Impacts of Water Loop Management on Simultaneous Heating and Cooling in Coupled Control Air Handling Units

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Abstract

The impacts of the water loop management on the heating and cooling energy consumption are investigated by using model simulation. The simulation results show that the total thermal energy consumption can be increased by 24% for a typical AHU in San Antonio, Texas when the differential pressure is increased from 7 psi to 21 psi for both chilled water and hot water loops. It appears that significant amounts of energy consumption can be reduced by improving the differential pressure of water loops. This paper presents the system model, methods, and the results.

Introduction

The coupled control unit is a typical single zone constant volume system. It is often packaged in the plant [ASHRAE, 96]. Figure 1 presents the systems diagram.

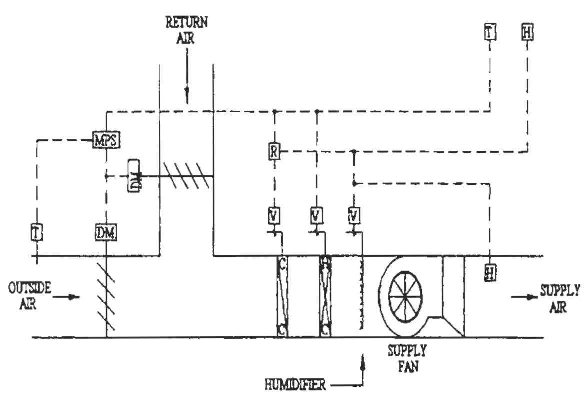

Figure 1: System Diagram of "Perfect Coupled Control Unit"

For a perfect coupled control unit, the thermostat controls the heating coil, ventilation dampers and cooling coil in sequence as thermal load varies in the conditioned space.
Ventilation
dampers are controlled for outdoor air cooling as first-stage cooling. When the outside air temperature rises to the point it can no longer be used for cooling, an outdoor air limit control overrides the signal to the ventilation dampers, and moves them to the minimum ventilation position, as determined by the minimum positioning switch. Where applicable, an enthalpy control system can replace the OA limit control for appropriate climate area. A zero energy band thermostat can separate the heating and cooling control ranges, thus saving energy [ASHRAE 1995].

Humidity is controlled by either a space humidistat or a return air humidistat. It controls a moisture-adding device for winter operation and can also override space thermostat control of the cooling coil for dehumidification during summer operation. In the latter case, the space thermostat activates the heating coil for reheat. The high-limit humidistat in the supply air duct prevents excessive duct moisture.

For a typical coupled control unit, humidistat is often not used. The unit is often controlled from a single space thermostat. Figure 2 presents the systematic control diagram.

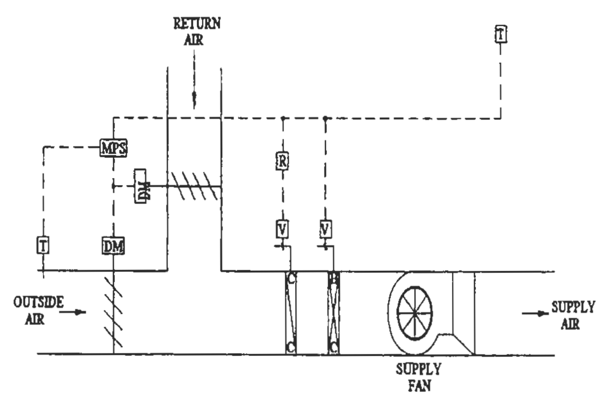

Figure 2: System Diagram of Typical Coupled Control Unit

The thermostat modulates both chilled water and hot water control valves to maintain room temperature. Both chilled water and hot water control valves receive the same control signal from the thermostat while one of the control valves is the direct action valve and another is the reverse action valve. To control the zone relative humidity level, an overlap of the action range is often used. Consequently, simultaneous heating and cooling occurs in this kind unit most of the time. It is well recognized that the range of over-lapping is critical for both comfort and building energy consumption.

