11

Air-Conditioning PID Control System with Adjustable Reset to Offset Thermal Loads Upsets

Takanori Yamazaki[1], Yuji Yamakawa[2],
Kazuyuki Kamimura[3] and Shigeru Kurosu[4]
[1]Oyama National College of Technology,
[2]Univ. of Tokyo,
[3]National Institute for Environmental Studies,
[4]Research Inst. "Crotech",
Japan

1. Introduction

The heating, ventilating, and air-conditioning (HVAC) systems have huge different characteristics in control engineering from chemical and steel processes. One of the characteristics is that the equilibrium point (or the operating point) usually varies with disturbances such as outdoor temperature (or weather conditions) and thermal loads. The variations of the operating point intend to vary parameters of a plant model. Thus, the HVAC control systems are extremely difficult to obtain an exact mathematical model (Kasahara 2000). Proportional-plus-integral (PI) controllers have been by far the most common control strategy as the complexity of the control problem increased (Åström 1995).
Today, a variable air volume (VAV) system has been universally accepted as means of achieving energy efficient and comfortable building environment. While the VAV control strategies provide a high quality environment for building occupants, the VAV system analysis rarely receives the attention it deserves. As a result, basic control strategies for the VAV system have remained unchanged up to now (Hartman 2003).
In addition, applying the model predictive control method to the HVAC systems, the control performance has been highly improved by pursuing the deviation from the operating point (Taira 2004). According to this report, recognizing the deviation from the operating point and calculating the optimal control inputs about the newly obtained operating point on next sampling time, the control system gives better responses than the traditional feedback control system.
Motivated by these considerations in these reports, we consider the room temperature and humidity controls using the adjustable resets which compensate for thermal loads upsets. One of the primary objectives of the HVAC systems is to maintain the room air temperature and humidity at the setpoint values to a high quality environment for building occupants. The room temperature and humidity control systems may be represented in the same block-diagram form as single-variable, single-loop feedback control systems because this interaction is weak relative to the desired control performance.

In some applications, disturbances can be estimated in advance before they entered the plant. Particularly, in the HVAC systems, it is possible that the outdoor thermometer detects sudden weather changes and the occupant roughly anticipates thermal loads upsets. Using this information, disturbances can be offset by the compensation of the reset, which is the exactly same function as an integral (I) control action. In the previous paper, the compensation method of the reset for PID controllers was proposed and the control system for room air temperature was often effective in reducing thermal loads upsets (Yamakawa 2010).
In this paper, of special interest to us is how to tune PID parameters more effective for the room temperature and humidity control. And the control performances for compensation of the adjustable reset are compared with the traditional method of the fixed reset. Namely, obtaining the approximate operating point using outdoor temperature and thermal loads profiles and adjusting the reset, the stabilization of the control system will be improved. The validation simulations will be demonstrated in terms of three performance indices such as the integral values of the squared errors, total control input, and PID control input.

2. Plant and control system

In this paper, we consider only the cooling mode of operation in summer and therefore refer to this system as a room air cooling system. The definition of variables in Equations is described in NOMENCLATURE.

2.1 Dynamics of air-conditioning system

To explore the application of PID controllers to the room temperature and humidity control system, we consider a single-zone cooling system, as shown in Figure 1. It is due to the fact that cooling and heating modes are found to perform nearly the same under most circumstances. The controlled room (the controlled plant) measures 10 m by 10 m by 2.7 m and is furnished with an air-handling unit (AHU) consisting of the cooling coil and the humidifier to control room air temperature and humidity. In general, since the responses of the AHU are faster than those of the controlled room, the dynamics of the AHU may be neglected for all practical purposes. Thus, as will be seen later, this rough assumption may be fairly validated. The model, however, possesses the important elements (the controlled room and the AHU) to analyze the air-conditioning system.
With this system, the room air temperature (θ) and relative humidity (φ) are measured with a thermometer and a hygrometer (sensors). The output signals from the sensors are amplified and then fed back to the PID controllers. Using the errors defined as the differences between the setpoint value (θ_r and φ_r) and the measured values of the controlled variables (θ and φ), the PID controllers generate the control inputs for the actuators (the supply air damper and the humidifier) so that the errors are reduced. The AHU responds to the control inputs (f_s and x_s (is adjusted by humidifier h)) by providing the appropriate thermal power and humidity to the supply airflow. Air enters the AHU at a warm temperature, which decreases as air passes the cooling coil, and then the humidifier supplies steam to cooled air if necessary. This occurs in a momentary period because there are a lot of times when the humidifier is not running. In this AHU, a dehumidifier is not installed, so an excessive demand for humidity is difficult to achieve.

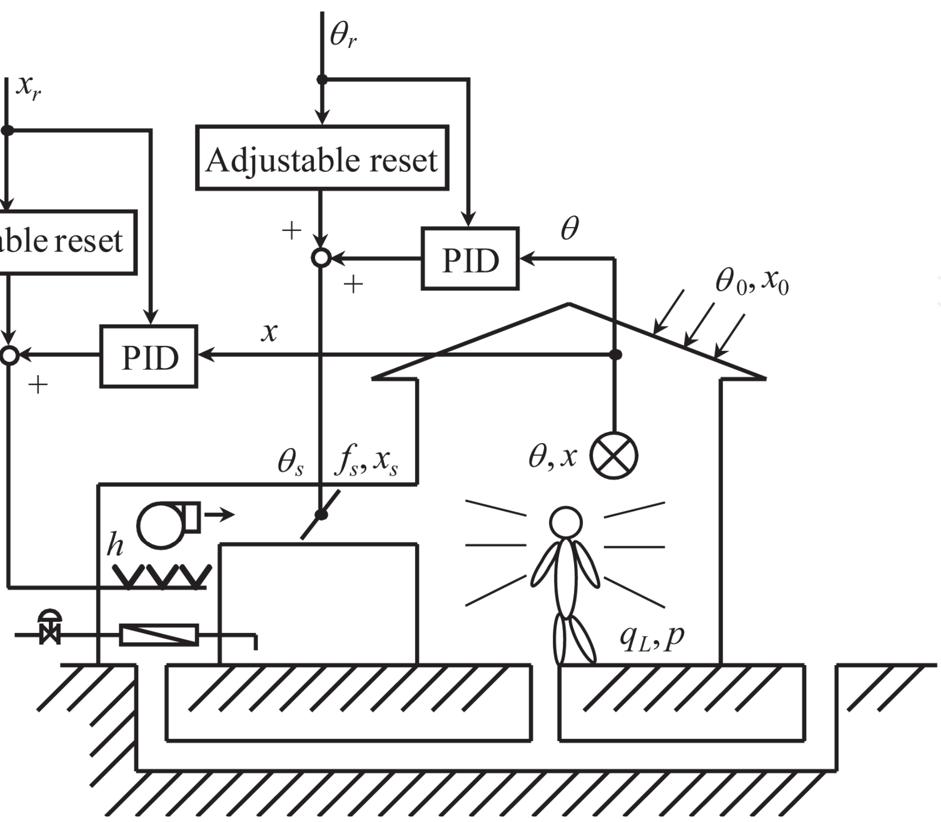


Fig. 1. Overall structure of a single-zone cooling system.

2.1.1 Room temperature model

Simplifying this thermal system to be a single-zone space enclosed by an envelope exposed to certain outdoor conditions is of significant interest to treat the fundamental issues in control system design (Zhang 1992, Matsuba 1998, Yamakawa 2009). This simplified thermal system (the room temperature model) can be obtained by applying the principle of energy balance,

$$C\frac{d\theta}{dt} = w_s\left(\theta_s - \theta\right) + \alpha\left(\theta_0 - \theta\right) + q_L \tag{1}$$

where

C = overall heat capacity of air-conditioned space [kJ/K],
α = overall transmittance-area factor [kJ/min K],
q_L = thermal load from internal heat generation [kJ/min],
$w_s = \rho_a c_p f_s$ [kJ/min K], which is heat of supply air flowrate,
ρ_a = density of air [kg/m^3],
c_p = specific heat of air [kJ/kg K],
f_s = supply air flowrate [m^3/min].

