Simulation on the Characteristics of the Control System of an Environmental Control Facility

Jong Suk Lee*

Key words : Environmental control facility, Control system, Simulation, Digital controller, Controlled variable, Manipulated variable

Abstract

Environmental control facilities are used to simulate an environment or combination of environments under which many kinds of research and tests can be performed. The design of the control system to maintain desired environmental conditions is essential to proper operation of the facility. A simulation model of the facility has been developed by analyzing each component of the system thermodynamically with necessary properties and heat transfer relations. Using the system simulation model, the required characteristics of the control system has been investigated. PI controller is considered as the most probable controller for this kind of the facility, and electric heater power is shown as the proper manipulated variable for temperature control.

1. Introduction

In studying and testing thermal systems, it is often necessary to maintain constant environmental conditions. Environmental control facilities are used to simulate an environment or combination of environments under which many kinds of research and tests can be performed. They are composed of two main parts: the air handling units and the rooms themselves. Air handling units provide cooling, dehumidification, heating, and moisture addition to balance the effects of the device under test, and rooms are where the test equipments are placed.

In addition to the design of the facility and selection of the air handling equipment, the design of the control system to maintain desired conditions is central to proper operation. Shavit and Brandt[1] modeled and analyzed a discharge air control system with a P-I controller. They

* Department of Precision Mechanical Engineering, Kangnung National University, Kangnung, Kangwon-do 210-702, Korea

stressed that the selection of optimal gains must be based on experience, analysis, and experimentation. The comtroller studied by them was an analog controller. Digital computers are commonly used to replace analog electronic controllers.

Originally, the term direct digital control (DDC) was used for systems that position the controlled device in a discrete stepping manner directly from digital output of a computer, in order to eliminate digital-to-analog converters and electronic-to-pneumatic transducers. Now, however, the term DDC is commonly used to mean simply programmable control using a digital computer. May et al.[2] discussed the application of direct digital control of a pneumatic chilled-water valve positioner through a digital-to-pneumatic interface. Nesler and Stoecker[3] studied proportional and integral gains in the direct digital control of discharge air temperature.

Thompson and Chen[4] investigated the effect of room and control system dynamics on energy consumption. They developed a mathematical model and used a computer simulation to observe the effect on energy consumption of various room, equipment, control system parameters. This study is to research the proper control system for the environmental control facility through computer simulation in a similar procedure as they did.

2. Description of the Facility

Figure 1 shows the components of the air handling unit, which is composed of a filter, two chilled water coils, an electric heater, a fan, a humidifier, and duct work.

For the operation and control of the environmental control facility, proper instrumentation is necessary. The purpose of the instrumentation is twofold: control and energy flow calculation. With the proper controller, the desired air conditions can be maintained closely. Then the heat rates of the test equipment are determined by the energy removal or addition by the air handling unit required to balance them.

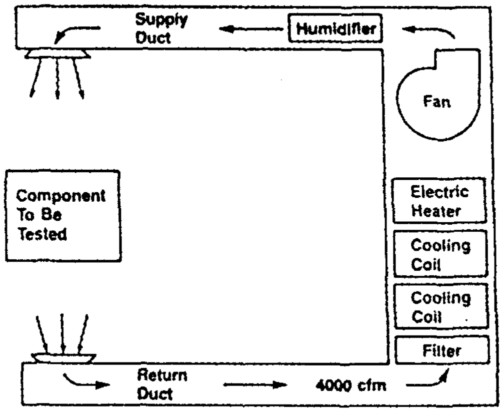


Fig. 1 Diagram of the air handling unit.

For the calculation of the energy removal by the cooling coil in the air handling unit, inlet and outlet temperatures of the cooling water, and its flow rate must be measured. For the energy addition by the electric heater, its power must be measured. The steam flow rate into the humidifier must also be known, but it is difficult to measure directly. Instead, it can be regarded as equal to the condensation rate, which may be calculated from the humidity ratio difference and the air flow rate. For the estimation of the heat flow between the facility and the surroundings, temperatures of outdoor, indoor, and the other room must be measured. All of these measurements are for determining instantaneous energy flow rates. In order to calculate the amount of energy flow, these measurements should be taken at regular time intervals to be integrated over time. Accordingly, a sophisticated data acquisition system is a very important part of the facility.

