Energy Performance Analysis of Coupled-Control Units With Both Thermostat and Humidistat

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Coupled control units typically condition a single zone with constant air volume. A thermostat controls the heating valve or cooling valve when the room relative humidity is below the set point. When the room relative humidity is higher than the set point, a humidistat controls the cooling coil to dehumidify air and the thermostat controls the heating coil to maintain room temperature. Theoretical modeling is performed to investigate the energy performance of coupled control units and potential improvement measures. The study shows that the annual thermal energy consumption of the coupled-control units is up to four times higher than the optimal thermal energy consumption. Thermal energy consumption can be reduced by (a) eliminating excessive airflows, (b) minimizing valve leakages, and (c) modulating airflows with a zone sensible load. This paper presents the simulation models and results and discusses improvement measures.
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Introduction

The coupled-control unit is a single zone constant air volume system. Due to its relatively low cost, it is still widely used in hotels, dormitory buildings, hospitals, and office buildings. The coupled-control unit uses reheat as the primary measure for room humidity control. When reheat is not provided, it cannot provide good humidity control. A case study [1] showed room relative humidity as high as 70% under normal occupancy conditions for a class room facility. A strong musty smell and mold was identified in the facility as well. When room comfort is properly maintained, the coupled control unit uses far more thermal and fan energy than a variable air volume system.

This paper presents the analytical system models, the simulated design energy performance, the impacts of key design and operating parameters, the influences of climates, and the improved operation and control measures. All numerical results are generated using a hypothetical generic space in this study.

System Modeling

Figure 1 presents the system schematic of coupled-control units. The coupled-control system conditions a single zone. A thermostat and a humidistat are installed in the conditioned space. When the room temperature is lower than the set point, the thermostat modulates the chilled water valve or the hot water valve to change the supply air temperature. The hot water valve will not open until the chilled water valve is completely closed. However, if the room relatively humidity is higher than the set point, the humidistat controls the chilled water valve to dehumidify the air. The thermostat controls the hot water valve to maintain the room temperature. The coupled-control unit provides constant airflow to the space. The outside air intake may be controlled based on the room temperature. The model simulation assumed constant minimum outside air intake. An economizer is not simulated; most coupled-control units do not have economizers due to their small sizes. For relatively large size units, however, economizers significantly impact system performances. The detailed system models are presented below.

The space is modeled by the zone sensible and latent loads. The zone sensible load is expressed as a function of outside air temperature according to ASHRAE research [2]. The moisture load is a constant.

$$Q_{sen}=a+bt_0 \tag{1}$$

$$W=c \tag{2}$$

Both the sensible thermal and moisture loads are expressed using the unit floor area. The values of constants a, b, and c are determined based on typical commercial building conditions: (1) the zone has zero cooling load when outside air temperature is 13°C (55°F), (2) the zone has a design sensible load when outside air temperature is 36°C (96°F), (3) the design room conditions are 23°C (74°F) and 50% for relative humidity, and (4) the design airflow is approximately 5 l/s·m^2 or 1 cfm/ft^2. Based on these assumptions, the constants a, b, and c are calculated as -26.2 Btu/ft2/h, 0.48 Btu/ft2/°F/h, 0.002 lbm/ft^2 h, for British units, and -34.20 W/m^2, 2.725 W/m^2/°C, 9.765 g/m^2 h for SI units. According to ASHRAE [3] the outside airflow rate is 0.5 L/m^2 s (0.1 cfm/ft^2).

For each simulation the cooling and heating design temperatures and wet bulb temperatures are selected from the ASHRAE Handbook [3]. The design sensible load is calculated using Eq. (1). The design airflow rate is determined based on the design sensible load, the design supply air temperature (12.8°C or 55°F), and the room temperature 23°C (74°F).

The bin weather is used to model partial load conditions. The zone sensible load is calculated under each temperature bin. The supply air temperature is calculated using Eq. (3).

$$t_s=t_r-\frac{Q_{sen}}{\dot{m}_a c_p} \tag{3}$$

The supply air humidity ratio must be able to maintain the room relative humidity at or below the design level. The required supply air humidity ratio can be calculated using the room design humidity ratio, the moisture production, and the airflow rate.

$$w_s=w_{r,d}-\frac{w}{\dot{m}_a} \tag{4}$$

If the mixed air humidity ratio is lower than the required value, no dehumidification is required. The thermostat modulates the

