Air Handling Unit Supply Air Temperature Optimization during Economizer Cycles

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ABSTRACT

Most air handling units (AHUs) in commercial buildings have an air economizer cycle for free cooling under certain outside air conditions. During the economizer cycle, the outside air and return air dampers are modulated to seek mixing air temperature at supply air temperature setpoint. Mechanical cooling is always required when outside air temperature is higher than the supply air temperature setpoint. Generally the supply air temperature setpoint is set at 55ºF for space humidity control. Actually the dehumidification is not necessary when outside air dew point is less than 55ºF. Meanwhile the space may have less cooling load due to envelope heat loss and/or occupant schedule. These provide an opportunity to use higher supply air temperature to reduce or eliminate mechanical cooling and terminal box reheat. On the other hand, the higher supply air temperature will require higher air flow as well as higher fan power. Therefore the supply air temperature has to be optimized to minimize the combined energy for fan, cooling and heating energy. In this paper a simple energy consumption model is established for AHU systems during the economizer and then an optimal supply air temperature control is developed to minimize the total cost of the mechanical cooling and the fan motor power. This paper presents AHU system energy modeling, supply air temperature optimization, and simulated energy savings.

INTRODUCTION

Several factors should be considered to determine AHU supply air temperature. Lowing supply air temperature can downsize AHU systems. However, occupants will feel discomfort and mechanical cooling systems will be enlarged if the supply air temperature is too low. On the other hand, the supply air temperature cannot be too high for the dehumidification purpose during the summer. Generally the supply air temperature is set at 55°F during humid seasons.

Typically AHUs have an economizer cycle to reduce or eliminate mechanical cooling under certain outside air conditions. During the economizer cycle, return air and outside air dampers are modulated to seek mixing air temperature at a design supply air temperature set point (ASHRAE 1999). High-limit shutoff control is chosen to automatically switch from the economizer cycle to the minimum outside air cycle when outside air intake no longer reduces cooling energy usage. ASHRAE (2001) provides several high-limit shutoff control methods for economizer operation, including the enthalpy control, the electronic enthalpy control and the dry bulb temperature control. Theoretically the outside air temperature range for economizer cycles can extend to the same temperature as the return air temperature. The mechanical cooling is completely eliminated when the outside air temperature is less than 55°F. The supply air temperature can be always maintained at its design setpoint (55°F) by modulating the outside airflow using the outside air and return air dampers. Since the lowest supply air temperature is already achieved without any mechanical cooling, supply air temperature optimization is not necessary in this outside air condition.

On the other hand, the mixing air temperature is always higher than 55°F even though the outside air damper is fully opened when the outside air temperature is higher than 55°F. The mechanical cooling is normally required in order to maintain the supply air temperature setpoint. The supply air temperature should be fixed at 55°F if the outside air is humid or in other words the outside air dew point is higher 55°F. However, the supply air temperature is not limited at 55°F if the outside air is dry and the space has partial cooling load due to envelope heat transfer and occupant schedule. In this case the AHUs still can maintain the space temperature with a variable supply air temperature rather than the design value (55°F). Adjusting the supply air temperature above the design setpoint will affect fan power, mechanical cooling and terminal box reheat. Increasing supply air temperature will reduce or eliminate the mechanical cooling and the terminal box reheat but will increase the fan airflow as well as fan power. The supply air temperature should be optimized by balancing the mechanical cooling and fan power consumption during the economizer cycle. However, the supply air temperature optimization is a fairly complex tradeoff due to the lack of energy consumption model. Several conventional reset sequences were recommended. Oregon Department of Energy (2007) suggested that the economizer free cooling is used first until space temperature cannot be maintained. CEC (2003) recommended that the supply air temperature is reset to maintain the highest zone cooling output at a setpoint of 90%. As a result, the chillers are only enabled until the fan speed reaches to 90% or 100% with the conventional supply temperature reset sequences.

The terminal reheat may not exist or may be independent with the supply air temperature for some AHU systems, such as single zone systems, and single duct systems serving interior areas or with fan powered terminal boxes. A simple energy consumption model can be established for these systems during the economizer cycle when the outside air temperature is higher than design supply air temperature setpoint and the outside air dew point is lower than 55°F. An optimal supply air temperature control is developed to minimize the total cost of the mechanical cooling and the fan motor power for a typical AHU with a chilled water cooling coil. This paper presents AHU system energy modeling, supply air temperature optimization, and simulated energy savings.