The loop differential pressure may be as critical as the range of over-lapping of valve range for the zone comfort and energy consumption. For example, the AHU is delivering 60°F air to the zone with both the chilled water and hot water valves 20% open. Suddenly, the chilled water loop differential pressure in increased from 10 psi to 30 psi due a failure of VFD operation. The increased loop differential pressure will increase the chilled water flow significantly and decrease the air temperature to the heating coil. In order to maintain the same discharge air temperature to the zone, the heating coil must open more to provide more heating. Eventually, the thermostat will place both chilled water and hot water control valves at new positions. It is clear that the increased chilled water differential pressure increases both chilled water and hot water energy consumption comparing with the initial condition.

In existing systems, the differential pressure of the water loop is often significantly higher than designed value in most units due to (1) poor water balance; and (2) extreme requirement in one unit. The differential pressure of water loop is significantly higher than the design value due to (1) inappropriate control set-point; (2) lack of control modulation; and (3) failure of control system. Moreover, there is no method or guide line available to select the differential pressure of the water loop for design engineers. The selection of over-lap is often based on rule of thumb. Consequently, either extra energy is wasted or the room comfort cannot be maintained. Authors did not find any documented studies on these issues. It appears that the optimal of energy and comfort performance is a matter of "luck". Therefore, it is important to manage the water loop differential pressure to maintain room comfort and minimize the thermal energy consumption as well.

In this paper, a model building is selected. The annual energy consumption is simulated under the bin weather of San Antonio for different water loop pressure. The impacts of the water loop management are discussed based on these simulations.

Building and System Model

Building Model: To study the impact of the water loop management, numerical method is used in the paper. The following presents the building and the system models used in the simulation.

The building floor area is assumed to be 5,000 square feet. The room temperature is 75°F. The building heating and cooling load varies from -60,000 (heating) to 110,000 Btu/hr (cooling) when the ambient temperature increases from 35°F to 105°F. The total air flow rate is determined as 5,000 cfm with 500 cfm outside air intake. Under the peak cooling demand, the supply air temperature is determined as 55°F. Under the peak heating demand, the supply air temperature is determined to be 85°F. Figure 3 presents room load and the supply air temperature to the room versus the ambient temperature. When the ambient temperature is lower than 60°F, heating is needed. When the ambient temperature is higher than 60°F, cooling must be provided.

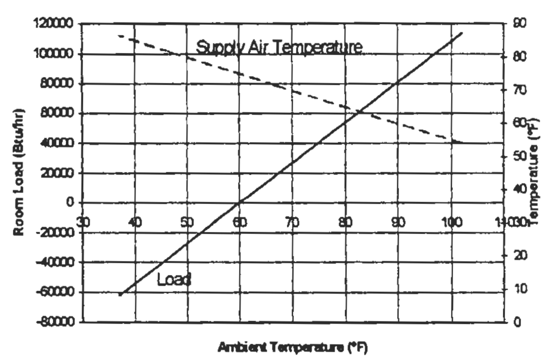

Figure 3: Building Load and the Supply Air temperature Versus the Ambient Temperature

This building is conditioned by a typical coupled control unit which consists of supply air fan, cooling coil, heating coil, chilled water control valve, heating water control valve, and a thermostat to modulate the control valve to maintain the suitable room temperature set-point. Both chilled water and hot water are supplied from a central plant.

Control Valve Model: A direct-react chilled water control valve is used. The chilled water control valve remains closed when the control signal is less than 8 psig. The chilled water control valve changes from fully closed to full open when the control signal increases from 8 psig to 13 psig.

A reverse acting valve is used for the heating water control. The heating water control valve remains closed when the control signal is higher than 10 psig. The heating water control valve changes from full closed to full open when the control signal decreases from 10 psig to 5 psig. When the control signal is lower than 5 psig, the heating water control valve is full open.

Figure 4 presents the sequence of the control valves. 2 psig over-lap is used in the model simulation.