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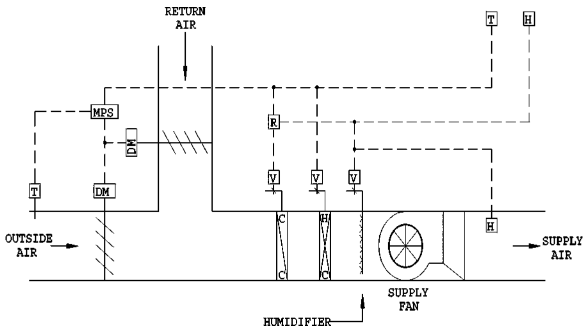

Fig. 1 System schematic of typical coupled-control unit

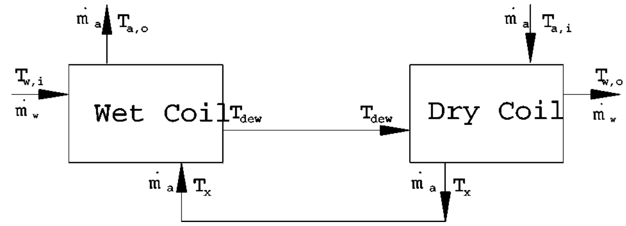

Fig. 2 Schematic of the modeling treatment of partially wet and partially dry coil

chilled water valve or the hot water valve to maintain the room air temperature. No simultaneous heating and cooling occurs if the valves do not leak.

If the mixed air humidity ratio is higher than the required value, dehumidification is required. However, simultaneous heating and cooling may not occur under certain conditions. Therefore, a hypothetical discharge air humidity ratio (w'_{dew}) is calculated using the coil model by assuming that the discharge air temperature equals the supply air temperature. If the hypothetical discharge humidity ratio is lower than the required supply air humidity ratio ($w_{dew,r,d}$), the discharge air temperature should be set to the supply air temperature. No heating is required. If the hypothetical discharge humidity ratio is higher than the required supply air humidity ratio, the discharge air humidity ratio is set to the required supply air humidity ratio, and the discharge air temperature is calculated using the coil model. The heating coil warms the air to the supply air temperature. Equation (4) models the required supply air humidity ratio of the cooling coil.

$$w_{dew,c}=\min(w_{dew,r,d},w'_{dew}) \tag{5}$$

The mixed air conditions can be determined based on outside air conditions, return air conditions, and the outside air intake ratio. Since the mixed air conditions depend on the room air conditions and vice versa, an iterative process is used to determine the mixed air conditions, the room air conditions, and the discharge air conditions. The heating and cooling energy consumptions are functions of airflow and entering and leaving coil air conditions.

$$E_c=m(h_m-h_c) \tag{6}$$

$$E_h=mc_p(t_s-t_c) \tag{7}$$

The fan power ratio (fan power over the design fan power) is modeled as a function of the airflow ratio (airflow over the design airflow). For a typical coupled control unit, the fan power equals the designed fan power. If a variable speed drive is used for energy performance improvement, the fan power ratio is proportional to the cubic power of the airflow ratio.

$$E_f=\left(\frac{m_a}{m_{a,d}}\right)^3 \tag{8}$$

The annual energy consumptions are calculated based on the hourly energy consumption and the number of hours in each temperature bin.

Both heating and cooling coils are modeled using the effectiveness-NTU method. The detailed techniques can be found in Ref. [4]. The cooling coil is designed under the wet condition since the entering air humidity ratio is much higher than the required supply air humidity ratio. The coil design load is calculated by Eq. (9).

$$Q_{c,d}=\dot{m}_a(h_{m,d}-h_{c,d}) \tag{9}$$

where

$$\dot{m}_a=\frac{Q_{sen,d}}{c_p(t_{r,d}-t_{c,d})} \tag{10}$$

The mixed air enthalpy is determined based on the outside air enthalpy, the room air enthalpy, and the outside air intake fraction. The discharge air conditions are 12.8°C (55°F) for the dry bulb temperature. The discharge air humidity ratio can be determined using Eq. (4). The discharge air enthalpy is then determined using the discharge air temperature and the moisture ratio.