3. System Simulation Model

A system simulation model was needed to study the characteristics of the control system of the environmental control facility. It was developed by considering proper control volume for each component of the system and using fundamental mass and energy balance equations with necessary properties and heat transfer relations. In doing so, the following general assumptions were made:

- kinetic and potential energies are negligible
- flow conditions are uniform at any cross-section
- air-water vapor mixtures are treated as ideal gases

For a control volume, the mass and energy rate balances are, respectively

$$\frac{dm_{cv}}{dt} = \sum \dot{m}_i - \sum \dot{m}_e \quad (1)$$

$$\frac{dU_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_i h_i - \sum \dot{m}_e h_e \quad (2)$$

where m is mass, U is internal energy, Q is heat, W is work, h is enthalpy, and i and e denote inlets and exits respectively.

By applying equation (1) and (2) to each component of the system, governing equations have been derived, some of which are algebraic equations and others first-order ordinary differential equations. These equations must be solved simultaneously.

Properties of air and water such as enthalpy, internal energy, and psychrometric properties are needed to be evaluated. The thermophysical properties of water and air are assumed to be constant for the expected range of operating conditions. Heat transfer rates in the derived governing equations are expressed in terms of heat transfer coefficients and temperature differences. The heat transfer coefficients for the chilled wate coil model are evaluated using appropriate empirical relations and geometrical considerations.

Modeling of the controlled space(room) as well as the air handling unit must be completed, and the details are described in references[5,6]. The system simulation model developed in this way is used for the design and performance analysis of the control system for the environmental control facility.

4. Description of the Control System

4.1 Digital control concept

In modern HVAC control, digital control using computers is commonly adopted. A schematic diagram of the main components of a typical control system is shown in Fig. 2, which illustrates how temperature is controlled by adjusting the chilled water flow.

The temperature is measured by thermocouples, and a voltmeter converts the analog electric voltage into a digital electric signal. This

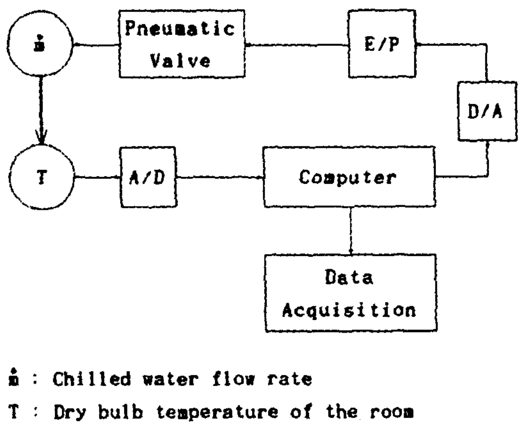


Fig. 2 Diagram of a typical control system.

signal is transmitted to the digital computer as an input. A suitable control program gives the necessary control signal as an output, while the data acquisition system records the incoming data. The digital output signal is converted again to an analog signal and then transmitted to the electro-pneumatic transducer, which renders it into a pneumatic pressure to actuate a pneumatic valve. The pneumatic valve controls the chilled water flow rate, which in turn controls the temperature. This is the feedback control system to maintain the desired temperature by adjusting the flow rate in the chilled water coil.

An advantage of a digital control system is that the control algorithms are software-based. Thus, proportional (P), proportional-integral (PI), or proportional-integral-derivative (PID) control can be implemented by just changing the associated gains without switching hardwares.

4.2 Control strategy

The variables which are to be controlled are temperature and humidity. The manipulated inputs to provide these controls are the water flow rates in the chilled water coils and/or the power input to the electric heater and the steam injection rate. Therefore, this system is a multiple-input, multiple-output(MIMO) system. Because MIMO systems are more complex than the single-input, single-output(SISO) system, it is advantageous to use SISO control strategy even for multivariable control systems if possible. To do this, it is necessary to characterize the amount of interaction present in the process to be controlled. If the interaction is slight, SISO control strategy can be applied to each individual loop with success.

Using the relative gain method, the process interactions were estimated[5]. The results indicate that the system is coupled, but the interactions are small enough to allow the use of SISO control strategy. Therefore, temperature and humidity controls are to be considered separately. Comparison of the two relative gain matrices also indicates that electric heater power may be a beeter control variable than the water flow rate for temperature control.