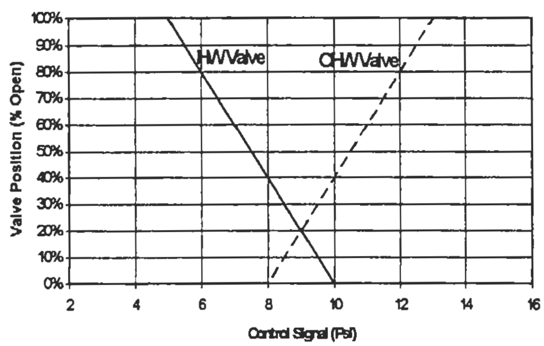

Figure 4: Control Sequence of the Chilled Water and Hot Water Control Valves

Both chilled water and hot water control valves are equal percentage type. Each equal increment of opening increases the flow by an equal percentage over the previous value. The flow ratio of the control valve is expressed by equation 1:

$$\dot{m} = \frac{1-\exp(kx)}{1-\exp(k)} \qquad (1)$$

Where $\dot{m}$ is the ratio of water flow rate to the maximum water flow when the valve is full open under the same differential pressure. x is the valve position which varies from 0 (full closed) to 1 (full open). k is the valve constant. In this study, the valve constant was selected to be 4 for both heating and chilled water control valves. The flow ratio is presented in Figure 5.

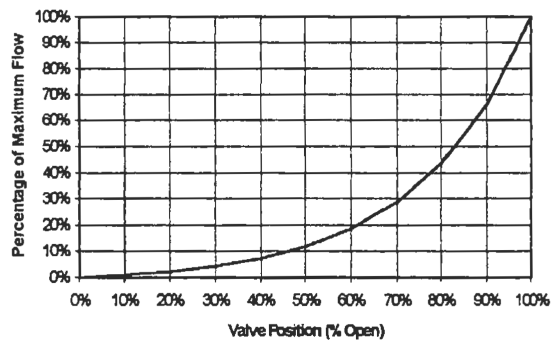

Figure 5: Flow Ratio Versus the Valve Position

Water Loop Model: It is assumed that when chilled water flow is 30 GPM, the pressure drop across the control valve is 5 psi and 2 psi across the coil and the pipe line. When the hot water flow is 20 GPM, the pressure drop across the hot water control valve is 5 psi and 2 psi for the coil and pipeline. The flow coefficient of the control valves are 9 GPM/psi$^{0.5}$ for hot water valve and 13 GPM/psi$^{0.5}$ for the chilled water control valve. The designed loop pressure is 7 psi for both chilled water and hot water loops. The water loop is modeled by Equations 2 to 5:

$$\Delta p_{coil} = \Delta p_{coil,deesign}\sqrt{\dot{m}} \qquad (2)$$

$$\Delta p_{valve} = \Delta p_{loop} - \Delta p_{coil} \qquad (3)$$

$$\dot{m} = C_v\sqrt{\Delta p_{valve}}\,\frac{1-\exp(kx)}{1-\exp(k)}m_{design} \qquad (4)$$

$$m = \dot{m} \times m_{design} \qquad (5)$$

Where: Δp_{coil} is the differential pressure across the coil under flow ratio of $\dot{m}$, $\Delta p_{coil,design}$ is the designed differential pressure across the

Coil Model: The heat transfer coefficients of heating coil is 3,000 Btu/°F/hr, and 12,000 Btu/°F/hr for the cooling coil. Both heating and cooling coil are simulated as a dry coil model by the NTU method, which are expressed by Equations 6 to 14 [Incropera and Witt, 1985]:

$$C_{air} = cfm \times 60 \times c_{p,air} \times \rho \quad (6)$$

$$C_{water} = m \times 60 \times c_{p,water} \quad (7)$$

$$\begin{aligned} C_{\min} &= \min(C_{air}, C_{water}) \\ C_{\max} &= \max(C_{air}, C_{water}) \end{aligned} \quad (8)$$

