The Office Air Handling Unit versus the Two Dedicated Air Handling Unit System

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ABSTRACT

Both analytical and the numerical methods have been developed to compares the energy performance of the OAHU systems and the two-AHU (TAHU) system. The OAHU system saves up to 1.85 kilojoules heating energy for each kilogram air supplied to the building, or 0.8 Btu for each pound air supplied to the building when the outside air dew point temperature is less than the room dew point temperature. When the outside air dew point temperature is higher than the room dew point temperature, the OAHU system saves up to 4.65 kilojoules heating and cooling energy for each kilogram air supplied to the building, or 2 Btu for each pound air supplied to the building. Both OAHU and TAHU consume the same amount of fan energy. The paper presents the system models, the simulation procedures and the simulation results.

INTRODUCTION

HVAC systems of large commercial buildings consume energy in excess of the sum total of the building loads. This excessive energy use is due to the fact that a single air-handling unit in a HVAC system, having to provide conditioned air at low humidity ratio and different supply temperatures to multiple zones in the building, can do so only by resorting to (a) low cold deck air temperature and (b) a certain amount of mixing of cold and hot air stream in dual-duct systems or to terminal reheating as in single duct systems. EDE, called energy delivery efficiency, was developed to rate the energy performance of HVAC systems on an absolute scale [1]. The study found that the energy consumptions of typical existing systems are twice higher than the summation of the heating and cooling loads even the systems are operated using optimized schedules [2]. The energy performance of HVAC system can be improved significantly by developing new AHU systems.

This paper compares the OAHU system, which was developed for large commercial buildings [3], with the TAHU system. The OAHU system supplies all outside air to the interior zone and recycle the return air from the interior zone to the exterior zone. It allows setting the supply air temperature based on the exterior zone sensible load only without humidity problems. It reduces or even eliminates reheat energy consumption. The TAHU system has one unit for the interior zone and the other for the exterior zone. This paper presents the analytical system models and the simulated thermal energy performance. An engineering method for evaluating the potential savings is also presented in the paper.

SYSTEM MODELS

This section presents the system principles, the optimal operation and control schedules, and the energy models. The fan energy models are not included since both systems consume the same fan energy under the optimal operation and control schedules.

<u>OAHU system</u>

Figure 1 shows the schematic of the OAHU system. The constant interior zone load is assumed. If the interior zone airflow (m_i) is higher than the total outside air demand for the entire building ($m \cdot \beta_{o,set}$) and the outside air dew point temperature is higher than the room dew point temperature, the OAHU system supplies all outside air to the interior zone. The exterior zone unit uses 100% return air. Consequently, the exterior zone supply air temperature ($T_{c,e}$) can be reset based on zone load ($\gamma_e \cdot Q_{e,d}$) only to minimize the reheat, as shown in Equation (1).

$$T_{c,e} = Tr - \frac{\gamma_e \cdot Q_{e,d}}{m_e \cdot C_p} \qquad (1)$$

When the outside air enthalpy (h_o) is higher than the return air enthalpy (h_r), the OAHU system supplies minimum outside airflow to the building. Otherwise, the OAHU system supplies more outside air to decrease the mechanical cooling. Equation (2) and (3) present the optimal interior zone and exterior zone outside air intake (β_i, β_e) schedules.

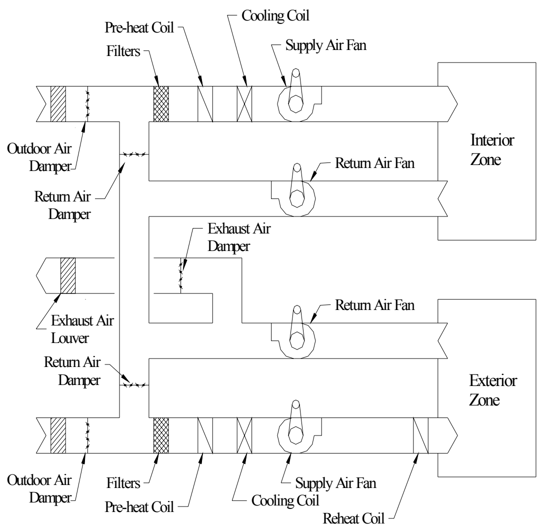


Figure 1: Schematic of the OAHU System

$$\beta_i = \begin{cases} 1 & (T_o \geq T_{c,i} \,\&\, h_o \leq h_r) \\ \max\left(\beta_{o,set}, \dfrac{(T_r - T_{c,i})}{(T_r - T_o)}\right) & (h_o > h_r) \\ \max\left(\beta_{i,\min}, \dfrac{(T_r - T_{c,i})}{(T_r - T_o)}\right) & (T_o < T_{c,i}) \end{cases} \quad (2)$$

$$\beta_e = \begin{cases} 1 & (T_o > T_{c,e} \,\&\, h_o \leq h_r) \\ \max\left(\beta_{e,\min}, \dfrac{(T_r - T_{c,e})}{(T_r - T_o)}\right) & others \end{cases} \quad (3)$$

where $\beta_{e,\min} = \max\left(\dfrac{\beta_{o,set} - \varphi \cdot \beta_i}{1-\varphi}, 0\right)$.

