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USE OF A GENERAL CONTROL SIMULATION PROGRAM TO EVALUATE HVAC CONTROL STABILITY

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to Evaluate HVAC Control Stability

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ABSTRACT

A general process engineering simulation program for a microcomputer is used to evaluate the response of a cooling system to changes in load, set point, and controller tuning parameters to better understand the control stability of a supply air cooling system used in the Mobile Windows Thermal Test (MoWiTT) Facility. A detailed system model is built out of standard components and is used to plot the steady state performance under different operating conditions. The response of the cooling coil output depends strongly on the transport delays. For a system with a linear control valve, the process gain of the cooling coil increases rapidly as the valve flow fraction decreases. A system which is stable at mid-range is shown to become marginally unstable at the low end of the control range. The application of both an equal percentage valve and reheat are shown to improve the control response of the system under low cooling load conditions.

INTRODUCTION

The Mobile Windows Thermal Test (MoWiTT) Facility [1,2] has been developed at LBL specifically to measure the dynamic net energy performance of windows under realistic field conditions. The principle of operation is that the thermal fluxes through the window system can be carefully measured. Each test cell is a carefully calibrated hotbox. The thermal loses through the envelope are largely eliminated by surrounding the test cell

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by a guard air plenum that is maintained at very close to the interior temperature conditions.

A separate heating and cooling system supplies or removes heat to maintain a constant interior temperature. By measuring the heat added or extracted from the test cell, and measuring the heat flux through the floor, ceiling, and walls (which should be small) a direct measure of the net thermal flux by the window system can be obtained. Crucial to the operation of the test facility is the requirement that the cooling and heating system respond to changes in thermal loading of the test cell and maintain a constant interior temperature. This work was undertaken to better understand the control stability and tuning of a supply air cooling system used in the Mobile Windows Thermal Test (MoWiTT) Facility to maintain the constant interior temperature of the test cell.

Powerful general purpose simulation tools are now becoming available for microcomputers. TUTSIM,™† a program for engineering design and optimization by simulation of continuous dynamic systems was developed by Twente University of Technology in The Netherlands. It is used to model the complex discharge air and room temperature control system. A detailed system model is built out of standard components and the steady state performance is plotted under different operating conditions. The time constants for system response to a step change in input are observed. The feedback control of the system is then modeled and stability of the system for various control parameters can be verified. Different approaches to control of the system including the application of an equal percentage valve and reheat are modeled to show the improvement in the control response under low cooling load conditions.

SYSTEM MODEL

A discharge air temperature cooling system has been modeled as shown in Figure 1. The cooling system consists of a three-way valve, bypass, pipe and pump, a fan coil unit

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heat exchanger, and a return pipe. Chilled water is provided at temperature θ_{w3} and flow rate F_w to the three way valve and bypass. The pump maintains a constant flow rate, f_w, in the fan coil unit. The fraction of the flow mixed by the three-way valve, x, controls the coil supply water temperature θ_{w1}. Because there is a significant length of pipe there is be a time delay between the temperature at the valve and bypass outlet, θ_{w1b}, and the temperature delivered to the fan coil unit, θ_{w1a}. The control diagram for this system is shown in Figure 2. Each functional block of the control diagram is discussed below.

Controller

The transfer function for the controller, $G_{control}(s)$, relates the controller output signal, V(s), to the temperature difference between the room and the set point.

$$V(s) = G_{control}(s) \cdot \left(\theta_s(s) - \theta_m(s)\right)$$

where V(s) is the controller output to the valve actuator, $\theta_s(s)$ is the set point temperature, and $\theta_m(s)$ is the room air temperature. We shall assume the controller output varies from 0 to 10.

Valve Actuator

The flow fraction of the three-way valve, x, depends on the valve characteristics and the actuator. We shall examine the performance of a linear and an equal percentage valve. $G_{actuator}$ in Figure 2 represents the relationship between flow fraction and valve actuator input. For a linear mixing valve the fraction of flow through the valve, x, is a linear function of the actuator input,

$$x = G_{actuator}(V) = 1.0 \cdot (V/V_o) \qquad (1a)$$

where V is the controller output signal, and V_o is the controller output for maximum valve opening. If controller output signal is too large or is negative, the valve will encounter the hard non-linearity in the valve action as the valve becomes fully open or closed. Unless specifically stated otherwise our analysis will apply to a system with a linear valve. For an equal percentage valve the fraction of flow through the valve, x, is a power function of the actuator input,

$$x = G_{actuator}(V) = 1.0 \cdot (V/V_o)^{3.3} \tag{1b}$$

The effect of the equal percentage valve will be discussed later.

Valve and Bypass

Assuming that $F_w < f_w$, the linear three-way mixing valve and bypass has an output given by

$$f_w \cdot \theta_{w1b} = F_w \cdot x \cdot \theta_{w3} + (f_w - F_w x) \cdot \theta_{w2b}, \text{ or}$$
$$\theta_{w1b} = (F_w/f_w)x\theta_{w3} + \left(1-(F_w/f_w)x\right)\theta_{w2b} \tag{2}$$

where, f_w, F_w are water mass flow rates and θ_{w1b}, θ_{w2b}, θ_{w3} are water temperatures as shown in Figure 1. G_{valve} in Figure 2 represents response of the output flow of the three-way valve and bypass as given by equation (2). The chilled water return temperature θ_{w4} is given by

$$F_w \cdot \theta_{w4} = F_w \cdot (1-x) \cdot \theta_{w3} + F_w \cdot x \cdot \theta_{w2b}, \text{ or}$$
$$\theta_{w4} = (1-x)\theta_{w3} + x\theta_{w2b}. \tag{3}$$

Pipe Transport

There is a time delay, τ_{pipe}, required for the fluid flow, f_w, to move a distance, L_{pipe}, through a pipe of area, A_{pipe} given by

$$\tau_{pipe} = \rho_w L_{pipe} \cdot (A_{pipe}/f_w).$$

If the two way valve is used to modulate the flow through the cooling coil then the transport delay time, τ_{pipe} will increase as the flow rate decreases. The transfer functions for the pipe delay is given by

$$G_{pipe1}(s) = G_{pipe2}(s) = e^{-s\tau_{pipe}}, \tag{4}$$

where τ_{pipe} is the delay time. The flow rate for the test facility is about 2 gal/minute (7.6 kg/min), of 50% glycol in a 1.9 cm (0.75 inch) diameter pipe. This corresponds to a fluid velocity of about 0.5 m/s. If the pipe length is 5 m, then the transport delay would be $\tau_{pipe} = 10$ s.

Water-side Cooling Coil

The steady state performance of the fan coil system is modeled using a simple effectiveness model for a counter flow heat exchanger.

$$\epsilon_w = \frac{\theta_{w1} - \theta_{w2}}{\theta_{w1} - \theta_{a1}} \tag{5}$$

Following Borresen [3], we simulate cooling coil dynamics with a single time constant, τ_{coil}.

$$\theta_{w2} = ((1-\epsilon_w)\cdot\theta_{w1a} + \epsilon_w\cdot\theta_{a1}) \cdot 1/(1+s\tau_{coil}) \,, \tag{6}$$

where, θ_{a1} is inlet air temperature and τ_{coil} is time constant of the heat exchanger. Since the flow rate, f_w, is assumed constant, the time constant, τ_{coil} is the time for the mass of the coil to respond to a step input to water temperature.

Air-side Cooling Coil

The air temperature of the room is being controlled by the application of cooling by the air side of the fan coil unit. In general, there can be both heating and cooling. When air passes through the heat exchanger, it is cooled by water. The heat flow rate from air to water is

$$Q_{coil} = F_a\cdot(C_a)\cdot(\theta_{a1} - \theta_{w1a})\cdot\epsilon_a\cdot 1/(1+s\tau_{coil}), \tag{7}$$

where F_a is the air mass flow and C_a is the air heat capacity. The air side effectiveness, ϵ_a, is given by the standard heat exchanger equation [4].

$$\epsilon_a = \frac{\theta_{a1} - \theta_{a2}}{\theta_{a1} - \theta_{w1}} = \frac{1 - e^{-C_o\cdot(1-M)}}{1 - (M)\cdot e^{-C_o\cdot(1-M)}},$$

where C_o is the ratio of heat transfer effectiveness of the coil, $A_o\cdot U_o$ to the heat transfer on the air side, $F_a \cdot C_a$,

$$C_o = A_o \cdot U_o / F_a \cdot C_a$$

and M is the ratio of mass flow capacitance,

$$M = C_{min} / C_{max} = F_a \cdot C_a / f_w \cdot C_w .$$

The heat capacity of the 50% glycol heat transfer fluid is C_w = 3.5 kJ/kg-C (0.84 Btu/lbm-F).