$$NTU = UA / C_{\min} \quad (9)$$

$$C_r = \frac{C_{\min}}{C_{\max}} \quad (10)$$

$$\varepsilon = 1 - \exp\left\{ \frac{NTU^{0.22}}{C_r} \left(\exp\left[-C_r NTU^{0.78} \right] - 1 \right) \right\} \quad (11)$$

$$Q = \varepsilon C_{\min} \left(T_{ws} - T_c \right) \quad (12)$$

The leaving air temperature from the heating coil can be calculated by equation 13:

$$T_s = T_c + \frac{Q}{C_{air}} \quad (13)$$

The enthalpy of the leaving air from the cooling coil is calculated by equation 14:

$$h_c = h_{mix} - \frac{Q}{C_{air}} \quad (14)$$

Using dry coil model will introduce certain error for the cooling coil simulation since the heat transfer coefficient is higher when the coil is wet.

Thermostat Model: The thermostat generates a pneumatic pressure signal from 3 to 15 psig. When the room temperature is lower than the set-point, the thermostat decreases the output control signal. When the room temperature is higher than the set-point, the thermostat increases the output control signal. In the model simulation, the supply air temperature is calculated based room sensible load first. Then, the hot water flow and the leaving water temperature are determined by the heat transfer model of the heating coil (Equations 6 to 12) by assuming entering air temperature to the heating coil. When the water flow rate is known, the position of the hot water valve can be determined using the water side model. Finally, the control signal is determined by equation 15:

$$PCS = HVC - x(HVC - HVO) \quad (15)$$

Where PCS is the control signal; HVC is the pressure corresponding the hot water valve is fully closed, HVO is the pressure signal corresponding the valve is fully open, and x is the position of the hot water control valve.

When the hot water valve is closed. The chilled water flow is determined first using the cooling coil model (Equation 6 to 12 and 15). Then, the control valve position is determined by the water side model. The control signal is then determined by Equation 16:

$$PCS = CVC - x(CVC - CVO) \quad (16)$$

Where CVC is the pressure signal with the chilled water valve full closed, and CVO is the pressure signal with the chilled water valve is full open.

Chilled Water and Hot Water Supply Temperatures: In this study, we assumed a constant chilled water temperature of 42°F. The hot water supply temperature decreases from 150°F to 110°F when the ambient temperature varies from 30°F to 100°F. Figure 6 presents both chilled water and hot water supply temperature versus the ambient temperature.

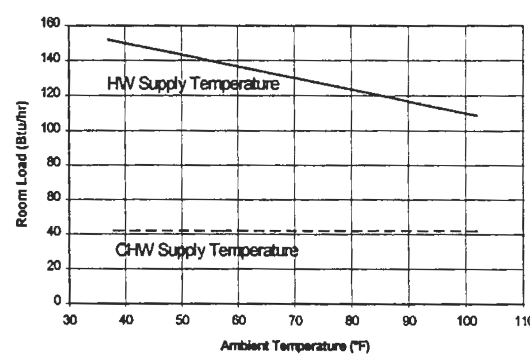

Figure 6: Chilled Water and Hot Water Supply Temperature Versus the Ambient Temperature

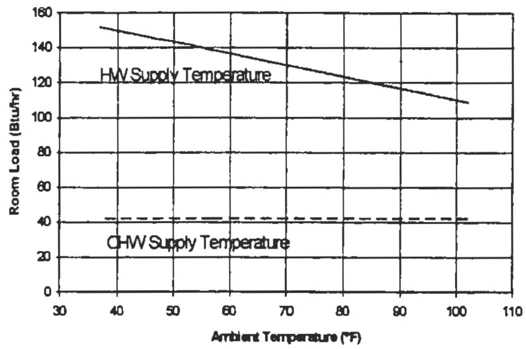

Figure 6: Chilled Water and Hot Water Supply Temperature Versus the Ambient Temperature

In this study, the impact of the supply air fan is neglected while the return air is assumed to be 2°F higher than the room air temperature to consider the heat gain through return air duct and return air fan.