MODELINGS

Figure 1 shows a schematic of single duct or single zone variable air volume (VAV) air handling units (AHUs) with a chilled water cooling coil. It is assumed that the single duct AHU either serve interior area only or use fan powered terminal boxes with a zero minimum airflow setpoint. With this assumption, the terminal box reheat can be excluded in energy consumption model. Both the supply fan and return fan have a VFD. The supply fan speed is modulated to maintain either the duct static pressure in the single duct system or the space temperature in the single zone system while the return fan speed is modulated to maintain the building pressure. The cooling coil valve is modulated to maintain the supply air temperature setpoint. The outside air damper is fully open when the outside air temperature is higher than the design supply air temperature (55°F) during the economizer cycle.

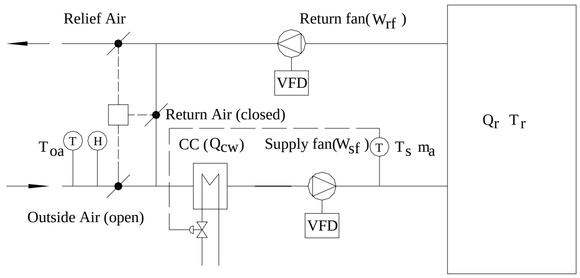

Figure 1: Schematic of AHUs

It is assumed that the outside air is dry with the dew point less than 55°F and the latent cooling load is ignored. Space sensible cooling load has to be removed by the AHU. The supply air temperature can be determined based on the space sensible cooling load and the supply airflow using Eq. (1).

$$Q_r = m_a c_p (t_r - t_s) \tag{1}$$

Since the AHU uses 100% outside air, the outside air is cool down through the cooling coil and then is heat up through the supply fan. The cooling coil covers the supply fan heat generation and the cooling energy required to cool down the air from the outside air temperature to the supply air temperature. The supply fan heat generation depends on the motor power and location. The motor installation factor (η) is used to indicate the motor location. It equals 1 if the motor is installed inside the AHU while it equals the motor efficiency if the motor is installed outside the AHU.

$$Q_{cw} = m_a c_p (t_{oa} - t_s) + W_{sf}\eta \tag{2}$$

Combine Eq. (1) and (2), the cooling coil load can be expressed in another format.

$$Q_{cw} = Q_r + W_{sf}\eta + m_a C_p [t_{oa} - t_r] \tag{3}$$

Equation (3) shows that the cooling coil load (Q_{cw}) covers the space cooling load (Q_r), the supply fan heat generation ($W_{sf}\eta$,) and the outside cooling load $(m_a C_p [t_{oa} - t_r])$. Since the outside air temperature is less than space temperature, the outside air load is negative or free cooling. In other words, the economizer cycle can reduce the mechanical cooling.

Both supply and return fans have electricity consumption by the motors. The total motor electricity consumption can be expressed as:

$$W_f = W_{sf} + W_{rf} \tag{4}$$

The total utility cost includes both the chilled water and the motor electricity consumption.

$$C = W_f P_{ele} + Q_{cw} P_{cw} \tag{5}$$

Substitute Eq. (3) and (4) into Eq.(5), the total utility cost can be expressed by Eq. (6).

$$\begin{aligned} C = W_{sf}[(P]_{ele} + \eta P_{cw}) + W_{rf} P_{ele} \\ + Q_r P_{cw} - m_a C_p (t_r - t_{oa}) P_{cw} \end{aligned} \tag{6}$$

The actual fan motor power can be determined based on the design motor power and the airflow ratio between the actual airflow and the design airflow using Eqs. (7) and (8). Ideally the exponent of motor power (*n*) is 3 based on the fan law if all dampers of AHUs have a fixed position and the motor efficiency is constant. Since the motor efficiency always varies with the load and the terminal box damper are always modulated, the actual factor of motor power will varies from 2 to 3.

$$W_{sf} = W_{sf,d} \left(\frac{m_a}{m_{a,d}} \right)^n \tag{7}$$

$$W_{rf} = W_{rf,d} \left(\frac{m_a}{m_{a,d}} \right)^n \tag{8}$$

The space design sensible cooling load is used to normalize the utility cost. The space design cooling is defined using the design airflow and the design supply air temperature, described by Eq. (9).

