An Integrated Air Handling Unit System for Large Commercial Buildings

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ABSTRACT

This paper presents an integrated air handling unit system (OAHU[*]) for large commercial buildings. The system introduces outside air into the interior section and circulates the return air to the exterior section. Detailed analytical models are developed to compare the energy performance and indoor air quality between the OAHU and conventional AHU systems (single AHU). The OAHU uses significantly less energy than the conventional system in both winter and summer. The OAHU also provides better indoor air quality for the interior zone.

INTRODUCTION

Commercial buildings have been the focus of energy conservation due to their large number and size. Several energy conservation measures are commonly implemented, including economizer, conversion of constant air volume (CAV) to variable air volume (VAV), and heat recovery [Haines 1987]. The potential savings has been simulated for different weather and building conditions [Spitler et al 1987]. The implementation recommendations have been discussed by Roberts [1991]. The implementation of these energy conservation measures has significantly decreased building energy consumption as well as peak demand.

Taking an integrated system approach, the authors have developed a new AHU system for large commercial buildings, named the OAHU system. The OAHU is superior to the conventional AHU. It uses less heating energy during winter, and uses less cooling and re-heat energy during transitional and summer months. It also improves indoor air quality of the interior zone.

In this paper, the analytical system models are developed. Numerical simulations are conducted and simulation results are analyzed.

MODELS

Thermal energy consumption models are developed for both the OAHU and conventional AHU systems. The energy savings is calculated as the difference of both systems under the same ambient and load conditions.

Conventional Air Handling Unit

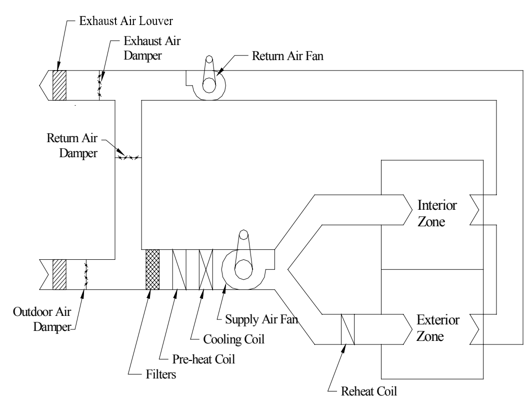

Figure 1: Schematic Diagram of the Conventional AHU System

Figure 1 shows the schematic diagram of the conventional air handing unit, a single duct AHU providing conditioned air to exterior and interior zones through two branches. The AHU has an enthalpy economizer. The cold deck temperature is 55˚F.

The mixed air temperature and mixed air enthalpy are expressed by Equations (1) and (2).

$$T_{mix} = \begin{cases} T_o & T_c \le T_o \,\&\, h_o \le h_r \\ \max\left(\beta_{o,set} \cdot T_o + \left(1-\beta_{o,set}\right) \cdot T_r , T_c\right) & others \end{cases} \tag{1}$$

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[*] Patent pending

$$h_{mix} = \begin{cases} h_o & T_c \le T_o \,\&\, h_o \le h_r \\ \max\left(\beta_{o,set} \cdot h_o + \left(1-\beta_{o,set}\right) \cdot h_r, h_c\right) & others \end{cases} \tag{2}$$

The cooling energy consumption is:

$$E^b_{cc} = \max\{m \cdot C_p \cdot (T_{mix} - T_c), m \cdot (h_{mix} - h_c), 0\} \tag{3}$$

The pre-heat energy consumption is:

$$E^b_{ph} = \max\{m \cdot C_p \cdot (T_c - T_{mix}), 0\} \tag{4}$$

The reheat energy consumption is:

$$E^b_{rh} = \max\{(1-\varphi) m \cdot C_p \cdot (T_{s,e} - T_c), 0\} \tag{5}$$

The total heating energy consumption of the conventional AHU system is:

$$E^b_h = E^b_{rh} + E^b_{ph} \tag{6}$$

OAHU system

Figure 2 shows the schematic diagram of the OAHU. The OAHU system has two separate supply air fans for interior and exterior zones. All outside air can be supplied directly to the interior zone. The return air from the interior zone can be supplied to the exterior zone. The OAHU transfers the heat from the interior zone to the exterior zone during winter. It allows higher supply air temperature to the exterior zone during summer and mild weather. The set point of the interior cold deck temperature is 55 ˚F. The exterior cold deck temperature is reset according to the zone load, when there is no need for dehumidification. The return air from both interior and exterior zones can be used to obtain an optimal mixed air temperature to minimize exterior zone heating and cooling.