Simulation and Results

A simulation program is developed to simulate the system operation and energy consumption. The Bin weather of San Antonio [ASHRAE, 1985] is used to investigated annual energy cost. The bin data is presented in Table 1.

To investigate the impacts of water loop differential pressure, two faults are introduced: (1) the chilled water loop is over pressurized from a differential pressure of 7 psi to 21 psi; and (2) the hot water loop is over-pressurized from differentials pressure of 7 psi to 21 psi. A total of 4 cases is constructed to investigate the impacts for each fault or the combination of the faults. Table 2 summarizes the simulation cases as well as the simulation results. The first column presents the case number. The second column presents the differential pressure of the chilled water. The third column presents the hot water differential pressure. Columns 4, 5, and 6 present the annual heating, cooling, and the sum of heating and cooling energy consumption.

Columns 7, 8, and 9 present the simulated annual energy increases for heating, cooling, and the sum of heating and cooling.

Case 1 is the base case where the water loop differential pressures are controlled at the design level for both hot and chilled water loop. The heating and cooling coils are designed properly. The annual heating and cooling energy consumption is 105 MMBtu and 388 MMBtu, respectively. The total energy consumption is 493 MMBtu or 98.6 kBtu/ft^2/yr. The system is operated 24 hours per day and 365 days a year.

When the differential pressure of chilled water loop is increased from 7 psi to 21 psi (case 2), the annual heating energy is increased by 24%, 9% for the cooling energy. The total annual energy consumption is increased by 12%. When the differential pressure of hot water loop is increased from 7 psi to 21 psi (Case 2), the annual heating energy consumption is increased by 15%, and 5% for the cooling energy. The total energy consumption is increased by 7%. When the loop water differential pressure is increased from 7 psi to 21 psi for both loops, the annual heating energy consumption is increased by 55%, and 17% for the cooling energy. The annual total energy consumption is increased by 25%. The simulation results shows significant impacts and the complex nature of the water loop management on the energy consumption. From these limited results it appears that (1) excessive differential pressure of the water loop can increase both heating and cooling energy consumption; (2) when the differential pressures are increased for both loops, the increase (25%) of the energy consumption is higher than the sum (17%) of the increase of the energy consumption when the differential pressure is increased separately; (3) the impact of chilled water loop differential pressure is different with the impact of the hot water loop differential pressure.

Table 1: Bin Weather Data of San Antonio

Dry Bulb (°F)	Dew Point (°F)	Wet Bulb (°F)	Time (hr)
102	59.3	73	3
97	64.0	74	136
92	64.7	73	465
87	67.1	73	570
82	66.1	71	827
77	68.4	71	1371
72	63.0	66	1133
67	59.2	62	1015
62	51.8	56	663
57	46.0	51	689
52	42.3	47	637
47	38.7	43	608
42	35.3	39	377
37	29.9	34	162
32	21.9	28	70
27	24.5	26	32
23	23.0	23	3

Table 2: Summary of Simulation Results

	ΔP_{chw}	ΔP_{hw}	Energy Consumption (MMBtu/yr)			Extra Energy Cost (MMBtu/yr)		
Case	(psi)	(psi)	Heating	Cooling	Total	Heating	Cooling	Total
1	7	7	105	388	493			
2	21	7	131	422	553	24%	9%	12%
3	7	21	121	405	527	15%	5%	7%
4	21	21	163	454	617	55%	17%	25%

Figures 7 a-h compare the detailed simulation results for Case 1 and Case 4. When the differential pressures are increased from 7 psi to 21 psi for both loops, the thermostat sends a higher control signal to the valves when the ambient temperature is lower than 62 °F (Figure 7a). Consequently, the chilled water control valve opens more even the differential pressure is 3 time higher than the base case (Figure 7b). When the ambient temperature is higher than 62°F, the thermostat sends lower signals to the control valves (Figure 7a), the hot water control valve opens more (Figure 7c) than in the base case. Consequently, the discharge air temperature of the cooling coil is lower than in the base case (Figure 7d) whenever both valves are open.