4.3 Digital controller

The facility is simulated by integrating the governing first-order differential equations with the time interval of 0.36 seconds while the digital controller is simulated with the sampling time of 3.6 seconds. Any kind of controller can be implemented by using the PID equation, as described in Brandt and Shavit's paper[7]. For a continuous control system, the PID equation is

$$O = M + K_P \cdot E + K_I \cdot \int E dt + K_D \cdot \frac{dE}{dt} \qquad (3)$$

where O is controller output, M is offset of controller output, E is deviation of sensed variable from setpoint, and K_P, K_I, K_D are proportional, integral, derivative gains respectively. Deviation E is that of temperature or humidity and output O is the adjusted amount of water flow rate, heater power, or steam injection rate.

To use the PID equation with a digital controller, the discrete form of the equation should be used. The integration is replaced by a summation and the derivative is approximated by the ratio of difference between the previous error and the current error to the sampling time(t_s), which result in

$$O_i = M + K_P \cdot E_i + K_I \cdot \sum E_i \cdot t_S + K_D \cdot \frac{E_i - E_{i-1}}{t_s} \qquad (4)$$

where i denotes values at time i, and Σ means summation from time zero through time i.

The above equation can be used for any combination of control modes by assigning each gain. For example, an integral controller is implemented by setting the proportional and derivative gains to zero. In order to design the controller, a heat pump of 3-ton cooling capacity was chosen as the test equipment.

5. Simulation Results and Discussion

5.1 Simulation method

To choose the proper values of the gains, two different kinds of simulations were performed: step changes in setpoint and step change in the test heat rate (or cooling rate). Standard operating conditions for rating the heat pump were used as the initial steady-state conditions: 80°F for cooling tests and 70°F for heating tests. A step change of ten degrees in setpoint temperature was tried. Also, a step change in the test heat rate was simulated as a major disturbance to the system. The initial conditions for these simulations were obtained by solving the governing equations for steady-state operation at the setpoint values.

5.2 Comparison of controllers

Cooling test simulations with a step change in setpoint temperature from 80°F steady-state operating condition to 90°F were used to determine the general system responses of proportional (P), interal (I), proportional-integral (PI), proportional-derivative (PD), and proportional-integral-derivative (PID) controllers.

Figure 3 shows the system responses using only proportional control with three different gains. With a relatively small gain, as shown in Fig. 3 (a), the response is slow and has steady-state error which is typical of propor-

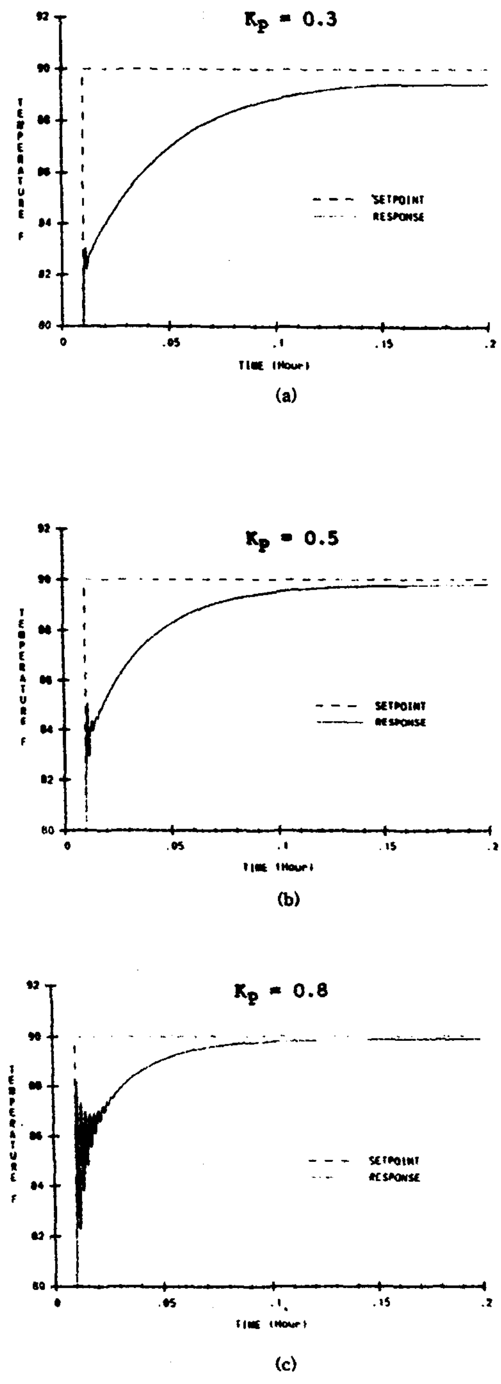


Fig. 3 Responses of the proportional controller.