The physical interpretation of Equation 1 is that the rate of change of energy in the room is equal to the difference between the energy supplied to and removed from the room. The first term on the right-hand side is the heat loss which is controlled by the supply air flowrate. The second term is the heat gain through the room envelope, including the warm air infiltration due to the indoor-outdoor temperature differential. The third term is the

thermal loads from the internal heat generation and the infiltration. In this simplified model, any other uncontrolled inputs (e.g., ambient weather conditions, solar radiation and inter-zonal airflow, etc) are not considered.
It should be noted that all variables such as θ, θ_s, θ_0, q_L and w_s in Equation 1 are obviously the function of a time t. For the sake of simplicity the time t is not presented. When realizing a digital controller, a deadtime exists between the sampling operation and the outputting time of control input, thus w_s, namely f_s, includes a deadtime L_P.
These plant parameters have been obtained by experimental results (National Institute for Environment Studies in Tsukuba, Yamakawa 2009). The room dynamics can be approximated by a first-order lag plus deadtime system from the experimental data (Åström 1995, Ozawa 2003). Thus, the plant dynamics including the AHU and the sensor can be represented by,

$$P(s) = \frac{K_P}{T_P s + 1} e^{-L_P s} = \frac{0.64}{18s + 1} e^{-2.4s} . \tag{2}$$

Comparing to Equation 1, the plant gain (K_P) and the time constant (T_P) can be given by,

$$K_P = \frac{\theta_s}{w_s + \alpha},\ T_P = \frac{C}{w_s + \alpha},\ w_s = \rho_a c_p f_s . \tag{3}$$

Therefore, K_P and T_P change with the control input (the supply air flowrate f_s). Similarly, it is assumed that L_P changes with the control input. Namely,

$$L_P = \frac{L_{P0}}{w_s + \alpha}, \tag{4}$$

where L_{P0} is determined so that L_P is equal to 2.4 [min] when f_s is equal to 50 [%]. From L_P = 2.4 [min], $w_s = \rho_a c_p f_s$ = 10.89 [kJ/min K] and α = 9.69 [kJ/min K], L_{P0} can be obtained to be equal to 49.4 [kJ/K]. It is easily be found that these parameters are strongly affected by the operating points. Carrying out an open-loop experiment in the HVAC field to measure K_P, T_P and L_P is one way to get the information needed to tune a control loop.
To get some insight into the relations between Equation 1 and Equation 2, we will describe a bilinear system in detail (Yamakawa 2009). Introducing small variations about the operating points and normalizing the variables, Equation 1 has been transformed to a bilinear system with time delayed feedback. A parametric analysis of the stability region has been presented.
The important conclusion is that the stability analysis demonstrated the validity of PID controllers and there was no significant advantage in analyzing a bilinear system for VAV systems. It was fortunate that the linear system like a first-order lag plus a deadtime system derived in Equation 2 often satisfactorily approximated to the bilinear system derived in Equation 1. The linear system is an imaginary system, but it does represent it closely enough for some particular purpose involved in our analysis.
Certainly the linear model derived in Equation 2 can be used to tune the PID controller and the physical model derived in Equation 1 can be used for numerical simulations. Over the range upon which this control analysis is focused, the relations between Equation 1 and Equation 2 are determined to be sufficiently close.

2.1.2 Room humidity model

The room humidity model can be derived by applying the principle of mass balance,

$$V\frac{dx}{dt} = f_s\left(x_s - x\right) + \frac{n}{\rho_a}p \tag{5}$$

where

V = room volume (10×10×2.7[m^3])
x = absolute humidity of the room [kg/kg (DA)]
x_s = absolute humidity of the supply air [kg/kg (DA)]
p = evaporation rate of a occupant (0.00133 [kg/min])
n = number of occupants in the room [-].

Equation 5 states that the rate of change of moisture in the room is equal to the difference between the moisture removed from and added to the room. The first term expresses a dehumidifying effect by the supply air flowrate. The second term is the moisture due to the occupants in the room. The absolute humidity x can be converted to the relative humidity φ as described in the next section.

In the same way as the room temperature model, the humidity model can be approximated by a first-order lag plus deadtime system as shown in Equation 2. Thus, the plant dynamics concerned with the room humidity model can be represented by,

$$P'(s) = \frac{K_{Ph}}{T_{Ph}s+1}e^{-L_{Ph}s} = \frac{1.0}{13.5s+1}e^{-2.4s}\,. \tag{6}$$

The gain constant K_{Ph} and the time constant T_{Ph} are given by,

$$K_{Ph} = \frac{f_s}{f_s} = 1\,,\; T_{Ph} = \frac{V}{f_s}\,. \tag{7}$$

Thus, K_{Ph} and T_{Ph} change with the supply air flowrate as same as those represented in the room temperature model. Similarly, the deadtime L_{Ph} is assumed to be changed with the supply air flowrate. Thus,

$$L_{Ph} = \frac{L_{Ph0}}{f_s}\,, \tag{8}$$

where L_{Ph0} is the constant. The deadtime L_{Ph} of the humidity model is assumed in the same way as one of the temperature model. Thus, the deadtime L_{Ph0} can be calculated by $L_{Ph}\times f_s$ = 2.4×8.33 = 19.99.

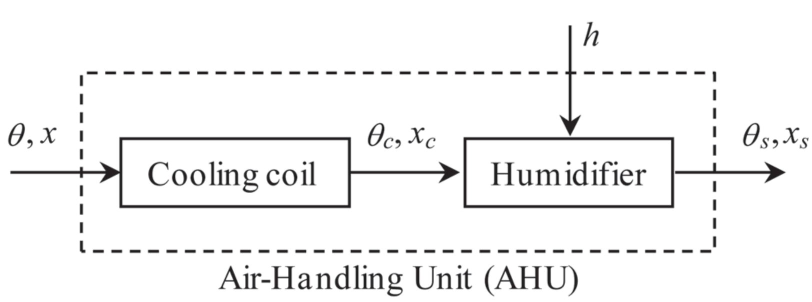


Fig. 2. Block diagram for AHU.

The room humidity can be determined by regulating the moisture of the supply air to the room. This implies that the room humidity can be indirectly controlled. Similarly the first-order lag plus a deadtime model by Equation 6 can be used to tune the PID controller and the physical model by Equation 5 can be used in numerical simulations. It does not mean that Equation 5 and 6 are mathematically equivalent.

2.1.3 Air-handling unit (AHU) model

Figure 2 shows the simple block diagram for the AHU that conditions supply air for the room. Air brought back to the AHU from the room is called return air. The portion of the return air discharged to the outdoor air is exhaust air, and a large part of the return air reused is recirculated air. Air brought in intentionally from the outdoor air is outdoor air. The outdoor air and the recirculated air are mixed to form mixed air, which is then conditioned and delivered to the room as supply air.

The AHU consists of a cooling coil, a humidifier, and a fan to control supply air temperature (θ_s) and humidity (x_s). The mixed air enters the cooling coil at a given temperature θ, which decreases as the air passes through the cooling coil. The temperature of the air leaving the cooling coil is θ_c. Since the responses of the cooling coil and the humidifier are significantly faster than those of the room (a principal controlled plant), it can be generally assumed that the cooling coil and the humidifier are static systems. Namely, it is common for the cooling coil to be controlled to maintain the supply air temperature at a setpoint value (θ_{sr}). Thus, the temperature (θ_c) and the absolute humidity (x_c) of the cooling coil can be given by;

$$\begin{aligned} &\theta_c = \theta_{sr} \\ &x_c = \begin{cases} x_{si} & (p_w \le p_{ws}) \\ \dfrac{0.622 p_{ws}}{P - p_{ws}} & (p_w > p_{ws}) \end{cases} \end{aligned} \tag{9}$$

where θ_{sr} is the setpoint of the supply air temperature, p_w is the partial pressure of water vapor, p_{ws} is the partial pressure of saturated vapor at temperature, P (=101.3 [kPa]) is the total pressure of mixed air, and x_{si} is the absolute humidity of the air entering the cooling coil. The humidity is divided into two calculations depending on the difference between p_{ws} and p_w. This constraint means that the relative humidity does not exceed 100 %.

The humidifier is the most important actuator to control the room relative humidity (φ) for heating mode in winter. Nevertheless, we are interested here in examining control characteristics in the operation mode of cooling. Note that the control input $h(t)$ does not have strong effect on the room relative humidity (φ) in cooling mode. From the energy and mass balances, the dynamics of the humidifier can be described by,

$$\begin{aligned} &C_{ad} \frac{d\theta_s}{dt} = w_s(\theta_c - \theta_s) + \alpha_d(\theta_0 - \theta_s) + q_B + q_d \\ &V_d \frac{dx_s}{dt} = f_s(x_c - x_s) + \frac{h}{\rho_a} \end{aligned} \tag{10}$$

where

C_{ad} = overall heat capacity of humidifier space [kJ/K],

V_d = room volume of humidifier [m³],
α_d = overall transmittance-area factor [kJ/min K],
q_B = fan load (59.43 [kJ/min]),
q_d = load by humidifier ((190.1 – 1.805θ_h)h) [kJ/min]), and
h = rate of moist air produced in the humidifier.

