERV ENG RES TECHNOL published online 28 June 2011
DOI: 10.1177/0143624411402637

The online version of this article can be found at:
http://bse.sagepub.com/content/early/2011/06/27/0143624411402637

A more recent version of this article was published on - Nov 2, 2011
Energy conservation potential of an indirect and direct evaporative cooling assisted 100% outdoor air system

MH Kim MSc, JH Kim MSc, OH Kwon MSc, AS Choi PhD and JW Jeong PhD MASHRAE
Department of Architectural Engineering, Sejong University, Seoul, 143-747, Korea

This study aims to present the fundamentals in which operation of a 100% outdoor air system integrates with indirect and direct evaporative cooling systems and to estimate its energy saving potential. The simulation of the proposed system is performed using a commercial equation solver program, and the annual operation energy saving potential with respect to a conventional variable air volume system is determined. This paper shows that significant operation energy savings (i.e. 21–51% less energy consumption) is possible principally by the pre-conditioning of supply air due to the waste heat recovery using the indirect evaporative cooler and the sensible heat exchanger units. By components, the proposed system shows a 16–25% less annual cooling coil load and an 80–87% reduced annual heating coil load with respect to the conventional variable air volume system, while there is no fan energy savings expected.

Practical applications: This paper provides practical insight on how the evaporative cooling based 100% outdoor air system operates and how each essential component, such as the indirect evaporative cooler, cooling coil, direct evaporative cooler, heating coil and sensible heat exchanger should be controlled during the seasons for realising energy conservation benefits. The sequence of operation presented in this paper can be implemented to actual control logic.

1 Introduction

Examining the latest trends of research for reducing energy consumption in buildings, focus lies in the following three areas: (1) use of renewable energy technology for buildings; (2) enhancing thermal performance of passive building elements, such as building insulation, windows and envelope materials; and (3) development of a high performance heating, ventilating, and air-conditioning (HVAC) system, such as a desiccant cooling system, a geothermal or water source heat pump, a variety of waste heat recovery, optimum system control and maintenance technologies.

For the last decade, research related to high performance HVAC systems have focused on developing the next generation HVAC systems that not only minimise energy consumption but also provide a comfortable, healthy indoor environment. An evaporative cooling system1–4, simultaneously pursuing environmental-friendliness and energy conservation, is one of them.

Cooling systems using the latent evaporative heat of water have been principally known as the suitable system in arid regions. However, open literature5,6 indicates that economic cooling could also be obtained in
Evaporative cooling based 100% outdoor air conditioning system

2.1 System configuration

As shown in Figure 1, the system proposed in this research is composed of an IEC, a cooling coil, and a direct evaporative cooler (DEC) at the supply air side. A heating coil and a sensible heat exchanger are located at the exhaust air side of the system. A double duct or a multizone system is applied in order to handle different thermal environment demands in each room. The supply air volume is adjusted depending on load variation like a VAV system.

2.2 Operating scheme of the system

During the cooling season, hot and humid outdoor air is primarily cooled by the IEC. Then, additional cooling and dehumidification is performed by the cooling coil to satisfy the cold deck supply air temperature and humidity ratio setpoint. The IEC consists of two channels: one is a dry (or primary) channel where the supply air passes and the other is a wet (or secondary) channel where the water is injected to the scavenger air. In principle, the cooling effect in the dry channel is enhanced when the wet bulb temperature of the scavenger air passing through the wet channel is low. Therefore, in the proposed system, the air with lower wet bulb temperature between the outside air and the exhaust air is selected as the scavenger air.

In intermediate seasons when the outside air is relatively dry, the cooling coil load could be additionally reduced by operating the DEC. The supply air passed through the DEC is divided into the cold deck and the neutral deck. The neutral deck supply air temperature setpoint is met by recovering waste heat from the exhaust air side via the sensible heat exchanger. If it is difficult to maintain neutral deck temperature setpoint with only recovered sensible heat, additional
heat is supplied through the installed heating coil.

The air of the cold deck and/or neutral deck is selected by the terminal box in the duel duct system depending on the indoor load condition or the occupants’ needs. In the multizone system, the air from the cold deck and neutral deck are mixed in the mixing box at required temperature before delivery to spaces served by the unit.

