PI CONTROL OF AIR TEMPERATURE AND HUMIDITY WITH A CHILLED WATER SYSTEM

Ghaleb M. Hussein

A Thesis
in
The Department
of
Mechanical Engineering

Presented in partial fulfillment of the Requirements
for the Degree of Master of Applied Science at
Concordia University
Montreal, Quebec, Canada

January 1996

National Library of Canada

Bibliothèque nationale du Canada

Acquisitions and Bibliographic Services Branch

Direction des acquisitions et des services bibliographiques

395 Wellington Street
Ottawa, Ontario
K1A 0N4

395, rue Wellington
Ottawa (Ontario)
K1A 0N4

Your file Votre référence

Our file Notre référence

ISBN 0-612-10861-9

Canada

ABSTRACT

PI Control of Air Temperature and Humidity with a Chilled Water System

Ghaleb M. Hussein

Accurate control of temperature and humidity is required to satisfy both thermal comfort and industrial processes requirements. Using a chilled water system, temperature and humidity control was achieved with a variable speed fan motor and variable position valves. Temperature was controlled by varying the flow rate of air through a cooling coil. Humidity was controlled by varying the flow rate of the coolant using a throttling valve, or the temperature of the coolant using a mixing valve. Multiple control strategies were identified. The control strategies were applied to systems with and without storage. Control systems were implemented using a proportional-plus-integral (PI) controller. Process sensitivities for the systems were defined and evaluated both experimentally based on open loop operating characteristics, and analytically based on the coil characteristics. Tuning the PI controller was performed by relating the desired response characteristics to process sensitivities and sampling intervals. The tuning method was developed from a thermo-fluid engineering perspective, rather than from a control engineering perspective. A method for windup prevention, to be used with digital PI controllers, was developed. Control strategies were used to implement comfort control techniques using a variable set point. The control strategies, with chilled water system, could be extended to multizone heating, ventilating, and air conditioning (HVAC) systems.

ACKNOWLEDGMENTS

The author wishes to express his gratitude and appreciation to his thesis supervisor, Prof. K. Krakow for his valuable guidance and support throughout the research. The author is indebted to Dr. S. Lin for his support.

The financial support of the National Research Council of Canada is also acknowledged.

Finally, the author is grateful to his family and friends for encouragement throughout his university years.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

A	Amplitude ratio
a	Attenuation factor
b	Constant defined by Equations 4.90 and 4.104
C	Heat capacity (BTU/ °F)
CFM	Cubic feet per minute
CRR	Recirculation ratio
c	Specific heat (BTU/lbm · °R)
D	Capacity sensitivity (BTU/h · %signal)
E	Error (%)
f(t)	Function of time
g	Generic load function
h	Enthalpy (BTU/lbm)
K_I	Controller integral coefficient
K_P	Controller proportional coefficient
l	Generic load function
M	Mass (lbm)
m	Mass flow rate (lbm/min)
P	Process variable (units of measured property)
PPM	Pound per minute

Q	Heat transfer rate (BTU/h)
S	Control signal (%)
S_o	Control signal at zero error (%)
SG	Signal variable
T	Temperature (°F)
t	Time (sec.)
U	Internal energy (BTU)
V	Volume (ft^3)
V	Volume flow rate (ft^3/ min)
v	Specific volume (ft^3 / lbm)
Γ	Response time characteristic (sec.)
Δt	Sampling interval (sec.)
σ	Process sensitivity (units presented in Table 6.1)
ϵ	Attenuation exponent
τ	Period (sec.)
δ	Logaritmic decrement
ξ	Damping ratio
ϕ	Relative humidity
ω	Specific Humidity (lbm_{WV}/lbm_{DA})

Subscripts

CR	Critical

CW	Chilled water
CWC	Chilled water coil
DA	Dry air
FA	Fresh air
F.O	First order
IN	Inlet
L	Latent
LD	Load
MA	Moist air
MIX	Mixed
MAX	Maximum
MIN	Minimum
N	Normalized
o	Reference
S	Sensible
SPC	Space
SP	Set point
S.O	Second order
SUP	Supply
SYS	System
T	Temperature
TR	Transmitted

V	Volume
WV	Water vapor
ω	Humidity

CHAPTER 1

INTRODUCTION

Simultaneous control of temperature and humidity in environmental spaces is required to satisfy thermal comfort and industrial process requirements. Advances in technology and the conditions of industrial environments propose crucial considerations. Temperature and humidity levels affect human comfort, health, manufacturing processes, artifact storage, natural produce storage,. . . etc. Simultaneous control of temperature and humidity can be achieved using chilled water coils. Temperature can be controlled by varying the air flow rate through the coil. Humidity can be controlled by varying the coolant mass flow rate or temperature through the coil. Control strategies can be implemented using a proportional-plus-integral controller. Chilled water systems can be extended to multizone systems.

1.1 The Need for Simultaneous Control of Temperature and Humidity

Temperature and humidity affect the rate of sweat evaporation. Human thermal perception depends on these two properties as well as air circulation. Comfort levels require a space temperature of 72°F and relative humidity of 50%, but other factors such as activity level must also be considered [1]. Low humidity levels cause drying of skin and mucous membranes. High humidity levels prevent evaporative cooling causing thermal stress even death [2]. Human thermal comfort is acquired by combining humidity with temperature levels. Low humidity levels can be combined with higher temperature levels to maintain acceptable human comfort conditions. Higher temperature conditions may lead to energy savings [3]. Temperature and humidity control is crucial in manufacturing processes that involve moisture absorbing materials like electronic manufacturing processes and environments for textile and paper processing using high speed machinery [4]. Artifact storage in museums and libraries require accurate control of temperature and humidity to prevent growth of mold and mildew. Changes in humidity and temperature levels shorten the lives of artifacts by causing cyclic stresses [5].

Achieving accurate control of temperature and relative humidity of environments requiring cooling is accomplished by varying the air flow rate and/or the chilled water flow rate, or temperature, through a cooling coil. Varying the air flow rate is performed either by varying a fan speed motor through a variable frequency inverter or by varying a damper position. Varying the chilled water flow rate, or temperature, is performed by varying a valve position. This strategy is suitable for a variable volume - variable temperature (VVVT) multizone system. To control motor speeds and positions, a proportional-plus-integral (PI) controller is required. PI controllers are widely used in HVAC control systems. The benefit from such controllers is achieved when they are properly tuned [6].

1.2 Literature Review

Current literature does not reveal research on systems using variable speed fans and variable position valves to control space temperature and humidity, simultaneously. However, some work has been done on similar but not identical heating, ventilating and air conditioning (HVAC) processes. Tuning PI controllers in HVAC systems may be performed by trial and error methods. Stoecker and Stoecker, and other sources, recommend trial and error methods [7,8]. These methods are not practical, their performance depends on highly skilled operators. One of the practical methods that assists the HVAC engineer, by eliminating trial and error techniques, is the Ziegler-Nichols method [9]. Two strategies are associated with this method. The first strategy employs an open loop test reaction curve and empirical relations to determine the controller coefficients. The second strategy is used when an open loop test cannot be performed. It consists of assuming a value of the proportional coefficient, then increase this value till hunting in the control system occurs. The critical proportional coefficient can be used to determine other coefficients. The parameters determined from reaction curves are not related to system thermal characteristics. The Ziegler-Nichols method, originated in 1942, is more reliable for analogue than for digital controllers. Tuning digital PI controllers requires the specification of controller coefficients and a sampling interval [10]. Digital controller responses do not include delays for some systems. If delays are absent, the Ziegler-Nichols method result on high coefficients. High proportional coefficients are not recommended in control systems for stability considerations. When digital controllers are used, the literature recommends the use of the same empirical relations developed by Ziegler-Nichols to tune analogue controllers. When Zeigler-Nichols method fails to determine good coefficients, the HVAC engineer or the installer is obliged to use either trial and error methods or complex conventional control theory methods.