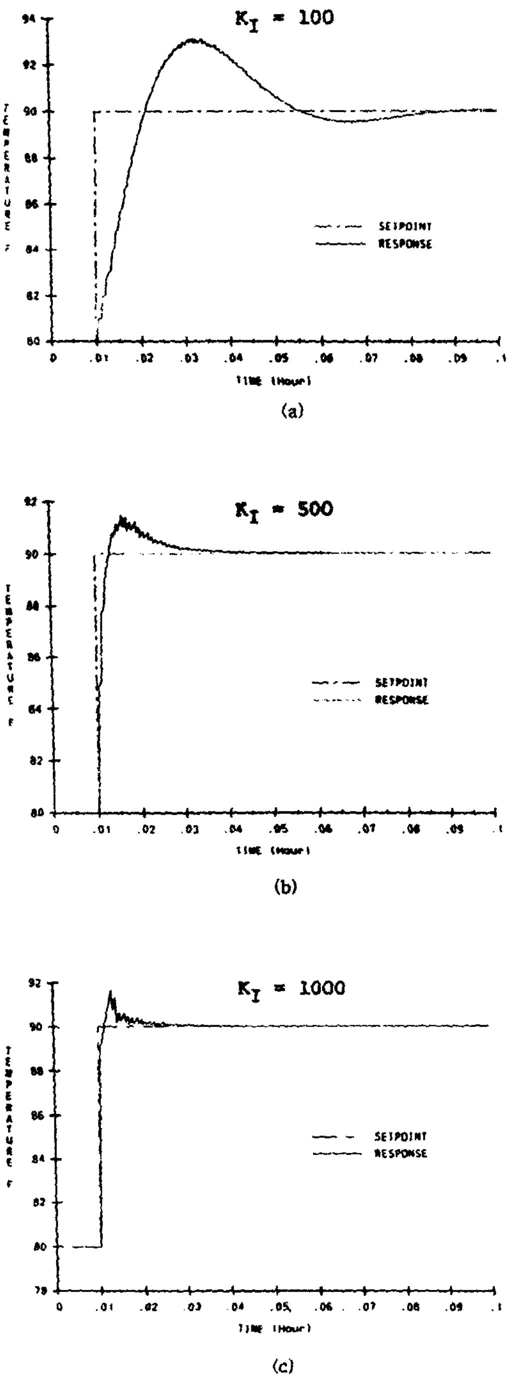


Fig. 4 Responses of the integral controller.

tional control. As the gain increases, as shown in Fig. 3 (b), the response becomes faster and the steady-state error decreases. A larger gain makes the response much faster and the steady-state error smaller, but it results in oscillations, as shown in Fig. 3 (c). Therefore, the desired proportional gain is to be compromised between small steady-state error and fast response with some oscillation.

Figure 4 shows the system responses for pure integral control with three different gains. Steady-state error is eliminated by this controller and the system responses are faster than those of proportional controllers. With a small gain, as shown in Fig. 4 (a), the response overshoots too much and it takes time to settle to the steady value. By increasing the integral gain, the responses are faster but more oscillatory, as shown in Fig. 4 (b) and (c). Further increase of the integral gain makes the response become unstable.

A PI controller is simulated by including both proportional and integral gains. Figure 5 shows the system responses for PI controllers with different gains.

Figure 5 (a) is for K_P=0.5 and K_I=500, which were nearly optimum for pure proportional and integral controls. The system response is fast and has zero steady-state error. The effects of excessive proportional and integral gains for the PI controller are shown in Fig. 5 (b) and (c), respectively. In each case, oscillations occur which might cause the system response to become unstable. In the actual system, attention should be paid to avoid such oscillations.