The chilled water supply and return temperatures are 5.6°C (42°F) and 12.2°C (54°F), respectively. The design chilled water flow rate is determined using Eq. (11).

$$\dot{m}_w=\frac{Q_{c,d}}{c_w(t_{w,l,d}-t_{w,e,d})} \tag{11}$$

The overall water side and airside heat transfer coefficients of the cooling coil are determined using the enthalpy effectiveness-NTU method based on these conditions. Under wet coil conditions, the overall heat transfer coefficient, defined as the ratio of heat rate to enthalpy difference of air stream, is correlated with the airside and the waterside heating transfer coefficients using Eq. (12).

$$\frac{1}{UA_{wet}}=\frac{C_{p,sat,d}}{UA_{\text{int}}}+\frac{C_{pa}}{UA_{ext,wet}} \tag{12}$$

Under dry coil conditions, the overall heat transfer coefficient, defined as the ratio of heat rate to temperature difference, of the coil can be calculated using Eq. (13).

$$\frac{1}{UA_{dry}}=\frac{1}{UA_{\text{int}}}+\frac{1}{UA_{ext,dry}} \tag{13}$$

Under partial load conditions, the waterside heat transfer coefficient is corrected using the water flow rate and the average water temperature.

$$UA_{\text{int}}=UA_{\text{int},d}\dot{m}_w^{0.8}\frac{1+0.11t_w}{1+0.11t_{w,d}} \tag{14}$$

When entering air conditions, entering water temperature, and one of the leaving air conditions (temperature or moisture ratio) are given, the water flow rate and the other leaving air condition can be determined using the heat transfer coefficients and dry/wet coil heat transfer models. Since the water flow rate is required in determining the waterside heat transfer coefficient, an iteration process is used to solve the coil heat transfer problem.

When the coil is not in either dry or wet conditions, part of the coil is assumed to be in a wet condition. The enthalpy effectiveness-NTU model is used. The other part of the coil is in a dry condition. The temperature effectiveness-NTU model is used. Figure 2 presents the schematic of the modeling treatment. The entire air flows through the dry coil first and then the wet coil. The chilled water is first introduced into the wet coil and then leaves the coil at the entering air dew point temperature. An iteration process is used to determine the dry coil fraction, the water flow rate, and the leaving air conditions based on the entering air conditions, the coil heat transfer coefficients, and one of the leaving air conditions.

The heating coil is sized based on the design airflow rate, the design entering and leaving air temperatures, and the design entering and leaving water temperatures. The design entering and leaving hot water temperature are 82°C (180°F) and 66°C (150°F), respectively. The design airflow is the same as the cooling coil design airflow. The entering air condition is the design cold deck temperature. The leaving air temperature is determined based on the design heating load

$$t_{h,d}=t_r+\frac{Q_{h,d}}{c_p} \tag{15}$$

The hot water temperature reset is a typical operational practice. When the outside air temperature is increased, the hot water supply temperature is decreased. In this study the hot water supply temperature is decreased from 82°C (180°F) to 49°C (120°F) when the outside air temperature is increased from 4°C (40°F) to 21°C (70°F). The constant chilled water supply temperature is used in this study.

When a valve is closed, water may leak through the valve. The actual leakage rate depends on both the valve quality or the design leakage rate, and the valve authority. The design leakage rate refers to the ratio of water flow rates when the valve is fully closed and fully open under the same differential pressure across the valve. The valve authority is defined as the ratio of the valve pressure loss over the entire loop loss under the design water flow rate. The actual leakage rate can be calculated using Eq. (16) if the loop differential pressure is maintained at a constant rate and the flow is kept at the same flow region (turbulent or laminar).

$$\lambda=\frac{\lambda_i}{\sqrt{k+\lambda_i^2(1-k)}} \tag{16}$$

When the control valve is closed, the flow is likely to be laminar. When the valve is fully open, the flow is likely to be turbulent. Therefore, Eq. (16) is not perfectly correct. However, the margin of error is negligible for typical calculations.

A simulation program is developed based on these models. The simulation program is capable of simulating the design energy performance and the impacts of each key operating and design parameter.

Design Energy Performance

Design energy performance is simulated under the following conditions: (1) the system (airflow and coils) is sized based on the space design sensible load; and (2) the control valves have zero leakage. San Antonio, U.S.A., bin weather is used [5]. The average coincident wet bulb temperature is used for each temperature bin. The true average energy performance may differ from the simulated results under average conditions since the wet bulb varies from hour to hour under the same bin temperature. However, the average bin weather data still provides a simple and yet reasonably accurate assessment of the system performance under true weather conditions.

The simulated hourly heating and cooling energy consumptions, as a function of the ambient temperature, are shown in Fig. 3. Simultaneous heating and cooling occurs when the ambient temperature is higher than 12°C (53°F). Significant amounts of heating and cooling energy are used to control room humidity. When the ambient temperature is lower than 12°C (53°F), the cooling energy consumption equals zero. Heating energy consumption equals the zone heating load. The system has no economizer. If an economizer is available, simultaneous cooling and heating should not occur until the outside air dew point temperature is higher than the zone design dew point.