Many other tuning methods are based on conventional control theory. Conventional control theory relies upon Laplace transforms and transfer functions. Techniques based on pole - zero cancellation, adaptive control [11,12], and Smith predictor [13], using more rigorous mathematics, are less practical. Pinella modeled every device involved in a fan pressure control system [14]. Mathematical modeling was required to determine the transfer functions of sensors, motors, ducts, . . . etc. In large HVAC systems, the determination of transfer functions of different components is not practical. It requires knowledge of components specifications.

Bekker presented a tuning method for first order processes [15]. He applied the method to control space relative humidity [16]. His method was based on root locus technique to determine the PI coefficients. The theory requires a critically damped target response. Critically damped response requires accurate system identification based on a unity damping ratio. Tests on the theory showed that an overdamped system response may result [17]. The theory requires knowledge of root locus techniques that may not be known by HVAC engineers or installers.

Krakow and al. [18] worked on simultaneous control of temperature and humidity by compressor and evaporator fan speed variation. Using a proportional, integral and derivative (PID) controller, they demonstrate accurate control of temperature and humidity with a refrigeration system. In a companion research, Krakow and al.[19] developed an analytical method to tune a PID controller for the aforementioned system. They relate the controller coefficients to target system response characteristics and system capacity. The system capacity was determined based on analytical performance of the refrigeration system. Some system performance parameters require experimentations. However, their method can be generally classified as an analytical tuning method.

One problem associated with the use of PI controllers is the integral windup

Different techniques are used for windup prevention. Current techniques are based on adjusting the feedback loop by including tracking time constants [20]. Time constants are hard to find for practical systems. Other literature suggested limiting the error signal to a controller to some limiting values [6]. The authors did not reveal what are these values or how can be found. A literature of survey did not reveal any specific method for windup prevention. It is known that the contractors use proprietary information to prevent windup. However, windup prevention is not common knowledge. If it were, it would be incorporated in all controllers.

1.3 Objectives

The main objective of the current study was to investigate the use of speed variation of a fan motor and variable valve position with chilled water systems to control temperature and specific humidity. The objectives of the investigation were to:

1) mathematically analyze the thermal systems,
2) develop control strategies based on the mathematical analysis and cooling coil characteristics,
3) develop a method for PI coefficients determination,
4) develop a windup prevention technique for use with PI controllers,
5) test the feasibility of control strategies, and
6) verify the mathematical analysis, experimentally.

CHAPTER 2

PI CONTROLS IN HVAC SYSTEMS

Actuators in HVAC applications can be modulated by either a pneumatic or an electric control signal. The signal is generated by a controller. The controller detects the error between a desired set point and the actual measured value of a controlled variable, and accordingly, it generates a control signal that modulates an actuator to eliminate the error. The actuator may be a positional motor such as a damper or valve motor, or it may be a rotational motor such as a fan or compressor motor. To satisfy the heating or cooling load requirements in HVAC systems, the control signal should be proportional to an indeterminate load. For this purpose, proportional-plus-integral (PI) controllers are the best known and most widely used control method.

PI controllers have several important functions. They provide feedback, by operating on the error between a set point and the measured value of the control variable. They can eliminate steady-state offsets, i.e., the difference between the final and desired values of the controlled variable, through the integral action. They can cope with actuators when they are functioning at maximum or minimum operating conditions termed " saturation limits." The PI algorithm is simple to be implemented. A digital PI controller is equivalent to software codes written for a computer program [20].

This chapter gives a short introduction to PI control. The basic algorithm is presented with a description of the properties of the controller based on intuitive arguments and representations of the controller. The phenomenon of windup, which occurs when an actuator saturates due to integral action, is discussed. A theory is developed to eliminate the integral windup by limiting the accumulated sum of error to that which will just saturate the controller. The limited sum of error is used in the summation formula to eliminate the windup effects. The theory will be derived, simulated numerically, and tested experimentally.

2.1 Basic PI Controller Algorithm

The PI algorithm, when implemented digitally, has the following form:

$$S = K_P \cdot E + K_I \cdot \sum_{i=1}^{n} E \tag{2.1}$$

The error E is expressed as

$$E = (P_{SP} - P) \cdot \frac{100}{\Delta P} \tag{2.2}$$

where

$$\Delta P = P_{MAX} - P_{MIN} \tag{2.3}$$

The error (E) and the control signal (S), are expressed as percentages. The control signal is limited to values between zero and 100, therefore

$$\text{if} \quad S < 0 \quad \text{then} \quad S = 0. \tag{2.4}$$

and

$$\text{if} \quad S > 100 \ \text{then} \ S = 100 \tag{2.5}$$

The control signal is thus a sum of two terms: the P-term (which is proportional to the error) and the I-term (which is proportional to the sum of the errors). The controller parameters are the proportional coefficient K_p and the integral coefficient K_I. The full benefit of the controller will be achieved when it is properly tunned, i.e., when the coefficients' values are determined accurately to eliminate the error within a reasonable time.

With proportional control, the control signal is directly proportional to the error. The controller then will provide a large corrective action when the error is large and a small corrective action when the error is small. For zero error, the controller has to maintain its previous output value to prevent zero value signals to the actuator. Proportional constant must have relatively slow response to the error signal, i.e., the proportional coefficient must be kept as low as possible. Theoretical and practical control applications show that an offset resulted whenever proportional control is used [21]. The control signal will be proportional to the load but the desired set point will not be met. Many HVAC systems, including terminal reheat and dual-duct systems, require extra heating or cooling when temperature set points are not met. To improve the system response by eliminating the offset, an integral term is introduced into Equation 2.1. The integration action, by holding a memory of all the error signals that previously occur, improves the response of the system by gradually eliminating the error.

The overall philosophy of PI control can be viewed in the following way. The integral term is included to satisfy control system requirements. The integral coefficient, K_I, insures the elimination of steady state error. The proportional coefficient, K_p, is then selected to achieve a stable closed-loop system.

An additional term, a derivative term, may be included in the control algorithm. This term contributes a control signal component that is proportional to the rate of change of the error. It acts as an accelerator and a brake, speeding and slowing the response of the system to disturbances. For systems experiencing electrical noise, a derivative controller may result on detrimental response. Proportional plus integral only control has been found to be satisfactory for most HVAC applications. The derivative mode was not implemented in the current study and will be excluded from further discussions.

The typical HVAC controller described by equation 2.1 will operate by performing functions sequentially and, therefore, transmit control signals at discrete time intervals. Such an interval is called the controller sampling interval. Within a sampling interval, the digital controller samples the process variable, calculates the error, and outputs the appropriate control signal that will modulate an actuator to reduce the error and eventually satisfy load requirements.

In Equation 2.1, the control signal modulates either motor position as for dampers and valves, or motor speed as for fans and compressors. Electronic hardware will convert the controller signal to a proportional voltage or current, which in turn will be transmitted to the actuator. A sample-and-hold circuit will maintain this control voltage or current until a new control output is calculated.

The implementation of the basic PI algorithm requires Equations 2.1 through 2.5. The value of the process variable is transformed by an analog-to-digital (A-D) converter. A digital-to-analogue (D-A) converter sends the control signal to the actuator.

2.2 Windup Prevention Algorithm

A controller using a PI algorithm may accumulate error in the summation term even though the actuator being modulated has reached its saturation limit. Saturation limits of an actuator refer to its operation either at minimum or at maximum levels. All actuators have limitations: a motor has limited speed, a valve cannot be more than fully open or fully closed. When a control system operates over a wide range of operating conditions, the control signal may reach the actuator limits. When this occurs, the control signal remains at one of its limits (0% or 100% control signal) for an extended time, the error in Equation 2.1 will continue to accumulate and may become very large, i.e., "winds up." It is then required that the error change sign for a long period before returning to normal operating conditions. The consequence is that any controller with integral action may give large transients and sluggish response when the actuator saturates.