Derivative control is not used by itself, but is combined with the other types of controllers. Proportional-derivative control doesn't change the system response much if the derivative gain is small, but high derivative gain causes the system response to be oscillatory or unstable.

Finally, a PID controller is obtained by adding the derivative term to the PI controller. Figure 6 shows the system responses using PID controllers with three different derivative gains. In each case, proportional gain is kept as K_P=0.5 while the integral gain is increased to K_I=700. Figure 6 (a) shows that PID controller responds slightly faster and the overshoot lessens, compared with PI controller(Fig. 5 (a)). This is because the derivative term allows higher integral gains by acting as a damping factor. However, the improvement is not so significant as to make PID controller more attractive than the PI controller.

The effects of lower and higher derivative gains are shown in Fig. 6 (b) and (c), respectively. The lower gain doesn't change the response much while the higher gain makes the response more oscillatory. According to Brandt and Shavit[7], the derivative calculation in a digital controller is sensitive to sensor noise. In the current study, the dynamics of the sensor was ignored for simplicity. In the actual system, however, sensor noise may not be negligible. This is another reason that PI control is preferred.

5.3 System responses to load change

For the test of controllers, a large step change in cooling load was simulated as a major disturbance to the system. For a step decrease in cooling load from 36,000 Btu/h to 26,000 Btu/h, the proper gains for each controller were slightly different from the values obtained for the simulations of setpoint change. Figure 7 (a) and (b) show the system responses of the properly tuned proportional and integral controllers, respectively. Proportional gain was increased to 0.7 from previously tuned

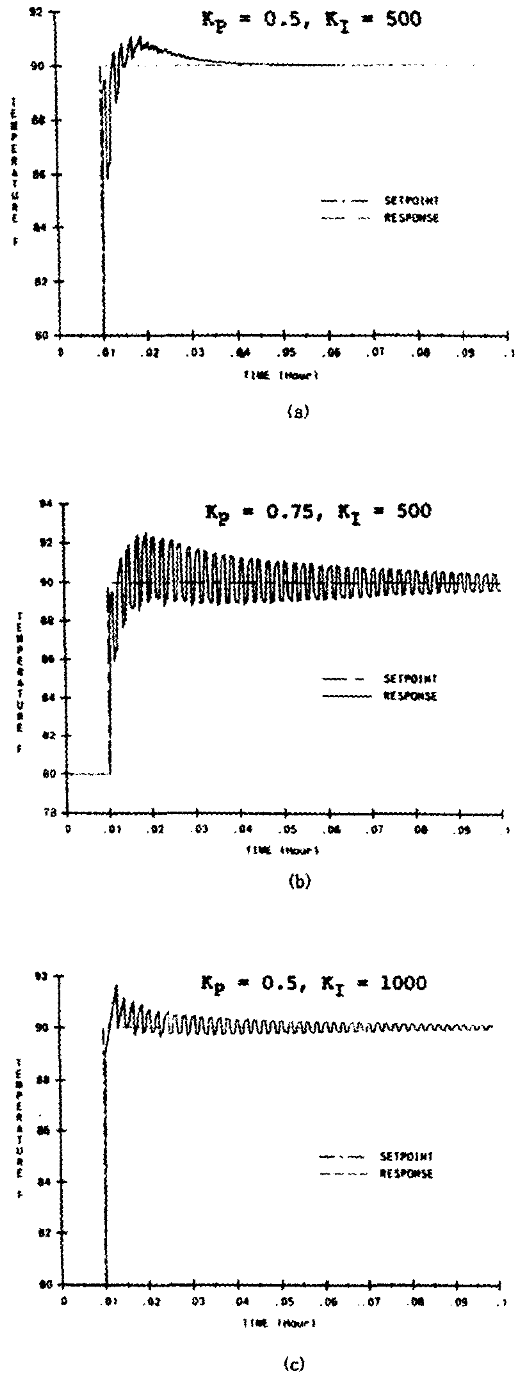
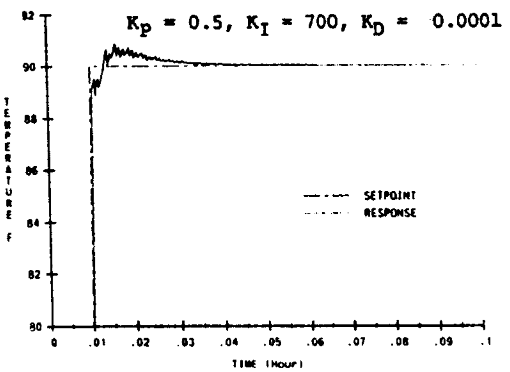
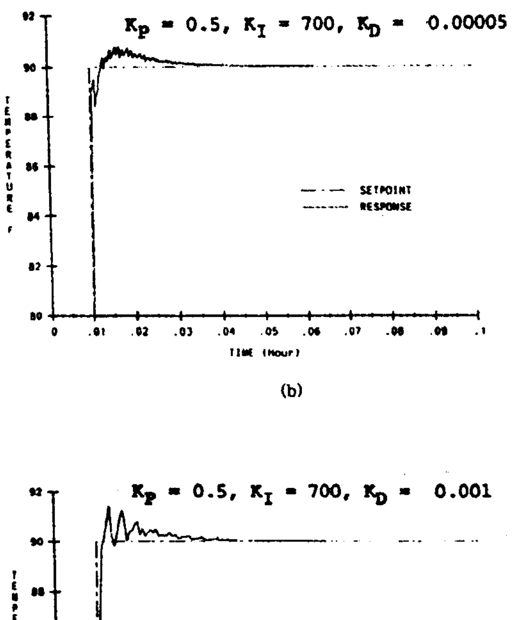


