SLIDING CONTROL OF A CHILLED WATER, FAN-COIL SYSTEM

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ABSTRACT

The control circuits used in conjunction with the chilled water, fan-coil system were investigated. The problems associated with their performances were discussed. The adaptive sliding control law was derived since robustness is the major consideration in the control strategy of the room temperature, especially in the presence of system nonlinearity, load disturbances, and uncertain time delays. The effectiveness of sliding control over the proportional, mixing, and injection control circuits was demonstrated.

NOMENCLATURE

A_o	Outside heat transfer surface area of coil, m^2.
B	Coil outside to inside surface area ratio.
c_{pa}	Specific heat of air at constant pressure, J/kg K.
c_{va}	Specific heat of air at constant volume, J/kg K.
c_w	Specific heat of water, J/kg K.
d(t)	Disturbance.
h_a	Air side heat transfer coefficient, W/m^2 K.
h_w	Water side heat transfer coefficient, W/m^2 K.
k_p	Controller gain, volt/°C.
m_a	Air mass flow rate, kg/s.
m_{chs}	Water mass flow rate from central chiller, kg/s.
m_{cos}	Water mass flow rate supplied to coil, kg/s.
m_r	Mass of air inside the room, kg.
NTU	Number of transfer units.
Q_{coil}	Cooling capacity of coil, W.
Q_L	Cooling load, W.
R	Heat capacity ratio.
s	Laplace time inverse, s^{-1}.
S(x,t)	Sliding surface condition, °C.
T_{a1}	Air inlet temperature to coil, °C.
T_{a2}	Air outlet temperature from coil, °C.
T_{cor}	Return water temperature from coil, °C.
T_{cos}	Supply water temperature to coil, °C.
T_r	Room temperature, °C.
$T_{r,d}$	Room design temperature, °C.
t_s	Settling time, s.
U_o	Overall heat transfer coefficient of coil, W/m^2 K.
v	Valve actuating signal, volt.
v_{max}	Valve maximum actuating signal, volt.
x	Mass fraction defined in equations(9),(10) and (11).
ϵ_a	Air side effectiveness.
ϵ_w	Water side effectiveness.
η	Fin efficiency.
τ_{coil}	Time constant of coil mass, s.
τ_p	Pipe transport delay time, s.

INTRODUCTION

The chilled water, fan-coil system is acknowledged in air conditioning applications where chilled water is already available and circulated in large quantities. These applications include industrial and chemical plants, hospitals, and hotels. Schematic arrangements of the fan-coil system and its conventional hydraulic circuits are shown in Figure (1).

Some problems are encountered in system responses of conventional circuits such as Hunting problem, long settling time, and maximum overshoots [1-4]. The difficulties in designing the fan-coil controller are the nonlinear characteristics of its model, the time varying nature of the system parameters, (especially time delays), and the unknown changes in the load. Sliding-mode control has been adapted as an effective methodology that can be used to control a large class of nonlinear system despite system parameter variations [5-7]. Using this approach, it is possible to achieve consistent robust fan-coil unit performance under

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This paper investigates the effectiveness and the feasibility of sliding-mode control of the fan-coil system.

