82 International Journal of Air-Conditioning and Refrigeration Vol. 11, No. 2 (2003)/pp. 82-90

A PID Control of Supply Duct Outlet Air Temperature in Personal Environment Module

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Key words: PEM (personal environment module), Electrical heater control, P control, PD control, PID control

ABSTRACT: The work presented here is a design and an implementation of PID control system to regulate a supply duct outlet air temperature in PEM (Personal Environment Module). In PEM, the air is heated to the required temperature while it flows through the supply duct without any mixing chamber. This makes the control of air temperature in PEM difficult. A simulation is done first to understand the relationship between a temperature distribution in working area, flow rate and the outlet air temperature of PEM. Then a linear dynamic model of heating process in PEM is derived. P, PD and PID type control systems, to provide the rapid response without overshoot and saturation in heater command voltage, are designed using a linear model obtained. Experimentally obtained data shows that the control system satisfies the design criteria and works properly in controlling the supply duct outlet air temperature.

Nomenclature

A_h : total heater surface area [m^2]
A_s : total surface area of supply duct [m^2]
A_w : total surface area of heating chamber [m^2]
$h_{o.d}$: convection heat transfer coefficient at supply duct outer surface [W/m^{2}K]
$h_{o.w}$: convection heat transfer coefficient at heating chamber outer surface [W/m^{2}K]
$\dot{m}$: air mass flow rate [kg/s]
$m_{ah}C_a$: air heat capacity of heating chamber [J/K]
m_dC_d : heat capacity of supply duct [J/K]
m_hC_h : heat capacity of heater [J/K]
m_sC_a : air heat capacity of supply duct [J/K]
m_wC_w : heat capacity of heating chamber [J/K]
T_a : heating chamber temperature [℃]
T_i : air temperature at the outlet of supply duct (control variable) [℃]
T_{rh} : temperature of heating chamber outer surface [℃]
T_{rs} : temperature of supply duct outer surface [℃]
T_∞ : inlet air temperature [℃]
U_h : overall heat transfer coefficient at heater surface [W/m^{2}K]
U_s : overall heat transfer coefficient at supply duct surface [W/m^{2}K]
U_w : overall heat transfer coefficient at heating chamber surface [W/m^{2}K]

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1. Introduction

PEM is an air conditioning module to control the thermal condition of working area, which is a narrow space defined near the occupant. A previous study[1] shows that PEM is better than the conventional air conditioning system in terms of energy saving and thermal comfort. Since PEM is used to control the narrow area, the air flow rate is usually small. Also it has a velocity limitation to prevent the draft effect. In PEM, regulating supply duct outlet air temperature is done generally by controlling the heat flux of electric heater installed in duct. The air is heated to the required temperature while it flows through the supply duct. For the conventional air conditioning system, a mixing chamber to get the uniform air temperature is used generally. However PEM cannot use such mixing device, because of the limitation of the size of PEM and space where PEM is installed. This makes the accurate control of air temperature difficult.

This paper presents an implementation of heat flux control of electric heater installed in a supply duct of PEM to control the supply duct outlet air temperature. We first simulate the temperature distribution of the office area where the under-floor air conditioning system and PEM are used simultaneously. Simulation is conducted to find the relationship between the thermal condition of working area, the air flow rate and the supply duct outlet air temperature of PEM, which will ultimately provide the required supply duct outlet air temperature of PEM to get a certain thermal condition at the working space. With the results obtained in the simulation, the control system to regulate the supply duct outlet air temperature is designed and implemented. In designing the control system, we first derive the mathematical model of heating process. Then P, PD and PID control systems to provide the fast response without overshoot are designed and implemented experimentally.

2. Simulation

To control the thermal condition of the working area properly using PEM, we need to understand how the flow pattern is formed and how the thermal conditions are varied after the air is discharged from the PEM diffuser.

Simulation is done to understand the relationship between the flow rate, supply duct outlet air temperature of PEM and varied thermal conditions of the working area. This information will provide how we adjust the supply duct outlet air temperature and flow rate to satisfy the required thermal condition in a working area.