If the interior zone airflow rate (m_i) is less than the total outside air requirement ($m \cdot \beta_{o,set}$), some of the outside air must be directly introduced to the exterior zone. The cold air temperature of the exterior zone ($T_{c,e}$) is kept at constant level to maintain required room humidity load if the outside air dew point temperature is higher than the room air dew point temperature. The outside air intake of the interior zone and the exterior zone is determined by Equation (4) and (5) respectively.

$$\beta_i = \min\left(1, \frac{m_{o,set}}{m_i}\right) \quad (4)$$

$$\beta_{e,\min} = \max\left(\frac{\beta_{o,set} - \varphi \cdot \beta_i}{1-\varphi}, 0\right) \quad (5)$$

If the outside air dew point temperature is less than the room dew point temperature, the outside air intake is determined based on the economizer principals.

The energy consumption can be calculated using Equations (6) ~ (10). When the airflow rates (m_i, m_e) are given, the mixed air conditions ($T_{mix,i}, T_{mix,e}, h_{mix,i}, h_{mix,e}$) are determined based on the optimal outside air intake ratio, the weather condition and the room air condition.

$$E_{ph,i} = \max\left(C_p \cdot m_i \cdot (T_{c,i} - T_{mix,i}), 0\right) \quad (6)$$

$$E_{cc,i} = \max\left(C_p \cdot m_i \cdot (T_{mixi} - T_{c,i}), m_i \cdot (h_{mixi} - h_{c,i}), 0\right) \quad (7)$$

$$E_{cc,e} = \max\left\{m_e \cdot C_p \cdot (T_{mix,e} - T_{c,e}), m_e \cdot (h_{mix,e} - h_{c,e}), 0\right\} \quad (8)$$

$$E_{ph,e} = \max\left\{m_e \cdot C_p \cdot (T_{c,e} - T_{mix,e}), 0\right\} \quad (9)$$

$$E_{rh,e} = \max\left\{m_e \cdot C_p \cdot (T_{s,e} - T_{c,e}), 0\right\} \quad (10)$$

where $T_{mix} = \beta \cdot T_o + (1-\beta) \cdot T_r$ (11)

$h_{mix} = \beta \cdot h_o + (1-\beta) \cdot h_r$ (12)

The total cooling/heating energy consumptions (E_{cc} / E_h) are the summation of the interior zone and exterior zone cooling/heating consumptions.

$$E_{cc} = E_{cc,i} + E_{cc,e} \quad (13)$$

$$E_h = E_{ph,i} + E_{ph,e} + E_{rh,e} \quad (14)$$

<u>TAHU System</u>

Figure 2 shows the schematic of the TAHU system, which consists of two independent AHU units, one serves the interior zone and one serves the exterior zone.

For the interior zone, the cold air temperature ($T_{c,i}$) is assumed to be 12.8ºC (55ºF). For the exterior zone, the TAHU system resets the cold air temperature ($T_{c,e}^b$) based on the zone sensible load and the humidity control. When the outside air dew point temperature ($T_{o,dew}$) is higher than the room air dew point temperature ($T_{r,dew}$), the cold air temperature is 12.8ºC (55ºF). When the outside air dew point temperature is lower than the room air dew point temperature, the cold air temperature is reset according to the zone sensible load, as shown in Equation (1). Equation (15) shows the exterior zone cold air temperature reset schedule.

$$T_{c,e}^{b} = \begin{cases} \max\left(T_{c,i}, T_{c,e}\right) & T_{o,dew} \le T_{r,dew} \\ T_{c,i} & T_{o,dew} > T_{r,dew} \end{cases} \qquad (15)$$

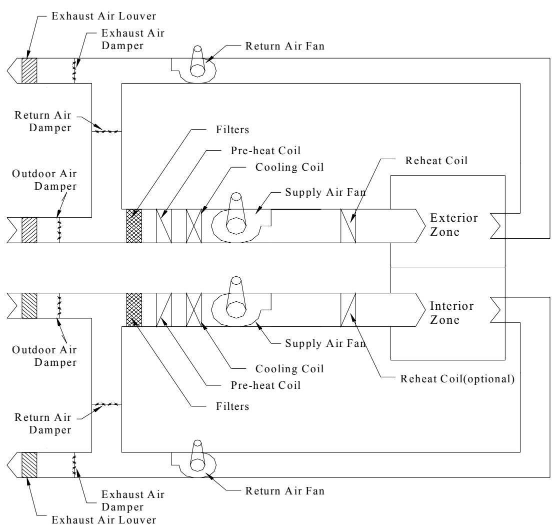


Figure 2: Schematic of the TAHU System

The interior zone and exterior zone outside air intake (β_i^b and β_e^b) is controlled by the enthalpy economizer as shown in Equation (16) and (17).