$$Q_{r,d} = m_{a,d} C_p \left(t_r - t_{s,d} \right) \tag{9}$$

Combine Eqs. (7) to (9) with Eq. (6), the normalized total utility cost can be rewritten in a dimensionless format.

$$\begin{aligned} \bar{C} = \bar{W}_{sf,d} \bullet \bar{m}_a^n \cdot \left(\bar{P}_{ele} + \eta \right) + \bar{W}_{rf,d} \cdot \bar{m}_a^n \cdot \bar{P}_{ele} \\ + \bar{Q}_r - \bar{m}_a \bar{t}_{oa} \end{aligned} \tag{10}$$

Where:

$$\bar{C} = \frac{C}{Q_{r,d} P_{cw}}$$

$$\bar{m}_a = \frac{m_a}{m_{a,d}}$$

$$\bar{Q}_r = \frac{Q_r}{Q_{r,d}}$$

$$\bar{t}_{oa} = \frac{t_r - t_{oa}}{t_r - t_{s,d}}$$

$$\bar{W}_{sf,d} = \frac{W_{sf,d}}{Q_{r,d}}$$

$$\bar{W}_{rf,d} = \frac{W_{rf,d}}{Q_{r,d}}$$

$$\bar{P}_{ele} = \frac{P_{ele}}{P_{CW}}$$

Equation 10 shows that the normalized cost ($\bar{C}$) is determined by the normalized airflow ($\bar{m}_a$), the normalized space cooling load ($\bar{Q}_r$), and the normalized outside air temperature ($\bar{t}_{oa}$) for a given system with a given normalized supply fan motor power ($\bar{W}_{sf,d}$) , a given normalized return fan motor power ($\bar{W}_{rf,d}$) and the normalized electricity price $\bar{P}_{ele}$).

OPTIMIZATION

The first derivative of the normalized cost with the normalized supply airflow is set to zero in order to determine the normalized critical supply airflow of the local minimum normalized cost.

$$\frac{d\bar{C}}{d\bar{m}_a} = n \cdot \bar{W}_{sf,d} \bullet \bar{m}_a^{n-1} \cdot \left(\bar{P}_{ele} + \eta \right) + n \cdot \bar{W}_{rf,d} \cdot \bar{m}_a^{n-1} \cdot \bar{P}_{ele} - \bar{t}_{oa} \tag{11}$$

Consequently the normalized critical supply air flow is expressed by Eq. (12).

$$\bar{m}_{a,cri} = \left(\frac{\bar{t}_{oa}}{n \bullet \bar{W}_{sf,d} \cdot \left(\bar{P}_{ele} + \eta \right) + n \cdot \bar{W}_{rf,d} \cdot \bar{P}_{ele}} \right)^{\frac{1}{n-1}} \tag{12}$$

However, the critical airflow may be out of the available supply airflow range, which is defined by the minimum and maximum airflow. The minimum supply airflow occurs when the supply air temperature is set to the design value (55ºF). Actually the normalized minimum supply airflow is equal to the normalized space cooling load.

$$\bar{m}_{a,min} = \bar{Q}_r \tag{13}$$

The maximum airflow under a given space cooling load occurs when the cooling coil is disabled. However, the maximum airflow cannot be higher than the design supply air flow. Therefore the normalized maximum airflow is selected by Eq. (14).

$$\bar{m}_{a,max} = \min\left(\bar{m}_{a,cal}, 1 \right) \tag{14}$$

Where

$$\bar{Q}_r + \bar{W}_{sf,d} \cdot \eta \cdot \bar{m}^n_{a,cal} = \bar{m}_{a,cal} \cdot \bar{t}_{oa}$$

The optimal airflow ratio can be determined based on the critical minimum airflow, the minimum supply airflow and the maximum supply airflow.

$$\bar{m}_{a,opt} = \begin{cases} \bar{m}_{a,max} & \text{when } \bar{m}_{a,cri} > \bar{m}_{a,max} \\ \bar{m}_{a,cri} & \text{when } \bar{m}_{a,min} < \bar{m}_{a,cri} < \bar{m}_{a,max} \\ \bar{m}_{a,min} & \text{when } \bar{m}_{a,cri} < \bar{m}_{a,min} \end{cases} \quad (15)$$