During winter: the optimal outside airflow ratio of the interior zone is a function of outside conditions and room conditions. The outside intake ratio is calculated as:

$$\beta_i^w = \begin{cases} 1 & \begin{pmatrix} T_o > T_c \,\& \\ h_o \le h_r \end{pmatrix} \\ \max\left(\beta_{o,\min}, {(T_r - T_c)}/{(T_r - T_o)}\right) & others \end{cases} \tag{7}$$

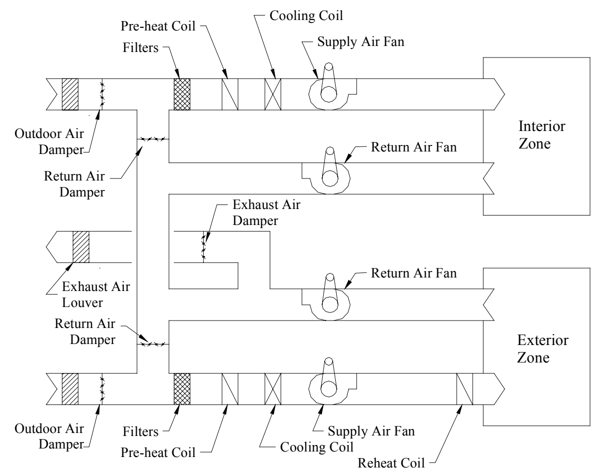


Figure 2: Schematic Diagram of the OAHU System

The outside airflow rate of the interior zone is $m_{o,i} = \beta_i \cdot m_i$.

For the exterior zone, the supply air temperature is:

$$T_s = T_r - \frac{\gamma_e \cdot Q_{e,d}}{m_e \cdot C_p} \tag{8}$$

The cold deck temperature is assumed to be the same as the supply air temperature:

$$T_{c,e} = T_{s,e} \tag{9}$$

If the interior outside airflow rate ($m_{o,i}$) is equal to or higher than the requirement for the entire building, the exterior outside damper can be closed. The mixed air temperature equals the return air temperature. If the interior outside airflow rate is less than the design outside airflow of the entire building, the outside airflow may be increased by one of the following methods. The first method is to increase the interior outside airflow rate to β_i^*, and the second is to increase the exterior outside airflow rate to β_e^*.

$$\beta_i^* = \frac{m_{o,set}}{m_i}, \beta_i \in \left(\beta_{i,\min}, \min\left(1, \beta_i^*\right)\right) \tag{10}$$

$$\beta_e^* = \frac{m_{o,set} - m_{o,i}}{m_{ed}}, \beta_e \in \left(0, \beta_e^*\right) \tag{11}$$

The total heating consumption is the sum of the interior zone and exterior zone heating consumption.

$$E_{ph} = \varphi \cdot m \cdot C_p \cdot \left(T_{c,i} - T_r\right) + C_p \cdot m \cdot \left(T_r - T_o\right) - Q_e$$

$$m = m_{ed} + m_i$$

where $\varphi = \dfrac{m_i}{m}$ (12)

$$\beta_{o,set} = \frac{m_{o,set}}{m}$$

Equation (12) shows that the choice of outside air increase has no impact on thermal energy consumption. Equation (11) is used in this study.

<u>During transition months and summer weather:</u> outside air humidity greatly affects energy consumption. If outside air enthalpy is less than the room air enthalpy, the interior zone uses 100% outside air. If outside air enthalpy is greater than room air enthalpy, all outside air intake will be supplied to the interior zone first and then re-circulate back to the exterior zone. Since the interior zone cooling coil removes moisture from the outside air, the exterior zone can use higher supply air temperature as needed without a high humidity problem. Consequently, significant heating and cooling energy can be saved under partial load conditions. At the same time, the indoor air quality of the interior zone is greatly improved. If the total airflow of the interior zone is less than the total outside air requirement, some of the outside air has to be directly introduced to the exterior zone. According to the interior outside airflow ratio, the exterior outside airflow ratio can be determined.