$$\beta_i^b = \begin{cases} \beta_{o,set} & \left(h_o > h_r\right) \\ 1 & \left(h_o \le h_r and T_o > T_{c,i}\right) \\ \max\left(\beta_{o,set}, \dfrac{T_r - T_{c,i}}{T_r - T_o}\right) & \left(h_o \le h_r and T_o \le T_{c,i}\right) \end{cases} \qquad (16)$$

$$\beta_e^b = \begin{cases} \beta_{o,set} & \left(h_o > h_r\right) \\ 1 & \left(h_o \le h_r and T_o > T^b{}_{c,e}\right) \\ \max\left(\beta_{o,set}, \dfrac{T_r - T^b{}_{c,e}}{T_r - T_o}\right) & \left(h_o \le h_r and T_o \le T^b{}_{c,e}\right) \end{cases} \qquad (17)$$

When the outside air enthalpy is greater than the room air enthalpy, both units use the design minimum outside air intake ($m \cdot \beta_{o,set}$). When the outside air enthalpy is less than the room air enthalpy and the outside air temperature is higher than the cold air set point, both units use 100% outside air. If the outside air temperature is lower than the cold air temperature, the mixed air temperature is maintained at the cold air set point when the outside airflow intake satisfies the indoor air quality requirement.

The heating and cooling energy consumption can be calculated using Equation (18) ~ (21).

$$E^b_{cc,i} = \max\left\{m_i \cdot C_p \cdot \left(T_{mix,i} - T_{c,i}\right), m_i \cdot \left(h_{mix,i} - h_{c,i}\right), 0\right\} \qquad (18)$$

$$E^b_{ph,i} = \max\left\{m_i \cdot C_p \cdot \left(T_{c,i} - T_{mix,i}\right), 0\right\} \qquad (19)$$

$$E^b_{cc,e} = \max\left\{m_e \cdot C_p \cdot \left(T_{mix,e} - T_{c,e}\right), m_e \cdot \left(h_{mix,e} - h_{c,e}\right), 0\right\} \qquad (20)$$

$$E^b_{h,e} = \max\left\{m_e \cdot C_p \cdot \left(T_{c,e} - T_{mix,e}\right), 0\right\} \qquad (21)$$

When the airflow rates $\left(m_i, m_e\right)$, the weather and room conditions are given, the mixed and supply air conditions is determined based on optimal cold air temperature and outside air schedules.

The total cooling/heating energy consumptions of the TAHU system (E_{cc}^b / E_h^b) are the summation of the two units.

$$E_{cc}^b = E_{cc,i}^b + E_{cc,e}^b \qquad (22)$$

$$E_h^b = E_{ph,i}^b + E_{h,e}^b \qquad (23)$$

The energy savings are determined by the energy consumption difference between the TAHU system and the OAHU system, as shown in Equation (24) and Equation (25).