2.3 Modes of operation

The operating modes of the 100% outside air system with the indirect and direct evaporative cooler can be more effectively understood on the psychrometric chart. As shown in Figure 2, the psychrometric chart is divided into four regions (i.e. Regions A, B, C and D) depending on the outdoor air condition. In each region, the system follows a specific mode out of four different modes of operation.

2.3.1 Operating mode 1

Region A refers to the climate condition in which the outside air dry bulb temperature is higher than 13°C and the humidity ratio is more than 9.37 g/kg (Figure 2). In this region, the cooling coil leaving air temperature is maintained at 13°C (i.e. cold deck setpoint), and the IEC operates at maximum effectiveness in order to reduce the cooling coil load as much as possible. In order to maximise the cooling effect of the IEC, the scavenger air passing through the secondary channel is selected as the air which has a lower wet bulb temperature between outside air and exhaust air. Greater evaporative cooling effect can be obtained at the primary channel as the wet bulb temperature of the air supplied to the secondary channel is low.

The IEC works mostly as a sensible cooling device for precooling the primary air (Figure 3(a)). However, it can also provide latent cooling when the secondary channel wet bulb temperature is lower than the dew point temperature of the primary channel, and condensation occurs on the primary channel surface (Figure 3(b)). After the IEC, the target supply air temperature (i.e. 13°C) and the humidity ratio (i.e. 9.37 g/kg) are satisfied by the cooling coil.

Once the 13°C of supply air is acquired through the IEC and the cooling coil, it is supplied through the cold deck and the neutral deck. The air in the neutral deck is reheated at neutral temperature via the sensible heat exchanger (SHE) when it is required.

2.3.2 Operating mode 2-a and 2-b

Region B (Figure 2) refers to the case in which the absolute humidity of the outside air
is less than 9.37 g/kg and its enthalpy is more than 36.7 kJ/kg.

- **Operating mode 2-a**: As shown in Figure 4(a), if the supply air does not reach 36.7 kJ/kg of enthalpy even after cooling through the IEC, it is cooled to the target enthalpy (i.e. 36.7 kJ/kg) by the cooling coil. Then, the DEC performs the cooling along 36.7 kJ/kg of enthalpy line until the supply air temperature reaches 13°C.

- **Operating mode 2-b**: If the enthalpy of the primary air is less than 36.7 kJ/kg after the IEC, the target dry bulb temperature (i.e. 13°C) is met by the DEC without operating the cooling coil (Fig. 4(b)).

The primary air at 13°C is passed through the cold deck as it is, and the SHE may reheat some of it.

2.3.3 **Operating mode 3**

Region C (Figure 5) refers to the case in which the outside air temperature is higher than 13°C and the enthalpy is less than 36.7 kJ/kg. In this region, the outdoor air can be cooled to 13°C, the target supply air temperature, only by the DEC without operating the IEC and the cooling coil.

2.3.4 **Operating mode 4-a and 4-b**

Region D on the psychrometric chart (Figure 6) refers to the case when the outside air temperature is less than 13°C. The sensible heat is recovered from the return air stream through the SHE in order to keep the neutral deck air temperature around 20°C, the neutral temperature setpoint. On the other hand, since the IEC works as a sensible heat exchanger when the secondary channel runs dry, the primary air passing the IEC is heated to 13°C by the sensible heat recovered from the exhaust air stream. If recovered, sensible heat is not enough to keep both primary air and neutral deck air temperature setpoints, the heating coil supplies make up heat. Until now, based on described operating method according to the outdoor air condition,
Figure 3 Psychrometrics in Region A: (a) no condensing at the primary channel of the IEC, (b) condensing at the primary channel of the IEC
summarised modes of operation are presented in Table 1.

- **Operating mode 4-a**: The supply air temperature passed through the IEC reaches 13°C without operating the heating coil.
- **Operating mode 4-b**: The supply air temperature through the IEC reaches 13°C by operating the heating coil.

**Figure 4** Psychrometrics in Region B: (a) operating mode 2-a, (b) operating mode 2-b
Figure 5 Psychrometrics in Region C

Figure 6 Psychrometrics in Region D
3 Energy simulation

3.1 Simulation model

Estimation of annual operating energy consumption is performed for the two different systems, the evaporative cooling based 100% outdoor air system (Figure 1) and the VAV system (Figure 7) serving four classrooms in a school building. Then, the energy saving potential of the outdoor air conditioning system over the conventional VAV system are predicted quantitatively.