The optimal supply air temperature is then determined from the optimal supply airflow using Eq. (16)

$$\bar{Q}_r = \bar{m}_{a,opt} \cdot \frac{t_r - t_s}{t_r - t_{s,d}} \quad (16)$$

The utility cost savings can be obtained by comparing the utility cost between the optimal supply air temperature and the conventional fixed supply air temperature. The normalized absolute savings can be expressed as:

$$\begin{aligned} \Delta\bar{C} &= \bar{C}_{m_{a,min}} - \bar{C}_{m_{a,opt}} \\ &= [\bar{W}_{sf,d} \cdot (\bar{P}_{ele} + \eta) + \bar{W}_{rf,d} \cdot \bar{P}_{ele}] \cdot \\ &\quad (\bar{m}^n_{a,min} - \bar{m}^n_{a,opt}) - (\bar{m}_{a,min} - \bar{m}_{a,opt})\bar{t}_{oa} \end{aligned} \quad (17)$$

Then the relative savings based on the conventional utility cost is calculated as follow

$$\Delta\overline{C(\%)} = \frac{\Delta\bar{C}}{\bar{C}_{m_{a,min}}} = \quad (18)$$

APPLICATION

Simulations were conducted on an existing single zone VAV AHU. The AHU has an enthalpy economizer cycle and a chilled water cooling coil. The chilled water is provided by a district cooling system. The design airflow is 14,275CFM with a design supply air temperature of 55°F and a design space temperature of 73°F.

The AHU has a 20HP supply fan and a 5HP return fan. Both the fans are installed inside the AHU and have a VFD. The measured motor power at the design speed is 11kW for the supply fan and 3kW for the return fan. The electricity price is $0.050/kWh and the chilled water price is $0.21/ton-hr. The factor of motor power is selected to be 3 in the simulation.
The optimal supply airflow and supply air temperature are obtained using Eqs. (15) and (16) with different outside air temperatures and different space cooling load ratio. Figure 2 shows the simulated optimal supply airflow versus the outside air temperature under different space cooling load ratio while Figure 3 shows the simulated optimal supply air temperature.

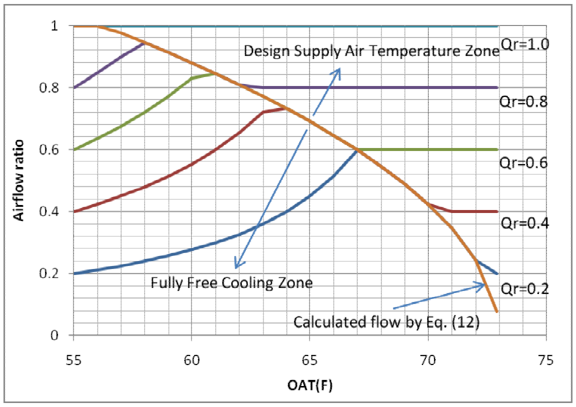

Figure 2: Optimal Supply air flow vs outside air temperature and space cooling load

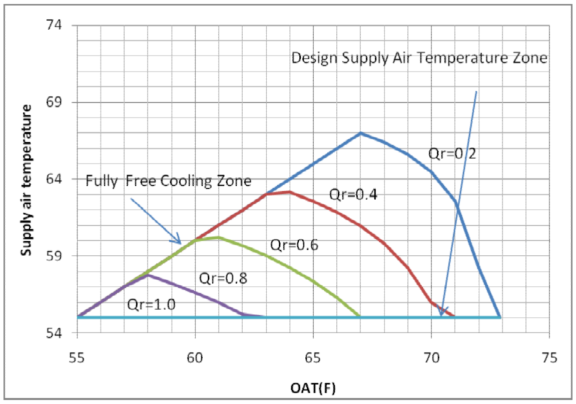

Figure 3: Supply air temperature vs outside air temperature and space load ratio

The results show that the optimal supply airflow and temperature control has three zones:

1. he mechanical cooling is completely eliminated and the fully free cooling is used by fully closing the chilled water cooling coil valve if the actual supply airflow is less than the critical airflow calculated by Eq. (12). The actual supply airflow can be measured using fan airflow stations. The supply air temperature is equal to the outside air temperature if the fan motor heat generation is ignored. The supply air flow increases as the outside air temperature increases with a constant space cooling load ratio.