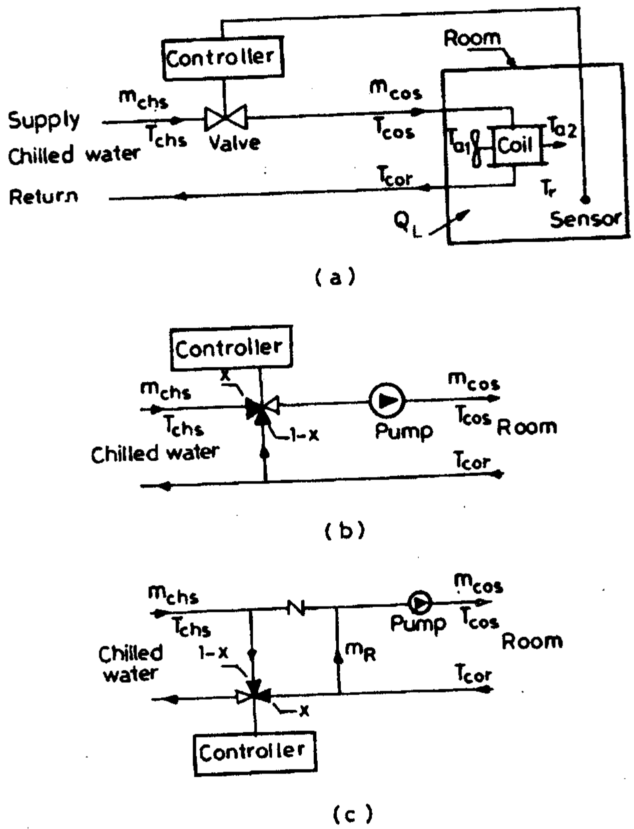

Figure 1. Schematics of the fan-coil system and its conventional control circuits.
a-proportional b-mixing c-injection

CHILLED WATER, FAN-COIL MODEL

The steady-state performance of the fan-coil system can be analyzed using the simple effectiveness model for a counter-flow heat exchanger [1]

$$\epsilon_w = (T_{cor} - T_{cos})/(T_{a1} - T_{cos})$$

$$\epsilon_a = (T_{a1} - T_{a2})/(T_{a1} - T_{cos}) \qquad (1)$$

$$\epsilon_w = R\,\epsilon_a$$

where $R = m_a\, c_{pa} / m_{cos}\, c_w$

ϵ_w and ϵ_a are the water-side and air-side effectiveness, respectively, T_{a1} and T_{a2} are the air inlet and outlet temperature, T_{cos} and T_{cor} are the inlet and outlet water temperature at the coil ends, m_a and m_{cos} are air and water mass flow rates supplied to the coil, and c_{pa} and c_w are the specific heat of air and water, respectively. The air-side effectiveness is given by the standard heat exchanger equation [3].

$$\epsilon_a = \frac{1 - e^{-NTU\,(1-R)}}{1 - Re^{-NTU(1-R)}} \qquad (2)$$

where

$$NTU = U_o\, A_o / m_a\, c_{pa}$$

$$U_o = 1/(1/(\eta\, h_a) + B/h_w)$$

η is the fin efficiency, B is the surface area ratio, h_a and h_w are the heat transfer coefficients of air-side and water-side, respectively. The heat transfer coefficient of water side depends on the water mass flow rate

$$h_w \propto (m_{cos})^{0.8} \qquad (3)$$

According to Borresen [4], cooling coil dynamics is given as :

$$T_{cor} = \frac{((1-\epsilon_w)T_{cos} + \epsilon_w T_{a1})}{(1 + s\,\tau_{coil})} \qquad (4)$$

where τ_{coil} is the time constant of the coil to respond to a step input water temperature which is inversely proportional to m_{cos}.

Because of the significant length of the pipe connecting the fan-coil to the actuating valve, there is a transport delay time τ_p. The transfer function of the pipe delay is

$$m_{cos} = m_{chs}\, e^{-s\,\tau_p} \qquad (5)$$

where m_{chs} is the water mass flow rate supplied from the central chiller and m_{cos} is the water mass flow rate supplied to the coil which is inversely proportional to τ_p.

Considering the case of all return air; i.e T_{a1} is equal to the room temperature T_r, then the dynamics of the room temperature is governed by the form

$$m_r\, c_{va}\, \dot{T}_{a1} = Q_L - Q_{coil} \tag{6}$$

where m_r is the mass of the air inside the room, Q_L is the cooling load, and Q_{coil} is the coil cooling capacity

$$Q_{coil} = m_{cos} c_w (T_{cor} - T_{cos})$$

$$= \frac{m_{cos} c_w \epsilon_w (T_{a1} - T_{cos})}{(1 + s\tau_{coil})} \tag{7}$$

HYDRAULIC CONTROL CIRCUITS

The function of the hydraulic control circuit connected to the fan-coil unit is to control the coil cooling capacity by modulating either the mass flow rate or the temperature of the water supplied to the coil. The difference between the measured room temperature and its desired value is the feedback signal. The most common circuits used in conjunction with the fan-coil unit are the proportional, mixing, and injection circuits, Figure(1).