In the modeled system, the mass flow rates, f_w and F_a, are constants so that the air side effectiveness, ϵ_a is a constant. The water- and air-side effectivenesses are related by the ratio of mass flow capacitance,

$$\epsilon_w = M \cdot \epsilon_a .$$

G_{coil} in Figure 2 represents the response to water temperatures, θ_{w1a}, and the room temperature, θ_m, of the outlet water temperature of the cooling coil, θ_{w2a}, as given by equation (6), and of the heat flow to the air, Q_{coil}, as given by equation (7).

Room Thermal Model

The thermal test cell consists of a room 2.4 m by 3.1 m by 2.4 meters, with a window in one wall. A cooling coil provides a net heat removal to maintain constant interior temperature. In this work no attempt has been made to identify the specific parameters of the test cell. Rather, estimates have been made to establish the magnitude of different parameters and their impact on control stability. The room has a volume of 17.8 m^3 and an assumed infiltration rate of 0.75 air changes per hour. The effective heat loss from infiltration is $U_{infiltration}$ = 3.7 W/C. The thermal mass of the room is assumed to be about the air mass in the room; $C_{room} = C_a * M_{room}$ = 5 kJ/C. For a south facing window at 40 degrees North Latitude in January, the solar gain through the 1.1 m by 1.1 m window has a maximum of about 400 watts. The effective conduction of the window, U_{cond} is about 3.9 W/C for single glazing, about 2.7 W/C for double clear glazing, and about 1.6 W/C for low-emissivity coated double glazing. Because of the constant temperature guard volume surrounding the test room, the conduction through the remaining volume is assumed to be negligible.

A simple model [3] is used to describe the temperature of the room in terms of the room transfer function, heat conduction, heat gain through the window, and heat extraction by the cooling coil. The room transfer function is given by

$$G_{room}(s) = 1/C_{room} \cdot s ,$$

where C_{room} is the room heat capacity.

The heat flow from outside through the window to the inside by conduction is given by

$$Q_{cond}(s) = G_{cond}(s) \left(\theta_o(s) - \theta_m(s)\right) \tag{8}$$

where $\theta_o(s)$ is the outside temperature, and $\theta_m(s)$ is the room air temperature. The transfer function of the window heat conduction is given by

$$G_{cond}(s) = U_{cond}/(1+s\tau_{cond})$$

where U_{cond} combines the effect of window conduction and infiltration.

For this analysis we have assumed that the total conduction, $U_{cond} = 8$ W/C and that the room thermal mass is $C_{room} = 5$ kJ/C. The time constant for the room to come to equilibrium with the outdoor temperature is

$$\tau_{room} = R_{Cond} \cdot C_{room} = C_{room}/U_{Cond} = 625\text{s} .$$

Solar gains through the window is represented by $Q_{win}(s)$, the heat flow from outside to inside in the form of radiation.

G_{sensor} in Figure 2 represents the transfer function for the measurement of the room temperature, θ_m. We simply assume $G_{sensor}(s) = 1$.

From Figure 2, we know that the output of the system, the room air temperature $\theta_m(s)$, is a complicated function of the chilled water supply temperature, $\theta_{w3}(s)$, the outdoor temperature, $\theta_o(s)$, the set point temperature, $\theta_s(s)$, and the solar gain through the window, $Q_{win}(s)$. The system is high order, nonlinear, multiple input, and difficult to analyze. We shall now develop a TUTSIM model of the system. There are hard non-linearities in the action of the control valves when they reach the fully open or closed position.

TUTSIM MODEL

The simulation model is developed by a series of functional blocks with the inputs, outputs, and initial conditions specified, which are then integrated with time to calculate the response of the system. The program has blocks representing summation (SUM) , multiply (MUL), constant (CON), gain (GAI), limit (LIM), delay (DEL), first order (FIO), or Euler integration (EUL) blocks[6].

The discharge air temperature open loop control system can be modeled as shown in Figure 3. Each functional element is represented by a group of blocks with constants and initial conditions specified. The simulation runs on a personal computer and plots its results to the screen. A simple model can be developed to predict the steady state performance. With non-zero delays and time constants, the response of the system to different disturbances and control and process gains can be observed. The parameters used in the model simulation are shown in Table 1. The program operates in the time domain and generates the time response of the system to programmed forcing functions.

STEADY STATE RESPONSE

From equations 2, 4, 6, 7, 8, and 9 for the valve and bypass, pipe, cooling coil, and room we can calculate the steady state output of the chilled water to the coil and the cooling delivered to the space at a constant space temperature as a function of the valve actuator input, V . We can also determine the steady state room temperature.

The coil cooling output depends on the temperature and flow rate of the chilled water entering the coil and the temperature and flow rate of the air entering the coil. The heat transfer is limited on the air side. As the air flow rate decreases, the air-side

Table 1.
Parameters Used for the Simulation

Water flow to cooling coil	f_w	0.13 kg/s (2.7 gal/min)
Water flow from chiller	F_w	0.10 kg/s (2 gal/min)
Flow ratio	F_w/f_w	0.75
Air flow rate	F_a	0.05 kg/s
Glycol density	ρ_w	1000 kg/m^3
Glycol and water heat capacity	C_w	3.5 kJ/kg-C (0.84 Btu/lbm-F)
Water heat capacity	C_{water}	4.186 kJ/kg-C
Air heat capacity	C_a	1.01 kJ/kg-C
Air-side coil effectiveness	ϵ_a	0.9
Water-side coil effectiveness	ϵ_w	0.1
Chilled water supply temperature	θ_{w3}	7 C
Pipe length	L_{pipe}	5 m
Pipe transport delay	τ_{pipe}	10 s
Coil time constant	τ_{coil}	15 s
Room Capacitance	C_{room}	5 kJ/C
Room Infiltration	$U_{infiltration}$	3.7 W/C
Window Conductance	U_{win}	3.9 W/C
Room Conductance	U_{cond}	8.0 W/C
Room Set point	θ_s	20 C
Outside temperature	θ_o	25,35 C
Solar Gain	Q_{win}	400,150 W

effectiveness increases giving closer approach to the inlet water temperature, but the waterside effectiveness decreases and the total cooling output is reduced. Table 2. shows the the air- and water-side effectivenesses, ϵ_a and ϵ_w, and the maximum cooling output as a function of air flow rate, F_a. For the analysis we shall assume that the air flow rate is $F_a = 0.05$ kg/s. Because the air flow has been greatly reduced compared to the water flow rate, the air-side effectiveness is large and the water side effectiveness is small. In the test facility the air flow was adjusted in an attempt to reduce the cooling coil output. Once a suitable condition was reached it was kept constant thereafter.

Table 2. Air- and Water- Side Effectiveness

F_a kg/s	ϵ_a	ϵ_w	Qmax W
0.04	0.94	0.083	490 W
0.05	0.90	0.100	590 W
0.06	0.85	0.11	670 W

Chilled Water Response

The steady state chilled water output temperature, θ_{w1b}, is shown in Figure 4 as a function of actuator signal, V, assuming that the room is at a temperature of $\theta_m = 20$ C, and the chilled water supply temperature, θ_{w3}, = 7 C. The output is shown for two different control valves: a linear valve and an equal percentage valve. The lower curve shows that the cooling output from a coil controlled by a linear valve changes rapidly at small actuator position. The upper curve shows the equal percentage valve which has its maximum slope at the middle of the control range.

Cooling Coil Output

The steady state cooling coil output depends on the temperature of the entering water and air and the coil effectiveness and is given by

$$Q_{coil} = \epsilon_a F_a C_a (\theta_m - \theta_{w1a}) \tag{9}$$

The cooling coil output to the room depends on the actuator signal V varies from 0 to about 500 W of cooling. For the linear control valve, to obtain a variation from 100 W to 400 W the control valve will go between 0.03 and 0.20. The gain of the control valve (the slope of the curve) becomes quite large at small flow fractions. An equal percentage valve would go from 0.33 to 0.63 to cover this same range in output. The maximum gain (slope) occurs in the middle of the range. For a system designed to cover a wide range in cooling loads, the equal percentage valve gives greater control capability. We shall first analyze the system in detail using a linear control valve. At the end we shall return to the equal percentage valve.

Steady State Room Temperature

The steady state room temperature is a balance between the cooling delivered by the coil, Q_{coil}, the solar gain through the window, Q_{win}, and the conductive heat gain, Q_{cond}. As the solar gain and outdoor temperature increase the room temperature at a constant valve position increases. In practice, the control system adjusts the control valve to maintain the room temperature at a set point, $\theta_s = 20$ C. To maintain a temperature of 20 C when the solar gain is small, 150 W, the linear control valve is almost completely closed, $V = 0.05\ V_o$, while the equal percentage valve under the same conditions is at mid-range, $V = 0.43\ V_o$, giving better control.