Considering the steady-state of the dynamics of the humidifier, the supply air temperature θ_s and the supply air absolute humidity x_s can be obtained by,

$$\theta_s = \frac{c_p \rho_a f_s \theta_c + \alpha_d \theta_0 + q_B + q_d}{c_p \rho_a f_s + \alpha_d}$$
$$x_s = x_c + \frac{h}{f_s \rho_a} \tag{11}$$

As can be seen in Equation 11, the supply air temperature (θ_s) can be influenced by the humidifier (h), so that the errors in the reset (f_{s0}) can be produced. Thus, the control performance may be deteriorated.

The air flowrate from the outdoor air is considered 25% of the total supply air flowrate. This ratio will be held constant in this study. Note that the pressure losses and heat gains occurring in the duct have negligible effects on the physical properties of air for simplification. The absolute humidity of mixed air entering the cooling coil can be described by,

$$f_s x_{si} = 0.25 f_s x_0 + 0.75 f_s x \ . \tag{12}$$

where x_0 and x are the absolute humidity of outdoor air and of indoor air, respectively. All the actual values of the plant parameters used in the numerical simulations are listed in Table 1. Since we assume that the supply air temperature for the cooling coil can be controlled so as to maintain the setpoint value (θ_{sr}) of the supply air temperature, the energy-balance of mixed air is not needed to consider.

C	370.44 [kJ/K]
V	270 [m³]
c_p	1.3 [kJ/kg K]
ρ_a	1.006 [kg/m³]
α	9.69 [kJ/min K]
α_d	0.1932 [kJ/min K]
q_L	121.72 [kJ/min]
f_{smax}	16.66 [m³/min]
f_{smin}	0.00 [m³/min]
h_{max}	0.33 [m³/min]
h_{min}	0.00 [m³/min]
θ_{sr}	13.1 [°C]

Table 1. Summary of significant parameters in the development of the room and the AHU

2.1.4 Calculation of relative humidity

In this section, the conversion from the absolute humidity to the relative humidity is briefly explained. The relative humidity is derived from the air temperature and the absolute humidity of the air (ASHRAE 1989; Wexler and Hyland 1983).
First, the air temperature must be converted to the absolute temperature as,

$$\Theta_a = \theta_a + 273.15\,, \tag{13}$$

where θ_a is the air temperature, and Θ_a is the absolute temperature of the air.
Second, to evaluate the supply air temperature θ_c reaches its dew-point temperature, the two partial pressures p_w and p_{ws} can be conveniently defined. The partial pressure of water vapor p_w can be obtained by,

$$p_w = \frac{Px_i}{0.622 + x_i}\,, \tag{14}$$

where x_i is the absolute humidity of water vapor and P is the total pressure of mixed air (101.3 [kPa]). And, the partial pressure p_{ws} of saturated vapor at temperature Θ_a can be given by,

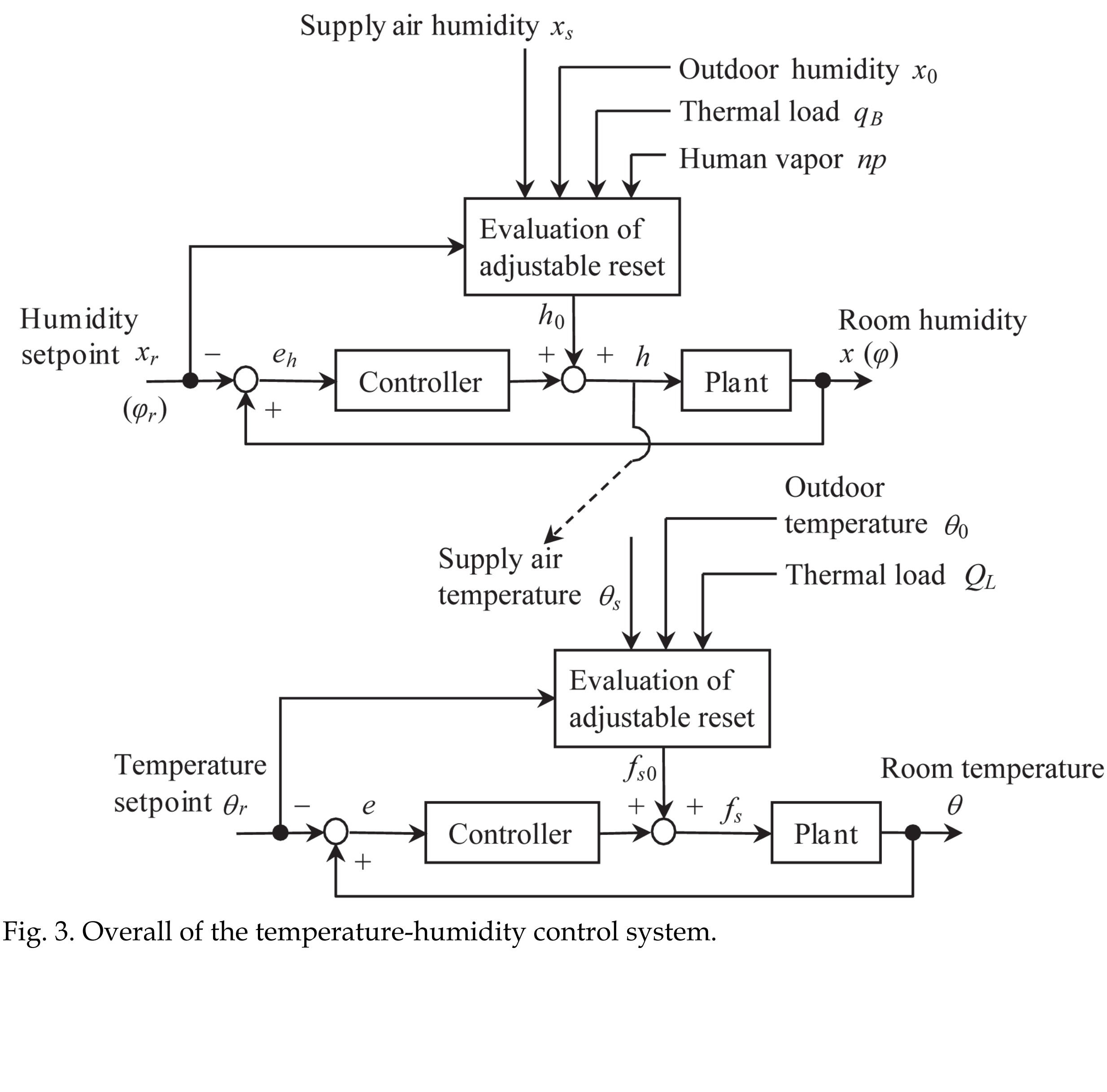


Fig. 3. Overall of the temperature-humidity control system.

$$
\begin{aligned}
\ln(10^3 p_{ws}) = & -0.58002206\times 10^4 / \Theta_a + 0.13914993\times 10 \\
& -0.48640239\times 10^{-1}\Theta_a + 0.41764768\times 10^{-4}\Theta_a{}^2 \\
& -0.1445293\times 10^{-7}\Theta_a{}^3 + 0.65459673\times 10\times \ln\Theta_a
\end{aligned} \tag{15}
$$

Finally, the relative humidity φ for the room can be given by,

$$\varphi = \frac{p_w}{p_{ws}}\times 100\,. \tag{16}$$

2.2 Control system

Figure 3 shows a block diagram of the room temperature and humidity control systems using adjustable resets which compensate for thermal loads upsets. In this figure, signals appear as lines and functional relations as blocks. The primary controlled plant is the room. The cooling coil, the humidifier and the damper are defined as the secondary controlled plants (to produce appropriate actuating signals). The following control loops are existed in our room temperature and humidity control system:

- Room air temperature control system
- Room air humidity control system

The control outputs of interests are room air temperature (θ) and relative humidity (φ). In order to maintain room air temperature and humidity in desirable ranges, traditional PID controllers have been used to reduce component costs. The control inputs that vary according to the control actions are the supply air flowrate (f_s) and the rate of moist air produced in the humidifier (h), which will be discussed in more detail.