Proportional Circuit

The output signal from a proportional controller, Figure (1-a), is used to actuate a 2-way valve to modulate the mass flow rate of the water supplied to the coil

$$\Delta v = k_p\,(\,T_{r,d} - T_{a1}\,) \tag{8}$$

where k_p is the controller gain, and $T_{r,d}$ is the desired room temperature. The valve performance is assumed to be linear, i.e.

$$x = \frac{v}{v_{max}} = \frac{m_{cos}}{m_{cos,max}} \tag{9}$$

Since R, h_w, NTU, ϵ_w, ϵ_a, τ_p, and τ_{coil} are function of m_{cos}, therefore their values will vary with time.

Mixing Circuit

The circuit, Figure(1-b), consists of a three-way valve, a pump, and a proportional controller. The pump maintains a constant water flow rate to the coil. The inlet water temperature, T_{cos}, and the coil cooling capacity is controlled by adjusting the fraction of the return flow mixed in the three-way valve.

$$T_{cos} = (1-x)\, T_{cor} + x\, T_{chs} \tag{10}$$

where

$$x = \frac{m_{chs}}{m_{cos}} = \frac{v}{v_{max}}$$

T_{chs}, and m_{chs} are the temperature and the mass flow rate of the water delivered from the central chiller. It should be noted that the effectiveness and the time delays remain constant at their design values because m_{cos} is constant.

Injection Circuit

The circuit, Figure (1-c), consists of a three-way valve, a by-pass, a pump, and a proportional controller. The water mass flow rate supplied to the coil is constant, but the supplied water temperature is controlled according to the relation:

$$T_{cos} = \frac{m_{chs}}{m_{cos}} x\, T_{chs} + \left(1 - \frac{m_{chs}}{m_{cos}} x\right) T_{cor} \tag{11}$$

where

$$x = \frac{m_{cos} - m_R}{m_{chs}}$$

m_R is the mass flow rate of the return water mixed with the supplied water to the coil.

SLIDING CONTROL

The nonlinearity of the fan-coil model is associated with air and water effectiveness. Moreover, there are uncertainties in the model parameters because of the imprecision of both the effectiveness and time delays model, aging, and unknown variations in the cooling load. Also inaccurate time delay presents a high frequency unmodeled dynamics. While the frequency content of the control action must be as large as possible to reduce the effects of parameters uncertainties, it must still be kept low enough not to excite the high frequency unmodeled dynamics. Due to the highly nonlinear nature of the model, and the need for robustness of the design, sliding control [6] is a likely choice which is similar in construction of propotional control, Figure (1-a).

The fan-coil model can be reduced to a simple first order nonlinear equation,

$$\dot{x} = f(x) + g(x)\,u + d(t) \tag{12}$$

with an output

$$y = x(t)$$

where

$$x = T_{a1} - T_{r,d}$$

$$f(x) = 0$$

$$g(x) = \frac{\epsilon_w\, m_{cos,d}\, c_w (T_{a1} - T_{cos})}{m_r c_{va} v_{max}} C$$

$$\Delta m_{cos} = u\, m_{cos,d} / v_{max}$$

$$d(t) = \Delta Q_L / (m_r c_{va})$$

d(t) represents the disturbances, and C is a parameter whose value is not exactly known and varying with time due to the variation of the effectiveness and time delays. The variation in g(x) is not exactly known but bounded. In order to design a robust feedback controller, a smooth version of the sliding control will be applied [7]. The sliding surface S(x,t) is defined as the deviation of the room temperature from its desired value

$$S(x,t) = T_{a1} - T_{r,d} \tag{13}$$

with the sliding condition:

$$\dot{S} = \dot{T}_{a1} - \dot{T}_{r,d} \leq -\lambda S \tag{14}$$

where λ is a positive constant related to the rate of tracking. The controller structure that satisfies condition (14) is

$$u = \frac{1}{\hat{g}} (-\hat{f} - K\,S) \tag{15}$$

where $\hat{g}$ and $\hat{f}$ are the model of g and f.