The outside airflow rate of interior zone is $m_{o,i} = \beta_i^{t,s} \cdot m_i$.

$$\beta_i = \frac{m_{o,i} + m_{o,e}}{m_i} \tag{13}$$

$$\beta_i^{t,s} = \max\left(\beta_i^w, \beta_i\right) \tag{14}$$

$$T_{mix,i} = \beta_i^{t,s} \cdot T_o + \left(1 - \beta_i^{t,s}\right) \cdot T_r \tag{15}$$

$$h_{mix,i} = \beta_i^{t,s} \cdot h_o + \left(1 - \beta_i^{t,s}\right) \cdot h_r \tag{16}$$

$$\beta_e = \begin{cases} \max\left(0, \dfrac{m_{o,set} - m_{o,i}}{m_{e,d}}\right) & (h_o > h_r)\, or\, (h_o \le h_r \;\&\; T_{o,dew} > T_{r,dew}) \\ \max\left(\dfrac{T_r - T_{s,e}}{T_r - T_o}, \dfrac{m_{o,set} - m_{o,i}}{m_{e,d}}\right) & (h_o \le h_r \;\&\; T_{o,dew} \le T_{r,dew} \;\&\; T_{s,e} > T_o) \\ 1 & (h_o \le h_r \;\&\; T_{o,dew} \le T_{r,dew} \;\&\; T_{s,e} \le T_o) \end{cases} \tag{17}$$

$$T_{mix,e} = \beta_e \cdot T_o + \left(1 - \beta_e\right) \cdot T_r \tag{18}$$

$$h_{mix,e} = \beta_e \cdot h_o + \left(1 - \beta_e\right) \cdot h_r \tag{19}$$

When the outside air is directly introduced into the exterior zone, the exterior cold deck temperature set point should satisfy both thermal load and dehumidification requirements.

$$T_{c,e} = \begin{cases} T_{c,i} & h_o > h_r or T_{o,dew} > T_{r,dew} \;\&\; m_{o,set} > m_{o,i} \\ T_{s,e} & others \end{cases} \tag{20}$$

The energy consumptions of the interior zone are calculated as:

$$E_{ph,i} = \max\left(C_p \cdot m_i \cdot \left(T_{c,i} - T_{mix,i}\right), 0\right) \tag{21}$$

$$E_{cc,i} = \max\left(C_p \cdot m_i \cdot \left(T_{mix.i} - T_{c,i}\right), m_i \cdot \left(h_{mix.i} - h_{c,i}\right), 0\right) \tag{22}$$

The energy consumptions of the exterior zone are calculated as:

$$E_{cc,e} = \max\left\{m_e \cdot C_p \cdot \left(T_{mix,e} - T_{c,e}\right), m \cdot \left(h_{mix,e} - h_{c,e}\right), 0\right\} \tag{23}$$

$$E_{ph,e} = \max\left\{m_e \cdot C_p \cdot \left(T_{c,e} - T_{mix,e}\right), 0\right\} \tag{24}$$

$$E_{rh,e} = \max\left\{m_e \cdot C_p \cdot \left(T_{s,e} - T_{c,e}\right), 0\right\} \tag{25}$$

The total energy consumptions are calculated as:

$$E_{cc} = E_{cc,i} + E_{cc,e} \tag{26}$$

$$E_h = E_{ph,i} + E_{ph,e} + E_{rh,e} \tag{27}$$

The energy savings are calculated as calculated as:

$$\Delta E_{cc} = E_{cc}^b - E_{cc} \tag{28}$$

$$\Delta E_h = E_h^b - E_h \tag{29}$$

SIMULATION RESULTS AND ANALYSIS

The numerical simulation is performed using Albuquerque bin data with different interior airflow rate and load ratios.

In order to analyze the impact of relative humidity, similar simulations are conducted by assuming different relative humidity of 40%, 60% and 80%.

All the simulations assume 75˚F and 50% relative humidity room conditions. The design cooling coil discharging conditions are 55˚F and 90% relative humidity when the mixed air humidity ratio is higher. The design minimum outside air ratio is 0.2 of the design airflow rate.

Figure 3 and 4 present contour lines of simulated heating and cooling energy savings (Btu/lbm.hr). Each contour line represents the savings versus the outside air temperature and exterior zone load ratio. The heating savings are divided into four regions.