HVAC systems are frequently designed such that the system capacity would not exceed the load 100% of the operating time therefore the systems may operate at maximum capacity (100% control signal) for extended periods. Systems may operate at maximum capacity at start-up. HVAC systems may also be operated at minimum capacity (0% control signal), rather than be shut down, for extended periods. It is therefore necessary that the controller is limited with subsequent control actions to correct the windup of the integral term.

One common method in use, to prevent windup problems of PI controllers, is by manual reset of the control system. The procedure tends to reset the sum of errors to its starting values by manually switching the controller off/on. This procedure requires continuous supervision of the system. Therefore, it is not convenient to HVAC applications subjected to unexpected load requirements. Other methods suggest the improvement of the feedback control loop by including terms, such as a tracking time constant, in the control loop [20]. The tracking time constant is defined as the time that determines how quickly the integral term is reset. The method requires detailed knowledge of control systems theory. The method is not practical because the HVAC system engineer or installer would not have such knowledge. It is not easy to determine the tracking time or any other time constants, for a real HVAC system. Other literatures refer to limiting the error to certain fixed values but do not reveal these values or how they are determined [6]. Current literature does not provide practical techniques to solve windup problems. A limitation procedure for the summation of error will be derived based on the conventional PI control algorithm. It is evident from Equation 2.1 that for

$$E \approx 0 \tag{2.6}$$

the control signal is related to the error summation by

$$S_0 \approx K_I \cdot \sum_{i=1}^{n} E. \tag{2.7}$$

Whereas

$$0 \leq S_0 \leq 100 \tag{2.8}$$

therefore

$$\text{if} \quad K_I > 0 \quad \text{then} \quad 0 \le \sum_{i=1}^{n} E \le \frac{100}{K_I} \tag{2.9}$$

and

$$\text{if} \quad K_I < 0 \quad \text{then} \quad \frac{100}{K_I} \le \sum_{i=1}^{n} E \le 0. \tag{2.10}$$

The following modifications to the basic PI algorithm will prevent windup:

$$\text{if} \;\; K_I > 0, \;\; S = 0, \;\; \text{and} \;\; \sum_{i=1}^{n} E < 0 \quad \text{then} \quad \sum_{i=1}^{n} E = 0 \tag{2.11}$$

$$\text{if} \;\; K_I > 0, \;\; S = 100, \;\; \text{and} \;\; \sum_{i=1}^{n} E > \frac{100}{K_I} \quad \text{then} \quad \sum_{i=1}^{n} E = \frac{100}{K_I} \tag{2.12}$$

$$\text{if} \;\; K_I < 0, \;\; S = 0, \;\; \text{and} \;\; \sum_{i=1}^{n} E > 0 \quad \text{then} \quad \sum_{i=1}^{n} E = 0 \tag{2.13}$$

and

$$\text{if} \;\; K_I < 0, \;\; S = 100, \;\; \text{and} \;\; \sum_{i=1}^{n} E < \frac{100}{K_I} \quad \text{then} \quad \sum_{i=1}^{n} E = \frac{100}{K_I}. \tag{2.14}$$

These conditions limit the summation of error to values appropriate to a zero error condition.

2.2.1 Simulation Results

Numerical simulations of a chiller system controlled with a windup prevented PI algorithm and with a conventional PI algorithm were performed [22]. Numerical simulations assume a chilled water supply controlled at 45°F. The results are shown in Figures 2.1 and 2.2, respectively. The chiller is required to operate at maximum capacity at start-up. Figure 2.1 illustrates the desirable characteristic resulting from the windup prevention algorithm. Figure 2.2 illustrates the undesirable system response characteristic resulting from the conventional PI algorithm. The scales for the temperature and the error summation are larger for Figure 2.2 than for Figure 2.1.

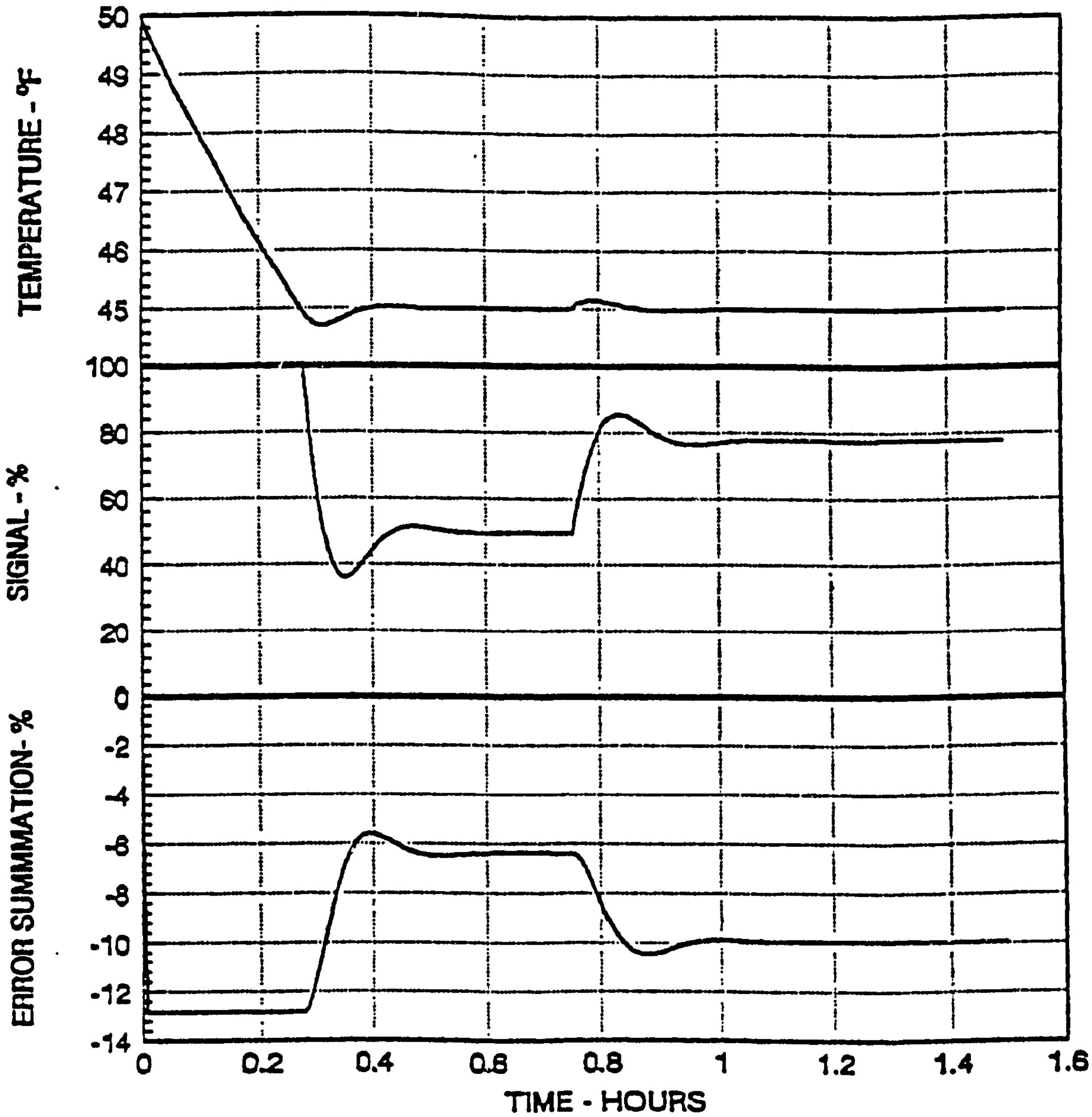

Figure 2.1. Simulated Response Characteristics of a Chiller Controlled with a PI Algorithm Modified for Windup Prevention.