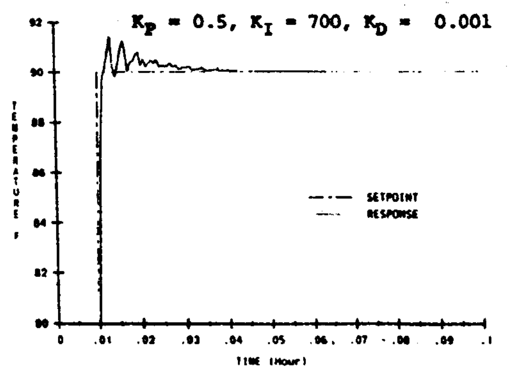
Fig. 5 Responses of the proportional-inegral controller.



(a)



(b)



(c)

Fig. 6 Responses of the PID controller.

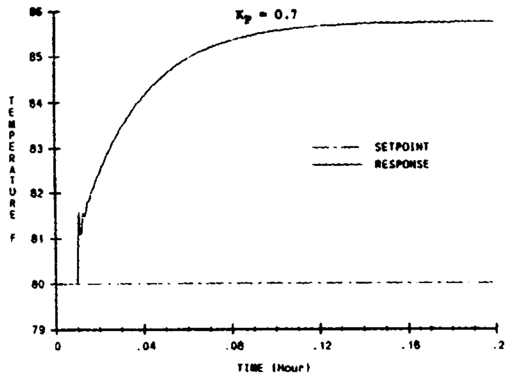
0.5. The proportional controller has steady-state error of more than 5°F, and it reaches steady state in about 8 minutes. The integral controller reaches its steady state in about 3 minutes and has zero steady-state error.

Responses of PI and PID controllers were similar to that of integral controller. In both cases, the system responses reach steady state in about 2 minutes with zero steady-state error. The PID controller adds some damping, however, the difference is too small to be significant. Considering that derivative term may become so sensitive to the sensor characteristics, it seems to be of no benefit to use PID control with the risk of instability.

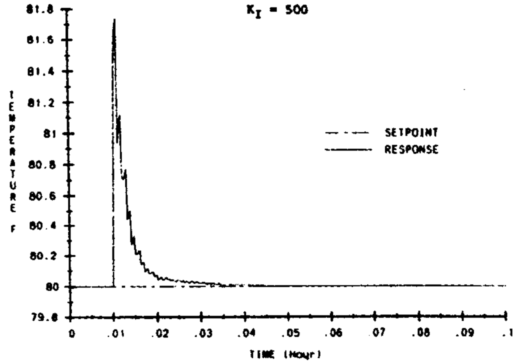
Summarizing the results of the cooling test simulations, a PI controller with properly tuned gains seems to be the most probable controller to be implemented for the environmental control facility.