Region I: outside air intake from interior zone satisfies the minimum outside air need for the entire building. The heating energy savings can be deduced as:

$$\Delta E_h = m_e \cdot C_p \cdot \left(T_r - T_{c,i}\right) \tag{30}$$

Both Equation (30) and simulation results show that the heating savings are independent upon the outside air temperature and load ratio. The savings is proportional to the airflow rate of the exterior zone.

Region II: outside air intake from interior zone cannot satisfy the minimum outside air need for the entire building. Outside air needs to be supplied directly to the exterior zone. The exterior supply air temperature is less than room temperature. In this case, the heating savings is:

$$\Delta E_h = m_e \cdot C_p \cdot \left(T_{s,e} - T_{c,i}\right) \tag{31}$$

Both Equation (31) and simulation results show that the savings is independent upon the outside temperature. It increases as the load ratio decreases. When the load ratio decreases, the exterior supply air temperature increases. Consequently, under the same ambient temperature heating savings increases. Since the major heating savings comes from the exterior zone, heating savings decreases as interior airflow rate ratio increases.

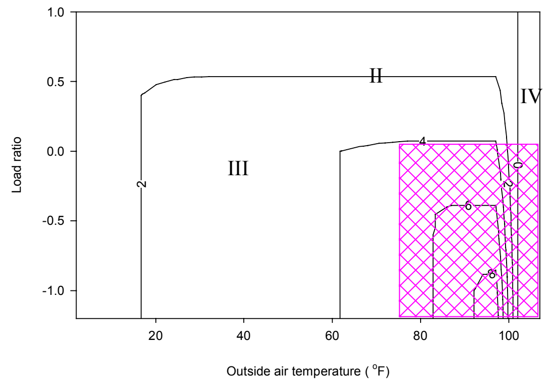
Region III: outside air from the interior zone is less than the whole building requirement. Outside air is directly supplied to the exterior zone. The mixed air temperature of the exterior zone is less than the exterior supply air temperature. The heating savings can be expressed by Equation (32).

$$\Delta E_h = m_e \cdot C_p \left(\frac{\beta_{o,set} - \varphi}{1 - \varphi} \cdot \left(T_o - T_r\right) + T_r - T_{c,i}\right) \tag{32}$$

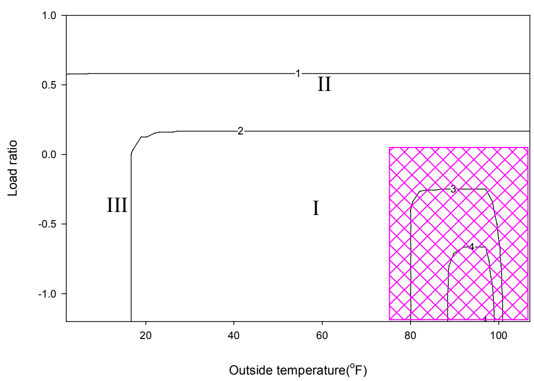
Both Equation (32) and simulation results show that the savings is independent upon load ratio. However, it is affected by outside air temperature and airflow rate of exterior zone, as observed from different figures.

Region IV: outside air enthalpy is larger than room air enthalpy. Outside air is directly supplied to the exterior zone. The cooling coils for both interior zone and exterior zone have to provide humidity control. Both systems, the conventional system and the OAHU system, need reheat for the exterior zone. There is no heating savings. It only exists in Figure 3(a) where the interior zone flow ratio is extremely small ($\varphi = 0.1$).

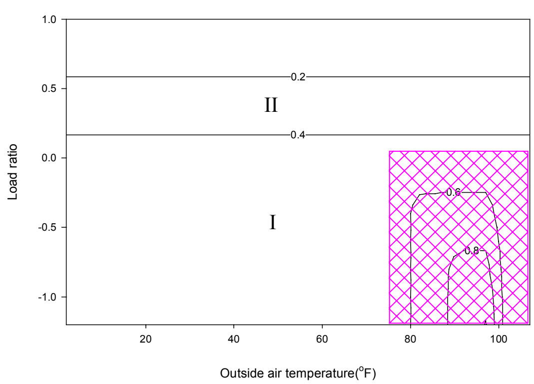
Figures 4(a), 4(b) and 4(c) show contour lines of the simulated cooling energy savings. Each contour line represents the same saving versus outside air temperature and exterior zone load ratio. Because enthalpy economizer is used for both the conventional system and the OAHU system, there is no mechanical cooling required when the outside air temperature is lower than 55˚F. The cooling savings can be divided into four regions.