$$\Delta E_{cc} = E_{cc}^b - E_{cc} \qquad (24)$$

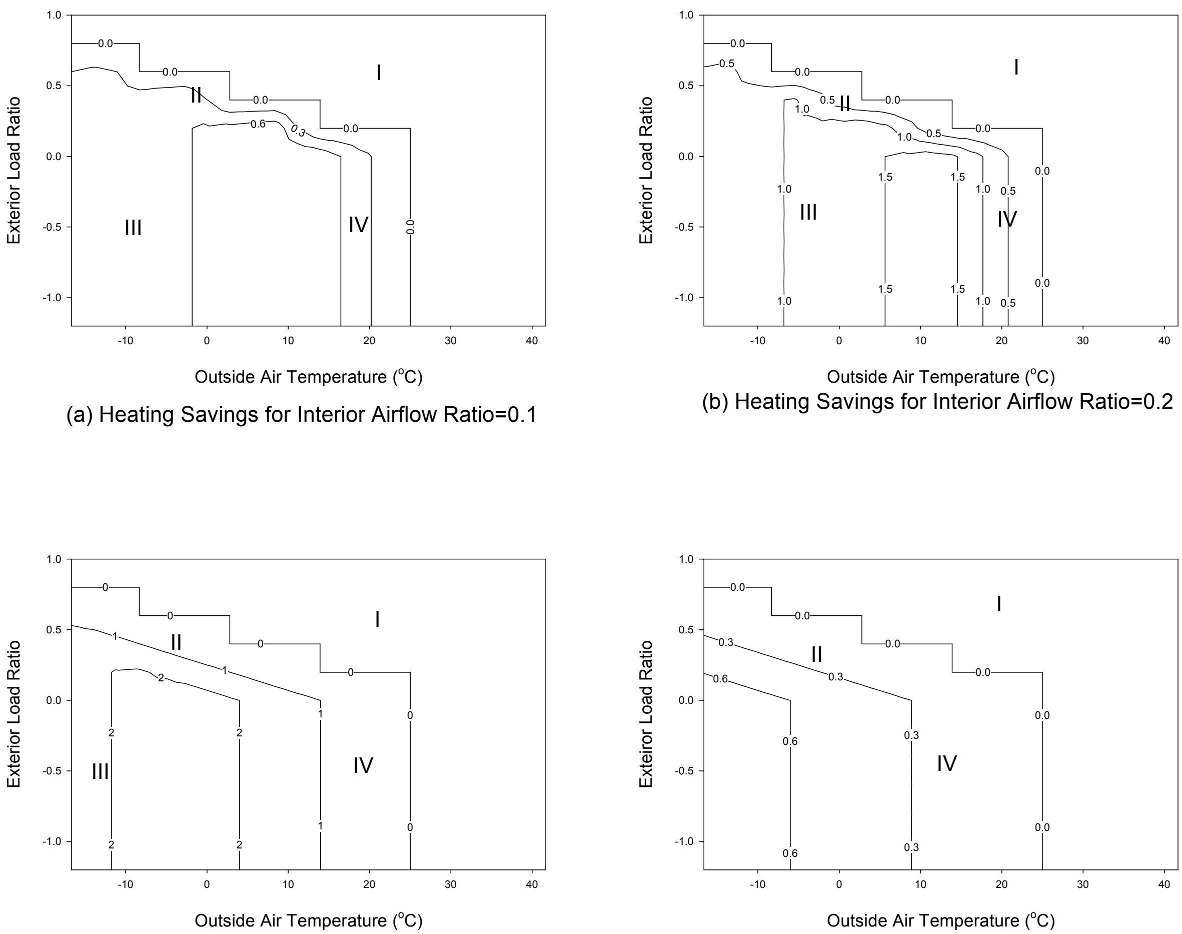
$$\Delta E_h = E_h^b - E_h \qquad (25)$$

ANALYSIS

The potential thermal energy savings depends on system types (VAV or CAV), outside air conditions, cooling loads, ratios of the interior zone to the exterior zone (IAR), cold air temperatures, room temperatures, minimum outside air intakes, and airflow rates.

The outside air temperature and the exterior zone load ratio (γ_e) are selected as the primary parameters. First, the potential thermal energy savings is simulated using dry weather conditions (Albuquerque bin data) under four different interior airflow ratios (IAR): 0.1, 0.2, 0.5 and 0.9. Then, the simulations are performed using three different outside air relative humidity ratio: 40%, 60% and 80% with 0.5 of IAR, since the outside air humidity ratio has significant impact on the enthalpy economizer operation.

All simulations are conduced using the same room conditions and the same minimum outside air intake ratio. The room temperature and relative humidity are 24.1ºC (75ºF) and 50%, respectively. The design minimum outside air ratio ($\beta_{o,set}$) is 20% of the design airflow rate. The cold air temperature and outside air intake are determined using the optimal control schedules. Both AHU systems are CAV system.



(a) Heating Savings for Interior Airflow Ratio=0.1

(b) Heating Savings for Interior Airflow Ratio=0.2

(c) Heating Savings for Interior Airflow Ratio=0.5

(d) Heating Savings for Interior Airflow Ratio=0.9

Figure 3: Simulated Potential Energy Savings (Albuquerque Bin Data with different IAR)
(Contour lines are smoothed by the plot software.)

Figure 3 presents the contour lines of simulated heating and cooling energy savings (kJ/kg) under dry climate (Albuquerque bin data). The abscissa represents the outside air temperature. The ordinate represents the exterior zone load ratio (γ_e). The chart is divided into four zones by a zero heating savings line (ZHS), a zero heating consumption line (ZHC), and a peak heating savings line (PHS).

The ZHS line presents the boundary where the mixed air temperature equals the exterior zone cold air temperature of the TAHU system. Above this line, the TAHU system uses no heating. Below this line, the TAHU system needs reheat to maintain the room temperature.

The ZHC line is below the zero heating savings line. It is the boundary where the mixed air temperature equals the exterior zone cold air temperature of the OAHU system. Above this line, the OAHU system uses no heating. Below this line, the OAHU system needs heating to maintain the room temperature.