$$\hat{g} = \left(\frac{\epsilon_w\, m_{cos}\, c_w}{m_r\, c_{va}\, v_{max}}\right)_{design} (T_{a1} - T_{cos})\, C \tag{16}$$

K must be chosen to satisfy equation (14) in the presence of a bounded model-error. If $|f - \hat{f}|$ or $|g - \hat{g}|$ becomes large, K should be made large to guarantee tracking accuracy. Reducing the model error, and consequently K, can be achieved by implementing an adaptive law for the parameter C in conjunction with sliding control.

C Adaptive Law

The adaptation law for the parameter C can be derived from a Lyapunov analysis [7]. If the error in C is ΔC, the Lyapunov function can be constructed as:

$$V = \frac{1}{2} (S^2 + \Delta C^2) \tag{17}$$

Taking the derivative of V and imposing constraints to ensure negative semi-definite results in the following stabilizing adaptive law :

$$\dot{C} = \frac{S}{C} \left(\dot{T}_{r,d} - \lambda\, S - \frac{\Delta Q_L}{m_r\, c_{va}} \right) \tag{18}$$

It should be mentioned that the parameter C will not converge to its actual value, but the adapted value will guarantee the tracking to the desired room temperature.

DISCUSSION OF RESULTS

The system model and various control circuits were analyzed numerically using time marching technique with time step 0.01 second. The effects of a 50% decrease in the cooling load on the system performance were investigated. The values of the system parameters used in the simulation are : m_a = 0.7214 kg/s ,m_{chs}

= 0.26 kg/s, T_{chs} = 6°C, B = 20, and η = 0.9. The design conditions are: Q_L =6525 watts, $T_{r,d}$= 24°C , x = 1, v_{max}= 10 volts, m_r= 70 kg, R = 0.5, ϵ_a= 0.545, h_w= 2800 W/m²K, h_a= 100 W/m²K, and NTU = 0.94 .

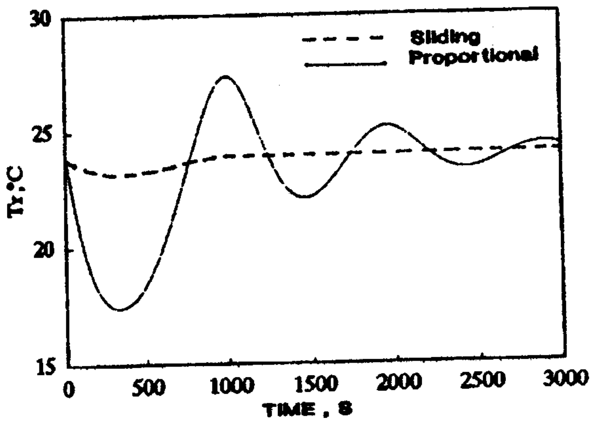

Figure 2. Responses of the room temperature with both proportional and sliding control.

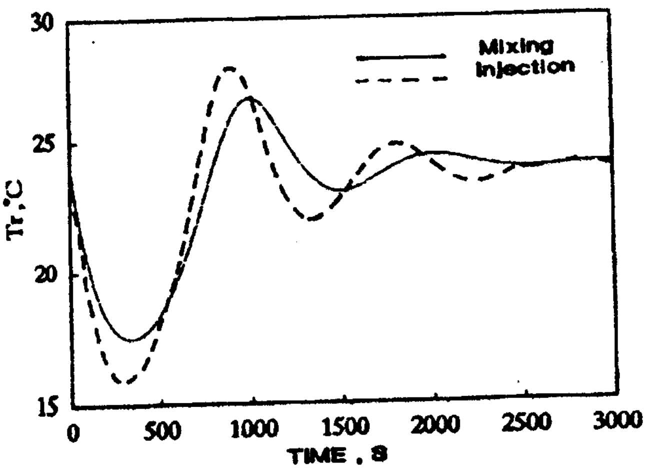

Figure 3. Responses of the room temperature with mixing and injection control circuits.