(a):Heating savings for interior airflow rate ratio = 0.1

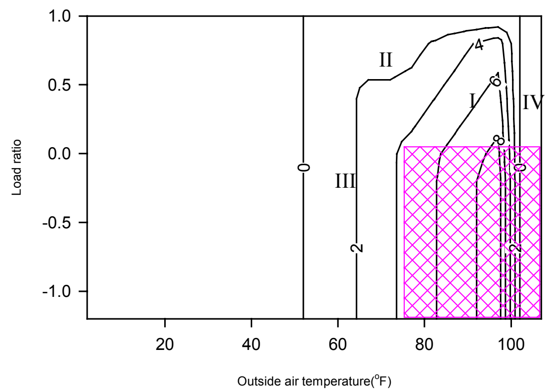


(b);Heating savings for interior airflow ratio=0.5

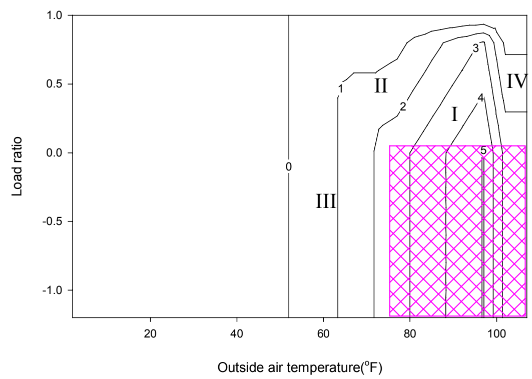


(c): Heating savings for interior airflow ratio ratio=0.9

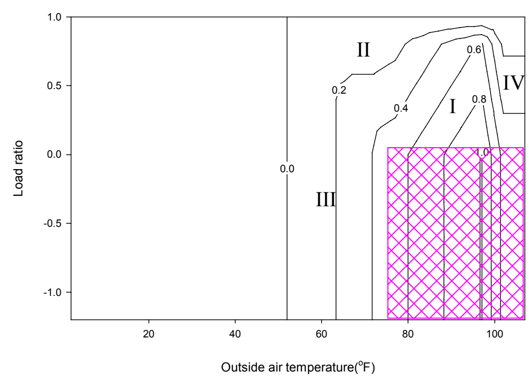
Figure3: Heating Energy Savings for Albuquerque Bin Data
(Contour lines are smoothed by the software)



(a):Cooling savings for interior airflow ratio=0.1



(b) Cooling savings for interior airflow rate ratio=0.5



(c): Cooling savings for interior airflow rate ratio=0.9

Figure4: Cooling Energy Savings for Albuquerque Bin Data
(Contour lines are smoothed by the software)

Region I: the supply air temperature is lower than the outside air temperature. The outside air temperature is greater than the room air temperature. 100% return air is supplied to the exterior zone. The cooling savings for this region is expressed as:

$$\Delta E_{cc} = m_e \cdot C_p \cdot \left(T_{s,e} + T_o - T_c - T_r\right) \tag{33}$$

Both Equation (33) and simulation results show that the cooling savings increases as outside air temperature increases, as load ratio decreases and as exterior airflow rate increases.

Region II: the supply air temperature is less than the outside air temperature. The outside air temperature is less than the room air temperature. 100% outside air is supplied to the exterior zone. The cooling savings can be expressed by Equation (34).

$$\Delta E_{cc} = m_e \cdot C_p \cdot \left(T_{s,e} - T_c\right) \tag{34}$$

Both Equation (34) and simulation results show that the savings is a function of supply air temperature and airflow rate of the exterior zone. The cooling savings increases as load ratio decreases, and it decreases as interior airflow rate ratio increases.

Region III: the exterior supply air temperature is higher than the outside air temperature. The outside air temperature is less than 75˚F. Free cooling can be used by the exterior zone with economizer. The cooling savings equation for this region is:

$$\Delta E_{cc} = m_e \cdot C_p \cdot \left(T_o - T_c\right). \tag{35}$$

Both Equation (35) and simulation results show that the savings depends on outside air temperature and the airflow rate of the exterior zone. The savings increases as both variables increase.