Fig. 1 shows the configuration of the office area used in the simulation. The office space consists of working and ambient areas. A diffuser of PEM is located on top of the desk. Also we put the four rectangular-shaped inlets of air conditioning (0.2×0.4 m) on the floor and two square-shaped outlets (0.3×0.3 m) at the ceiling. PEM diffuser (0.25×0.2 m) is installed in 26 degrees downward direction at the center of the space. The office area is separated from the outdoor by a 0.1 m thickness window glass. We assume that the airflow and temperature distribution be at steady state and incompressible. A $\kappa-\varepsilon$ model is applied with non-uniform grids ($75 \times 46 \times 26$ in each direction) and a finite volume method in computation. Simulation is done for various flow rates (less than 1 m/s) and various supply duct outlet air temperatures (less than 45℃).

Fig. 2 shows one of the simulation results, showing constant temperature lines and buoyancy effect due to PEM. In computation, the

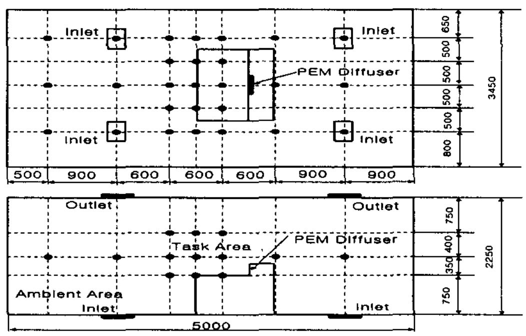

Fig. 1 Dimensions of office area.

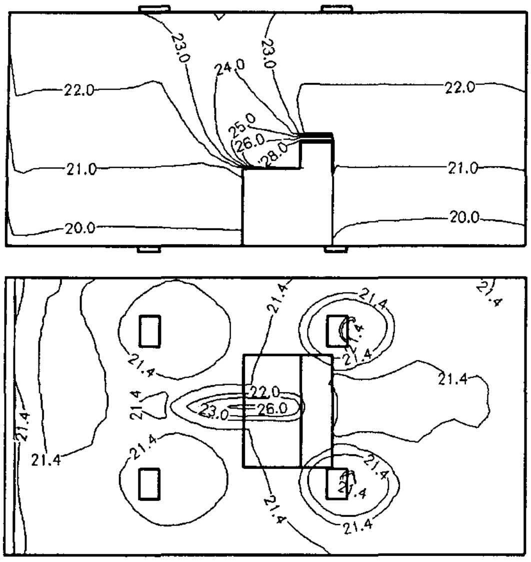

Fig. 2 Temperature distribution in office area.

inlet air temperature at the floor and the supply duct outlet air temperature in PEM are set at 24℃ and 30℃ respectively, while the outdoor temperature is set at 0℃. Also the air flow velocity is 0.0625 m/s at the floor and 0.4 m/s at the PEM respectively. As shown in the figure, if the temperature in the PEM is high and the flow rate is low, the heated air discharged from PEM will flow away before reaching a working area. This indicates that the flow rate and air temperature in PEM needs to be set properly. Using the result obtained from the simulation, the supply duct outlet air temperature in PEM to provide the required thermal condition in the working area could be determined.(2)

3. Mathematical model of heating process

3.1 Governing heat transfer equations

The heating process in PEM is nonlinear, thus a nonlinear model is required to describe the heat transfer phenomena accurately. How-

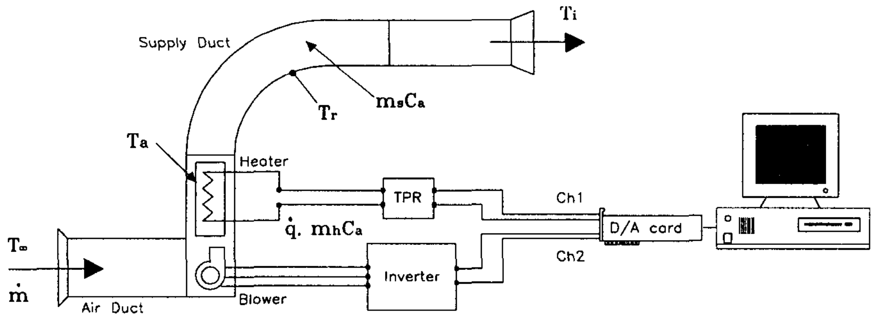

Fig. 3 Schematic diagram of experimental apparatus.