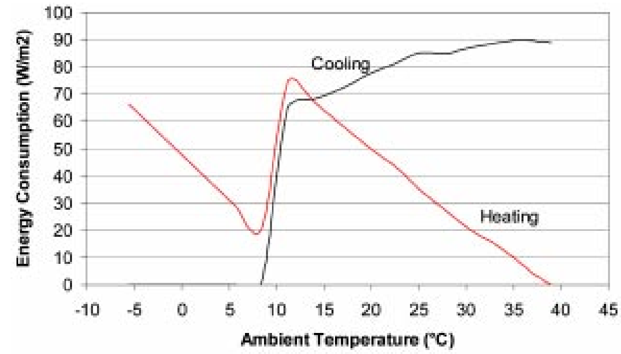

Fig. 3 Simulated hourly heating and cooling energy consumption under ideal design and ideal operation conditions

The annual thermal energy performance is evaluated using the concept of the Energy Delivery Efficiency (EDE) [6], which is the ratio of the optimal thermal energy consumption to actual thermal energy consumption. For a single zone system, the optimal annual thermal energy consumption equals the sum of the heating and the cooling loads. Since the heating or the cooling load equals the difference of the heating and cooling coil energy consumption, the EDE is calculated using Eq. (17).

$$EDE=\frac{\sum N_i|E_{h,i}-E_{c,i}|}{\sum N_i(E_{h,i}+E_{c,i})} \tag{17}$$

The annual EDE is calculated to be 0.33. The design annual thermal energy consumption is three times higher than the optimal thermal energy consumption. The low EDE value is primarily due to the constant airflow requirement. When the space load is decreased, the supply air temperature has to be kept at the design value in order to maintain room humidity control. A significant amount of reheat is used simultaneously to maintain room temperature.

Typical Energy Performance

The actual systems are often oversized. The control valves may leak, and they also may be over-pressurized. The impacts of design and operating parameters are simulated using San Antonio, U.S.A. bin weather.

System Size Impact. The space heating and sensible cooling design loads are assumed to be 10, 20, 30, and 40% higher than the actual design loads. The outside air intake is also proportionally increased. The valve leakage is assumed to be zero.

Figure 4 presents the simulated hourly cooling and heating energy consumption versus the ambient temperature. Both hourly cooling and heating energy consumptions increase proportionally with system oversize. The simultaneous heating and cooling occurs at a higher outside air temperature when the total airflow rate is higher. When the airflow is higher than the designed value, the outside air intake is also proportionally increased. The excessive outside air intake reduces the room relative humidity level when the outside air dew point is lower than the room design dew point. Consequently, the excessive outside air intake delays the simultaneous heating and cooling to a higher outside air temperature.

Table 1 presents the additional annual energy consumption caused by system oversize. The base case assumes zero oversize. The EDE values are also listed. System oversize increases the total airflow proportionally. A 10% excessive airflow (10% oversize) causes 33% excessive annual fan energy for the same duct design, 9% in annual cooling energy, and 14% in annual heating energy. A 40% excessive airflow (40% oversize) causes 174% more fan power, 26% more cooling energy, and 41% more heating

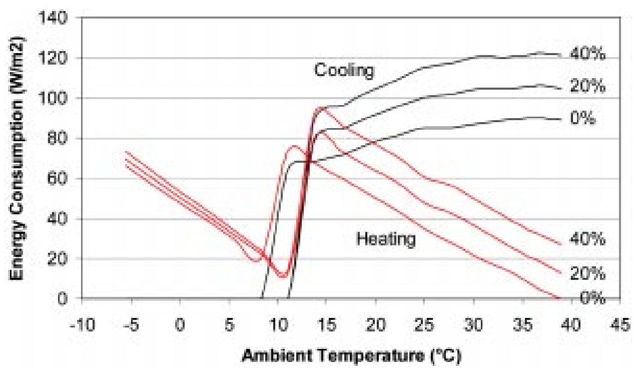

Fig. 4 Simulated heating and cooling energy consumption versus ambient temperature for different sizes of the space heating and cooling design loads

energy. The EDE is as low as 0.27. The annual thermal energy consumption is approximately four times higher than the optimal thermal energy consumption.