3.1.1 Outline of model building

The model building is a school, and four classrooms facing the south are selected for the energy simulation. Each classroom has 60-m² floor area and accommodates 30 students. The average value of overall heat transfer coefficient (i.e. U-value) and the envelope area product (i.e. UA value) is 120 W/°C; the UA value of the roof slab is 17 W/°C; and the equipment heat density is 22 W/m². All the classrooms are regularly used from 8AM to 7PM for 5 days a week, and the required ventilation is established as 17 m³/h per person and 2.2 m³/h per unit floor area with respect to ASHRAE Standard 62.1-2007.

3.1.2 Operating scenarios

It is assumed that both systems operate for 12 different operating scenarios generated by changing the number of occupants and the solar load in each classroom as shown in Table 2. Annual operating energy consumption of the proposed system and the VAV system are calculated using the BIN method.

Table 1 Modes of operation

<table>
<thead>
<tr>
<th>Region</th>
<th>Operating mode</th>
<th>IEC</th>
<th>Cooling coil</th>
<th>DEC</th>
<th>Heating coil</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1</td>
<td>ON</td>
<td>ON (outlet Temperature: 13 °C)</td>
<td>OFF</td>
<td>OFF</td>
</tr>
<tr>
<td>B</td>
<td>2-a</td>
<td>ON</td>
<td>ON (outlet enthalpy: 36.7 kJ/kg)</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td></td>
<td>2-b</td>
<td>ON</td>
<td>OFF</td>
<td>ON (outlet temperature: 13 °C)</td>
<td>OFF</td>
</tr>
<tr>
<td>C</td>
<td>3</td>
<td>OFF</td>
<td>OFF</td>
<td>ON (outlet temperature: 13 °C)</td>
<td>OFF</td>
</tr>
<tr>
<td>D</td>
<td>4-a</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>OFF</td>
</tr>
<tr>
<td></td>
<td>4-b</td>
<td>ON</td>
<td>OFF</td>
<td>ON (IEC outlet temperature: 13 °C)</td>
<td></td>
</tr>
</tbody>
</table>
In both systems, the indoor temperature is set to 24°C for cooling, 20°C for heating. The occupancy rate is selected between 25–100%, and the solar load is set to 100%, 50% and 10% of the maximum solar radiation load (2.93 kW) estimated for each room.

### 3.1.3 Operating condition of the proposed system

The evaporative cooling based 100% outdoor air system shown in Figure 1 could be operated with a variety of operating conditions determined by the climate conditions and the building uses. However, this research establishes the following operating conditions for the simulation of this system, which could be generally applied in conventional buildings.

- **IEC**: If the wet bulb temperature of outdoor air is higher than 13°C, the water is injected to the wet channel. The air with lower wet bulb temperature between outside air and exhaust air is selected as scavenger air passing through the wet channel for maximising the sensible cooling effect in the dry channel. If the dry bulb temperature of the outside air is less than 13°C, the wet channel operates without injecting the water and the amount of scavenger air is modulated to keep the dry channel leaving air temperature at 13°C.
- **Cooling coil**: If the humidity ratio at the coil inlet is higher than 9.37 g/kg (i.e. 13°C saturated condition), the coil leaving air temperature should be maintained at 13°C. If the coil inlet humidity ratio is less than 9.37 g/kg and the enthalpy is more than 36.7 kJ/kg (i.e. 13°C saturated condition), the cooling coil should be modulated to maintain the enthalpy of the coil leaving air to 36.7 kJ/kg.
- **DEC**: When the enthalpy of the DEC inlet air is less than 36.7 kJ/kg and the dry bulb temperature is more than 13°C, the DEC unit maintains the outlet air temperature at 13°C. At any other inlet air condition, the DEC unit should not be activated.
- **SHE**: The neural deck supply air temperature should be kept around 20°C by recovering sensible heat from the exhaust air using the SHE with the bypass damper.
- **Heating coil**: If all exhaust air is passed through the SHE, but the neutral deck temperature is less than 20°C, the exhaust air is heated by the heating coil to maintain the neutral deck air temperature setpoint. In addition, when all exhaust air passes through the wet channel of the IEC without sprayed water, but the IEC leaving supply air temperature is less than 13°C, the exhaust air should be additionally heated by the heating coil to keep the IEC leaving supply air temperature setpoint (i.e. 13°C).
- **Fans**: They are controlled as the variable air volume depending on indoor sensible heat load. The minimum amount of supply air for each room is established as the required amount of ventilation recommended by the ASHRAE Standard 62.1-2007.