The impacts of hourly and annual energy consumption are presented in Figures 7e, 7f, 7g, and 7h.

Since the discharge air temperature of the cooling is impacted by the water loop pressure, the room relative humidity is affected as well when the humidistat is not used. If the differential pressure of chilled water loop is too low, the AHUs will loss the capacity to remove the moisture while still be able to provide suitable discharge air temperature. Therefore, it is critical to have a suitable water loop differential pressure to maintain the room relative humidity.

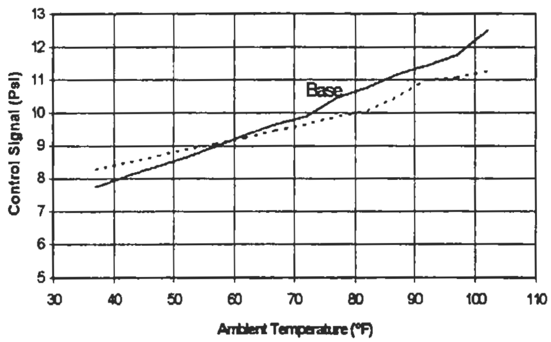

Figure 7a: Comparison of Control Signal From the Thermostat Under Different Loop Pressure (7 psi for case 1 or base and 21 psi for case 4)

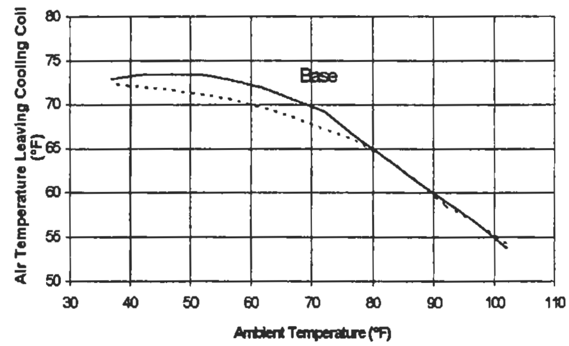

Figure 7d: Comparison of Discharge Air Temperature of Cooling Coil Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

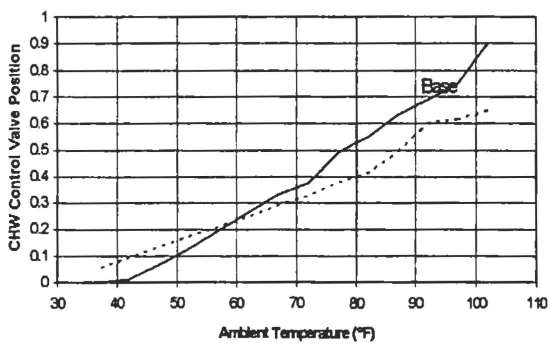

Figure 7b: Comparison of Position of Chilled Water Control Valve Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

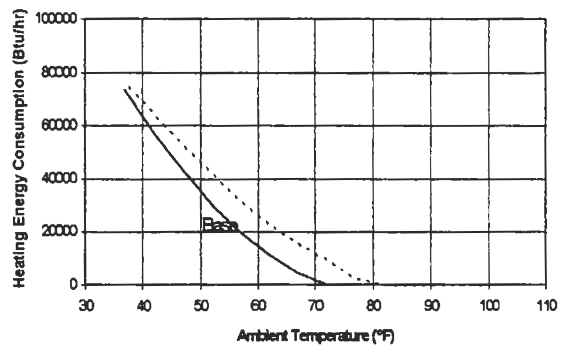

Figure 7e: Comparison of Heating Energy Consumption Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

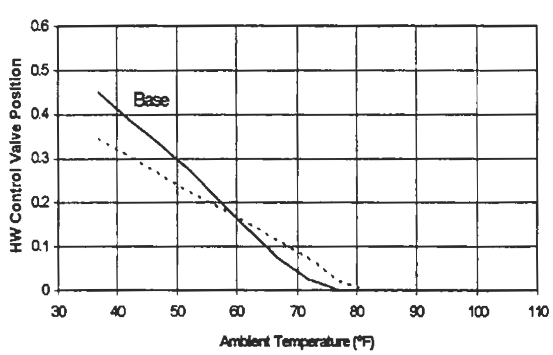

Figure 7c: Comparison of Position of Hot Water Control Valve Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