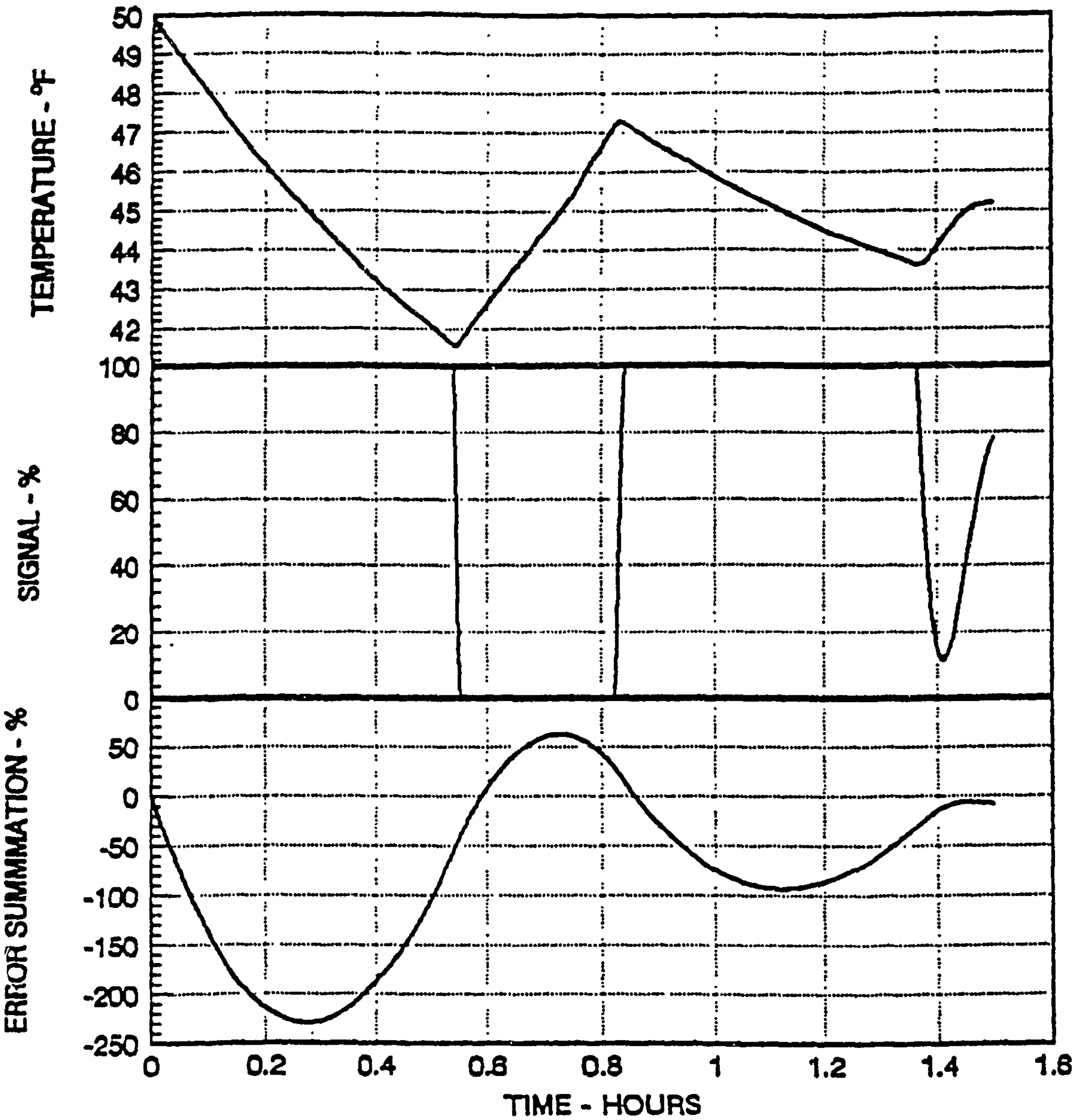

Figure 2.2. Simulated Response Characteristics of a Chiller Controlled with a Conventional PI Algorithm.

2.2.2 Experimental Testing

Experimental testing of the windup prevention algorithm was perfomed on a system similar to that shown in Figure 2.3. The controller was programmed to modulate a fan motor speed varying the air flow rate into the space. Varying the air flow rate into the room controlled the space temperature. Two tests were done. In the first test, the conventional PI algorithm was used (Figure 2.4). The proportional and the integral coefficients used were -33 and -2.9 respectively while the desired set point temperature was 72 °F. In the second test (Figure 2.5), the same room conditions exist but the windup prevention algorithm was added to the PI algorithm. The scale for the sum of error is larger for Figure 2.4 than for Figure 2.5.

The windup phenomenon is illustrated in Figure 2.4. The initial error was so large that the fan motor saturates at the high limit. The sum of error decreased initially because the error was negative, and it reached its lowest value at about one hour and seven minutes when the error went through zero. The output remained saturated at this point providing 100% signal because of the large negative value of the sum of error, and it did not leave the saturation limit until the error had been positive for a sufficiently long time to let the sum of error increases to higher levels. The net effect is a large overshoot, which is clearly noticeable in the figure. The integral windup results from the large error at the start up or from large load variations. The set point was never maintained constant even after more than four hours resulting in a sluggish system response.

Figure 2.5 shows the response when the controller with anti-windup is applied to the physical system. The same conditions, PI coefficients, and set point were used as for

Figure 2.4. Using K_I = - 2.9 (a negative value), the higher limit of the sum of error at S = 0 is calculated by Equation 2.13 as

$$\sum_{i=1}^{n} E = 0 \qquad (2.15)$$

and the lower limit at S = 100 is calculated by Equation 2.14 as

$$\sum_{i=1}^{n} E = \frac{100}{K_I} = \frac{100}{-2.9} = -34.48 \qquad (2.16)$$

Notice that at the start-up the room temperature was higher than the set point, the controller signal then reaches 100%. At this point the sum of errors decreased till it reaches the value of -34.48. The sum of error remained at this limit although the controller saturated. After about half an hour, the set point is reached and the error was zero, the sum of errors changed gradually causing the control signal to decrease according to the system requirements.

The same phenomenon occurred when the space temperature was lower than the set point. For this case the control signal remained at zero, the sum of errors increased until it reaches the higher limit of zero. At this point, there was no continuous build up of the sum of error. The sum of error changed gradually when the set point is reached. This phenomenon also occurred at about two hours time when more heat was added with a heater located in the room. The room temperature increased and the controller responded automatically without any windup effect.