### Table 2 Operating scenarios

<table>
<thead>
<tr>
<th>Scenario</th>
<th>Occupancy rate (%)</th>
<th>Solar load (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Room 1</td>
<td>Room 2</td>
</tr>
<tr>
<td>S1</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S2</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>S3</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S4</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>S5</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S6</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>S7</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S8</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>S9</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S10</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>S11</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>S12</td>
<td>80</td>
<td>80</td>
</tr>
</tbody>
</table>
3.1.4 Operating conditions of VAV system
The operating conditions of the VAV system (Figure 7), considering the general operating conditions of conventional VAV systems, are established as follows.

- **Cooling coil**: If the dry bulb temperature at the coil inlet is higher than 13°C, the supply air temperature setpoint of 13°C is maintained by the cooling coil.
- **Heating coil**: If the air temperature at the heating coil inlet is less than 13°C, it is operated to keep the established temperature of 13°C at the coil outlet. However, the heating coil should not operate with the cooling coil.
- **Fans**: The supply air volume is controlled depending on indoor sensible heat load. The minimum amount of supply air for each room is established as the required amount of ventilation recommended by the ASHRAE Standard 62.1-2007.
- **Air-side economiser**: If the enthalpy of outside air is lower than the enthalpy of the exhaust air and the dry bulb temperature of the outside air is more than 13°C, the return damper is completely closed and the air conditioning is done by 100% outside air. If the outside air temperature is less than 13°C, the outside damper and return damper are modulated to the temperature of mixed air as 13°C.

3.1.5 Outside air BIN data
The BIN data of outside air is generated at the dry bulb temperature interval of 1°C during the system operating period. The standard meteorological data for Seoul, South Korea, provided by the Korean Solar Energy Society, is used for generating the BIN data.

The proposed system and the VAV system are simulated under the given operating scenarios by modelling each system using the EES program, enabling mathematical modelling and analysis of various thermal systems. And then, annual heating and cooling energy consumption of both systems are compared to each other to estimate the energy saving potential of the proposed system with respect to the conventional VAV system. For calculating the fan energy consumption, the static pressure of the supply fan and the exhaust fan are assumed as 1.25 and 0.75 kPa, respectively.

3.1.6 Effectiveness of IEC, DEC and SHE
The operating performance of proposed system is mostly affected by the effectiveness of the heat exchange components including IEC, DEC and SHE. The effectiveness of IEC varies depending on whether the wet channel is operated under wet (i.e. with sprayed water) or dry (i.e. without sprayed water) conditions. Condensation, occasionally formed in the dry channel, also affects the IEC effectiveness. When the water is sprayed in the wet channel and the wet bulb temperature of the scavenger air is lower than the dew point temperature of outside air passing through the dry channel, one may expect that condensation will occur in the dry channel and the effectiveness to increase about 5%. In Table 3, the effectiveness values of three different manufacturers’ IEC units are presented for the possible combinations of dry channel and wet channel operating conditions. The three IEC units selected represent ‘the best’, ‘the normal’ and ‘the worst’ performance unit one may find in the market. They are used for predicting IEC leaving supply air conditions in the simulation of the proposed system and show the impact of the effectiveness value on the performance of the whole system.

The IEC leaving primary air temperature is estimated by Equation (1) or Equation (2) based on the operating condition of the wet channel (i.e. wet or dry operation).

\[
DBT_{p,o} = DBT_{p,i} - \varepsilon_{IEC}(DBT_{p,i} - WBT_{s,i})
\]  

[Wet operation]  

\[
DBT_{p,o} = DBT_{p,i} - \varepsilon_{DEC}(DBT_{p,i} - WBT_{s,i})
\]  

[Dry operation]  

\[
DBT_{p,o} = DBT_{p,i} - \varepsilon_{SHE}(DBT_{p,i} - ST_{s,i})
\]  

[Supply air condition]  

DBT, o: IEC leaving primary air temperature
DBT, i: Primary air temperature
DBT, s: Supply air temperature
WBT, s: Supply air wet bulb temperature
ST, s: Supply air wet bulb temperature

[Equation 1]
where $\varepsilon_{IEC}$ is the effectiveness of IEC under wet or dry operation in the wet channel, $DBT_{p,i}$, the IEC inlet primary air dry-bulb temperature (°C), $DBT_{p,o}$, the IEC leaving primary air dry-bulb temperature (°C), $DBT_{s,i}$, the IEC inlet secondary air dry-bulb temperature (°C) and $WBT_{s,i}$, the IEC inlet secondary air wet-bulb temperature (°C).