Region IV: the outside air enthalpy is greater than the room air enthalpy. If the outside air from the interior zone is less than the entire building requirement, the cooling energy savings is zero. If the interior zone airflow rate is higher than or equal to outside airflow rate for the entire building requirement, the cooling savings is:

$$\Delta E_{cc} = m_e \cdot C_p \cdot \left(T_{s,e} - T_c\right) \tag{36}$$

Both Equation (36) and simulation results show that the cooling savings decreases as exterior zone supply air temperature decreases, when the interior zone airflow is higher than design total outside airflow.

The preceding analyses are based on Albuquerque bin data, for which there is no dehumidification requirement. In order to investigate the impacts of the relative humidity on energy savings, simulations are conducted using different outside air humidity ratios. Figure 5 and 6 show the results. The interior airflow rate ratio is kept at 0.5.

Figures 5(a), 5(b) and 5(c) show that the heating savings is independent upon relative humidity, since the heating savings comes from exterior reheat savings determined by differences in temperature, not in enthalpy.

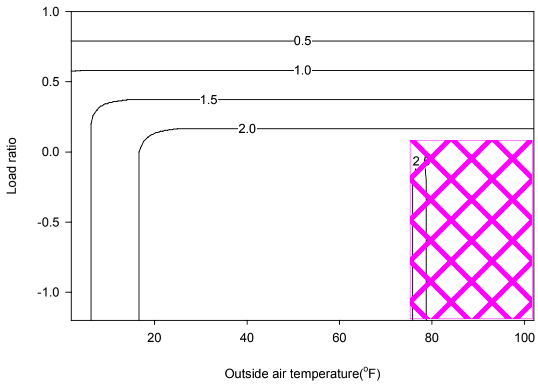
Figures 6(a), 6(b) and 6(c) show how relative humidity affects cooling savings. When the outside enthalpy is greater than the room air enthalpy, the cooling savings will not change with relative humidity. Minimum design outside air ratio is supplied for both cases. For the OAHU system, all the outside air is cooled down for dehumidification by an interior zone cooling coil. 100% return air is supplied to exterior zone. The cooling savings is determined by Equation (37).

$$\Delta E_{cc} = m_e \cdot C_p \cdot \left(T_{s,e} - T_c\right) \tag{37}$$

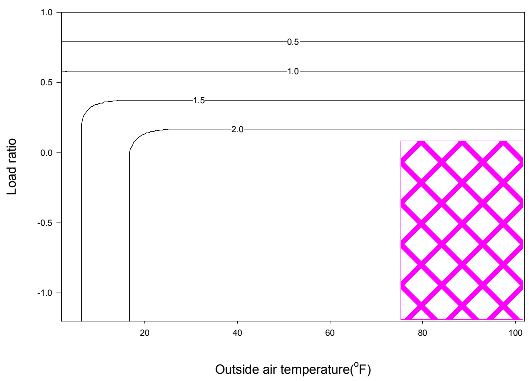
Both Equation (37) and simulated results show that the cooling savings decreases as the load ratio increases and decreases as interior airflow rate ratio increases, as shown in the figure.

When the outside air enthalpy is less than the room air enthalpy, cooling savings increases with relative humidity in general. There are some situations that outside air humidity ratio is greater than room air humidity ratio. Mechanical cooling is needed for cooling 100% return air down for exterior zone instead of mixing room air and outside air by economizer. Less cooling savings can be arrived. That is why some dips, even negative savings, occur in Figures 6(b) and 6(c).

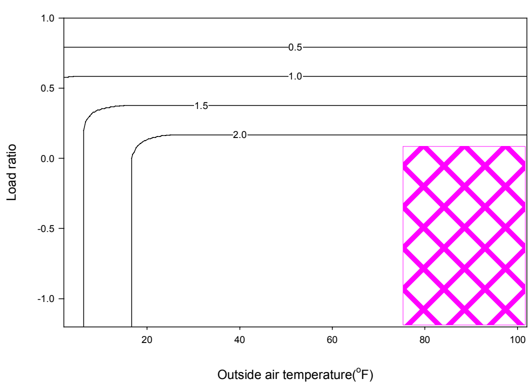
In the shaded areas in Figure 3, 4, 5 and 6, the outside air temperature is greater than 75 ˚F and the exterior zone has heating load. It cannot be true in practical situation.