2.2.1 Room temperature control system

Taking the PID control algorithm into account, one of control inputs, related to the room air temperature (θ) can be given by,

$$f_s(t) = k_p e(t) + k_i\int_0^t e(\tau)d\tau + k_d\frac{de(t)}{dt} + f_{s0}(t) \tag{17}$$

where $f_{s0}(t)$ is the manual reset. In electronic controllers, the manual reset is often referred to as "tracking input". The error $e(t)$ can be defined by,

$$e(t) = \theta(t - L_P) - \theta_r, \tag{18}$$

where θ_r is the setpoint value of the room air temperature, and L_P (= 2.4 [min]) is the deadtime. The PID parameters (the proportional gain k_p, the integral gain k_i, and the derivative gain k_d) can be determined by the well-known tuning method. The inherent disadvantage of the I action, which easily causes instabilities, can be reduced by varying the reset $f_{s0}(t)$ to compensate for thermal loads upsets (disturbances). In some cases of HVAC systems, the reset $f_{s0}(t)$ can be estimated by knowledge of the plant dynamics.

Equation 17 can be given in a discrete-time system when control input and error signal are respectively assumed to be $f_s(k)$ and $e(k)$ at time kT (T is the sampling period).

$$f_s(k) = k_p e(k) + k_i T\sum_{j=0}^{k}\frac{e(j-1)+e(j)}{2} + \frac{k_d}{T}\{e(k)-e(k-1)\} + f_{s0}(t) \tag{19}$$

This is called the position algorithm because $f_s(k)$ typically represents the position of an actuator (Takahashi 1969).
From Equation 1, the operating point at its steady-state can be written:

$$w_s\left(\theta_s - \theta\right) + \alpha\left(\theta_0 - \theta\right) + q_L + Q = 0 \;\; (w_s = \rho_a c_p f_s). \tag{20}$$

The reset (f_{s0}) of the supply air flowrate can be obtained by,

$$f_{s0}(t) = \frac{q_L(t) + q_{th}(t) + \alpha(\theta_0(t) - \theta_r(t))}{c_p \rho_a (\theta_r(t) - \theta_s(t))}. \tag{21}$$

In Equation 21, the supply air temperature (θ_s), the outdoor temperature (θ_0), and the setpoint (θ_r) can easily be measured. However, thermal loads cannot be specified in advance. Thus, it is recommended that occupants must roughly estimate thermal loads to improve the control performance at adequate sampling interval. For example, three of the rough estimates for compensation can be used as:

the maximum (75%), the medium (50%), and the minimum (25%),

where 100 % means the maximum supply air flowrate 16.66 [m^3/min].At any given point of operation, the reset (f_{s0}) to offset thermal loads can be easily calculated using Equation 21. Thus, it can be concluded that the controller with lower I action is superior to that with no I action, and is also called a PD controller.

2.2.2 Room humidity control system

To control the room air relative humidity, another one of control inputs that vary according to the control actions is the rate of moist air produced in the humidifier $h(t)$. The control input can be given by,

$$h(t) = k_{ph} e_h(t) + k_{ih} \int_0^t e_h(\tau) d\tau + k_{dh} \frac{de_h(t)}{dt} + h_0(t)\,, \tag{22}$$

where $h_0(t)$ is the reset. The error $e_h(t)$ can be defined by,

$$e_h(t) = \varphi_r - \varphi(t - L_{Ph}) \tag{23}$$

where φ_r is the setpoint value of the room air relative humidity and L_{Ph} (= 2.4 [min]) is the deadtime. The hygrometer in the room can detect the room air relative humidity (φ), but not the absolute humidity (x). Therefore, the relative humidity is used in the error $e_h(t)$ for the calculation of the control input $h(t)$. However, the humidity model can be described by the relational expression of the absolute humidity. And, the derivation of the humidity model parameters from the experimental results in terms of the relative humidity may be extremely difficult. As a result, PID parameters (proportional gain k_{ph}, integral gain k_{ih}, and derivative gain k_{dh}) must be determined by trial and error under the consideration that the absolute humidity cannot be directly measurable. In this study, for the sake of simplicity, it is assumed that the basic relation of the humidity model is invariant even if the variable in the humidity model is changed the absolute humidity into the relative humidity. For this reason, the traditional tuning method (Ziegler and Nichols 1942) for the first-order lag plus deadtime system as shown in Equation 6 (the plant parameters is described by Equations 7 and 8) can be used.

Since the supply air temperature (θ_s) can be affected by the rate of moist air produced in the humidifier (h), the reset of the supply air flowrate (f_{s0}) arising from moist air variations must be accounted. This means that good control performance for heating mode can be expected.

The reset $h_0(t)$ for the humidifier can be obtained from Equations 5, 9, 11, and 12 as follows: First, taking the humidity model (Equation 5) at the steady-state and the setpoint value x_r of the absolute humidity into account, the following equation can be obtained by,

$$f_s(x_s - x_r) + \frac{n}{\rho_a} p = 0\,. \tag{24}$$

Second, substituting Equation 24 into Equation 9, 11 and 12, Equation 24 can be rewritten by,

$$f_s\left(x_c + \frac{h_0}{\rho_a f_s} - x_r \right) + \frac{n}{\rho_a} p = 0$$

$$f_s\left(0.25x_0 + 0.75x_r + \frac{h_0}{\rho_a f_s} - x_r \right) + \frac{n}{\rho_a} p = 0$$

$$0.25x_0 f_s - 0.25x_r f_s + \frac{h_0}{\rho_a} + \frac{n}{\rho_a} p = 0$$

$$\frac{h_0}{\rho_a} = 0.25x_r f_s - 0.25x_0 f_s - \frac{n}{\rho_a} p$$

$$h_0(t) = 0.25 f_{s0} \rho_a (x_r - x_0) - np \tag{25}$$

However, as will be seen in Equation 25, the first term is small in comparison to the second term and $h_0(t)$ may be negative. The adjustable reset $h_0(t)$ can be found to be nearly zero under most circumstances in the present work.

Table 2 provides PID parameters tuned by the traditional ultimate sensitivity method (Ziegler and Nichols 1942) and the empirical modified PID method.

The ultimate sensitivity method is simple and intuitive. It has been still widely used, either in its original form or in some modification. Since it only gives “ball-park” values, it is necessary to make manual tuning to obtain the desired performance. Our empirical modified PID controller can help improve the time response of a control system because thermal loads and operating conditions are changing continuously in HVAC systems.

In modified PID parameters for room air temperature control, the proportional gain (k_p) is about 80 % of that of the conventional tuning method. The integral gain (k_i) is one-fourth of that of the conventional tuning method. The derivative gain (k_d) is nearly the same as that of the conventional tuning method. In modified PID parameters for room air relative humidity control, all gains (k_{ph}, k_{ih}, and k_{dh}) are nearly one-tenth of those of the conventional tuning method.

3. Simulation results in daily operation

To illustrate the control performance of the room temperature and humidity control systems, several simulation runs are made. Representative outdoor temperature and thermal loads profiles for one-day (between 08:00 in the morning and 08:00 in the next morning) are assumed as shown in Figure 4. These profiles are based on the experimental data obtained from the National Institute for Environmental Studies in Tsukuba, Japan. In the right hand side of Figure 4, the dashed line depicts the artificial estimated value of the thermal load. At the start-up (at 08:00 in the morning), the feedback control system takes over and controls the room air temperature and relative humidity. These simulation runs are carried out under the same conditions mentioned above. Figure 5 depicts the adjustable reset (f_{s0}) of the supply air flowrate for daily operation calculated using Equation 21. The computational interval of 1 hour (60 min) for adjusting the reset is used in this control. These simulation runs are made on MATLAB which is an effective tool for field engineers in control engineering.
The following control configurations are used in our room temperature and humidity control. These abbreviations are common throughout the remainder of this paper.

	k_p	k_i (T_i)	k_d (T_d)
Conventional PID	11.65	2.55 (4.57)	13.26 (1.16)
Modified PID	8.73	0.8 (10.9)	10 (1.15)

(a) Temperature control

k_{ph}	k_{ih} (T_{ih})	k_{dh} (T_{dh})
1.22	0.26 (4.65)	1.41 (1.16)

(b) Humidity control

(T_i, T_{ih}: integral times, T_d, T_{dh}: derivative times for temperature and humidity controls, respectively)

Table 2. PID parameters.