It is well known that a commercial DEC unit generally has 70–80% effectiveness. This research assumes the effectiveness of the DEC to be 70% to perform a simulation. In addition, it is supposed that the effectiveness of the SHE reclaiming the sensible heat from the exhaust air is also 60%, which is a general value. The DEC leaving supply air temperature is determined by Equation (3).

$$DBT_{p,o} = DBT_{p,i} - \varepsilon_{DEC}(DBT_{p,i} - WBT_{s,i})$$ 
[Dry operation] (2)

where $\varepsilon_{DEC}$ is the effectiveness of DEC unit, $DBT_{p,i}$, the DEC inlet supply air dry-bulb temperature (°C) and $WBT_{s,i}$, the DEC inlet supply air wet-bulb temperature (°C).

$$DBT_o = DBT_i - \varepsilon_{DEC}(DBT_i - WBT_i)$$ (3)

4 Simulation Results

4.1 Comparison of operating energy consumption

In Figure 8, the annual cooling coil load, heating coil load, and fan energy consumption of the proposed system and the conventional VAV system, acquired from the simulation, are compared for each operating scenario.

As shown in Figure 8(a), the VAV system may have higher annual cooling coil load in every operating scenario compared with the proposed system. The lower annual cooling coil load in the proposed system is principally due to the pre-cooling effect of the IEC. A greater reduction of cooling coil load can be expected as the effectiveness of IEC increases. In the proposed system with ‘A’ manufacturer’s IEC, which has the highest effectiveness, the annual cooling coil load is reduced by 25% on average compared with the conventional VAV system. In the cases with ‘B’ and ‘C’ manufacturers’ IECs, the annual cooling coil loads are reduced by 21% and 16%, respectively.

The benefit of the IEC is very significant in annual heating coil load reduction (Figure 8(b)). Because the IEC reclams waste sensible heat from the exhaust air during the heating season, one may expect significant annual heating coil load reduction with respect to the conventional VAV system in every operating scenario.

On average, the proposed system with ‘A’ manufacturer’s IEC shows a reduced annual heating coil load of 87% compared with the VAV system. In other cases with the ‘B’ and ‘C’ manufacturers,’ IEC units also provide about 80% reduction of the annual heating coil load.

As for the annual fan energy consumption (Figure 8(c)), one may see that there is no recognisable fan energy reduction in the proposed system. Since the supply air volume of the proposed system

---

**Table 3 IEC effectiveness**

<table>
<thead>
<tr>
<th>Manufacture</th>
<th>Wet channel operating condition</th>
<th>Dry channel operating condition</th>
<th>Heat exchange effectiveness (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Wet</td>
<td>Wet</td>
<td>85</td>
</tr>
<tr>
<td></td>
<td>Wet</td>
<td>Dry</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>Dry</td>
<td>Dry</td>
<td>70</td>
</tr>
<tr>
<td>B</td>
<td>Wet</td>
<td>Wet</td>
<td>75</td>
</tr>
<tr>
<td></td>
<td>Wet</td>
<td>Dry</td>
<td>70</td>
</tr>
<tr>
<td></td>
<td>Dry</td>
<td>Dry</td>
<td>60</td>
</tr>
<tr>
<td>C</td>
<td>Wet</td>
<td>Wet</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>Wet</td>
<td>Dry</td>
<td>55</td>
</tr>
<tr>
<td></td>
<td>Dry</td>
<td>Dry</td>
<td>60</td>
</tr>
</tbody>
</table>

aWater sprayed in wet channel: Yes – Wet, No – Dry.

bCondensation in dry channel: Yes – Wet, No – Dry.
is modulated depending on the internal thermal load like the conventional VAV system, there cannot be any significant difference in the fan energy consumption between the systems.