STEP RESPONSE

To examine the time behavior of the system we examine the response of the supply water temperature to a step change in the valve flow fraction, x, the response of room air temperature to a change in supply water temperature, and finally the combined response of both systems. These responses will be examined by isolating portions of the TUTSIM simulation, applying a step change, and observing the results.

Chilled Water Temperature

The chilled water temperature depends on the supply and return flow rates and temperatures, the room air temperature, the transport delays in piping, the time constant response of the cooling coil, and the flow fraction of the three way mixing valve. Figure 5 shows the time response of the coil supply water temperature to a step in the valve flow fraction in the case where the time constant for the cooling coil is $\tau_{coil} = 0$. At t = 0 s the flow fraction changes from 0.1 to 0.2 or 0.6. There is a 10 s delay before this change appears at the inlet to the cooling coil. There is an additional 10 s delay before the cooler water returns from the coil to the mixing valve. There is another 10 s delay before the new chilled water reaches the coil inlet. Thus the chilled water changes by incremental steps of 20 seconds duration as the slug of fluid circulates. When the thermal mass of the cooling coil, $\tau_{coil} = 15$ s, is included in the analysis the steps are smoothed out.

For a linear control valve the effective time response depends on the fraction of valve flow. Borresen [5] has analyzed the problem and found that the effective time constant for a cooling coil in a three way valve configuration is given by $\tau_{eff} = 2 \cdot K \cdot \tau_{pipe}$ where

$$K_{eff} = -1/\ln[(1-\epsilon_w)\cdot(1-q)]$$

where $q = (F_w/f_w)\cdot x$. For $\epsilon_w = 0.1$, the effective time constant is $\tau_{eff} = 75$ s for x = 0.2 and 28 s for x = 0.6, respectively. This is in general agreement with the observed decay times from Figure 5 of 82 s and 27 s respectively.

Room Air Temperature

When the coil inlet water temperature changes suddenly, the response of the room air temperature is dominated by the time constant of the room air. The coil response is very short. Figure 6 shows the response of the room temperature, θ_m as the coil inlet water temperature θ_{wla} changes from 19 C to 8 C. The time constant can determined from the maximum slope of Figure 6 is about 95 s. The cooling coil outlet is a function of the room temperature as given by equation (9). The room temperature is driven toward equilibrium as given by

$$C_{room} \frac{d\theta_m}{dt} = Q_{coil} + Q_{cond} + Q_{win} = \epsilon_a F_a C_a \cdot (\theta_{wla} - \theta_{al}) + U_{cond} \cdot (\theta_o - \theta_{al}) + Q_{win}$$

The time constant to reach equilibrium is then given by

$$\tau_{room}' = \frac{C_{room}}{(\epsilon_a F_a C_a + U_{cond})} = 93 \text{ s}$$

which is close to the time constant observed in Figure 6.

Combined Response

The combined response of the valve mixing, transport delays, coil response, and room response when the valve flow fraction, x, goes from 0.1 to 0.2 is shown in Figure 7. The slow response of the cooling coil inlet water conditions at small x, $\tau_{eff} = 82$ s, combines with the response of room temperature to changes in coil water temperature, τ_{room}' = 90 s, to give a long decay time, $\tau' = 320$ s.

CONTROL SIMULATION

The control block diagram drawing including the controller and feedback is shown in Figure 2. The controller compares the measured variable, the room temperature, θ_m, with the set point or reference temperature, θ_s, and provides an output voltage, V(s), to the valve actuator. $G_{control}$ represents the control block which compares the observed room temperature θ_m with the set point temperature, θ_s and generates a control signal, V(s). $G_{control}(s)$ is controller transfer function. The control output is given by

$$V(s) = G_{control}(s) \cdot (\theta_s(s) - \theta_o(s))$$

Proportional Control

When proportional feedback control is used, $G_{control}(s) = K_p$, where K_p is the proportional gain. If the $K_p = 10/C$, then the actuator valve signal would change by 10 units for every degree change from the set point temperature. Since the range of the control actuator is V from 0 to 10, $V_{range} = 10$, the throttling range of the controller is given by

$$TR = V_{range}/K_p = 10/(10/C) = 1 \text{ C}$$

The response of the system with a linear control valve under high solar gain, 400 W, to change in zone temperature set point is shown in Figure 8. The system is stable and the initial swing in room temperature is damped within a few cycles. Because of the rather long response of the room air temperature, as shown previously in Figure 7, the period of the oscillations is about 200 s. Because proportional control is used there is an offset from the set point in the final room temperature of about 0.2 C. If the solar gain is reduced to 150 W the system is close to marginal stability, as shown in Figure 9. As the control valve cycles from closed to about 20% open, the coil inlet temperature and room temperature also cycle with a period of about 140 seconds. The system will not settle down. At small valve openings the gain of the cooling coil system is quite large, that is the change in coil output for a small change in valve output, the slope of Figure 5, is large. Qualitatively this type of oscillation has been observed in the MoWiTT facility during initial operation and is quite unacceptable for detailed calorimetric experiments.

Proportional plus Integral (PI) Control

The control response can be made more stable by reducing the control gain, K_p to perhaps 3/C. This increases the throttling range of the controller to TR = 3.3 C. This increases stability, but it also gives rise to an increased control offset required to maintain the control signal. The control offset can be eliminated by the use of proportional plus integral (PI) control.

When proportional plus integral feedback control is used, the transfer function for the controller is given by

$$G_{control}(s) = K_p\left(1 + \frac{1}{T_i \cdot s}\right) ,$$

where K_p is the proportional gain and T_i is the integral time constant. The integral term accumulates a signal that slowly resets the control set point to eliminate offset.

When the control gain has been reduced to $K_p = 3$ /C, on the system with a linear control valve under low solar gain, 150 W, the response to an upset is stable. The integral term effectively eliminates the offset. Because of the smaller control gain the control valve cycles does not go through such large swings and the system settles down, as shown in Figure 10. Again the smaller gain reduces the swings in the valve flow fraction giving more stable operation.

IMPROVED CONTROL OPTIONS

To demonstrate the usefulness of control simulation we have evaluated two modifications to the system that can make the system easier to control. The main problem with the linear valve system is that the process gain of the cooling coil is very large at small valve openings. Application of an equal percentage valve or introduction of resistive heating at small valve flow fractions can greatly improve the control stability and controllability.

Equal Percentage Valve

The coiling coil inlet water temperature, θ_{wl}, as a function of actuator signal, V, has been shown previously in Figure 4, for the linear and equal percentage valve. The equal percentage valve is designed to give a uniform increase in cooling output for a given change in valve actuator signal, V. Unlike the linear valve, at small openings the rate of change of cooling output for change in actuator signal decreases. When an equal percentage valve is used, a system which is stable in the middle of the control range will remain stable at each end of the range. The proportional control response of a system with a control gain $K_p = 10/C$, and a solar gain of 150 W, is shown in Figure 11. This system with a linear valve was shown to be unstable in Figure 9. As the actuator signal, V, increases the valve flow fraction, x, increases slowly. Because the proportional gain is large the final control offset is only 0.3 C. Addition of integral gain would remove this offset. The test facility was built using electrically operated linear control valves and changing valves at this time would interfere with ongoing experiments.

Reheat

Electric resistance heating can easily be controlled in a linear manner. If heating is introduced into a system with a linear cooling valve when the cooling load is reduced, the cooling coil output will stay larger. In addition if the room temperature is too low, the heater can drive the room temperature upward much faster than heat gain from conduction. The heater input is modeled as

$$Q_{heat} = (3-V)*100 \text{ W} ; 0 < V < 3 .$$

The heater does not come on until the control signal falls below 3. This has the effect of greatly reducing the slope of the cooling coil output for small values of the control signal V. Reheat is commonly used in HVAC to establish good temperature control and has been successfully applied in the MoWiTT facility.

The response of a system with electric reheat is shown in Figure 12. The heater, Q_{heat}, comes on initially and drives the room temperature, θ_m upward. As the temperature rises the heat input decreases and steady state is quickly reached. This is the same system that is shown to be unstable in Figure 9 with a linear control valve. If the actuator signal is above 3, the heater is off, but there are significant solar gains to drive the room temperature.

CONCLUSIONS

Computer simulation is a powerful tool to solve control engineering systems. A general control simulation program, developed for control engineering analysis on a microcomputer, has been used to simulate the control of discharge air temperature of a cooling system consisting of a valve, pipe and a heat exchanger with multiple time constants and interacting components. The analysis has been applied to better understand the control stability and tuning of a supply air cooling system used in the Mobile Windows Thermal Test (MoWiTT) Facility.

Microcomputer simulation is a powerful tool for modeling complex discharge air and room temperature control systems. A detailed system model can be built out of standard components and can be used to plot the steady state performance under different operating conditions. The time constants for system response to a step change in input can be observed. The response of the cooling coil output depends strongly on the transport delays in agreement with Borresen. The response of the room temperature to changes in coil inlet temperature depends on the room decay time.