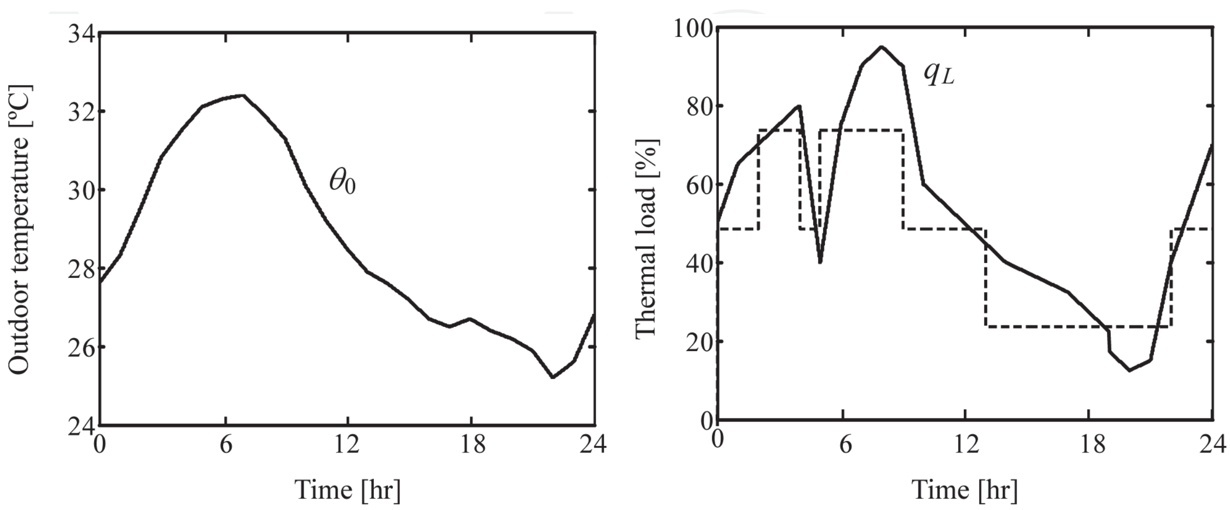


Fig. 4. Outdoor temperature and thermal loads profiles.

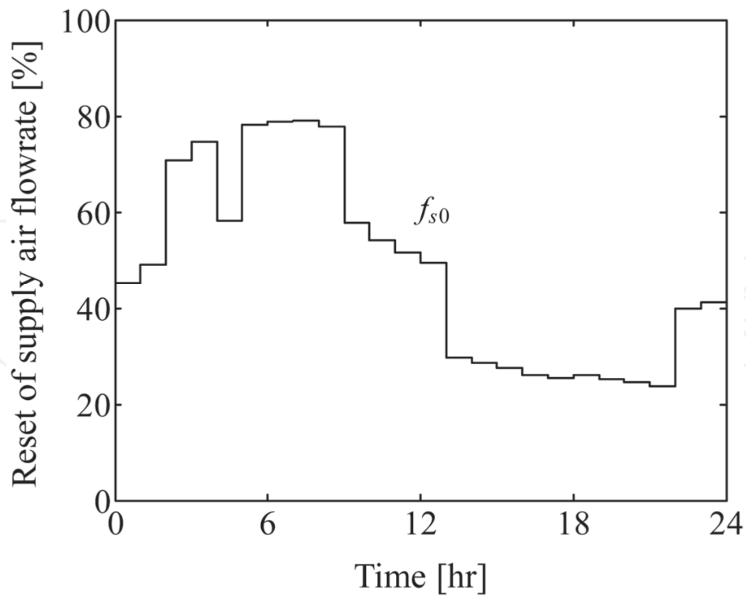


Fig. 5. Reset of supply air flowrate.

- Number of control outputs of interest
  Room temperature and humidity control
  This refers to the room air temperature and relative humidity control.
- Setpoints of control outputs

Regarding the room air temperature θ_r:

1. Fixed setpoint
   The setpoint θ_r is fixed at 24 °C for daily operation.
2. Variable setpoint
   The setpoint θ_r are varied within the range, that θ_r is set at the value ($\theta_0 - 4$) °C where θ_0 is the outdoor temperature, and θ_r is limited to the minimum 20 °C and the maximum 28 °C.

Regarding the room air relative humidity φ_r:

The setpoint φ_r is usually fixed at 55 % for daily operation.

- Control strategies for the reset

1. Conventional PID control
   This refers to conventional PID control with the fixed reset (f_{s0} = 50 %).
2. Modified PID control
   This refers to modified PID control with the adjustable reset (Figure 5).

- Performance indices

The control performance should be evaluated by defining three performance indices.

1. ISE (the integral of squared error)
   $$\text{ISE} = \int_0^{24} e^2 dt$$
2. ICI (the integral of control input)
   $$\text{ICI} = \int_0^{24} f_s dt$$
3. IPID (the integral of control input produced in PID controller only)
   $$\text{IPID} = \int_0^{24} (f_s - f_{s0}) dt$$

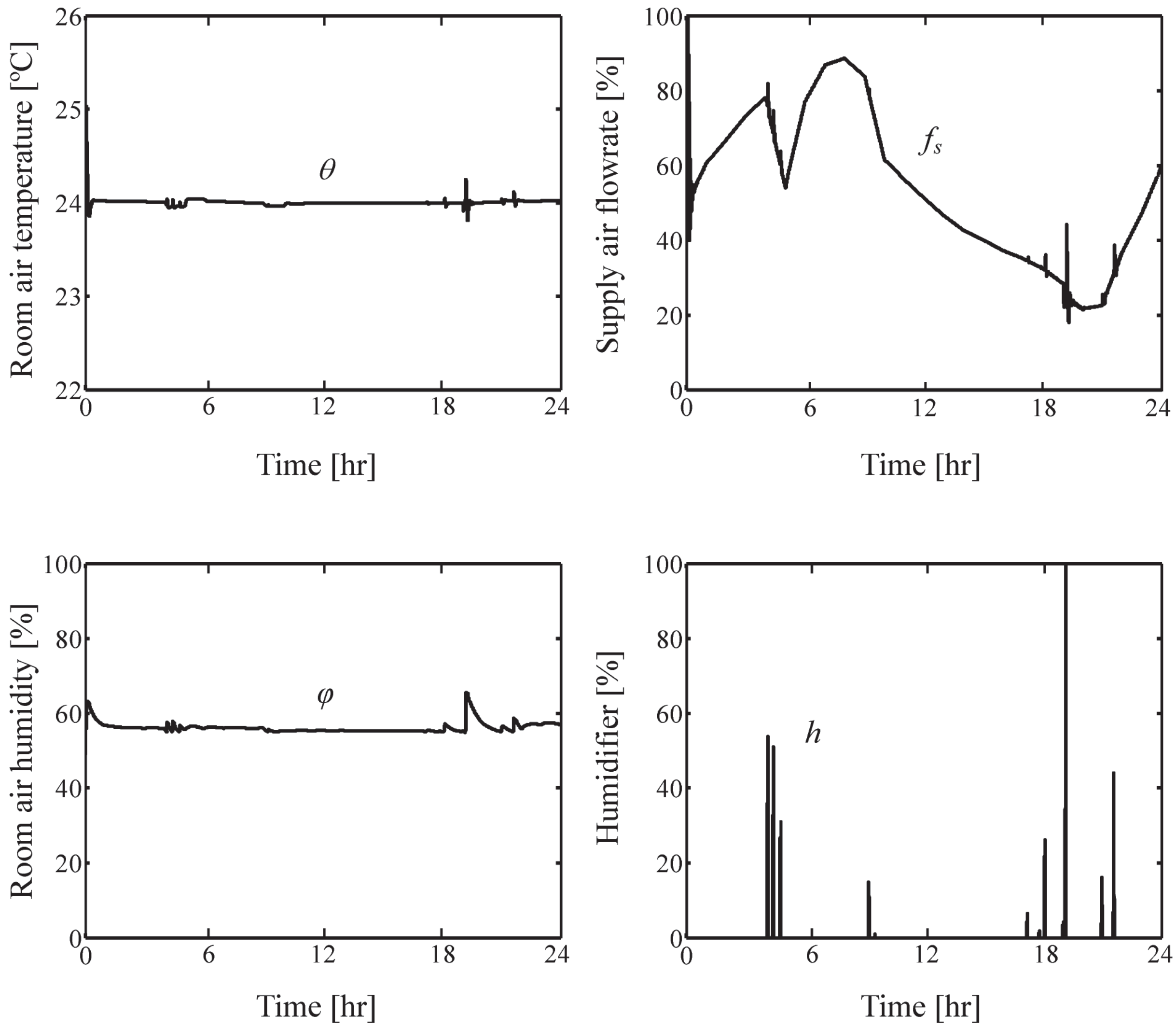


Fig. 6. Simulation results of conventional PID.