Figures (2) and (3) show the transient response of room temperature. The proportional, mixing, and injection circuits undergo oscillatory responses. The amplitude of oscillation is larger with injection circuit than those associated with the proportional and mixing circuits. The maximum overshoots are 8°C in the injection circuit, and 6.5°C in both the proportional and mixing circuits. Damping in the mixing circuit is higher than damping in other conventional ones. The settling time based on 2% error is 2800 s in the mixing circuit compared to 3000 s and 3800 s in the injection and proportional circuits, respectively. Meanwhile, upon implementing the sliding circuit, which is similar in construction with the proportional, Figure(1-a), the oscillations are damped. Since the room time constant is long, and because of the sharp decrease in the cooling load, an overshoot of 1°C which is practically accepted is traced and the settling time is 600 s upon 2% error. However, for a gradual decrease in the load, the sliding controller keeps the room temperature at the desired value for all time.

The responses of water mass flow-rate supplied to the coil are plotted in Figure (4). The steady-state value is 0.06879 kg/s for the proportional and sliding controller. Water mass flow rate is kept constant at 0.26 kg/s, and 0.3464 kg/s for the mixing and the injection circuits, respectively. This explains the maximum overshoot in room temperature experienced with the injection circuit. Water outlet temperature from the cooling coil responses are illustrated in Figures (5), and (6). The oscillatory behavior is exhibited upon using the conventional circuits. The steady-state outlet temperature is 17.33°C when either the proportional or the sliding controller is applied, while it reaches 18.13°C and 19.34°C for the mixing and injection circuits, respectively. This increase in the outlet temperature is attributed to the increase in water mass flow rate.

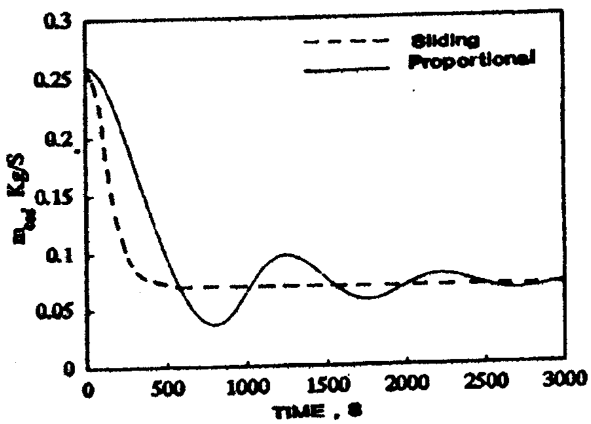

Figure 4. Responses of water mass flow-rate with both proportional and sliding control.

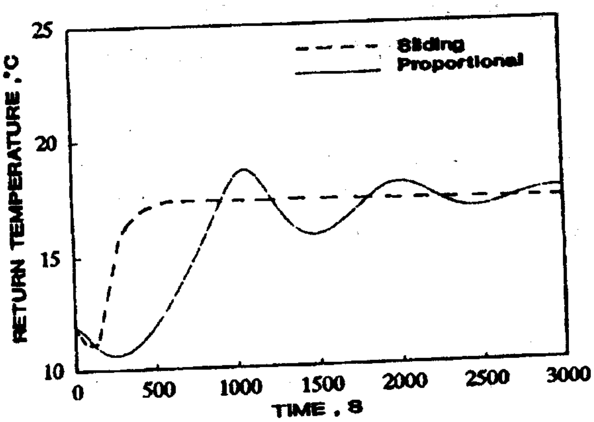

Figure 5. Responses of return water temperature with both proportional and sliding control.