The feedback control of the system has been modeled and stability of the system to various control parameters has been investigated. The model predicts a control offset for proportional control and increasing instability as the control gain is increased. Application of integral gain is shown to remove the control offset. For a system with a linear control valve, the process gain of the cooling coil increases rapidly as the valve flow fraction decreases. A system which is stable at mid-range can go unstable at the low end of the control range. These qualitative results assist in understanding the control of the facility.

Different modifications to system control have been modeled. The application of both an equal percentage valve and reheat are shown to improve the control response of the system under low cooling load conditions. The qualitative results presented here are based on estimates of system parameters. Future work should include identification of system parameters and comparison with measured system behavior.

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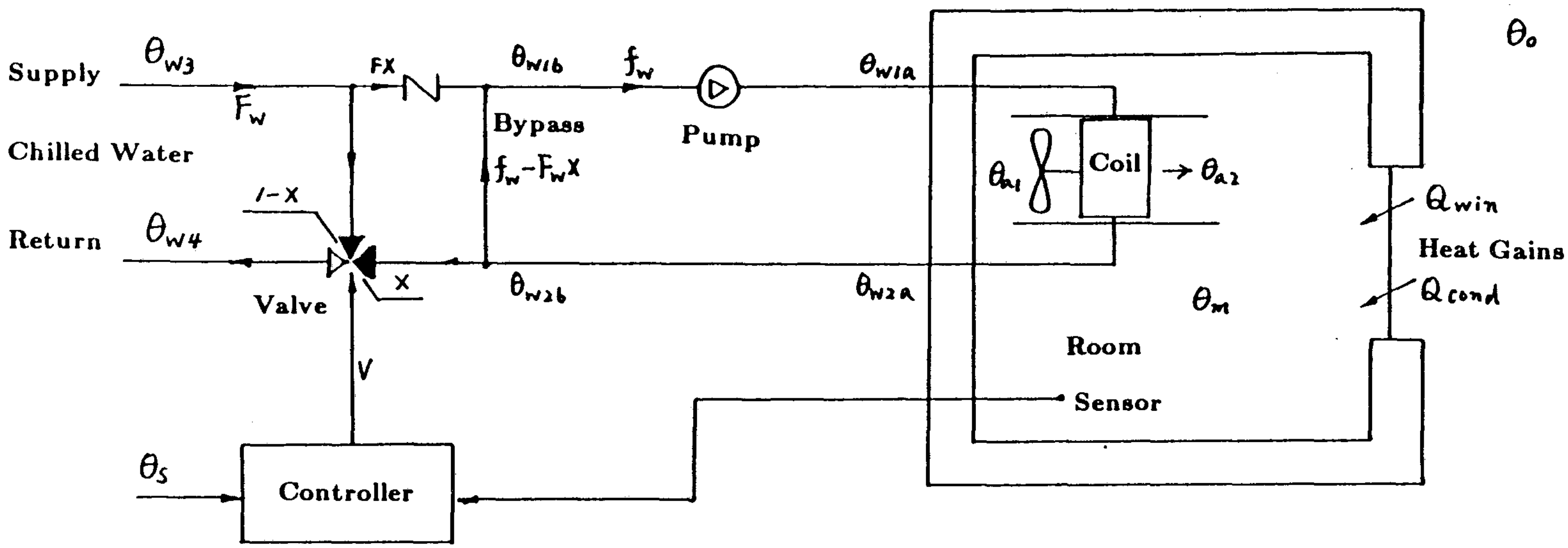
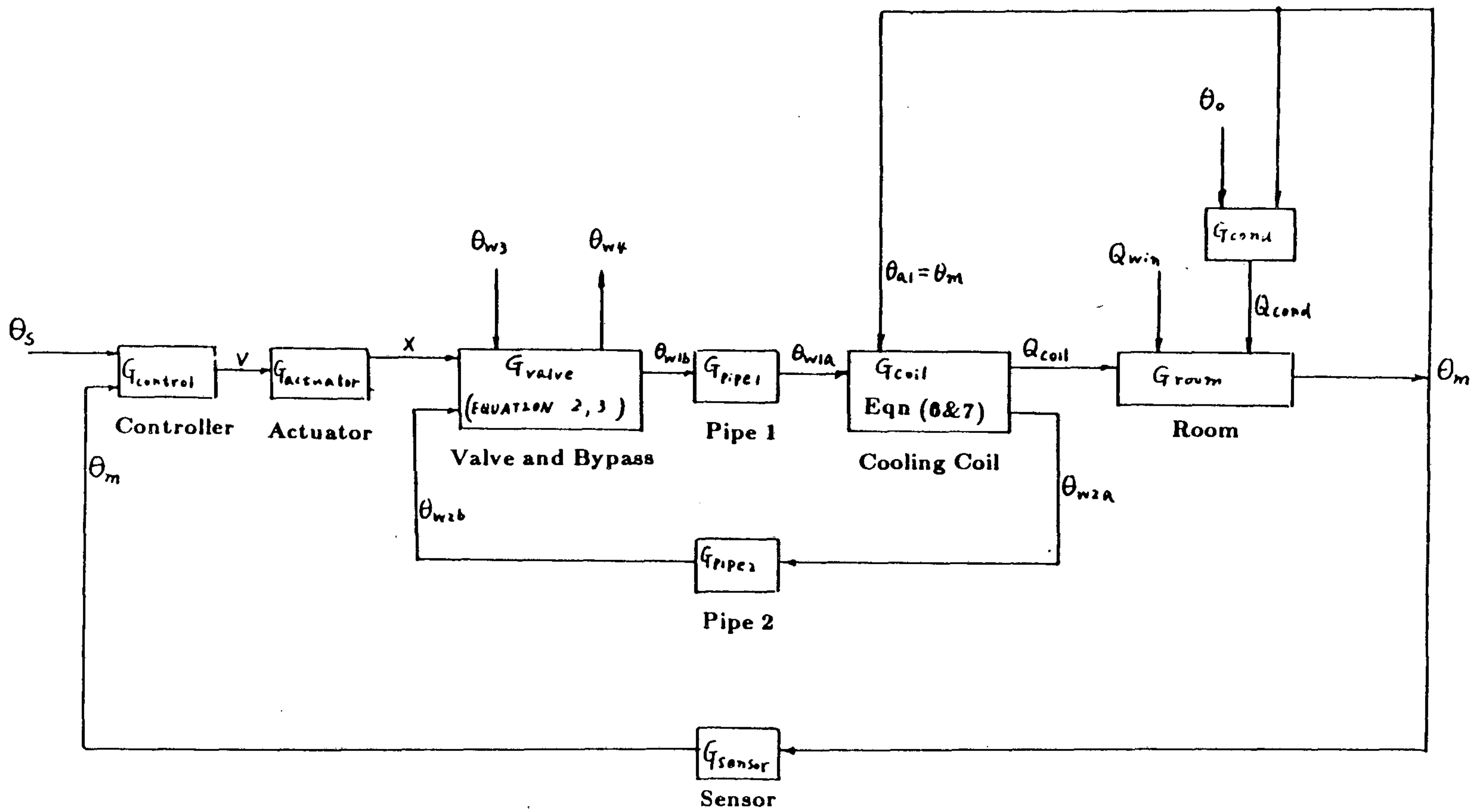