Room temperature and humidity control

Typical daily simulation results show that the conventional PID and the suitably modified PID controllers can maintain the room air temperature and relative humidity close their respective setpoints irrespective of variable thermal loads. The method of determining PID parameters for the modified controller is practical for room temperature and humidity control systems.

Fixed setpoint

Figure 6 and 7 show the responses to the fixed setpoint of the room air temperature for the cases of the conventional PID and the modified PID controls, respectively.
In Figure 6, there are sudden changes in θ and φ during the initial few hours, which then settle to setpoints. We can expect that, since the transient responses of θ and φ will also change rapidly, θ and φ are very close to their setpoints even though θ_0 and q_L are varied. The supply air flowrate illustrates instabilities locally due to humidifier working.
When looking over results of Figures 6 and 7, it should be noted that the responses (θ and φ) of the conventional PID control and the modified PID control are somewhat different.

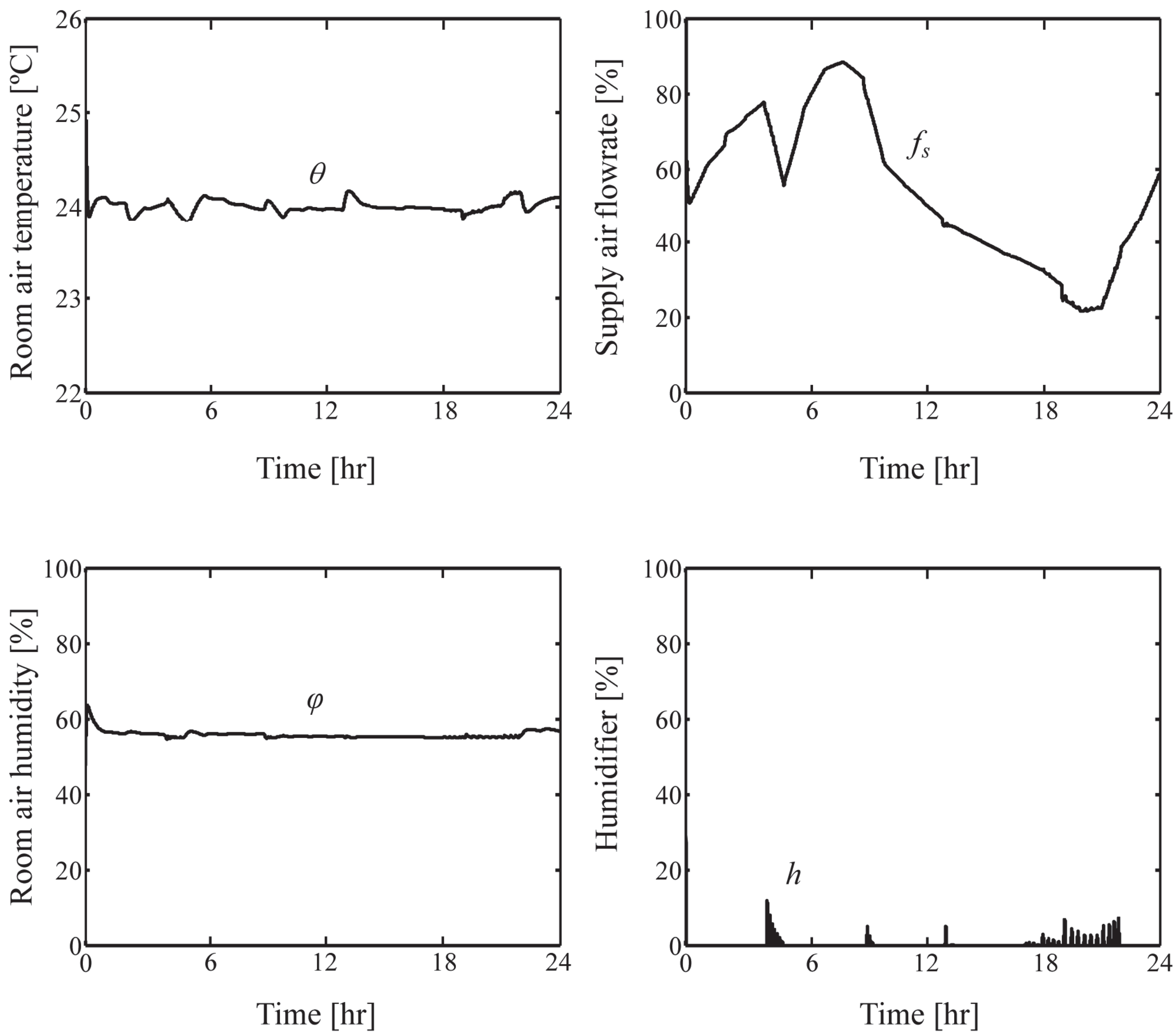


Fig. 7. Simulation results of Modified PID.

	Conventional PID	Modified PID
ISE	3.09	7.95
ICI	$2.08{\times}10^4$	$2.08{\times}10^4$
IPID	8742	2734

Table 3. Comparison of control performance indices to fixed setpoint.

Because the reset for the modified PID control can be adjusted very often, it becomes difficult to maintain θ and φ at the setpoints, so θ fluctuates around the setpoint. It is clear that the results for modified PD control cannot represent an improvement over those for the conventional PID control. For small values of the integral gain (k_i) for the modified PID control, θ creeps slowly towards the setpoint. However, as will be seen in the near future, this disadvantage may be clearly solved.

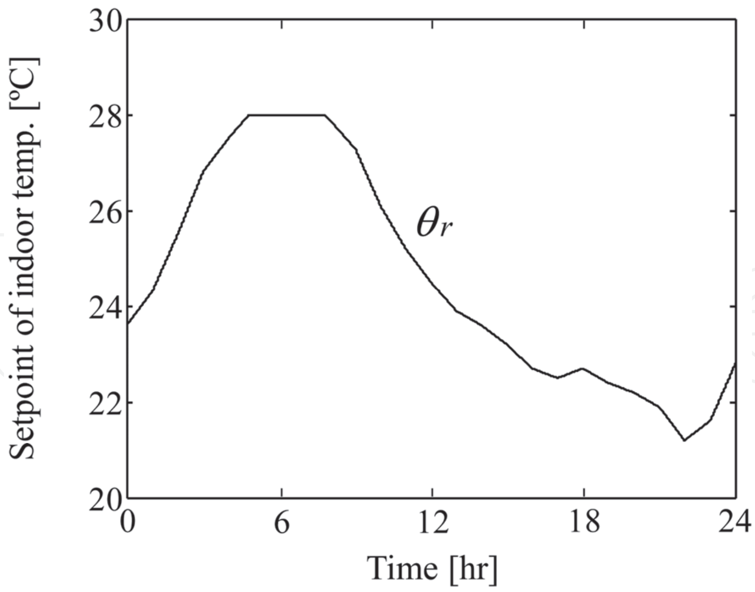


Fig. 8. Variable setpoint profile.

Table 3 shows that the results of the validation simulations in terms of three performance indices. For the ISE (tracking accuracy), it is evident that the sharply change of the reset aggravates the tracking accuracy of θ for the modified PID control, but it is enhanced by increasing the integral gain (k_i). Further investigation into the total amount of control inputs (ICI and IPID) can lead to some interesting results. It is recognized that the ICI is exactly the same for the two control strategies. The physical interpretation of this fact is that there is no difference of supply air flowrates between two control strategies. However, for the IPID, the modified PID control clearly represents an improvement over the conventional PID control.
As a matter of fact, the merit of the modified PID control becomes obvious when the maximum capacity of the controller is limited.