The PHS line is below the zero heating consumption line, and perpendicular to the abscissa. It presents an outside air temperature, where the mixed air temperature equals the cold air temperature set point when the interior zone has the entire building minimum outside air intake. On the left of this line, the OAHU system supplies some outside air to the exterior zone directly to satisfy the minimum outside air intake.

Region I is above the zero heating savings line. The heating savings is zero since both the OAHU and the TAHU system use no heating.

Region II is the area between the zero heating savings line and the zero heating consumption line. The heating savings equals the heating consumption of the TAHU system. It increases as the load ratio decreases and as the outside air temperature decreases. The heating savings can be expressed by Equation (26).

$$\Delta e_h = (1-\varphi)\cdot C_p \cdot \left(T_{s,e} - \beta_{o,set}\cdot T_o - T_r + \beta_{o,set}\cdot T_r\right) \qquad (26)$$

Region III is under the zero heating savings line, and left of the peak savings line. The heating savings decreases as the outside air temperature decreases. The heating savings can be expressed by Equation (27).

$$\Delta e_h = \begin{cases} (1-\varphi)\cdot C_p \cdot \beta_{o,set}\cdot (T_r - T_o) - \varphi \cdot C_p \cdot \left(T_{ci} - T_r + \dfrac{\beta_{o,set}}{\varphi}(T_r - T_o)\right) & \varphi \geq \beta_{o,set} \\ \varphi \cdot (1-\beta_{o,set})\cdot C_p \cdot (T_r - T_o) - \varphi \cdot C_p \cdot (T_{ci} - T_o) & \varphi < \beta_{o,set} \end{cases} \qquad (27)$$

Region IV is under the zero heating savings line, and right of the peak savings line. The heating savings decreases as the outside air temperature increases. The heating savings can be expressed by Equation (28).

$$\Delta e_h = \begin{cases} (1-\varphi)\cdot C_p \cdot \beta_{o,set}\cdot (T_r - T_o) & \varphi \geq \beta_{o,set} \\ \varphi \cdot (1-\beta_{o,set})\cdot C_p \cdot (T_r - T_o) & \varphi < \beta_{o,set} \end{cases} \qquad (28)$$

Both analytical and numerical results (Figures 3(a), 3(b), 3(c) and 3(d)) show that the maximum savings occurs when the IAR equals 0.5. The closer the IAR to 0.5, the higher the potential savings is. For example, the maximum savings is 0.805kJ/kg for IAR of 0.1, 1.72kJ/kg for IAR of 0.2, 2.68kJ/kg for IAR of 0.5 and 0.814kJ/kg for IAR of 0.9. For typical commercial buildings, IARs are fairly close to 0.5 since the interior zone areas are slightly smaller than the exterior zone area. The potential heating savings may vary from 1.6KJ/kg to 2KJ/kg. Under the dry climate, the OAHU system has a minimal impact on the cooling energy consumption. Although the savings are positive, the magnitude is negligible.