In an existing system it is important to balance the airflow based on the actual peak load. If the airflow is higher than required, the fan pulley should be adjusted or changed. This can significantly reduce fan and thermal energy consumption.

Leakage Rate Impact. The simulation is performed under the following conditions: zero system oversize and 50% valve authority. The designed valve leakage rate varies from 0.02 to 0.10.

Table 2 summarizes the simulated actual valve leakage rate and the extra annual energy consumption. The base case has zero valve leakage. The actual leakage rate is 40% higher than the design valve leakage rate due to the increased differential pressure on the control valve. The annual energy consumption increases from 0.6% to 7.7% for heating and 0.4% to 2.6% for cooling. The EDE values are approximately 0.33.

The valve leakage rate should be below 0.04 in order to keep the excessive thermal energy consumption below 1%. Good quality valves are recommended.

Valve Authority Impact. The simulations are conducted under the following conditions: 0.04 design valve leakage and zero system oversize. The valve authority varied from 10% to 50%. The simulated actual valve leakage rate and extra energy consumption are summarized in Table 3. The base case assumes zero valve leakage and zero oversize. The actual valve leakage rate varies from 0.056 to 0.126 depending on the valve authority. When the valve authority is 10%, the actual leakage rate is four times higher than the design valve leakage rate. The annual cooling energy is increased from 0.9% to 2.4%. The annual heating energy is increased from 1.5% to 3.9%. The valve authority has little impact on the EDE value. The EDE values are approximately 0.33.

The low valve authority significantly increases excessive energy consumption, even if good quality valves are used. Using a higher valve authority can reduce excessive thermal energy. However, it significantly increases the pump power. To improve system performance without penalty to the pump energy, it is suggested that the loop pressure be reset according to the water flow rates. If loop pressure is adjusted proportional to the square of the water flow rate, the valve leakage rate can be maintained at or below the design leakage rate. Excessive thermal energy consumption can be minimized and pump power also can be reduced.

Table 3 Extra annual energy consumption versus valve authority (4% valve leakage, zero valve overlap, and zero space load oversize)

Valve authority	10%	20%	30%	40%	50%
Actual leakage rate	0.126	0.089	0.072	0.063	0.056
Cooling	2.4%	1.5%	1.2%	1.0%	0.8%
Heating	3.9%	2.5%	2.0%	1.7%	1.4%

Climate Impact

Simulations are performed for four representative locations: Atlanta GA, Chicago IL, Miami FL and San Antonio TX. Design temperatures are listed in Table 4.

The simulation assumes 25% system oversize, 4% valve design leakage rate, and 30% valve authority. The base case assumes zero valve leakage and zero oversize. Table 5 summarizes the simulation results. Climate significantly impacts the annual extra thermal energy consumption and the EDE. The annual extra thermal energy consumption ranges from 15% to 45% for heating and 13% to 20% for cooling. The extra fan power is 95% due to the 25% airflow oversize. The EDE varies from 0.28 to 0.52. Further, the coupled-control units perform poorly in a humid climate. In Miami, for example, the extra energy consumption is 45% for heating and 20% for cooling. The EDE value is as low as 0.28. In a relatively cold climate, however, the coupled-control units have a reasonable performance. In Chicago, the extra energy consumption is 15% for heating and 13% for cooling. The EDE value is 0.52.

Table 1 Summary of the extra annual energy consumption versus space load oversize

	10%	20%	30%	40%
Cooling energy	9%	10%	18%	26%
Heating energy	14%	16%	29%	41%
Fan power	33%	73%	120%	174%
EDE	0.31	0.31	0.29	0.27

Table 2 Summary of extra annual energy consumption versus valve leakage (50% valve authority, zero space load oversize, and zero valve overlap)

Design leakage rate	0.02	0.04	0.06	0.08	0.100
Actual leakage rate	0.028	0.056	0.085	0.113	0.141
Heating	0.6%	1.4%	5.2%	6.3%	7.7%
Cooling	0.4%	0.8%	1.3%	2.0%	2.6%

Table 4 Space and system design parameters (°C/°F)

Parameters	Miami	San Antonio	Atlanta	Chicago
Cooling dry bulb	33/92	37/98	34/93	33/92
Cooling wet bulb	26/78	24/75	24/75	23/74
Heating temperature	8/46	−3/26	−8/18	−20/−4

Table 5 Summary of extra energy consumption (4% of valve leakage, 30% of valve authority)