Variable setpoint

Figure 8 depicts the setpoint profile (($\theta_0 - 4$) [°C]) of room air temperature depending on the outdoor temperature on a typical day. The responses to the variable setpoint for the cases of the conventional PID and the modified controls are shown in Figure 9 and 10, respectively. The room air temperature and humidity follow their respective setpoint profiles even though thermal loads are variable. It is apparent from Figure 9 that the solid areas indicate rapidly oscillating values due to hunting when the humidifier is positioned between 0 % and 100 %. Subsequently, the room air temperature can be oscillated with the occurrence of such huntings. The same trend is also apparent in the supply air flowartes.
It can be seen from Figure 10 that suitably tuned modified controller can maintain the room air temperature and humidity close to their respective setpoints suppressing such huntings. The effectiveness of the modified PID control can be confirmed. By comparing these responses with those of Figure 6 and 7, it is clear that the humidifier is turned on very often and the hunting of the room air temperature may occur simultaneously.
Fig. 9 and 10 demonstrate locally rapid oscillation of the humidifier when the indoor relative humidity φ becomes below the setpoint 55 %. This is due to the fact that the humidifier is very sensitive to control inputs. There are also many technological problems to be solved when we make positive use of the humidifier in cooling operation.

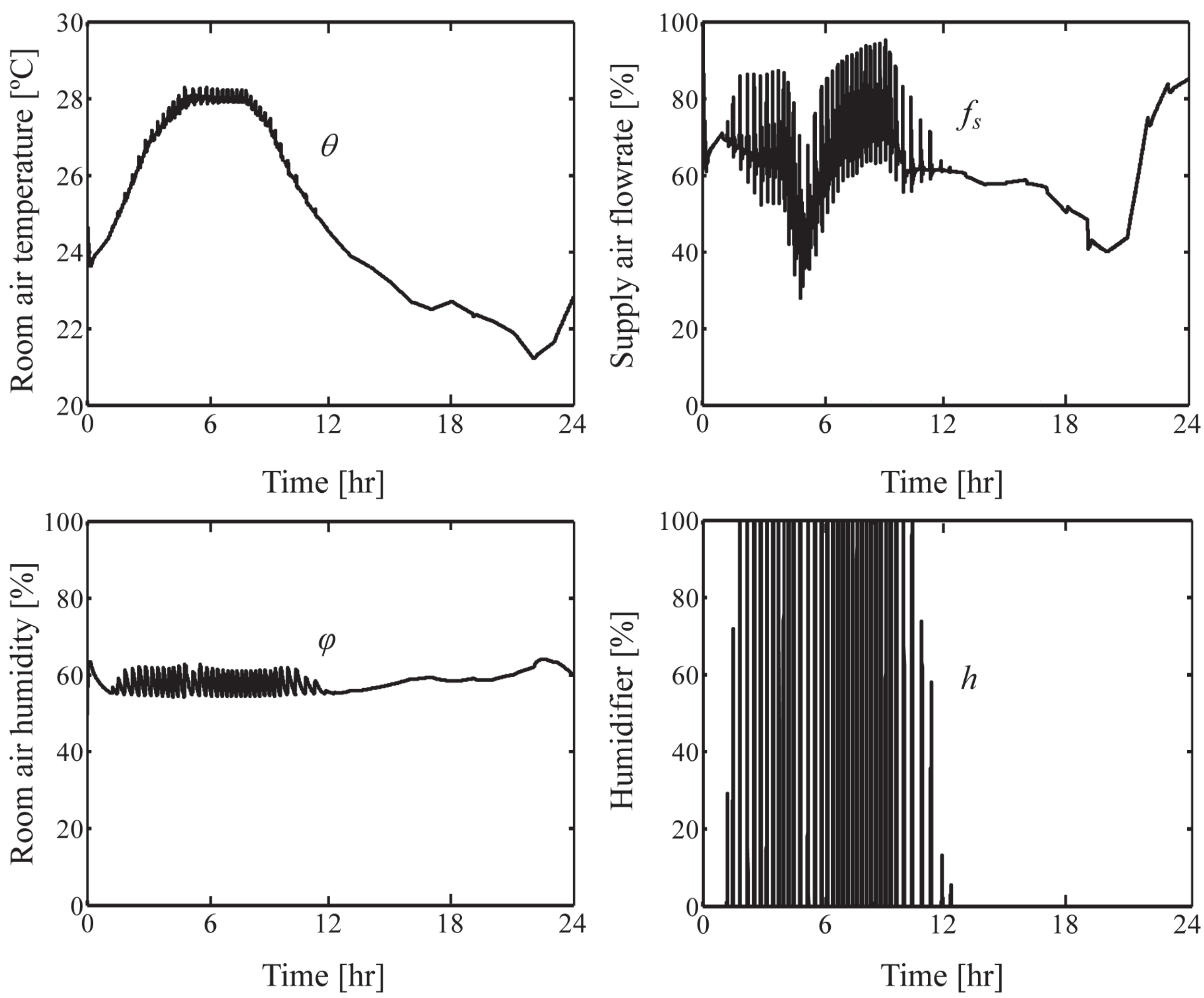


Fig. 9. Simulation results of conventional PID.

	Conventional PID	Modified PID
ISE	5.98	8.83
ICI	1.92×10^4	1.92×10^4
IPID	4276	3005

Table 4. Comparison of control performance indices to variable setpoint.

Table 4 represents the control performance indices obtained by typical daily simulation results. A comparison with Table 3 shows that there is very little difference in performance between the fixed setpoint and the variable setpoint. For the ISE, the ISE for the modified PID is larger than that for the conventional PID. This means that the I action is effective for not only elimination of offset (steady-state error) but also disturbance attenuation. Tracking accuracy and disturbance attenuation will be enhanced by selecting high integral gain.
For the ICI, it is striking that the ICI values are exactly the same for two control strategies. For the IPID, the modified PID control gives slightly better results than the conventional PID control. It is concluded that the modified PID control should be also incorporated by limiting the maximum control input available to the controller.

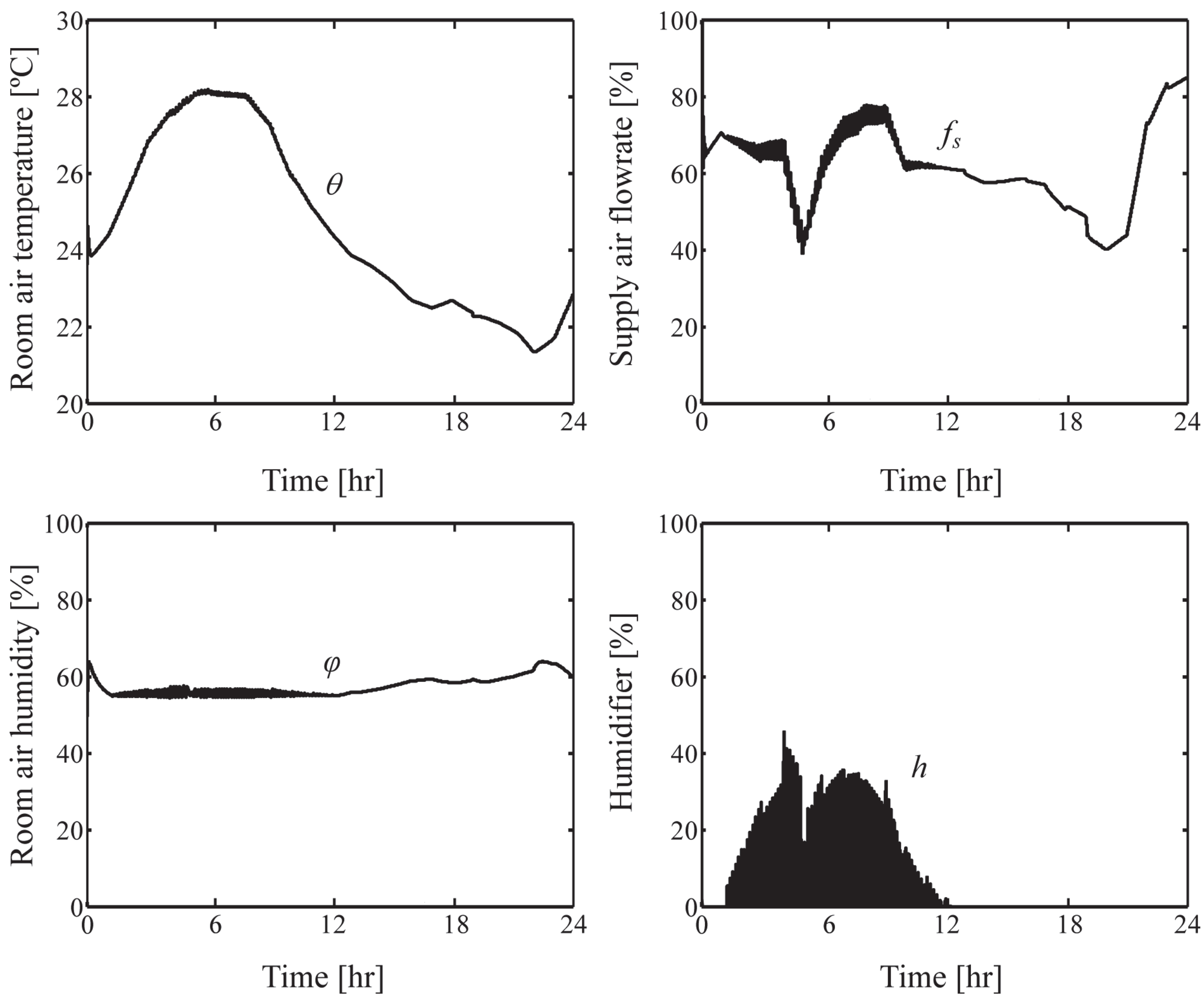


Fig. 10. Simulation results of Modified PID.