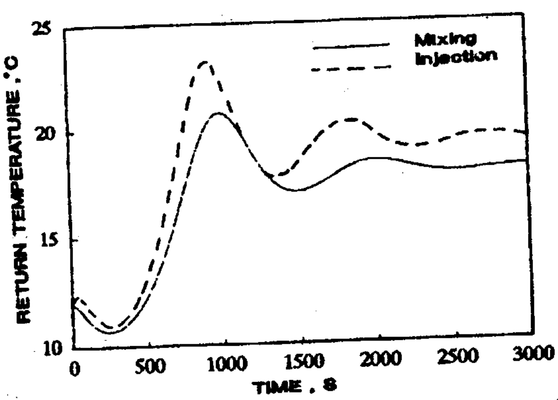

Figure 6. Responses of return water temperature with mixing and injection control circuits.

The effects of a 50% decrease in room cooling load on the coil cooling capacity are presented in Figures (7) and (8). Hunting problem is experienced when proportional, mixing, and injection circuits are used. The sliding control results in smooth operation. The cooling energy consumed until the steady state is reached, and the energy consumed during the first 3000 seconds for different circuits are compared in Table (1). The energy consumed until the steady-state is much smaller in the case of sliding control than the other controls since the settling time resulted from the sliding control is the smallest. However, it is clear from the table that the energy consumed during the first 3000 seconds is almost the same for all circuits. That is attributed to the fluctuations about the steady-state cooling load in the conventional circuits. On the other hand, in the case of sliding control, there is a high demand placed to adjust the room temperature at the desired value and to increase robustness rather than minimizing the control effort.

Table (1). Comparison between the cooling energy consumed in different circuits.

Type	$\int_0^{t_s} Q\,dt$, KJ	$\int_0^{3000} Q\,dt$, KJ
Proportional	12371	9761
Mixing	9134	9786
Injection	9796	9796
Sliding	2447	9787

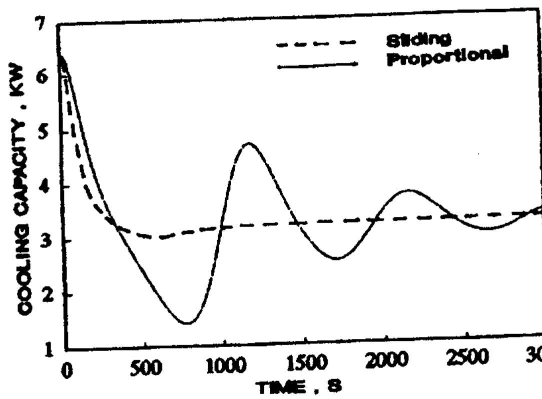

Figure 7. Responses of coil cooling capacity with both proportional and sliding control.

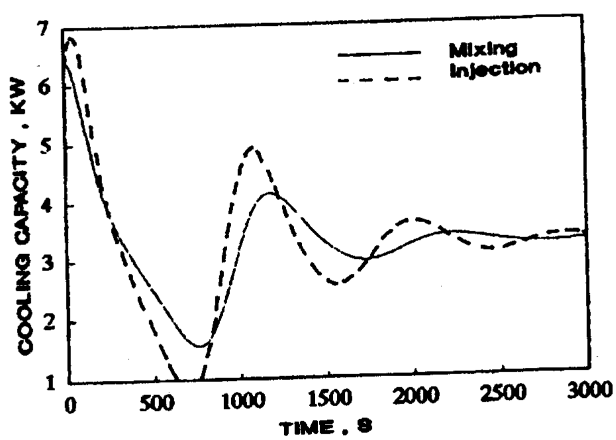

Figure 8. Responses of coil cooling capacity with mixing and injection control circuits.

CONCLUSIONS

The control of a chilled water, fan-coil system was examined. The conventional circuits (proportional, mixing, and injection) used in conjunction with fan-coil unit resulted in oscillatory responses, long settling time, and large overshoots. Adaptive sliding control was introduced to adjust the room temperature at the desired value in the presence of load disturbances, system nonlinearity, and uncertainty in time delays. The effectiveness and the superiority of the sliding control over the conventional circuits were demonstrated. The performance of the fan-coil system with adaptive sliding control is smooth and no oscillatory behavior is exhibited.

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