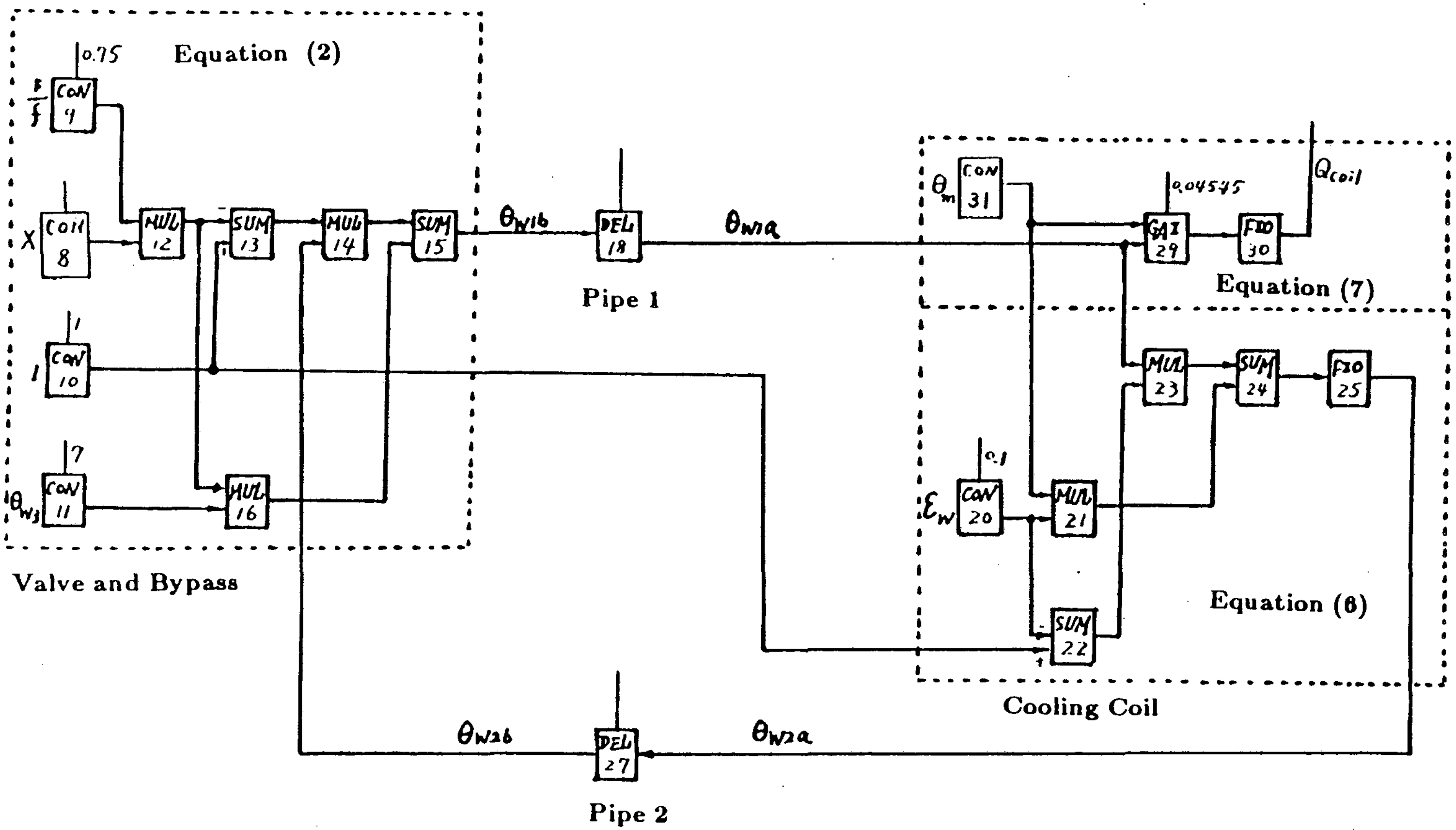
XBL 878-3567

Figure 1. Schematic of Test Facility showing measured room temperature, θ_m, room temperature set point, θ_s, chilled water temperatures, θ_w, air temperatures, θ_a, window solar gain, Q_{wind}, and conduction and infiltration gain, Q_{cond}.



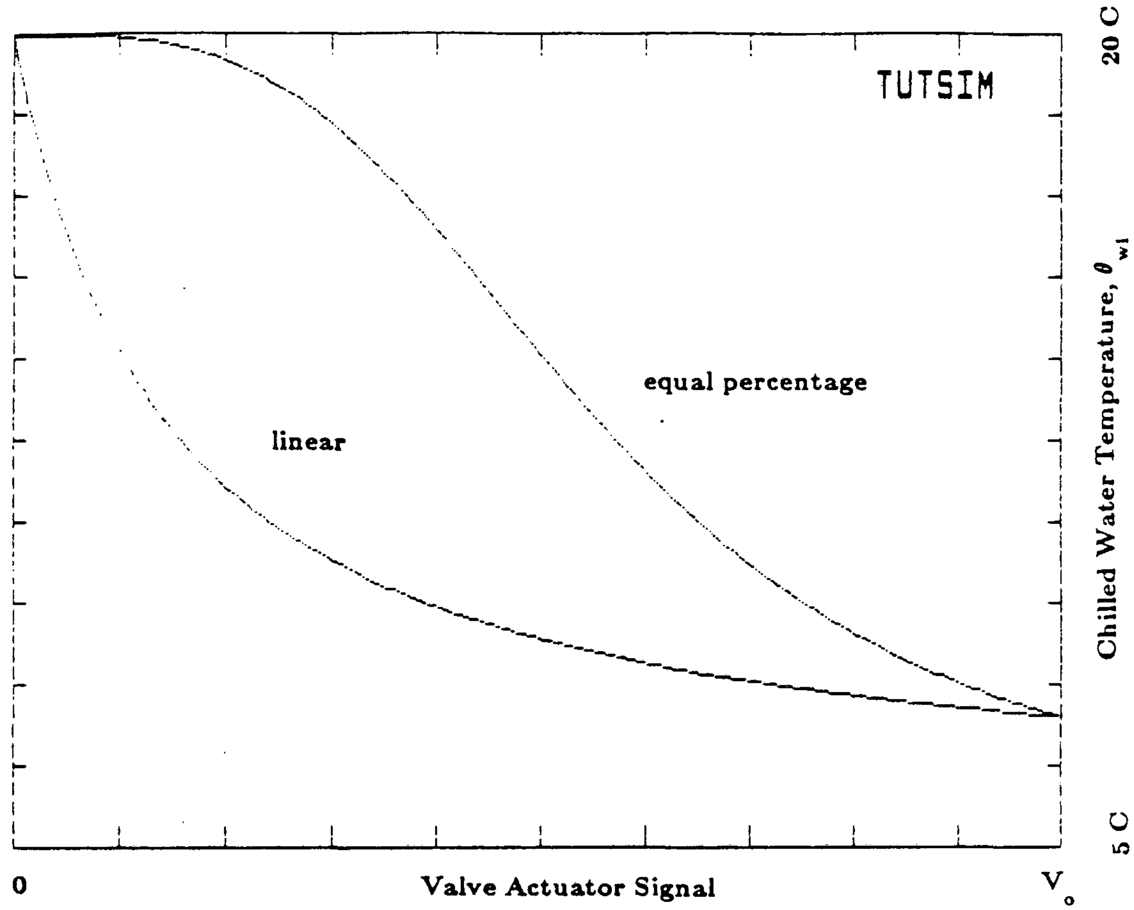
XBL 878-3568

Figure 2. Control Diagram for Test Facility Cooling System.



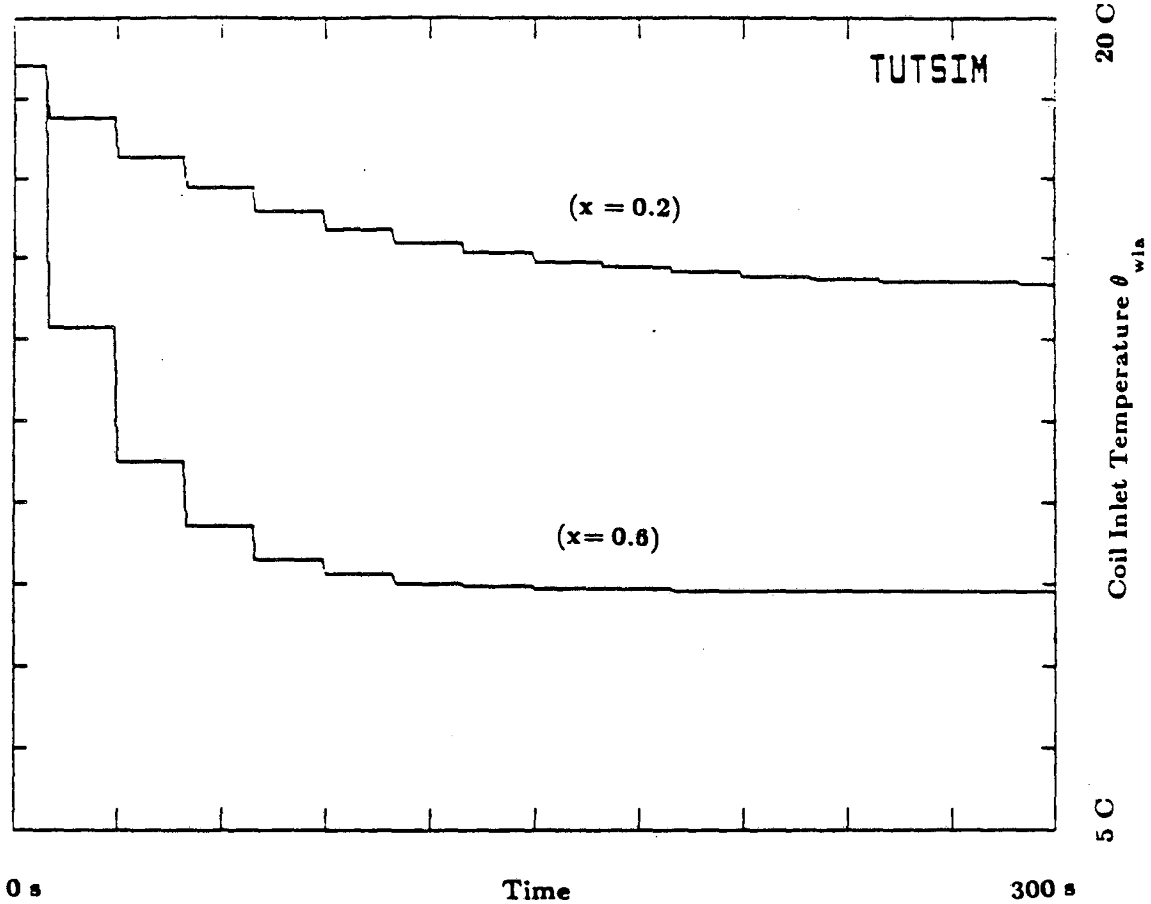
XBL 878-3569

Figure 3. TUTSIM™ simulation for open loop control of a discharge air temperature system used to evaluate the steady state and step response of the system.