5.4 Comparison of manipulated variables

During heating tests, the air handling unit must remove heat to balance the heat input by the test equipment. The chilled water coils are used for this purpose. In order to maintain the desired temperature, the water flow rate through the coils can be controlled by the proper controller. Alternatively, a cooling and reheating method may be employed. That is, the air may be cooled down to a lower temperature than desired and then reheated by the electric heater. Electric power input is usually easier to control than water flow rate, because water flow rate control includes an electro-pneumatic transducer and a valve with some hysteresis. Also this method may make it easier to control humidity, because steady condensation is anticipated with the constant water flow rate.

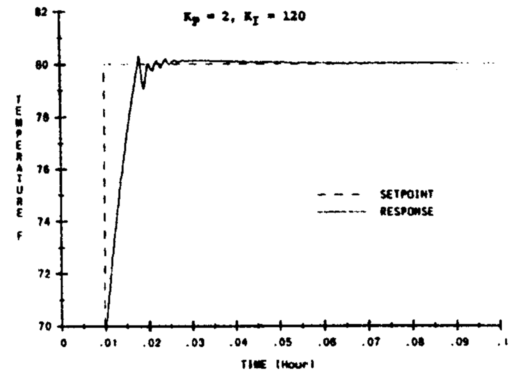


(a) Proportional controller

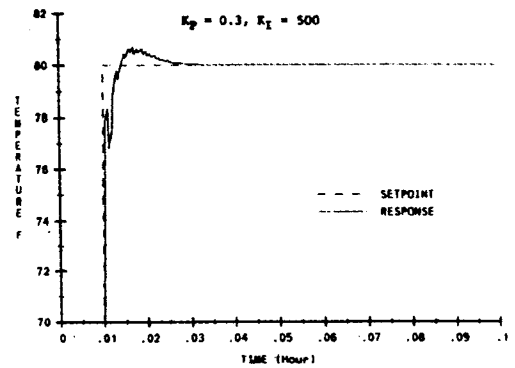


(b) Integral controller

Fig. 7 Responses to step change in cooling load.



(a) Water flow rate (cooling only)



(b) Electric heater power (cooling and reheating)

Fig. 8 Responses of PI controller using different manipulated variables in heating test.

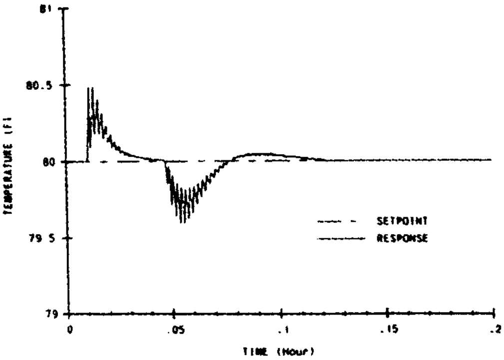
These two methods were compared with PI controllers. Figure 8 (a) and (b) show the system responses to the setpoint change when water flow rate and electric heater power were used as manipulated variables, respectively. Comparing the responses of the two different methods, the second method, cooling with reheat, appears to be a little faster than controlling the water flow rate alone. This result confirms the intuition that electric heater can be controlled faster than chilled water coil. It should be noted, however, that the simulation ignores any thermal dynamics of the heater itself.

5.5 Humidity control and interactions

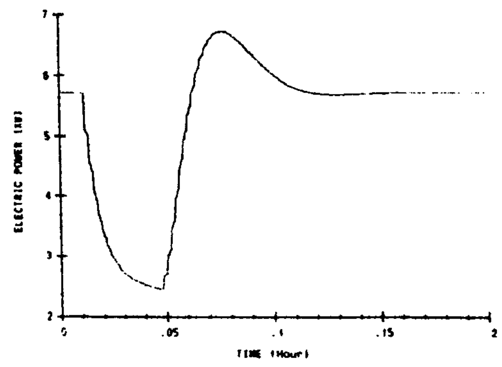
In the previous sections, temperature control was considered by assuming that humidity is controlled perfectly. In this section, the humidity controller is designed using the properly tuned temperature controller. Humidity ratio is used as the controlled variable for the humidity control. For cooling tests, standard operating conditions are dry bulb 80 °F and wet bulb 67 °F, in which humidity ratio is 0.011 lbw/lba. A step change of setpoint in humidity ratio from 0.011 to 0.015 was simulated for the design of the humidity controller. A PI controller was found to give the best response as in the temperature control, while it took more time for humidity to reach steady state than temperature.