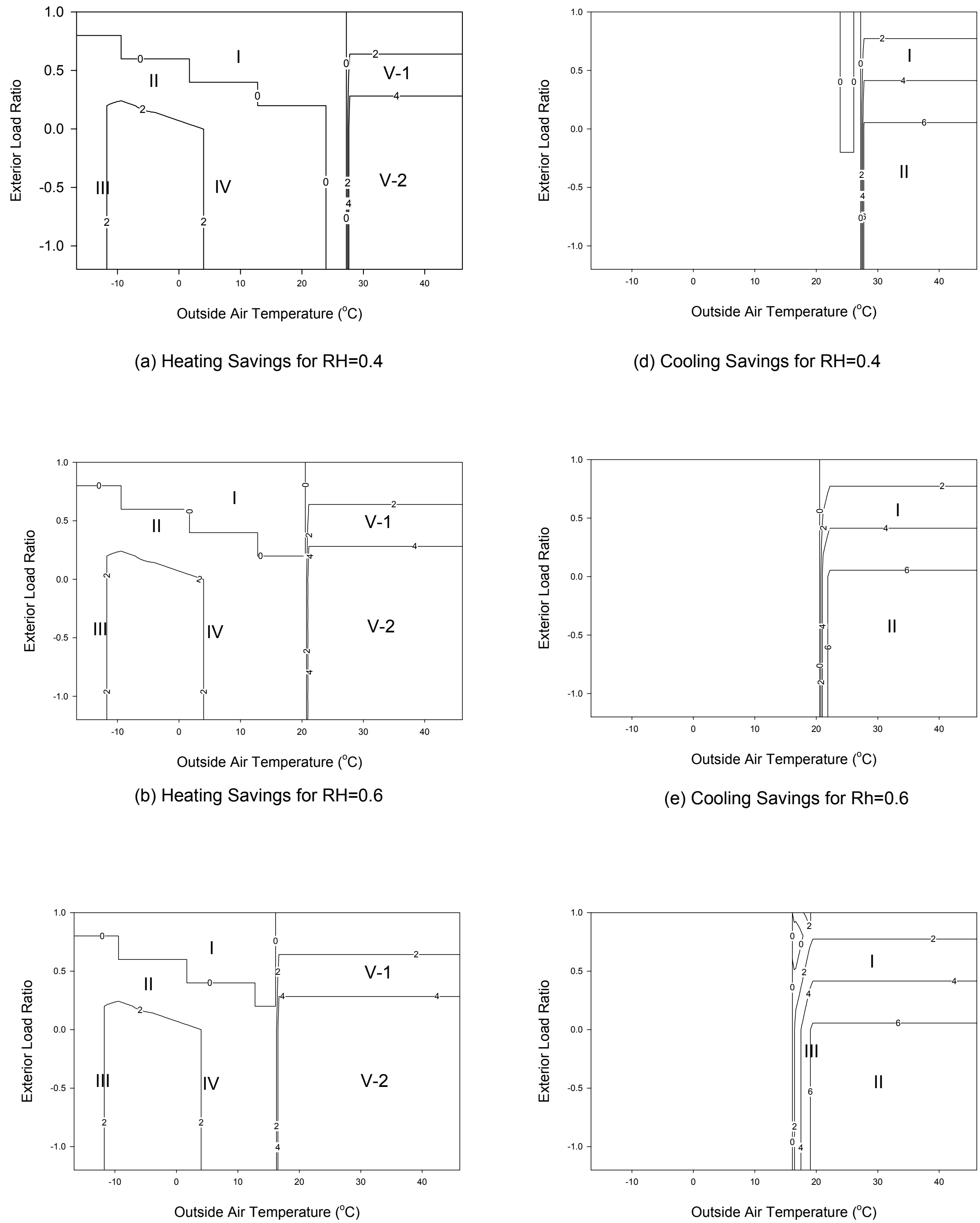
Figure 4 shows the simulated potential heating and cooling energy savings when the outside air relative humidity is assumed to be fixed at different levels: 40%, 60% and 80%. The interior airflow rate ratio (IAR) is assumed to be 0.5.

In Figure 4 (a), 4 (b) and 4 (c), a new heating savings area, Region V, is separated from the other areas by the room humidity ratio line (RHR). On the left of this line, the outside air humidity ratio is less than the room air humidity ratio, and on the right of the line the outside air humidity ratio is greater than the room air humidity ratio. The patterns of the potential heating savings are the same as in the dry climate in regions I, II, III and IV. When the outside air humidity ratio is less than the room humidity ratio, the potential savings is not impacted.

Region V is divided into two sub-regions, Region V-1 and Region V-2, by the zero load ratio line (ZLR). Above this line, the exterior zone needs cooling. Below this line, the exterior zone needs heating.

In Region V-1, the heating savings increases as the exterior load ratio decreases. The potential savings is independent on the outside air temperature. The heating savings can be expressed by Equation (29).

$$\Delta e_h = (1-\varphi)\cdot C_p \cdot \left(T_{s,e} - T_c\right) \qquad (29)$$



(a) Heating Savings for RH=0.4

(d) Cooling Savings for RH=0.4

(b) Heating Savings for RH=0.6

(e) Cooling Savings for Rh=0.6

(c) Heating Savings for RH=0.8

(f) Cooling Savings for RH=0.8

Figure 4: Simulated Potential Energy Savings under Different Relative Humidity
(Contour lines are smoothed by the plot software.)

In Region V-2, the heating savings is constant. It is independent of the outside air temperature and the exterior load ratio. The heating savings is expressed in Equation (30).

$$\Delta e_h = (1-\varphi)\cdot C_p \cdot (T_r - T_c) \tag{30}$$

Figure 4(d), 4(e) and 4(f) show significant cooling energy savings when the outside air humidity ratio is higher than the room humidity ratio. Two regions are separated by the ZLR line.

In Region I, the cooling savings increases as the exterior load ratio decreases. The cooling savings can be expressed by Equation (31).

$$\Delta e_{cc} = (1-\varphi)\cdot (C_p \cdot (T_{s,e} - T_r) + h_r - h_c) \tag{31}$$

In Region II, the cooling savings is constant. It is independent of the outside air temperature and the exterior load ratio. The cooling savings is expressed by Equation (32).

$$\Delta e_{cc} = (1-\varphi)\cdot (h_r - h_c) \tag{32}$$

Figure 4 shows significant heating and cooling energy savings when the outside air dew point temperature is higher than the room dew point temperature. For example, the potential savings is as high as 3.49kJ/kg, including 1.4kJ/kg for heating and 2.09kJ/kg for cooling when the load ratio (γ_e) is 0.75.