Location	Chicago	Atlanta	San Antonio	Miami
Heating	15%	20%	25%	45%
Cooling	13%	14%	15%	20%
Fan Power	95%	95%	95%	95%
EDE	0.52	0.38	0.36	0.28

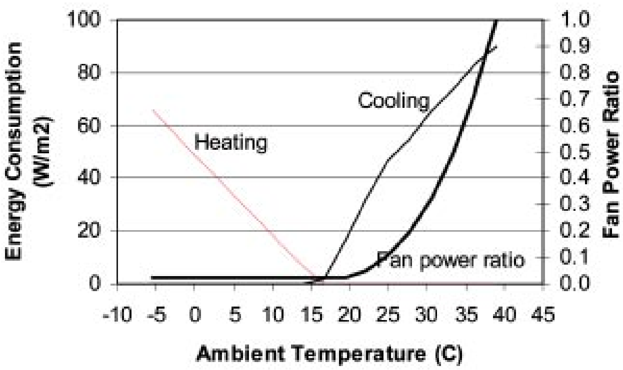

Fig. 5 Simulated energy performance of VAV coupled-control units (San Antonio Bin Weather)

VAV Coupled-Control Units

The coupled-control unit consumes constant fan power and uses a significant amount of heating and cooling for humidity control. Converting the unit into a VAV system can significantly decrease fan and thermal energy consumption. It is assumed that the VAV coupled-control unit maintains the designed discharge air temperature until the airflow is reduced to the minimum airflow rate. The constant outside air intake is maintained regardless of the total airflow rate. When the zone load ratio is less than the minimum airflow ratio, airflow is maintained at the minimum level. The supply air temperature is modulated to maintain room temperature and the humidity level.

A simulation is performed using the San Antonio U.S.A. bin weather. The minimum airflow is set at 30% of the design airflow. The valve leakage is assumed to be zero. Figure 5 presents the thermal and fan energy consumption as functions of the ambient temperature.

The simulation results show that the VAV coupled-control unit uses no simultaneous heating and cooling under the assumed conditions. The VAV coupled-control unit has an EDE value of 1. The VAV coupled-control unit decreases annual cooling energy consumption by 59%, and decreases the annual heating energy consumption by 85% and fan energy by 89%.

Simultaneous heating and cooling may still exist within a narrow outside air temperature range depending on zone load and climate characteristics. For example, if the zone needs heating when the outside air temperature is 16°C (60°F), the system will remove moisture and therefore use cool and warm air to maintain room temperature.

Conclusions

Couple-control units have poor energy delivery efficiency. The typical EDE value varies from 0.28 (Miami) to 0.52 (Chicago). Thermal energy consumption is two to four times higher than optimal energy consumption. Even when the units are properly designed, installed and operated, thermal energy consumption is three times higher than optimal thermal energy consumption under San Antonio weather conditions for typical commercial buildings.

Airflow oversize and valve leakage cause excessive thermal and fan energy consumption. To improve energy performance, the following measures are recommended: (1) balance the airflow based on the actual peak load; (2) use a high-quality valve and reset the water loop pressure to decrease the water leakage.

Converting the coupled-control unit into a VAV system is recommended. A VAV coupled-control unit will reduce and/or eliminate simultaneous heating and cooling and significantly reduce fan energy consumption.

Nomenclature

a = Constant
b = Constant
E = Energy (GJ/day, or MMBtu/day)
EDE = Energy Delivery Efficiency
N = Number of hours
Q = Space sensible load (W/m^2, or Btu/ft^2 h)
UA = Heat transfer coefficient (W/m^2 °C, or Btu/ft^2 h °F)
C_p = Specific heat of air (J/kg °C, or Btu/lbm °F)
C_{pa} = Specific heat of dry air (J/kg °C, or Btu/lbm °F)
$C_{p,sat,d}$ = Effective specific heat of saturated air (J/kg °C, or Btu/lbm °F)
W = Moisture load (kg/h, lb/h)
h = enthalpy (J/kg, Btu/lbm)
k = valve authority (0 to 1)
t = Temperature (°C or °F)
x = Position of the control valve (0 to 1)
w = Humidity ratio
$\dot{m}$ = Ratio of the actual and design flow (0 to 1)
λ = Leakage rate of the control valve (0 to 1)

Subscripts

a = Air
c = Cooling
d = Design
e = Entering
ext = Airside
dew = Dew point
h = Heating
int = Waterside
l = Leaving
m = Mixing
o = Outside
r = Room
s = Supply
sen = Sensible

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