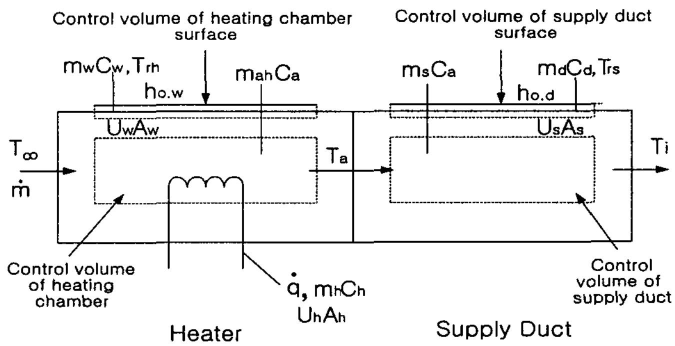

Fig. 4 Control volume of heater and supply duct.

ever, due to the difficulties in analysis and implementation,[3] a linear system model is derived.

Fig. 3 shows the heating device of PEM used in the experiment. Also Fig. 4 shows the control volumes used to derive the linear governing heat transfer equations. Using the lumped thermal mass concept, we assume that the system can be represented by five different temperatures. Also the temperature and thermal capacity at each part are assumed to be constant.

The heat transfer analysis was done for a heating chamber and a supply duct respectively. Eqs. (1) ~ (3) are governing heat transfer equations for a heating chamber. Eq. (1) and (2) are derived from the control volume defined inside the heating chamber. Eq. (3) is obtained using the control volume defined at the surface of a heating chamber.

$$m_{ah} C_a \frac{dT_a}{dt} + m_h C_h \frac{dT_h}{dt} = \dot{q} - \dot{m} C_a (T_a - T_\infty) - U_w A_w (T_a - T_{rh}) \tag{1}$$

$$U_h A_h (T_h - T_a) = m_{ah} C_a \frac{dT_a}{dt} + \dot{m} C_a (T_a - T_\infty) + U_w A_w (T_a - T_{rh}) \tag{2}$$

$$m_w C_w \frac{dT_{rh}}{dt} = U_w A_w (T_a - T_{rh}) - h_{o.d} A_w (T_{rh} - T_\infty) \tag{3}$$

Governing heat transfer equations in the supply duct can also be derived using the control volumes defined in a supply duct as follows.

$$m_s C_a \frac{dT_i}{dt} = \dot{m} C_a (T_a - T_i) - U_s A_s (T_i - T_{rs}) \tag{4}$$

$$m_d C_d \frac{dT_{rs}}{dt} = U_s A_s (T_i - T_{rs}) - h_{o.s} A_s (T_{rs} - T_\infty) \tag{5}$$

To get the transfer function, which is required in the analysis of dynamic characteristics of the heating device and in the design of a control system, we take the Laplace transformations of Eqs. (1) ~ (5).

$$\begin{aligned} \tilde{T}_a(s) &= G_1(s)\,\dot{q}(s) + G_2(s)\,\tilde{T}_{rh}(s) \\ G_1(s) &= \frac{1}{D(s)} \\ D(s) &= \frac{m_h C_h m_{ah} C_a}{U_h A_h} s^2 + \left[m_h C_h + m_{ah} C_a + \frac{m_h C_h}{U_h A_h} (\dot{m} C_a + U_w A_w) \right] s \\ &\quad + \dot{m} C_a + U_w A_w \\ G_2(s) &= U_w A_w G_1(s) \end{aligned} \tag{6}$$

$$\begin{aligned} \tilde{T}_{rh}(s) &= G_3(s)\,\tilde{T}_a(s) \\ G_3(s) &= \frac{U_w A_w}{m_w C_w s + U_w A_w + h_{o.w} A_s} \end{aligned} \tag{7}$$

$$\begin{aligned} \tilde{T}_i(s) &= G_4(s)\,\tilde{T}_a(s) + G_5(s)\,\tilde{T}_{rs}(s) \\ G_4(s) &= \frac{\dot{m} C_a}{m_s C_a s + \dot{m} C_a + U_s A_s} \\ G_5(s) &= \frac{U_s A_s}{m_s C_a s + \dot{m} C_a + U_s A_s} \end{aligned} \tag{8}$$

$$\begin{aligned} \tilde{T}_{rs}(s) &= G_6(s)\,\tilde{T}_i(s) \\ G_6(s) &= \frac{U_s A_s}{m_d C_d s + U_s A_s + h_{o.d} A_s} \end{aligned} \tag{9}$$

$$\tilde{T}_a(s) = T_a - T_\infty, \quad \tilde{T}_i(s) = T_i - T_\infty$$
$$\tilde{T}_{rh}(s) = T_{rh} - T_\infty, \quad \tilde{T}_{rs}(s) = T_{rs} - T_\infty$$