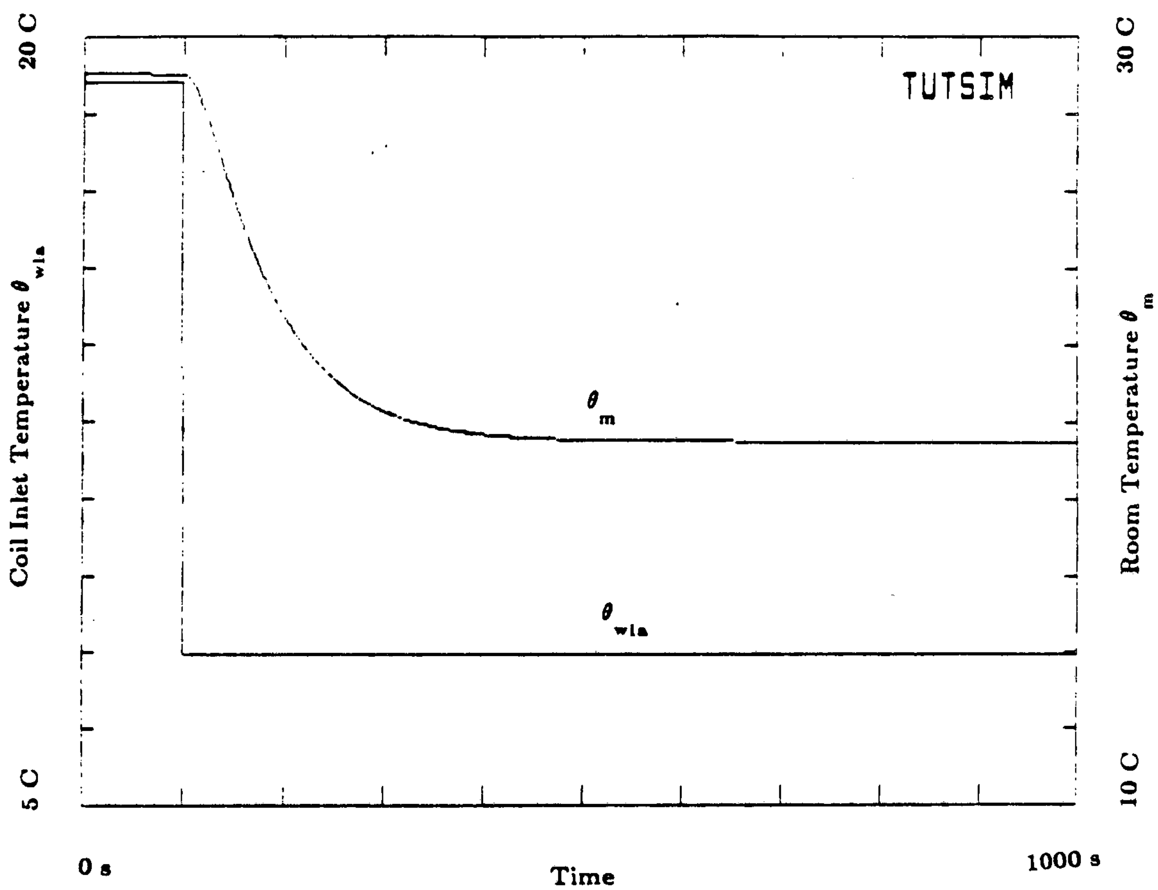
XBL 878-3570

Figure 4. Steady state chilled water output temperature to coil as a function of valve actuator signal, V, for linear and equal percentage valves assuming a supply chilled water temperature of 7 C and a constant room temperature of 20 C.



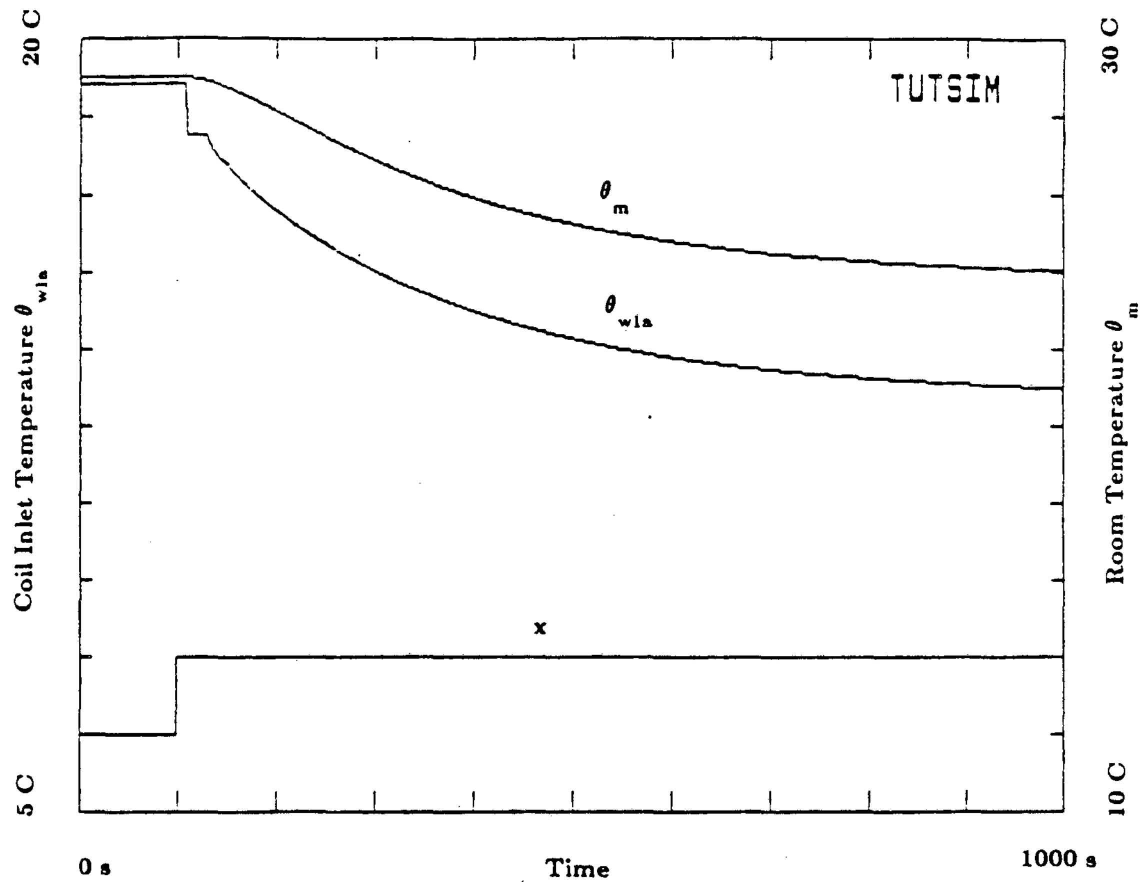
XBL 878-3571

Figure 5. Time response of the coil inlet water temperature to a step change in valve flow fraction. For t $<$ 0, x = 0.1. For t $>$ 0, x= 0.2 or 0.6. τ_{coil} = 0.



XBL 878-3572

Figure 6. Time response of the room air temperature, θ_m, to a step change in coil inlet temperature. For t < 100 s, θ_{wia} = 19 C. For t > 100 s, θ_{wia} = 8 C. τ_{coil} = 15 s.



XBL 878-3573

Figure 7. Combined response of the coil inlet temperature ,θ_{wia}, and room temperature, θ_m, when the valve flow fraction is changed from x = 0.1 to x = 0.2.