2. he supply airflow is maintained at the critical airflow by modulating the chilled water valve until the supply air temperature drops to the design supply air temperature setpoint, 55°F. Both the supply airflow and temperature decrease as the outside air temperature increases for a constant space cooling load ratio.

3. he supply air temperature is maintained at the design supply air temperature by modulating the chilled water valve. The supply airflow is adjusted based on the space cooling load. Actually the supply airflow ratio is always equal to the space cooling load ratio under the design supply air temperature.

The normalized cost savings and percentage cost savings are calculated using Eqs. (17) and (18). Figure 4 shows the normalized cost savings while Figure 5 shows the percentage cost savings.

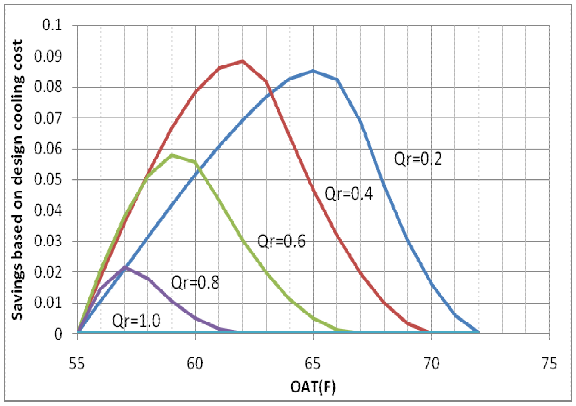

Figure 4: Cost savings based on the design space cooling cost

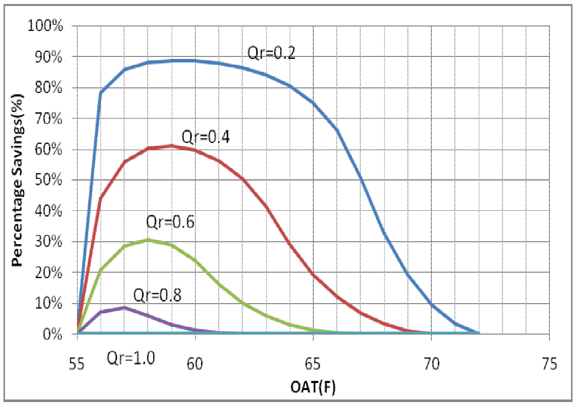

Figure 5: Percentage savings

The cost savings varies with the outside air temperature and the space cooling load ratio. The cost savings can reach up to 9% of the design space cooling cost when the actual space cooling load is between 20% and 40%. Compared with the conventional control, the savings will reach up to 90% when the actual space cooling load is 20%.

CONCLUSIONS AND FUTURE WORK

An energy cost model for AHU systems has been developed in the paper. The energy cost covers the fan motor electricity and the cooling coil chilled water. The chilled water consumption includes the space cooling load, the supply fan heat generation and the outside air free cooling load. The optimal supply airflow and supply air temperature is deduced from the developed energy cost model. The energy performance is evaluated between the conventional and optimal supply air temperature controls. The simulation results show the energy savings can reach to 90% under certain space loads and outside air conditions.

Future work will conduct experiments on an AHU to test the theoretical model developed. The results of these experiments will be presented in a subsequent paper.

NOMENCLATURE

C - total energy cost, including chilled water and motor electricity

$\overline{C}$ - normalized total energy cost based on chilled water cost associated with design space cooling load

c_p - air constant pressure specific heat

m_a - air flow rate, m^3/s

$\overline{m}_a$ - normalized air flow rate based on design supply airflow.

n - factor of motor power

P_{cw} - chilled water price, \$/kW

P_{ele} - electricity price, \$/kW

$\overline{P}_{ele}$ - normalized electricity price based on chilled water price

Q - cooling load, kW

$\overline{Q}$ - normalized cooling load based on design space cooling load

t - air dry bulb temperature, °C

$\overline{t}_{oa}$ - normalized outside air temperature

W - fan motor power, kW

$\overline{W}$ - normalized fan motor power based on design space cooling load

η - motor installation factor

$\Delta\overline{C}$ - normalized total energy cost savings

$\Delta\overline{C}(\%)$ - percentage energy cost savings

Subscripts:

cri - critical
cw - chilled water
d - design
f - both fans
max - maximum
min - minimum
oa - outside air
opt - optimal
r - room air
rf - return air fan
s - supply air
sf - supply air fan

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