Rearranging the Eqs. (6) ~ (9), the heating chamber transfer function $G_h(s)$ and the supply duct transfer function $G_s(s)$ are defined as follows.

$$G_h = \frac{G_1}{1 - G_2 G_3}, \quad G_s = \frac{G_4}{1 - G_5 G_6} \tag{10}$$

3.2 Determination of the coefficients

Heat capacity, surface area and heat transfer coefficients in the transfer function can be obtained theoretically. The air heat capacities of heating chamber and supply duct can be computed by the product of air density, air specific heat and volume of each part. The heat transfer coefficients can be determined from the geometry such as thickness and length and material information of the system. In determining the heat transfer coefficients, we considered a conduction and a convection heat transfers simultaneously. Conduction heat transfer coefficients are computed using the conduction heat transfer properties selected from the thermal conductivity table[4] according to the shape and material of the heating chamber and supply duct. For the convection heat transfer coefficients, we first assume that the air flow in each control volume is turbulent. Then Nusselt number is computed and the convection heat transfer coefficient is computed using Dittus-Boelter equation.[4]

Heating chamber total volume: $V_h=0.0056\,m^3$

Supply duct total volume: $V_s=0.0123\,m^3$

Air mass flow rate: $\dot{m}=0.0168$ kg/s

Air heat capacity in heating chamber: $m_{ah}C_a=6.614$ J/K

Overall heat transfer coefficient at the surface of heating chamber: $U_w=2.468\,W/m^2K$

Total heating chamber surface area: $A_w=0.189\,m^2$

Air heat capacity of supply duct: $m_sC_a=14.35$ J/K

Overall heat transfer coefficient at the surface of supply duct: $U_s=1.861\,W/m^2K$

Total supply duct surface area: $A_s=0.393\,m^2$

Heat capacity of electric heater: $m_hC_h=1385.44$ J/K

Overall heat transfer coefficient at the surface of electric heater: $U_h=225.76\,W/m^2K$

Surface area of electric heater: $A_h=0.0314\,m^2$

Heat capacity of the heating chamber: $m_wC_w=726.84$ J/K

Convection heat transfer coefficient at the outer surface of heating chamber: $h_{o.w}=644.27\,W/m^2K$

Heat capacity of the supply duct: $m_dC_d=259.58$ J/K

Convection heat transfer coefficient at the outer surface of supply duct: $h_{o.d}=0.648\,W/m^2K$

The transfer functions in Eqs. (6)~(10) are now expressed using the coefficients obtained as follows.

$$G_1=\frac{0.000774}{(s+0.00363)(s+3.7017)}$$

$$G_2=\frac{0.00549}{(s+0.00363)(s+3.7017)}$$

$$G_3=\frac{0.000642}{s+0.1682} \qquad G_4=\frac{1.1790}{s+1.230}$$

$$G_5=\frac{0.0509}{s+1.230} \qquad G_6=\frac{0.0028}{s+0.0038}$$

$$G_h=\frac{0.000774(s+0.1682)}{(s+3.7017)(s+0.003624)(s+0.1681)}$$

$$G_s=\frac{1.1790(s+0.0038)}{(s+0.00368)(s+1.230)}$$

3.3 Comparison between model and actual system

Fig. 5 shows the measured and simulated supply duct outlet air temperatures simultaneously. As shown in the figure, even though there is some discrepancy between the experimentally

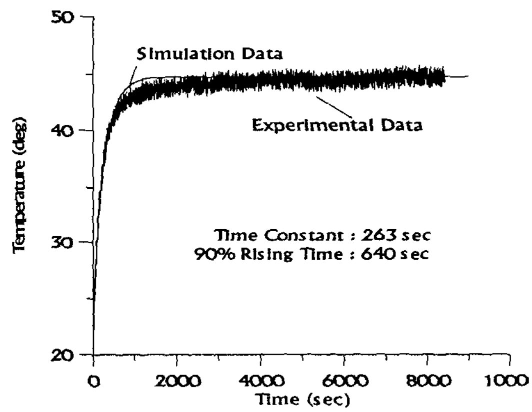

Fig. 5 Experimental data vs. simulation data.

obtained data and simulated data, the linear model represents the dynamics characteristics of the heating device very closely.

Fig. 5 also shows that the supply duct outlet air temperature in PEM is oscillating very rapidly. This is due to the fact that PEM cannot use a mixing chamber. If such fluctuating phenomena should be expressed exactly, the mathematical model will be nonlinear and very complicated which may not be very useful in terms of the implementation.