4. Conclusions

In this paper, the room temperature and humidity control systems with the conventioanl PID control using fixed reset or the modified PID control using adjustable resets which compensate for thermal loads upset are examined. The simulation results for one-day operation based on practical outdoor temperature and thermal loads profiles provide satisfactory control characteristics. The results of validation simulations are demonstrated in terms of three performance indicies (as three integrals of squared error (ISE), control input (ICI), and control input in PID controller only (IPID)).

The results obtained in this study are summarized in the following:

1. The room air temperature and humidity illustrate instabilities locally due to humidifier working.
2. By changing the setpoint of the room air temperature on the basis of the outdoor temperatures profile, the control performance can be remarkably improved.
3. In daily operation, when the reset is adjusted at every hour, the sharply change of the reset aggravate the response of the room air temperature. The response can be improved by proper selection of the computational period.

4. The proposed control strategy for the adjustable reset cannot be effective for energy-savings, but has a possibility in case that there exists a limitation of the maximum control input available to the controller.

Finally, the results given in this paper were motivated by the desire to obtain satisfactory performance with adjustable reset better than that with fixed reset. Consequently, it is concluded that there is little inherent advantages in designing the modified PID controller with adjustable reset. However, since this modified PID control lightens the total amount of control input produced in the controller, it can be good candidates for the next HVAC controllers.

The work reported here is being continued to validate several conclusions obtained by experimental results.

5. Acknowlegdment

This research was partially supported by the National Institute for Environment Studies in Tsukuba. The authors would like to acknowledge staffs of Controls Group for their contribution to this study.

6. Nomenclature

C = overall heat capacity of air-conditioned space (kJ/K)
C_{ad} = overall thermal capacity of humidifier space (kJ/K)
c_p = specific heat of air (kJ/kg K)
a = overall transmittance-area factor (kJ/min K)
a_d = overall transmittance-area factor outside humidifier (kJ/min K)
θ = indoor air temperature (°C)
θ_r = setpoint of indoor air temperature (°C)
θ_s = supply air temperature (in humidifier) (°C)
θ_{sr} = setpoint of supply air temperature (in humidifier) (°C)
θ_c = supply air temperature (in cooling coil) (°C)
θ_0 = outside temperature (°C)
ρ_a = density of air (1.3 kg/m^3)
c_p = specific heat of air (kJ/kg K)
f_s = supply air flowrate (m^3/min)
f_{s0} = reset of supply air flowrate of room (m^3/min)
w_s = $c_p \times \rho_a \times f_s$, heat of supply air flowrate (kJ/min K)
V = room volume (10×10×2.7 m^3)
V_d = room volume of humidifier (m^3)
x = indoor absolute humidity (kg/kg (DA))
x_s = absolute humidity of supply air (kg/kg (DA))
x_{si} = return air absolute humidity at the inlet of air-handling unit (kg/kg (DA))
x_0 = outdoor absolute humidity (kg/kg (DA))
φ = indoor relative humidity (%)
φ_r = setpoint of indoor relative humidity of room (%)
h = rate of moist air produced in humidifier (kg/min)
q_L = thermal load from internal heat generation (kJ/min)

q_B = fan load (59.43 kJ/min)
q_d = load by humidifier ($(190.1 - 1.805\theta_c)h$ kJ/min)
p = evaporation rate of a occupant (0.00133 kg/min)
P = total pressure of mixed air (101.3 kPa)
p_w = partial pressure of water vapor at the inlet of air-handling unit (kPa)
p_{ws} = partial pressure of saturated vapor at temperature θ_c (kPa)
h_0 = reset of rate of moist air produced in humidifier (kg/min)
n = number of occupants in the room (-)
K_P = plant gain of room temperature dynamics
T_P = time constant of room temperature dynamics
L_P = deadtime of room temperature dynamics
K_{Ph} = plant gain of room humidity dynamics
T_{Ph} = time constant of room humidity dynamics
L_{Ph} = deadtime of room humidity dynamics

7. References

Kasahara. M. et al. (2000). Physical Model of an Air-Conditioned Space for Control Analysis, *ASHRAE Transactions*, Vol. 106, Part II: pp. 304-317

Åström. K. & Hägglund. T. (1995). PID controllers: Theory, design and tuning, pp. 164-166 and 196-198, *Instrument Society of America*.

Hartman. T. (2003). Improving VAV Zone Control, *ASHRAE Journal*, pp. 24-32

Taira. U. (2004). Save-Energy Type Temperature-Humidity Simultaneous Control using Model Predictive Control Method, *Journal of Society of Instrument and Control Engineering*, Vol. 43-9, pp. 686-690

Yamakawa. Y. et al. (2010). Compensation of Manual Reset to Offset Thermal Loads Change for PID Controller, *ASHRAE Transactions*, Vol. 116, Part 1, pp. 303-315

Zhang. Z. & Nelson. R.M. (1992). Parametric analysis of a building space conditioned by a VAV system, *ASHRAE Transactions*, Vol. 98, Part 1, pp. 43-48

Matsuba. T. et al. (1998). Stability Limit of Room Air Temperature of a VAV System. *ASHRAE Transactions*, Vol. 104, Part II: pp. 257-265

Yamakawa. Y. et al. (2009). Stability of Temperature Control System in VAV Systems, *ASHRAE Transactions*, Vol. 115, Part 1, pp. 613-621

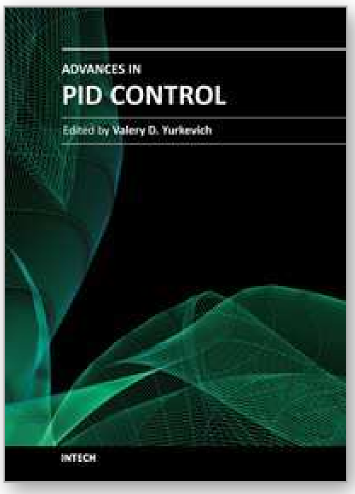
Ozawa. K. et al. (2003). A Tuning Method for PID Controller Using Optimization Subject to Constraints on Control Input. *ASHRAE Transactions*, Vol. 109, Part 1, pp. 3-11.

ASHRAE. (1989). 1989 ASHRAE Handbook–Fundamentals, *American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc*.

Wexler. A. & Hyland. R.W. (1983). ASHRAE Transactions 89(2A): 500.

Takahashi. Y., M. J. Rabins & D. M. Auslander. (1969). Control and dynamic systems, *New York*, *Addison-wesley*

Ziegler. J. G. & Nichols. N. B. (1942). Optimum settings for automatic controllers, *Transactions of the American Society of Mechanical Engineers*, Vol. 64-8: pp. 759



Advances in PID Control

Edited by Dr. Valery D. Yurkevich

ISBN 978-953-307-267-8

Hard cover, 274 pages

Publisher InTech

Published online 06, September, 2011

Published in print edition September, 2011

Since the foundation and up to the current state-of-the-art in control engineering, the problems of PID control steadily attract great attention of numerous researchers and remain inexhaustible source of new ideas for process of control system design and industrial applications. PID control effectiveness is usually caused by the nature of dynamical processes, conditioned that the majority of the industrial dynamical processes are well described by simple dynamic model of the first or second order. The efficacy of PID controllers vastly falls in case of complicated dynamics, nonlinearities, and varying parameters of the plant. This gives a pulse to further researches in the field of PID control. Consequently, the problems of advanced PID control system design methodologies, rules of adaptive PID control, self-tuning procedures, and particularly robustness and transient performance for nonlinear systems, still remain as the areas of the lively interests for many scientists and researchers at the present time. The recent research results presented in this book provide new ideas for improved performance of PID control applications.