The simulation results show that the OAHU system is superior to the TAHU system for typical commercial buildings. The OAHU system uses much less heating energy when the outside air dew point temperature is lower than the room dew point temperature, and uses much less heating and cooling when the outside air dew point temperature is higher than the room dew point temperature. The annual average thermal energy savings is approximately 1.85kJ/kg. For a 10,000 square meter building, the annual potential thermal energy could be as high as 3.31MJ/yr (3138 MMBtu/yr).

For the OAHU system, the exterior zone relative humidity level will be higher than the interior zone under low cooling load since no moisture is removed from the cooling coil. For a typical commercial building, the internal moisture production is very limited. The room relative humidity level of the exterior zone can be maintained at the comfort range although it may be several percent higher than the interior zone.

APPLICATIONS

To accurately determine the potential thermal energy savings, for a specific case, a complete model simulation can be conducted based on the analytical models presented in this paper. However, the potential energy savings can also be determined using Figure 4. When the specific conditions differ from the assumptions made in Figure 4, corrections are mandatory as shown in the following procedures.

Temperature set up correction

If the room temperature and the supply air temperature are different from the assumption in Figure 4, the equivalent load ratio (γ) should be determined using Equation (33).

$$\gamma = \gamma_{e,a} \frac{T_{r,a} - T_{s,e,a}}{T_r - T_{s,e}} \tag{33}$$

The potential savings are then determined using Figure 4 based on the outside air temperature and the exterior zone load ratio.

System type correction

For VAV systems, there is no savings until the exterior zone load ratio is less than the equivalent minimum airflow ratio ($\alpha_{\min}$). The equivalent minimum airflow ratio can be determined using Equation (34).

$$\alpha_{\min} = \alpha_{\min,a} \frac{T_r - T_{s,e}}{T_{r,a} - T_{s,e,a}} \tag{34}$$

If the exterior zone load is less than the equivalent minimum airflow, determine the equivalent load ratio using Equation (35). The potential savings is then determined using Figures 4 based on the outside air temperature and the equivalent load ratio.

$$\gamma = \frac{\gamma_a}{\alpha_{\min}} \tag{35}$$

IAR Correction

If the IAR is different from the assumed value (0.5) in Figure 4, get the thermal energy savings $\Delta e_{h,a}$ and $\Delta e_{c,a}$ from Figure 4 based on the exterior zone load ratio and the outside air temperature first.

For heating savings correction, the new Peak Heating Savings line shall be calculated using Equation (36).

$$T_{phsl} = \frac{T_c - \left(1 - \frac{\beta_{o,set}}{\varphi}\right) \cdot T_r}{\frac{\beta_{o,set}}{\varphi}} \tag{36}$$

If the outside air temperature is on the left of both the new Peak Heating Savings line and the Peak Heating Savings line in Figure 4, the heating savings is corrected by Equation (37).

$$\Delta e_h = \Delta e_{h,a} \frac{\varphi}{\varphi_a} \tag{37}$$

If the outside air temperature is on the right of both the new Peak Heating Savings line and the Peak Heating Savings line in Figure 4, the heating savings is corrected using Equation (38).

$$\Delta e_h = \frac{(1-\varphi)}{(1-\varphi_a)} \cdot \Delta e_{h,a} \tag{38}$$

If the outside air condition is in the middle of the two Peak Heating Savings lines, the heating savings is corrected by Equation (39).

$$\Delta e_h = \Delta e_{h,a} - C_p \cdot \beta_{o,set} \cdot (T_r - T_o) \cdot (\varphi - \varphi_a + 1) + \varphi \cdot C_p \cdot (T_r - T_c) \tag{39}$$

The cooling savings under different IAR follows the same pattern as in Figure 4, so the cooling savings can be corrected by Equation (40) directly.

$$\Delta e_c = \frac{(1-\varphi)}{(1-\varphi_a)} \cdot \Delta e_{c,a} \tag{40}$$

Minimum outside air intake correction

If the design minimum outside air ratio ($\beta_{o,set}$) is different from the assumed value (20% of the design airflow rate) in Figure 4, get the thermal energy savings $\Delta e_{h,a}$ and $\Delta e_{c,a}$ from Figure 4 based on the outside air temperature and the exterior zone load ratio first.

When the outside air humidity ratio is higher than the room air humidity ratio, no correction is required.

When the outside air humidity ratio is less than the room humidity ratio, the economizer starting line (ES) should be determined using Equation (41). Left of this line, there is no heating savings, because the same amount of the outside air is supplied to both the TAHU and OAHU system and then the same heating is consumed by both systems.

$$T_{esl} = \frac{T_c - (1-\beta) \cdot T_r}{\beta} \tag{41}$$

Right of this line, there is the same heating savings pattern as in Figure 4, but the new Peak Heating Savings line should be calculated using Equation (36).