The linear model obtained may not represent the dynamics of the heating process exactly. However, we think that it is accurate enough to be used in the design of the control system.

4. Control of supply duct outlet air temperature

Fig. 6 shows the control system block diagram used in this work. The controller $G_c(s)$ is implemented digitally in PC using VISUAL C^{++}. The supply duct outlet, inlet and the ambient air temperatures are measured by thermocouples. The air flow rate are measured using the hot wire type velocity meter. Then the measured information is transmitted from a data logger to the controller through LAN. The controller computes the required heater voltage, V_c in Fig. 6, to maintain the air temperature to the set value. The bias voltage V_{ss} is a voltage required to keep the steady state heat generation of the electric heater. It is added to the computed voltage, when P (proportional) and PD (proportional and derivative) control systems are used. Finally the electric heater command voltage, V_t in Fig. 6, which should be less than 5 V, is supplied to the electric heater current regulator (TPR) by the 12 bit D/A converter installed in PC. The bias voltage V_{ss} can be computed by letting the derivative terms in Eqs. (1)~(5) zero. In Eq. (11), the conversion factor K_h is 140.3 W/V, which is obtained experimentally.

$$\begin{aligned}\dot{q}_{ss} &= V_{ss}K_h \\ &= 17.24(T_i+273)-17.22(T_\infty+273)\end{aligned} \tag{11}$$

The control system is designed such that it provides the rapid response without overshot and saturation of electric heater command voltage. We test the control system to raise the air temperature from 23℃ to 43℃, while it flows through the heating chamber and supply duct.

4.1 P (Proportional) control system

The P type controller is a control system when $G_c(s)$ in Fig. 6 is as follows.

$$G_c(s) = K_P \tag{12}$$

Generally when the gain is increased in the proportional type control system, system damping decreases, thus overshoot and the oscillation around the steady state value are increased. Using the root locus techniques, we select K_P $=0.108$ which will provide the rapid response without overshoot. Fig. 7 shows the system re-

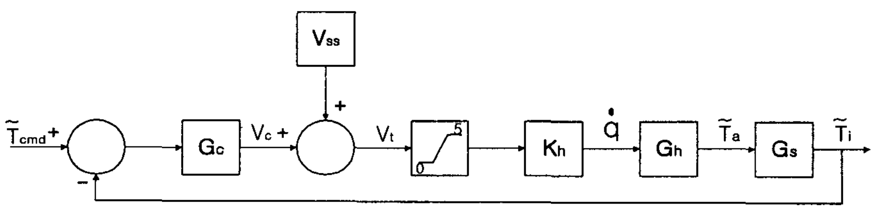

Fig. 6 Control system block diagram.

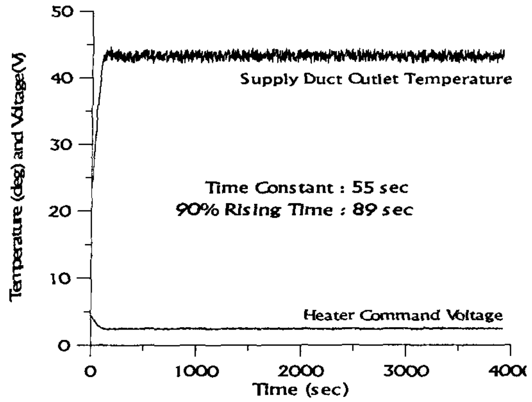

Fig. 7 System response when K_P=0.108.

sponse. Compared with the system response shown in Fig. 5, the proportional control system increases the system response about 7 times faster than that of the system which is not controlled.

4.2 PD (Proportional and Derivative) control system

The PD control system is given as follows.

$$G_c(s) = K_P + K_D s \tag{13}$$

The PD type control system generally increases the speed of system response compared with that obtained using P type control system. It also increases the system stability, since PD control system can be a pure phase lead type controller.

Fig. 8 shows the system response using PD control system. The control system gains are selected using root locus method such that the controller will not saturate the electric heater command voltage and, at the same time, the system response does not produce any overshoot. The system response with PD controller improves the system performance in reducing the rising time by 15% compared with that of P type control system. However there is a large oscillation in heater command voltage as shown in Fig. 8. This is due to the zero located near the origin of s-plane. If we select PD controller having a zero far from the origin of s-plane, the heater command voltage oscillation will be reduced, but the system response will be slow down compared with the results shown in Fig. 8.