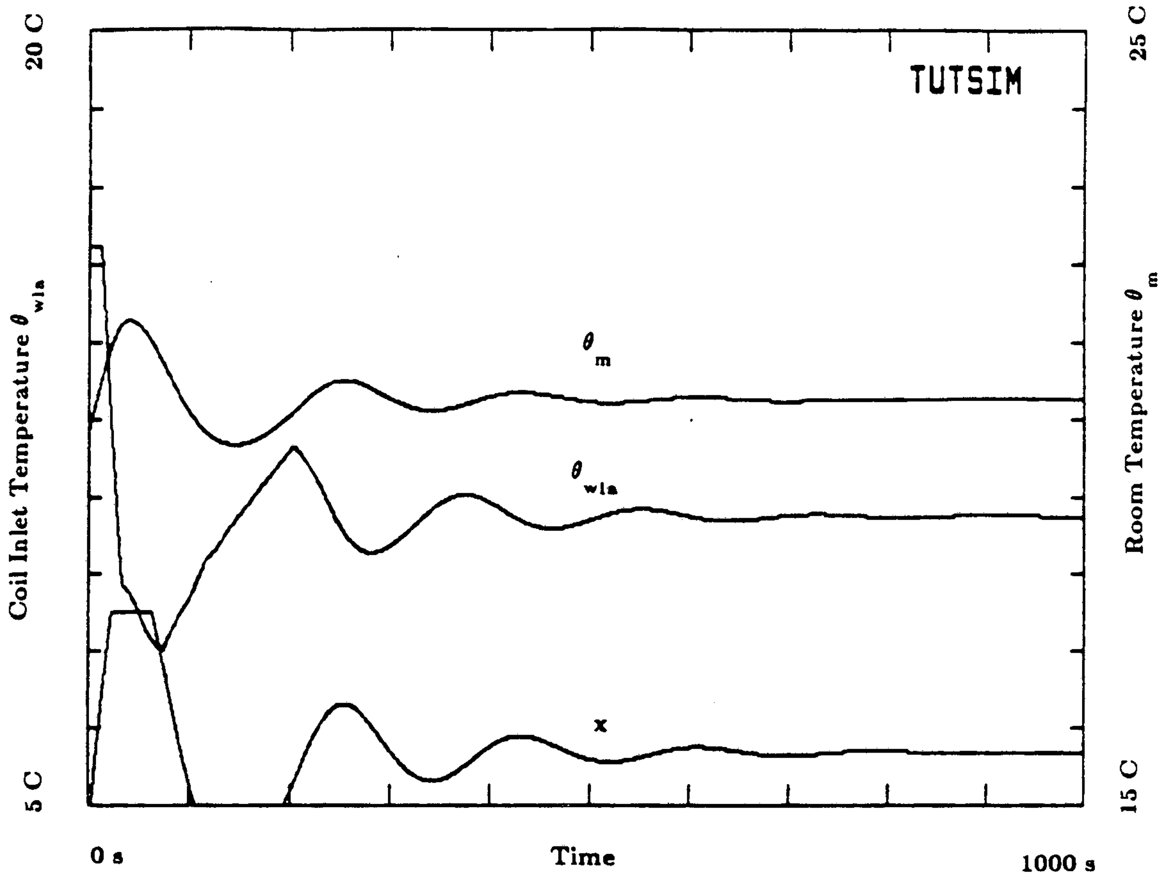


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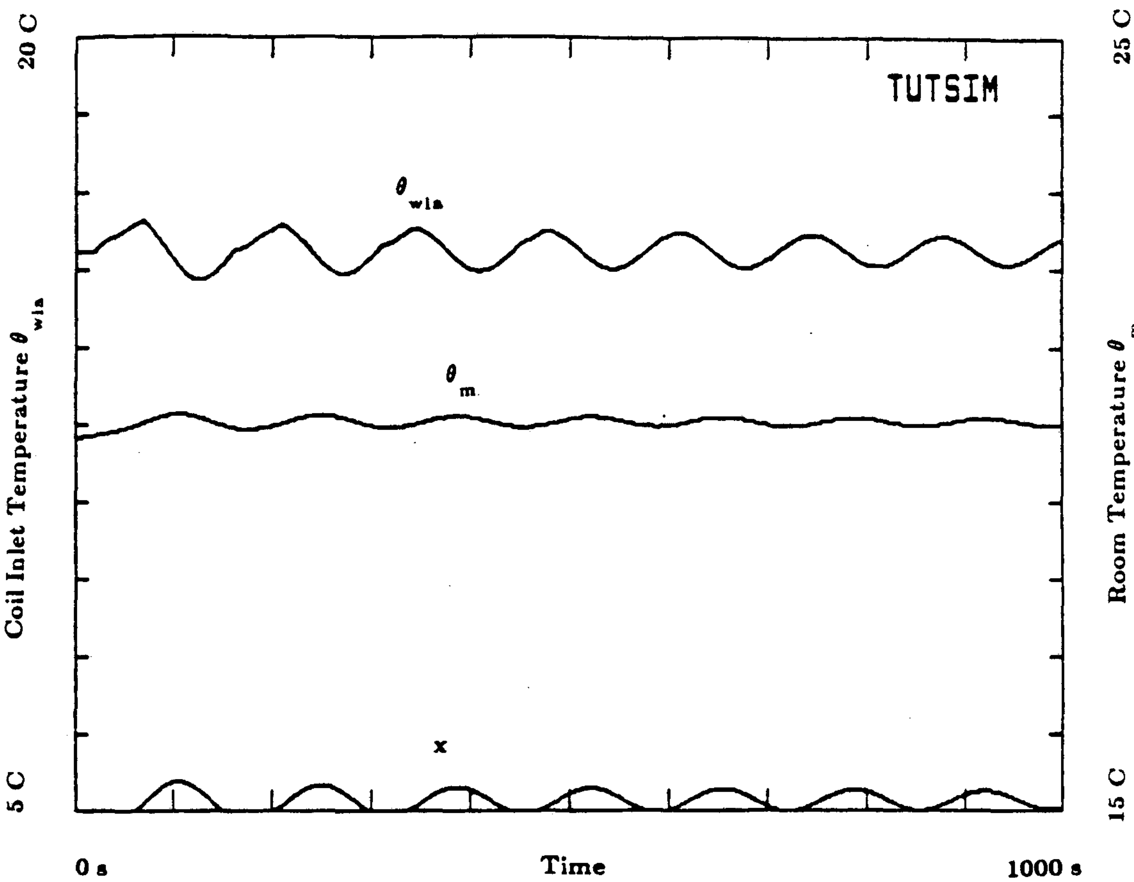
Figure 8. Proportional control showing damped response to upset. Room temperature, θ_m, coil input temperature, θ_{wia}, and valve flow fraction, x, as a function of time. $K_p = 10/C$, $Q_{win} = 400$ W, $\theta_o = 25$ C, and $\theta_s = 20$ C.



XBL 878-3575

Figure 9. Proportional control showing unstable response to upset of room temperature, θ_m, coil input temperature, θ_{wia}, and valve flow fraction, x, as a function of time. $K_p = 10/C$, $Q_{win} = 150$ W, $\theta_o = 25$ C, and $\theta_s = 20$ C.