(a): Heating savings for RH=40%

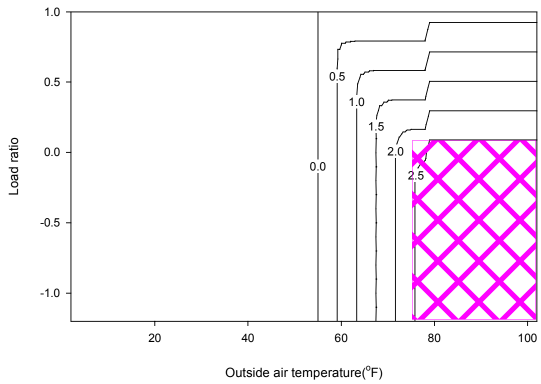


(b): Heating savings for RH=60%

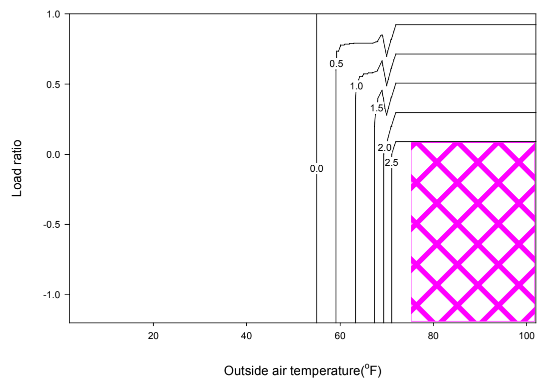


(c): Heating savings for RH=80%

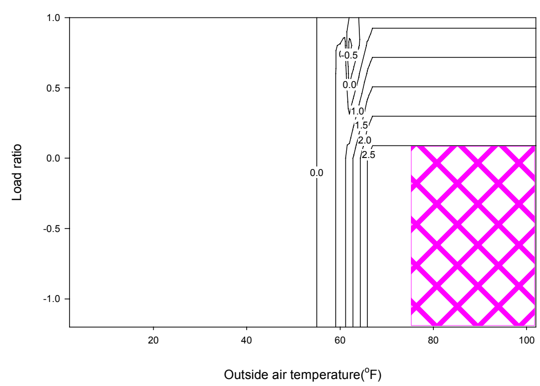
Figure 5: Simulated Heating Energy Savings versus Outside Air Temperature and Load Ratio (Contour lines are smoothed by software)



(a): Cooling savings for RH=40%



(b): Cooling savings for RH=60%



(c): Cooling savings for RH=80%

Figure 6: Simulated Cooling Energy Savings versus Outside Air Temperature and Load Ratio (Contour lines are smoothed by software)

CONCLUSIONS

A numerical simulation and investigation of the conventional system and the OAHU system has been performed based on thermal energy consumption models. The results show that the OAHU system uses significant less heating energy in winter and less re-heat and cooling energy in summer. At the same time, it provides better indoor air quality for interior zones.

The presented models and recommended procedure can be applied in any weather conditions and to any large commercial buildings to calculate potential thermal energy savings.

This study is limited to constant volume systems.

ACKNOWLEDGEMENT

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NOMENCLATURE

b = Outside air ratio
C_p = Specific heat for dry air (J/(kg·ºC) or Btu/(lbm·ºF))
E = Energy consumption (W or Btu/ (lbm·hr))
h = Air enthalpy (J/kg or Btu/lbm)
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

Subscripts

b = Base case systems, single unit systems
c = Cold deck set point
cc = Cooling
d = Designed
dew = Dew point
e = Exterior zone
i = Interior zone
mix = Mixed
min = Minimum
o = Outside air
ph = Pre heating, hot deck
rh = Reheat heating
r = Room air
set = Minimum design value
s = Supply air

Superscripts

b = Base case (the conventional system case)

s = Summer

t = Transition

w = Winter

REFERENCES

Haines R. W. 1984. Retrofitting Reheat-Type HVAC Systems for Energy Conservation. ASHRAE Transactions, vol.90, part 2B, 185-192.

Roberts J. W., P.E. September 1991. Outdoor Air and VAV Systems. *ASHRAE Journal*, vol.33, no.9, 26-30.

Spitler J. D., Hittle D C., Johnson D L., Pedersen C O. 1987. A Comparative Study of the Performance of Temperature-Based and Enthalpy-Based Economy Cycles. ASHRAE Transactions, vol.93, Part 2. 13-22.