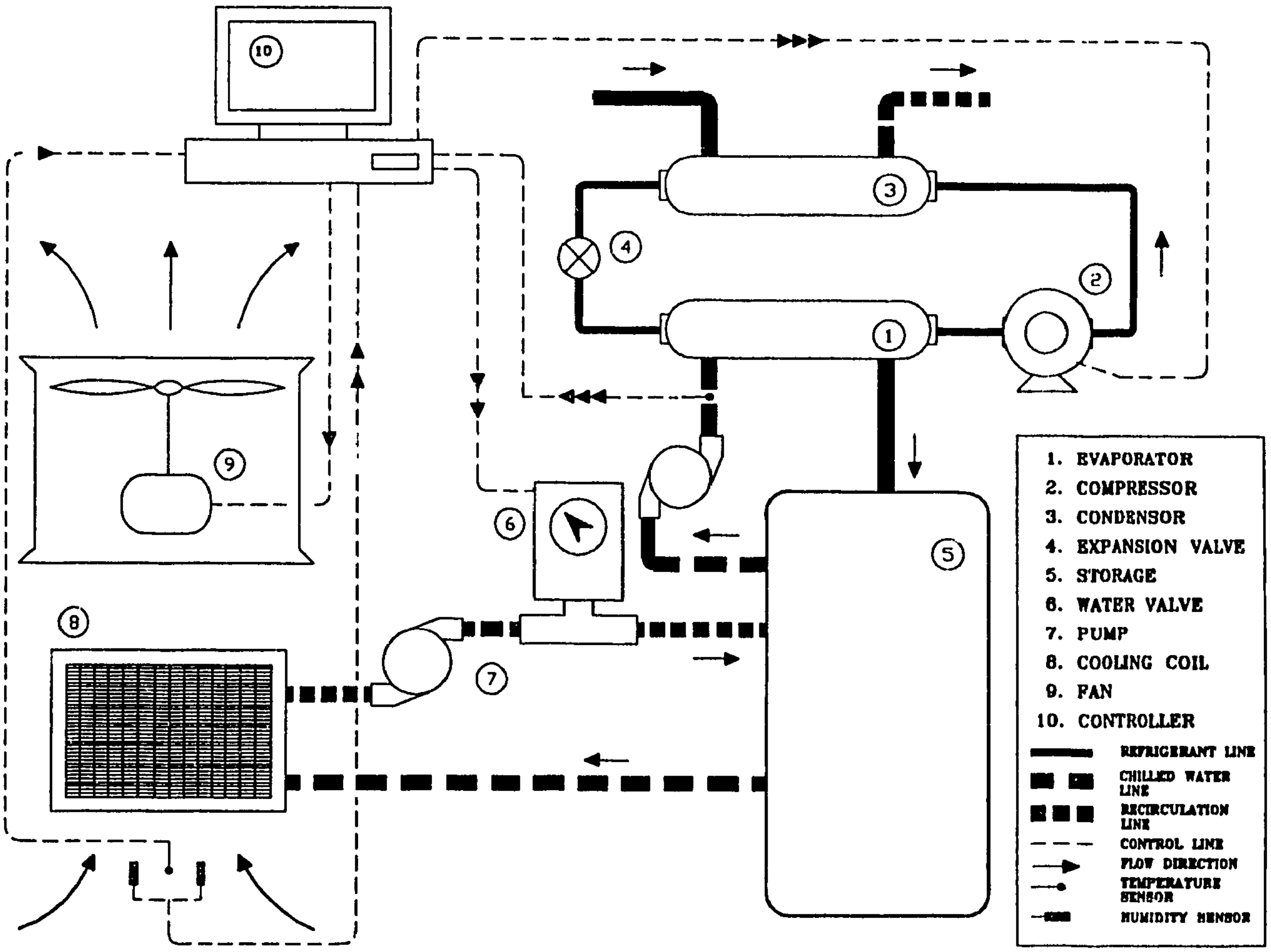

Figure 2.3 Schematic of the Experimental System used for Windup Prevention Testing.

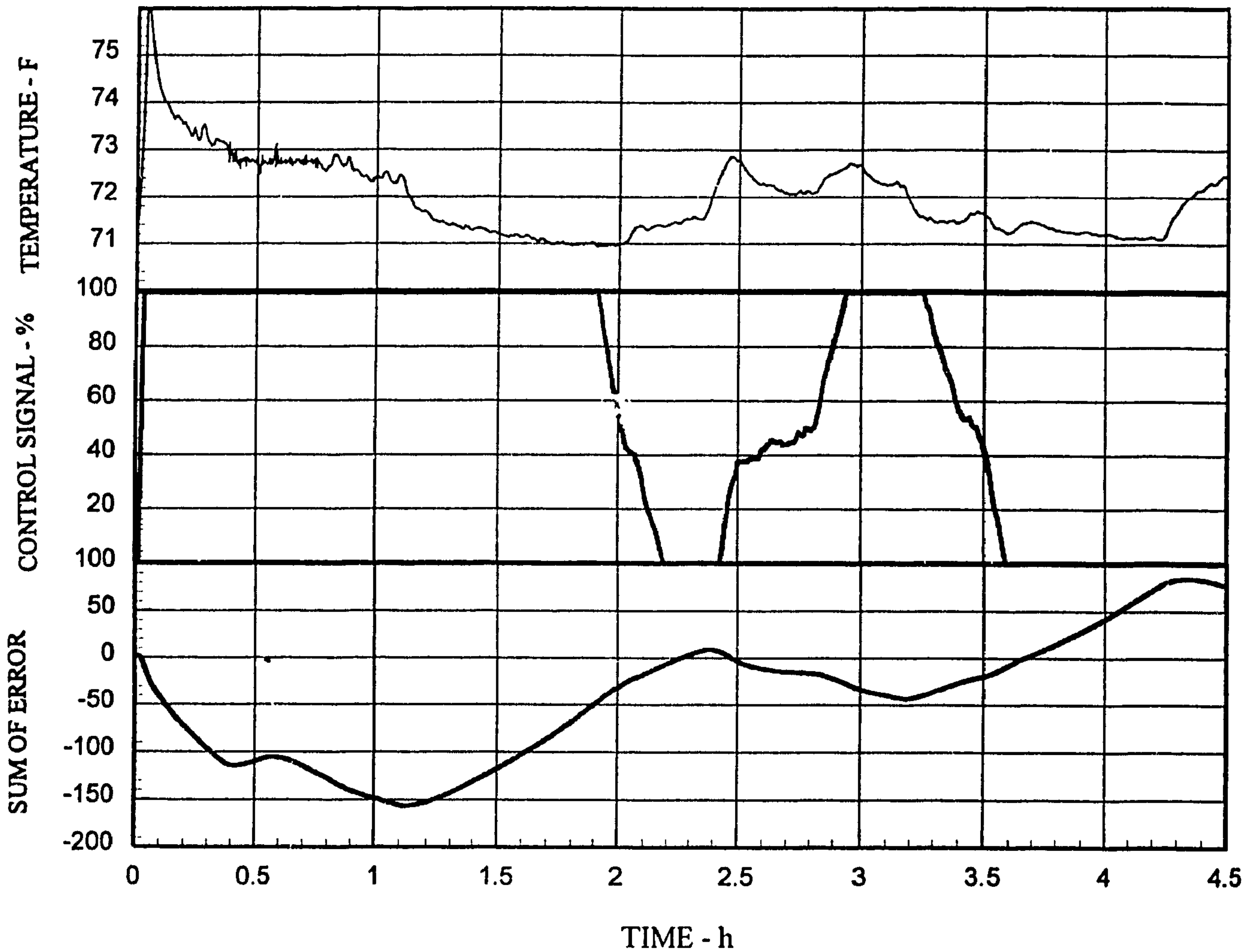

Figure 2.4. Experimental Response of A Space Temperature Controlled with a Conventional PI Algorithm.