In designing humidity controller, the temperature setpoint was remained unchanged. However, air temperature in the controlled space is affected by the humidity control due to coupling in the energy balances. Figure 9 (a) shows the temperature response affected by the humidity setpoint change. Owing to the steam injection, the temperature increases at first. The temperature controller reduces the electric heater power in order to keep the setpoint temperature, as shown in Fig. 9 (b). As a result, the temperature returns to the setpoint shortly. As the steam flow starts to decrease, as shown in Fig. 9 (c), after overshooting the desired humidity ratio, the temperature decreases at the same time. Thus, at this time temperature controller increases the electric heater power to raise the temperature. Finally, the air temperature in the controlled space settles to its setpoint value.

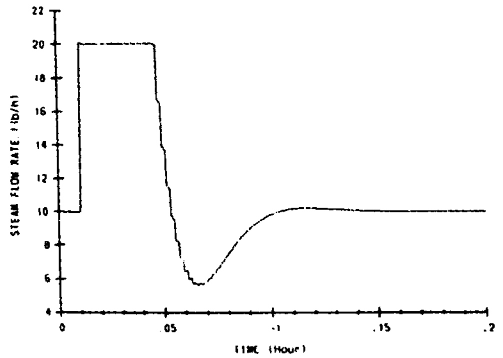
Even though there are obvious interactions between temperature and humidity controls, each controller can be designed independently and both temperature and humidity are shown to be controlled well within reasonably small tolerances(0.5°F). Therefore, the SISO control strategy used in the current study can be accepted as reasonable.



(a) Temperature response (with PI controller)



(b) Electric heater power



(c) Steam flow rate

Fig. 9 Interactions between temperature control and humidity control.

6. Conclusions

In order to study control system of the environmental control facility, a system simulation model was developed. For the development of the model, each component of the environmental control facility was analyzed thermodynamically with necessary properties and heat transfer relations.

In the design of the control system, a direct digital control concept and control strategy were addressed, and controlled and manipulated variables were considered. For the typical operating conditions, simulations of several different controllers were performed using the system simulation model developed in this study. Different controllers were compared by analyzing the simulation results.

In conclusion, among the various kinds of controllers, a PI controller seems to be the most probable controller for the facility, and electric heater power the proper manipulated variable for temperature control. Also, temperature and humidity can be controlled independently with success.

The purpose of the current study was to determine the probable control method and controller type for the environmental control facility by comparing simulationresults of different controllers. For the optimal controller design of the actual facility, all of the characteristics of each component of the control system should be taken into account.

References

(1) Shavit, G. and Brandt. S. G., 1982, "The dynamic performance of a discharge air-temperature system with a P-I controller" , *ASHRAE Transactions* Vol. 88 Part 2, pp. 826-838.

(2) May, W. B., Borresen, B. A., and Hurley, C. W., 1982, "Direct digital control of a pneumatically actuated air handling unit" , *ASHRAE Transactions* Vol. 88, Part 2, pp. 857-874.

(3) Nesler, C. G. and Stoecker, W. F., 1984, "Selecting the proportional and integral constants in the direct digital control of discharge air temperature" , *ASHRAE Transactions* Vol. 90, Part 2B, pp. 834-845.

(4) Thompson, J. G. and Chen, P. N. T., 1979, "Digital simulation of the effect of room and control system dynamics on energy consumption", *ASHRAE Transactions* Vol. 85, Part 2, pp. 222-237.

(5) Lee, J. S., 1987, Design and Analysis of an Environmental Control Facility, M. S. Thesis, Iowa State University, Ames, Iowa.

(6) Lee, J. S., 1997, "Design and modeling of an environmental control facility", Proceedings of the SAREK '97 Annual Summer Conference(I), pp. 150-155.

(7) Brandt, S. G. and Shavit, G., 1984, "Simulation of the PID algorithm for direct digital control applications", Proceedings of Workshop on HVAC Controls Modeling and Simulation, Atlanta, Ga.