4.3 PID (Proportional, Derivative and Integral) control system

The PID control system has the following form of controller.

$$G_c(s) = K_P + K_D s + \frac{K_I}{s} \tag{14}$$

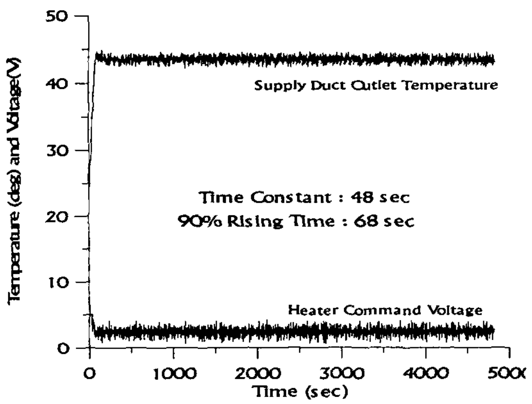

Fig. 8 System response when K_P=0.2, K_D=1.0.

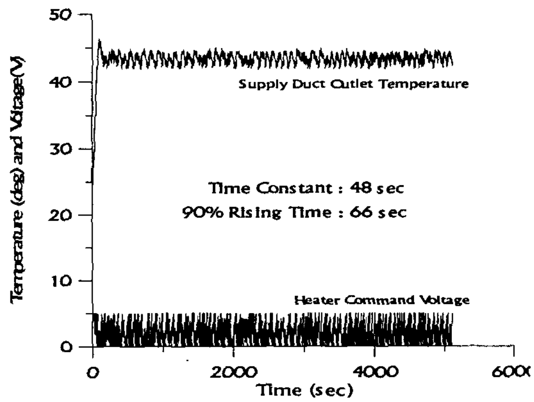

Fig. 9 System response when K_P=0.2, K_D=1.0, K_I=0.013.

PID controller is used to reduce the steady state error while other system performance, such as rising time, remains similar to the case of using P or PD type controller.

Generally the use of integrator in feedforwad path decreases the system stability, thus may cause the overshoot and oscillation in system response.

Fig. 9 shows the system response when $K_P = 0.7, K_D = 2.19, K_I = 0.013$. The PID control system is designed using root locus method such that the closed loop system poles are on the negative real axis, thus the system will have the overdamped response, while the overall performance is very similar to those of using P or PD type controller. However when PID controller is implemented, the system response shows the overshoot and a large electric heater command voltage oscillation. This is due to the saturation in heater command voltage. The saturation in command voltage contributes the generation of overshoot in response and oscillation in input voltage. When we reduce the integrator gain, we can get the system response with less overshoot and oscillation. However this reduces the speed of system response further.

4.4 Analysis of experimental results

The control system improves the system performance compared with that of the system not being controlled.

The experimentally obtained data shows that the supply duct outlet air temperature could be controlled in satisfactory manner using P type control system only. A PD type control system, which makes the system response faster by increasing the gain of P type control system, while the overshoot caused by the higher P gain is suppressed, can improve the system performance further. However the experimental data shows that the PD type control system may produce the oscillation in electric heater command voltage, thus needs to be cautious in use. The integral type control system located in feedforward path increases the system type so that it reduces the steady state error. However it also decreases the system stability. The experimental result shows that supply duct outlet air temperature in PEM is oscillating very rapidly, implying that the heating process in PEM is in relatively weak stable condition. Use of integral type control system in such weak stable system could make the system more unstable. Even though PID control system improves the system performance in reducing the steady state error, it also produces a saturation in electric heater command voltage, thus increase the oscillation in supply duct outlet air temperature.

5. Conclusions

Using a mathematical model and the simulation results describing the relationship between the supply duct outlet air temperature, flow rate and working area temperature distribution, P, PD and PID control systems are designed and implemented. The results obtained are summarized as follows.

(1) The simulation results show that PEM maintains only working area at required temperature rather than the total office area, thus can improve the thermal comfort and energy saving compared with the conventional air conditioning system.

(2) High temperature air discharging from PEM will be blown up before it reaches working area due to the buoyancy effect, if the flow rate is small. Thus the diffuser incident angle and flow condition should be selected properly and carefully based on the temperatures in the working area and in the outlet of PEM.

(3) Linear mathematical model obtained describes the actual dynamic characteristics of heating device very closely.

(4) PID controller can improve the system response.

Acknowledgements

This paper was supported by the research fund of Seoul National University of Technology.

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