If the outside air temperature is on the right of both the simulated Peak Heating Savings line in Figure 4 and the new Peak Heating Savings line, a supplement factor needs to be added to $\Delta e_{h,a}$, as shown in Equation (42).

$$\Delta e = \Delta e_{h,a} + (1-\varphi) \cdot C_p \cdot (\beta_a - \beta) \cdot (T_r - T_o) \tag{42}$$

If the outside air temperature is on the left of the both Peak Heating Savings line, the same supplement factor needs to be deducted from $\Delta e_{h,a}$, as shown in Equation (43).

$$\Delta e = \Delta e_{h,a} - (1-\varphi) \cdot C_p \cdot (\beta_a - \beta) \cdot (T_r - T_o) \tag{43}$$

If the outside air temperature is between the two Peak Heating Savings lines, the potential savings should be calculated using Equation (44).

$$\Delta e_h = \Delta e_{h,a} + (1-\varphi) \cdot C_p \cdot \left((\beta_{new} - \beta_{simul}) - \frac{\beta_{new}}{1-\varphi} + \frac{T_r - T_c}{T_r - T_o} \cdot \frac{\varphi}{1-\varphi} \right) \cdot (T_r - T_o) \tag{44}$$

Zone definition correction

Previous simulations assume the interior zone load is constant. In an existing building, the interior zone load varies greatly with the occupancy schedules. The reheat may actually be required for both the interior zone and the exterior zone. This does not impact the potential energy savings since the reheat is required by both the base system and the OAHU system. If the interior zone cooling load varies more than exterior zone cooling load, the OAHU system considers the exterior zone as "interior zone" and the interior zone as "exterior zone". All the preceding simulation results are still valid by redefining the exterior zone as the "interior zone" and the interior zone as the "exterior zone".

CONCLUSIONS

The optimal energy performances of the OAHU and the two dedicated AHU system are compared and analyzed. The results show that the OAHU system uses significant less heating energy (up to 1.85kJ/kg) when the outside air dew point temperature is less than the room dew point temperature. When the outside air dew point temperature is higher than the room dew point temperature, the OAHU system saves more thermal energy (up to 4.65kJ/kg). The OAHU system uses less thermal energy than TAHU in all climate conditions. The savings are much higher for the area, where the outside air dew point temperature is higher than the room dew point temperature for large amount of hours.

The potential energy savings can be evaluated using the models or the numerical results presented in the paper. The paper also provides a complete procedure to adjust the numerical results to a special case.

NOMENCLATURE

c_p	= Specific heat for dry air (kJ/(kg·ºC) or Btu/(lbm·ºF))
Δe	= Specific energy savings (kJ/kg or Btu/lbm)
E	= Energy consumption (kJ or Btu)
h	= Air enthalpy (kJ/kg or Btu/lbm)
IAR	= the ratio of the interior zone airflow rate to the total airflow rate of the entire building
m	= Airflow rate (kg/s or lbm/hr)
T	= Air temperature (ºC or ºF)
φ	= Interior airflow rate ratio
Rh	= Relative humidity
Q	= Heating or cooling load for exterior zone
γ	= Exterior zone load ratio
β	= Outside air intake ratio

<u>*Subscripts*</u>

a	= Simulation parameter and results in Figure 4 with some assumptions
b	= Base case systems, single unit systems
c	= Cold deck temperature set point
cc	= Cooling
d	= Designed condition
dew	= Dew point temperature
e	= Exterior zone
esl	= Economizer starting line
h	= Heating energy consumption
i	= Interior zone
mix	= Mixed
min	= Minimum
o	= Outside air
ph	= Pre heating, hot deck
phsl	= Peak heating savings line
rh	= Reheat heating
r	= Under applied condition different from the simulated parameter
s	= Supply air
set	= Minimum design value

<u>*Superscripts*</u>

b	= Base case system (the TAHU system)

REFERENCES

T. A. Reddy, M. Liu, S. Katipamula, D. E. Claridge, 1998. Extending the Concept of Energy Delivery Efficiency (DED) of HVAC Systems, ASHRAE Transactions, 1998, Vol. 104, Pt. 1.

T. A. Reddy, J. K. Kissock, S. Katipamula, and D. E. Claridge, 1994. An energy delivery efficiency index to evaluate simultaneous heating and cooling effects in large commercial buildings. *ASME Journal of Solar Energy Engineering, 16:79.*

Li Song and Mingsheng Liu. 2002. An Integrated Air Handling Unit System for Large Commercial Buildings (Submitted to Journal of Building and Energy).