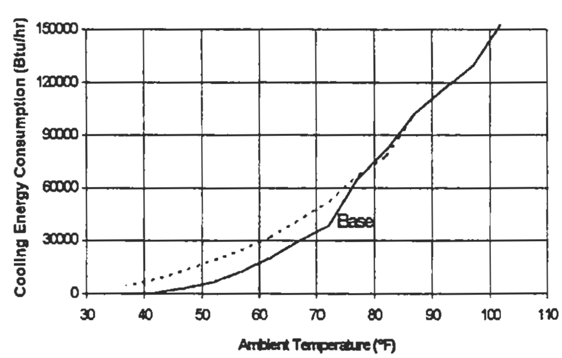

Figure 7f: Comparison of Cooling Energy Consumption Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

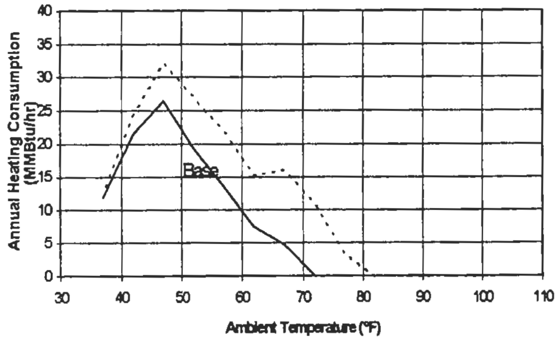

Figure 7g: Comparison of Annual Heating Energy Consumption Versus the Ambient Temperature Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

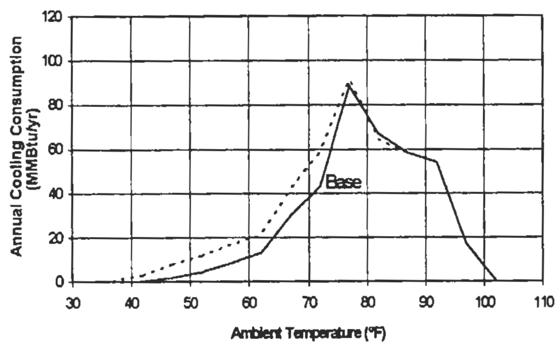

Figure 7h: Comparison of Annual Cooling Energy Consumption Versus the Ambient Temperature Under Different Water Loop Differential Pressure (7 psi for case 1 or base and 21 psi for case 4)

In this investigation, only water loop differential pressure is investigated. However, other parameters may also have important effects on the annual energy consumption. For example, the size of the cooling and heating coils, type of control valves, chilled water supply temperature, hot water supply temperature, and the amount of overlap.

The dry coil model is used in this analysis for the cooling coil. This can cause some error for the chilled water energy consumption. However, it tends to make the increase of energy increase appears smaller than it should be. Consequently, the increase of the energy increase is very conservative. The humidifier and economizer may need to be investigated as well.

Conclusion

The impacts of the differential pressure of water loops on the heating and cooling energy consumption are investigated by using model simulations. The simulation results show that the total thermal energy consumption can be increased by 25% for a typical AHU in San Antonio, Texas when the differential pressure is increased from 7 psi to 21 psi for both chilled water and hot water loops. It appears that significant amounts of energy consumption can be reduced by improving loop management in the existing system.

Two parameters are investigated in this study. The impacts of the following parameters need to be investigated systematically: the size of the cooling coil, type of control valves, chilled water supply temperature, hot water supply temperature, the over-lap, the supply air flow rate and outside air intake, and the load profile and climate. This investigation is focused on the thermal energy. Future studies should include the room relative humidity control as well. An optimized water loop management schedule should be able to be developed when the nature of operation is fully understood. The research results will also be useful for design engineers to design the system properly.

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