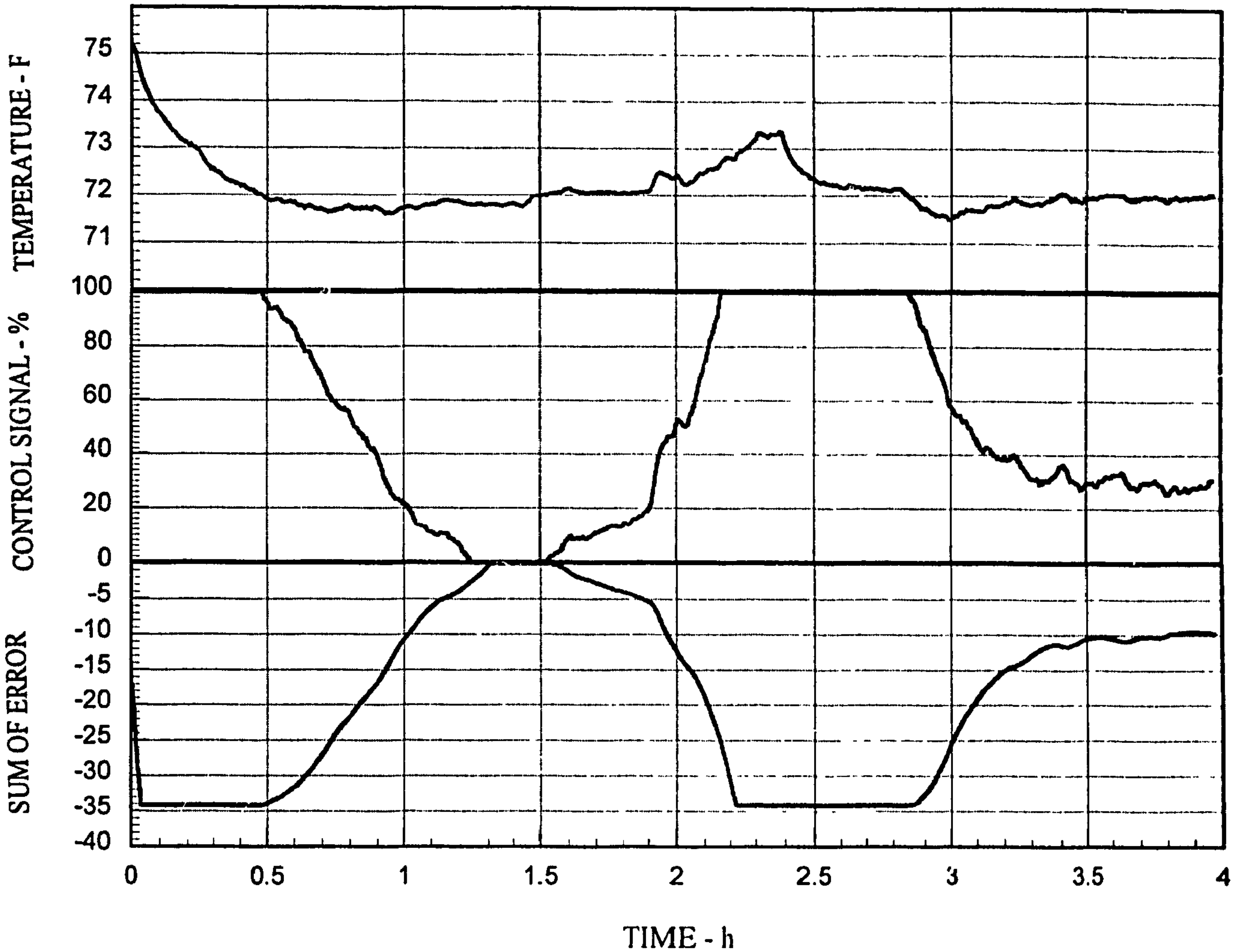

Figure 2.5. Experimental Response of A Space Temperature Controlled with a Windup Protected PI Algorithm.

2.2.3 Conclusions

The clear improvement in performance of the room temperature control system with windup prevention algorithm compared with the basic PI algorithm illustrated in Figure 2.4 leads to the following conclusions:

Windup

- It is a major problem in the control loop when PI control is used.
- It occurs at start-up conditions with large error, or when large disturbances are introduced to the system.
- It leads to sluggish system response, and loss of the desired system response.
- It may cause actuators damage

Windup prevention

- The proposed method is simple to implement
- Once the PI coefficients are determined, Equations 2.11 through 2.14 can be used to specify the higher and lower limits of the sum of error.
- Desired system response will be achieved in a shorter period.

The windup prevention technique presented and tested in the current investigation is feasible for any application that involves a PI control algorithm implemented digitally.

CHAPTER 3

THERMAL SYSTEM MODELLING

3.1 The Control Problems

Two control problems are associated with the current work. The first problem was the control of space air temperature and humidity. The second problem was the control of supply air temperature and humidity. Simultaneous control of temperature and humidity require minimum interactions between the control loops and between the control variables. Relative humidity is defined as the ratio of the density of water vapor to the saturation density of vapor at the same temperature. The specific humidity or humidity ratio is the ratio of vapor mass to dry air mass in a mixture. When space temperature varies, the relative humidity may vary while the specific humidity remains constant. The simultaneous control of air temperature and specific humidity is an alternative to control of air temperature and relative humidity.

The main purpose of an HVAC system installed in an environmental space is to control the temperature and humidity levels at standard values. Standard values are specified for human comfort, industrial processes, and energy conservation requirements. To maintain acceptable levels of temperature and humidity, a chilled water system is often used, especially for multizone systems applications. The sensible and latent components of the chilled water coil capacity must equal the sensible and latent components of the total (space and fresh air) system load, respectively. The sensible and latent components of the system load are time dependent, hence motor speeds and positional actuators used in HVAC systems must be varied with time. The signals to control the speeds and positions must be proportional to the sensible and latent space loads. These loads cannot be monitored hence they cannot be used to provide control signals. Control signals can be provided by monitoring the space temperature and humidity. Monitoring the temperature and humidity can be performed by temperature and humidity sensors located in the space.

Some industrial applications require the control of temperature and humidity of air leaving a thermal unit. This control problem was implemented by controlling the temperature and specific humidity of air leaving the cooling coil (i.e. supply air). Supply air temperature and humidity were monitored at the coil outlet.

3.2 Cooling Coil Characteristics

Cooling and dehumidifying air in a space may be performed using chilled water coils. Such coils will transfer heat to the chilled water from the air in the space. The performance of a chilled water coil depends on the coil geometry, the chilled water inlet temperature, chilled water mass flow rate, the air inlet temperature and humidity, and the air flow rate. Sensible and latent coil capacities, which depend on the aforementioned parameters, could be termed as coil characteristics.

Chilled water cooling coils are finned-tube heat exchangers that are used to cool and dehumidify air in space air-conditioning systems. Figure 3.1 shows a typical four row coil. The air enters the coil on one side, as shown, and flows through passages formed by closely spaced fin surfaces. Chilled water flows in a cross-counterflow arrangement, cooling and dehumidifying the warm air.

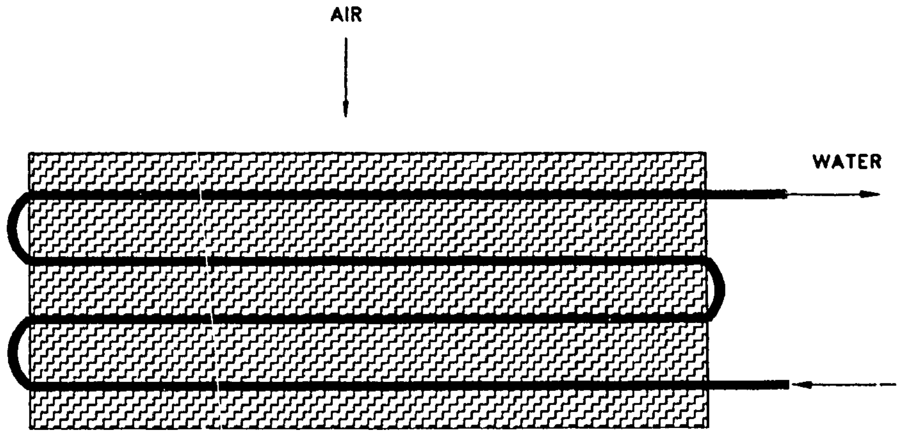

Figure 3.1. Schematic of a Four-row Chilled Water Cooling Coil.