Consequently, the annual operating energy reduction in the evaporative cooling based 100% outdoor air system is principally caused by the pre-conditioning of the primary air through the IEC.
Figure 9 shows the ratio of the total operating energy consumption of the proposed system with three different IEC units to the VAV system for each operating scenario. One may see that the proposed system requires about 72–79% operating energy of the VAV system under scenario 1 (S1), which is the highest indoor thermal load condition and about 49–56% under scenario 12 (S12) which is the lowest indoor thermal load condition. This means that 21–51% of the operating energy can be saved by the proposed system with respect to the conventional VAV system.

One may also find that the energy saving potential of the proposed system is enhanced when the effectiveness of IEC unit increases. In each operating scenario, the case with ‘A’ manufacturer’s IEC unit shows more improved operating energy savings, at 6–10%, with respect to the case with ‘C’ manufacturer’s unit.

In the proposed system, additional pumps, which are not required in the conventional system, need to be used in the evaporative cooler side of the system, and they have impact on the energy saving potential. However, the conventional system consumes more chilled water and cooling water pumping energy because of the higher cooling coil load with respect to the proposed system. It is a trade-off of pumping energy between the two systems, so this paper discusses the thermodynamic energy requirement only.

### 4.2 Cooling contribution ratio of each system component

In the proposed system, essential components for cooling and dehumidifying the supply air during the cooling season are the IEC, cooling coil and DEC. Figure 10 shows the contribution ratio of these essential components for cooling the supply air under each operating scenario. One can see that the cooling coil provides the largest contribution for all operating scenarios, and the IEC follows it. The contribution ratio of DEC is not significant because the period satisfying
the DEC’s operating condition mentioned in section 3.1 is relatively short.

By analysing the results shown in Figure 10, acquired from the simulation of the proposed system with ‘A’ manufacturer’s IEC, it can be found that the IEC deals with about 30% of the total cooling on average. The cooling coil and the DEC provide about 67% and 3% of the total cooling, respectively. This observation is also valid for the system with ‘B’ or ‘C’ manufacturer’s IEC.

4.3 Heating contribution ratio of each component

During the heating season, the proposed system reduces the operating load of heating coil required to maintain the supply air temperature setpoint by reclaiming the sensible heat from the exhaust air through the IEC. However, since the proposed system supplies only the minimum supply air volume required for ventilation like a conventional air conditioning system in the winter, the indoor temperature may not be maintained only by the supply air volume; so the operation of additional heating device is required. This is called parallel heating, and it is added into the total heating coil load in Section 4.1. This heating coil load is separated into the proposed system and the parallel heating device, and the heating contribution ratio of each system component is estimated.

Figure 11 shows the contribution ratio of each system component for maintaining the supply air temperature and the indoor temperature setpoints during the heating season under each operating scenario. It is found that the IEC takes the most significant role during the heating period.

By analysing the results shown in Figure 11 acquired from the simulation of the proposed system with ‘A’ manufacturer’s IEC, one may see that the IEC accommodates about 79% of the total heating on average by reclaiming the sensible heat from the exhaust air. The heating coil and the parallel heating device deal with 14% and 7% of the total heating, respectively.

This observation is also valid in the proposed system with ‘B’ or ‘C’ manufacturer’s IEC. The DEC should be off during the heating season so it does not contribute to the heating.
5 Conclusion

It is usually assumed that a high efficient air conditioning system consuming less energy and more environmental benefits with respect to a conventional system would be realised by implementing expensive, advanced new technologies. However, this research showed that excellent energy saving potential could be expected from the evaporative cooling based 100% outdoor air system consisting of conventional air conditioning devices, such as IEC, DEC and SHE.

In the proposed system, significant operation energy savings are principally caused by pre-conditioning (i.e. pre-cooling and pre-heating) of supply air due to the sensible heat exchange effect of IEC and SHE units. The proposed system shows 16–25% less annual cooling coil load and 80–87% reduced annual heating coil load with respect to the conventional VAV system. However, there is no particular fan energy reduction factor in the proposed system compared with the conventional VAV system.

Acknowledgement

This work was supported by Mid-career Researcher Program through NRF grant No. 2010-0000298 funded by the Korea Ministry of Education, Science and Technology.

References