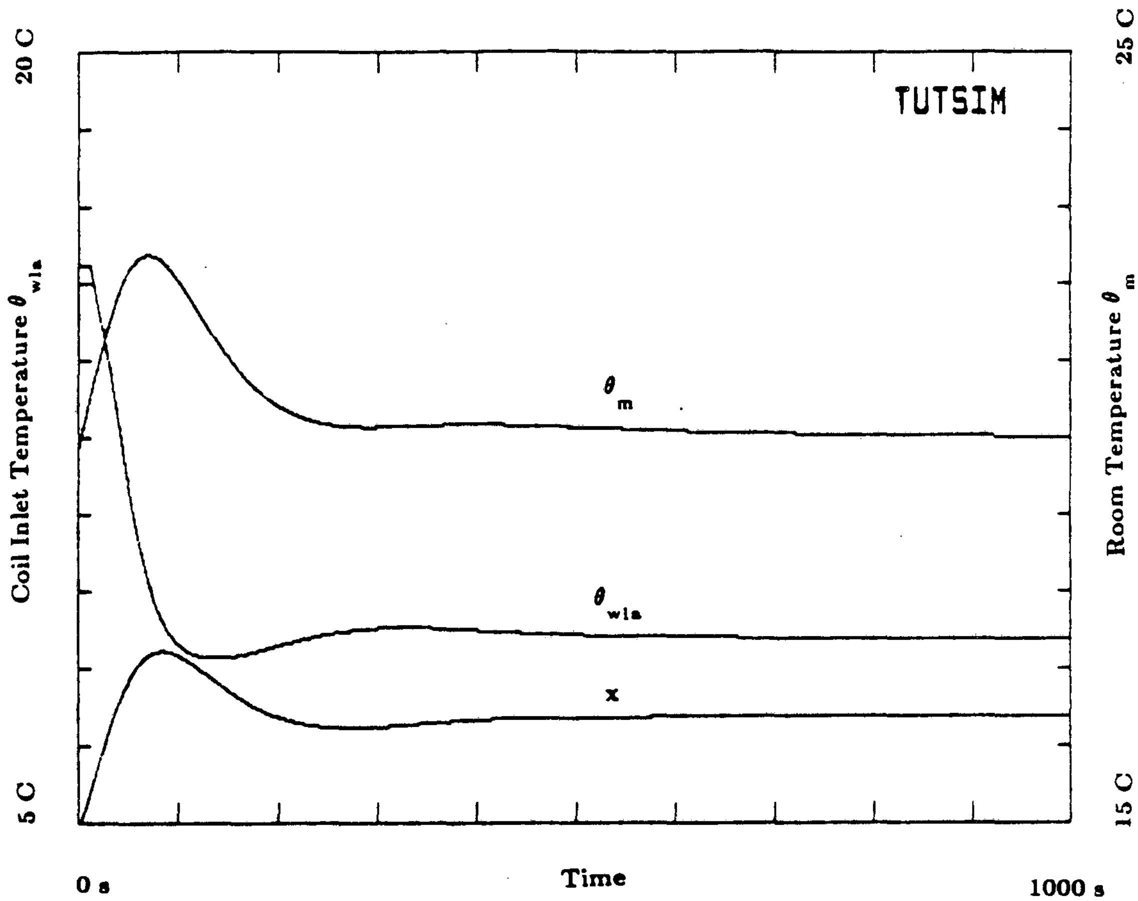


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Figure 10. Proportional-Integral control showing stable response to upset of room temperature, θ_m, coil input temperature, θ_{wia}, and valve flow fraction, x, as a function of time. $K_p = 3/C$, $T_I = 200$ s, $Q_{win} = 400$ W, $\theta_o = 25$ C, and $\theta_s = 20$ C.

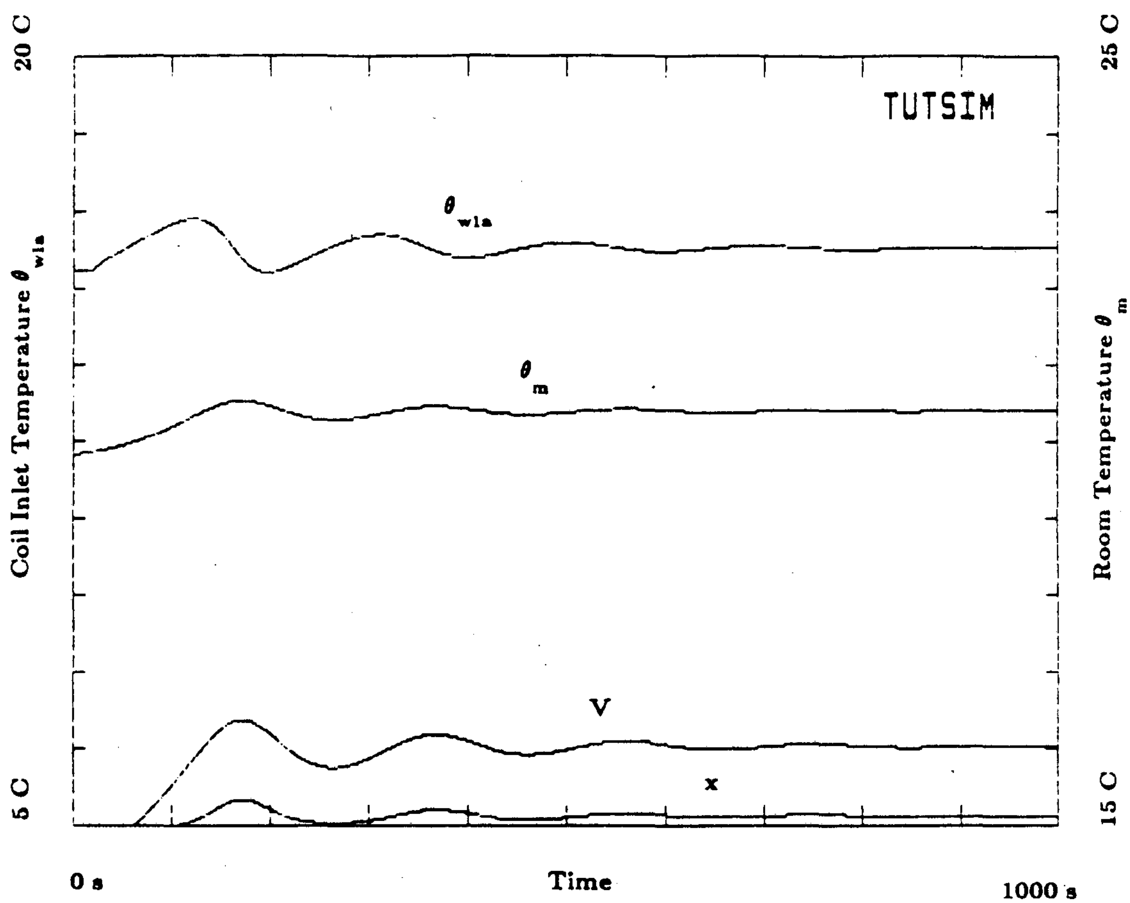
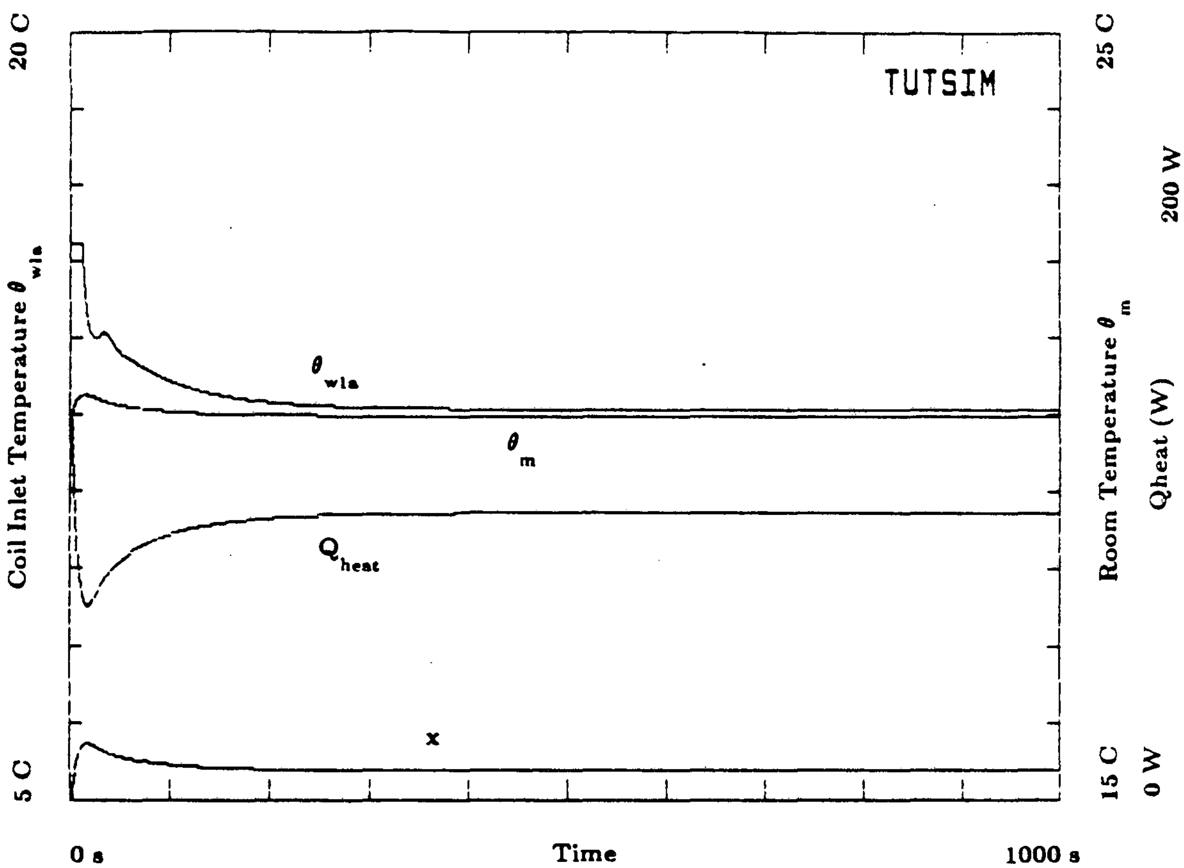


Figure 11. Proportional control with an equal percentage valve showing stable response to upset of room temperature, θ_m, coil input temperature, θ_{wia}, and valve flow fraction, x, as a function of time. $K_p = 10/C$, $Q_{win} = 150$ W, $\theta_o = 25$ C, and $\theta_s = 20$ C.



XBL 878-3578

Figure 12. Proportional-Integral control of a system with reheat showing stable response to upset of room temperature, θ_m, coil input temperature, θ_{wia}, and valve flow fraction, x, as a function of time. $K_p = 10/C$, $T_1 = 100$ s, $Q_{win} = 150$ W, $\theta_o = 25$ C, and $\theta_s = 20$ C.

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