Chilled water coil characteristics vary with the type of water valve existing at the coil inlet or outlet. Figures 3.2 and 3.3 show a coil with a throttling water valve and with a mixing water valve respectively. Space temperature and humidity are affected by the water temperature and flow rate in the coil and by the coil geometry. The heat transfer rate of a chilled water coil is proportional to the heat transfer coefficient, the area, and the air temperature to water temperature difference. Predicting coil performance requires thermal modelling of the coil. Different modelling techniques are in use [23]. For the purpose of this paper, simulated performance maps of chilled water coils were obtained from a model code appearing in an ASHRAE publication [24]. The ASHRAE model was written for pure water. The performance maps (Figures 3.4 and 3.5) indicate the sensible and latent capacity variations as functions of air and chilled water flow rates. Coil characteristics will vary with the type of chilled water valve at its inlet. When a mixing valve is used with the coil, some of the water return will be recirculated and mixed with the chilled water supply resulting on a mixed water flow going into the coil. Coil characteristics will depend on the

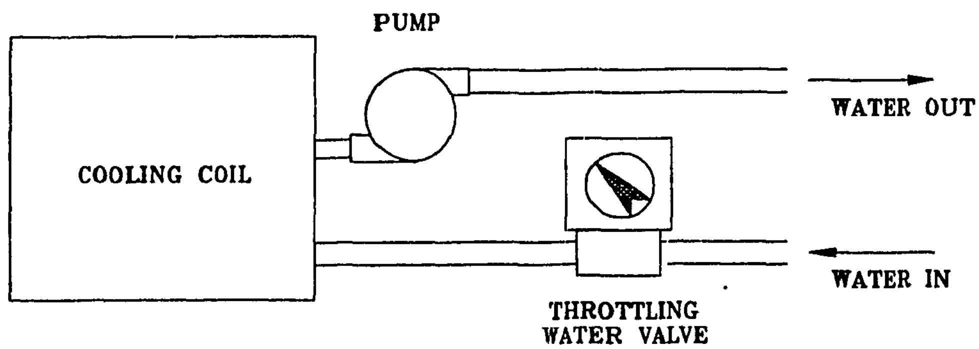

Figure 3.2. Cooling Coil with Throttling Water Valve.

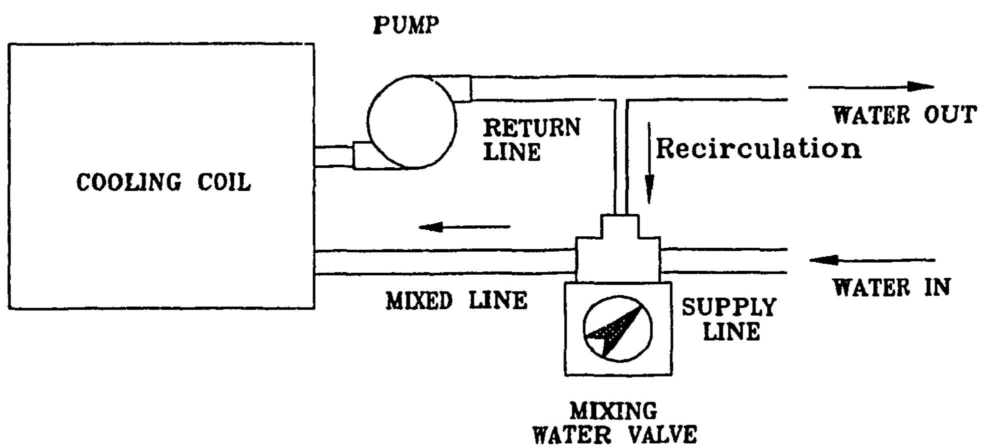

Figure 3.3. Cooling Coil with Mixing Water Valve.

recirculation ratio, CRR, defined as the ratio of the recirculated water to the supply water and it is given by

$$CRR = \frac{\dot{m}_{MIX} - \dot{m}_{SUPPLY}}{\dot{m}_{MIX}} = 1 - \frac{\dot{m}_{SUPPLY}}{\dot{m}_{MIX}} \tag{3.1}$$

The recirculation varies the effective water temperature into the coil.

The coil characteristics with a throttling water valve are shown in Figure 3.4. The coil characteristics with mixing water valve are shown in Figure 3.5. Both characteristics are derived at design operating conditions, i.e., at air entering temperature of 72 °F, air entering specific humidity of 0.009 lbm_{WV} /lbm_{DA} and entering chilled water temperature of 37 °F. Figure 3.4 shows that varying the air flow rate from 400 cfm to 800 cfm varies the sensible heat capacity by a mean of 5200 BTU/h while the latent heat capacity varies by a mean of 400 BTU/h. And varying the flow of chilled water into the coil from 25 lbm/min to 45 lbm/min, varies the latent heat capacity by a mean of 2200 BTU/h and the sensible heat capacity by a mean of 2000 BTU/h. The effect of varying the cfm is more significant on the sensible heat capacity while the effect of varying the chilled water flow rate is more significant on the latent heat capacity.

Figure 3.5 shows that varying the flow rate from 400 cfm to 800 cfm varies the sensible heat capacity by a mean of 5200 BTU/h while the latent heat capacity varies by a mean of 350 BTU/h. Varying the recirculation ratio from 0.52 to 1.0 varies the latent heat by a mean of 2000 BTU/h and the sensible heat by a similar amount of BTU/h. The effect of varying the cfm is more significant on the sensible heat capacity and the effect of varying the recirculation ratio is more significant on the latent heat capacity.

The coil performance maps (Figures 3.4 and 3.5) show that for a constant water flow rate line, and constant latent heat capacity, two values of the sensible capacity may exist. The effect of varying the water flow rate is less significant on the sensible capacity. Therefore, for this type of coils, the mass flow rate of chilled water cannot be used to control temperature. The mass flow rate of chilled water can be used to control humidity.

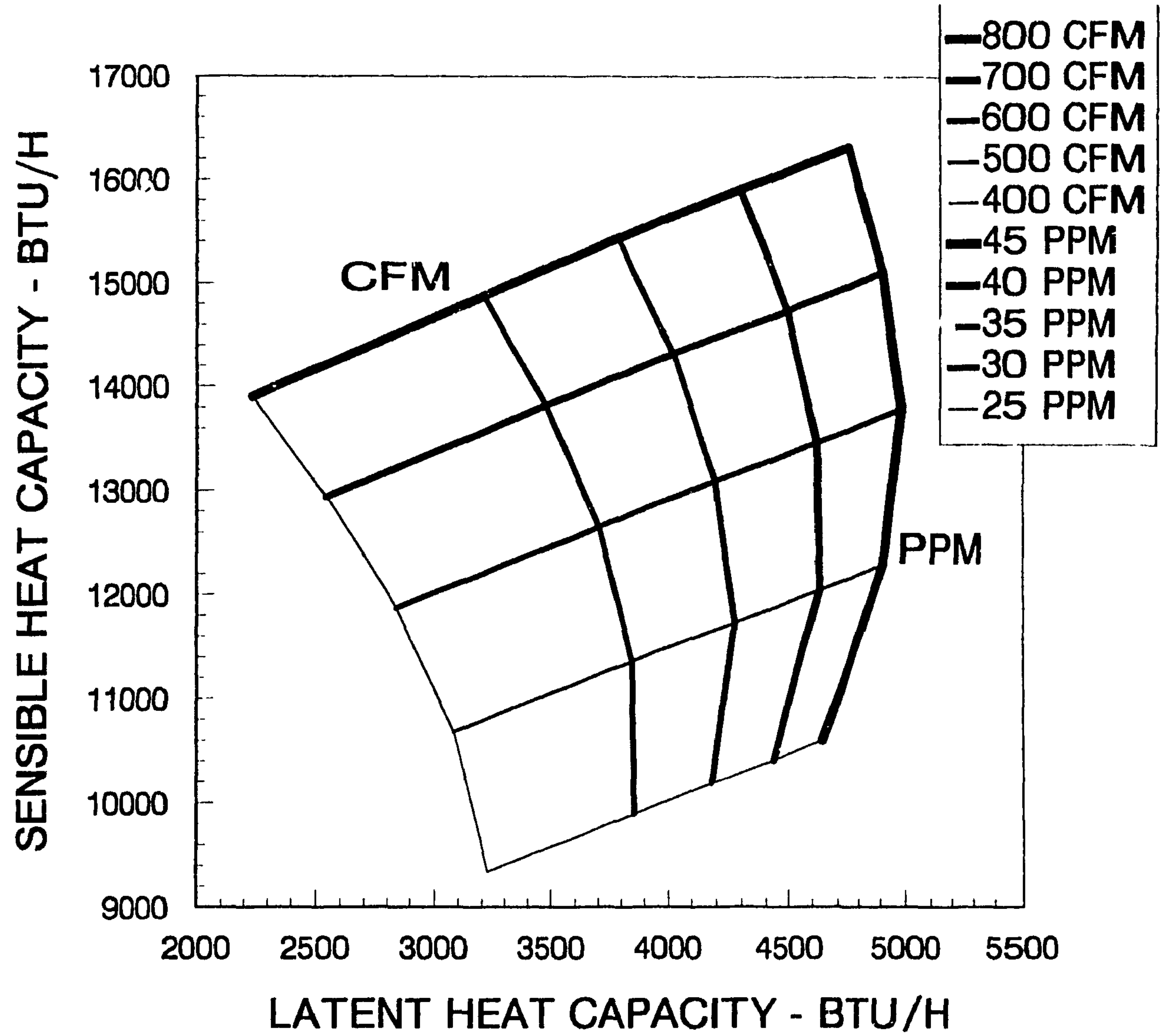

Figure 3.4. A Simulated Performance Map of A Chilled Water Coil with Throttling Water Valve.

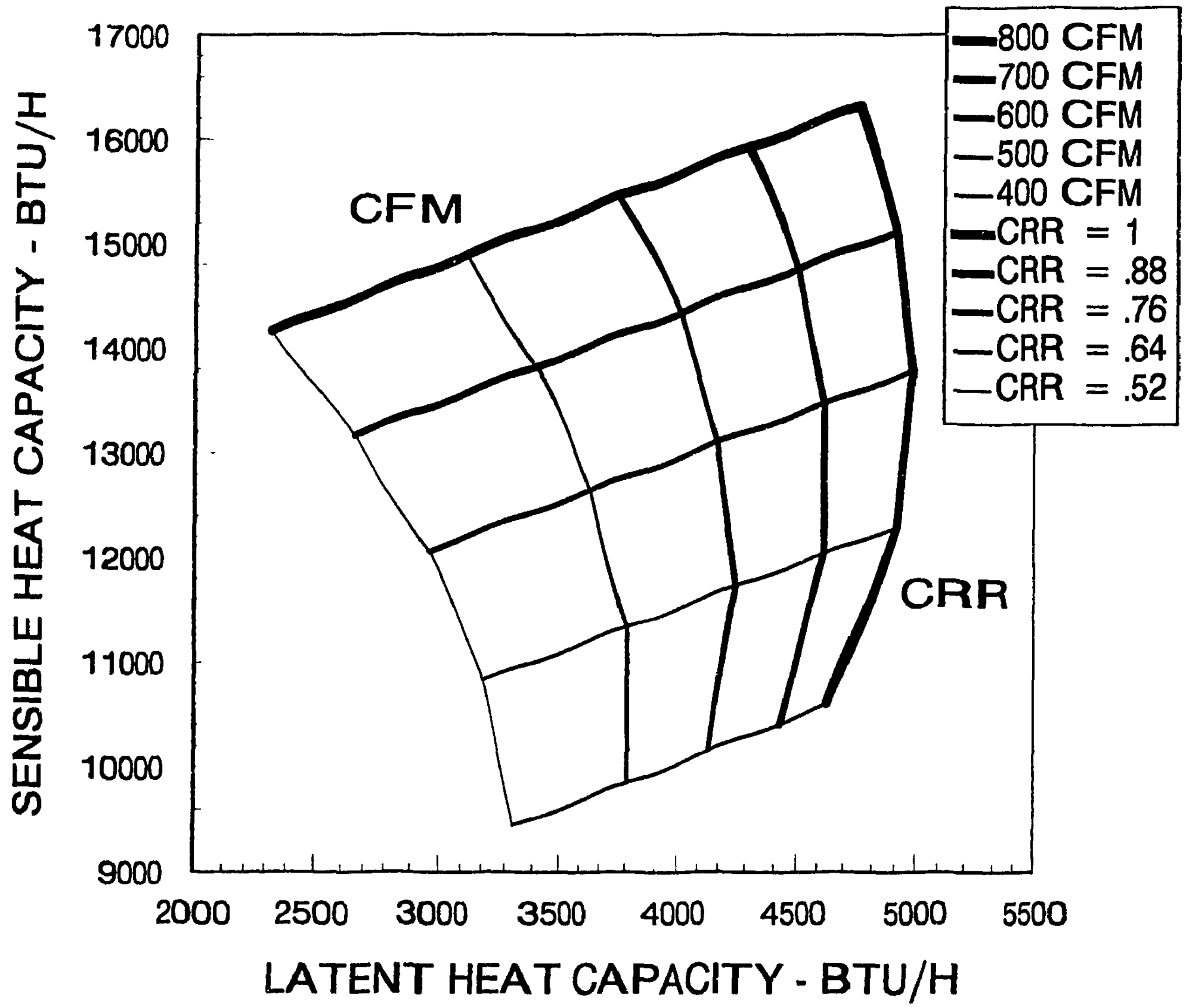

Figure 3.5. A Simulated Performance Map of A Chilled Water Coil with Mixing Water Valve.

3.3 Thermal System Description

3.3.1 Control Strategies

Theoretically, two control strategies are possible. The first strategy is to control the space temperature by modulating the conditioned air flow rate through the coil, and to control the space specific humidity by modulating the chilled water flow rate or temperature through the cooling coil. The second control strategy is to control the space temperature by modulating the chilled water flow rate through the cooling coil, and to control the space specific humidity by modulating the conditioned air flow rate through the coil. Experimental work and the coil characteristics presented in Section 3.2 show that the first control strategy is more feasible.

3.3.2 Control system Description

A schematic diagram of an environmental control system using chilled water system is shown in Figure 3.6. The system consists of five subsystems: chiller, conditioned space, chilled water coil, fan speed control, water valve motor control. The system shown in Figure 3.6 is for the implementation of the first control strategy. Temperature and humidity sensors are connected to a signal processor consists of a data logger and a computer. The signal processor is also connected to a variable frequency inverter which modulates the fan motor speed. The chilled water valve is directly connected to the signal processor. The fan motor speed will be modulated to control temperature. The water valve position will be modulated to control humidity. The sensors monitor the process variables, i.e., temperature and humidity at their locations and provide proportional analogue signals to the data logger.

The data logger generates equivalent digital signals and sends them to the computer. The computer processes the data using the programmed PI algorithm and supplies the resulting outputs as digital signals to the data logger that in turn transform the signals into analogue ones to modulate the corresponding actuators.

3.3.2.1 First and Second Order Thermal Systems

HVAC systems may be classified from a thermo-fluids perspective as systems with energy and/or mass storage and as systems without storage. Systems with storage are systems in which temperature and/or humidity of a space or a storage tank are controlled. Systems without storage are systems in which temperatures or pressures of fluid leaving components are controlled. Systems with storage are classified as second order systems, and systems without storage are classified as first order systems. The terms second and first order refer to the differential equations representing the thermo-fluid apparatus and the PI control system. The process variable sensors could be placed before or after the coil (Figure 3.6). When they are placed in the space or return duct, they sense the space temperature and humidity resulting in a second order system. When they are placed after the cooling coil, in the supply duct, they sense the temperature and humidity of the air leaving the coil, i.e., the supply air, resulting in a first order system. These thermo-fluid systems may be represented by flow diagrams composed of basic modules as shown in Figures 3.7 and 3.8. The closed loop of the modules signifies a feedback control system.

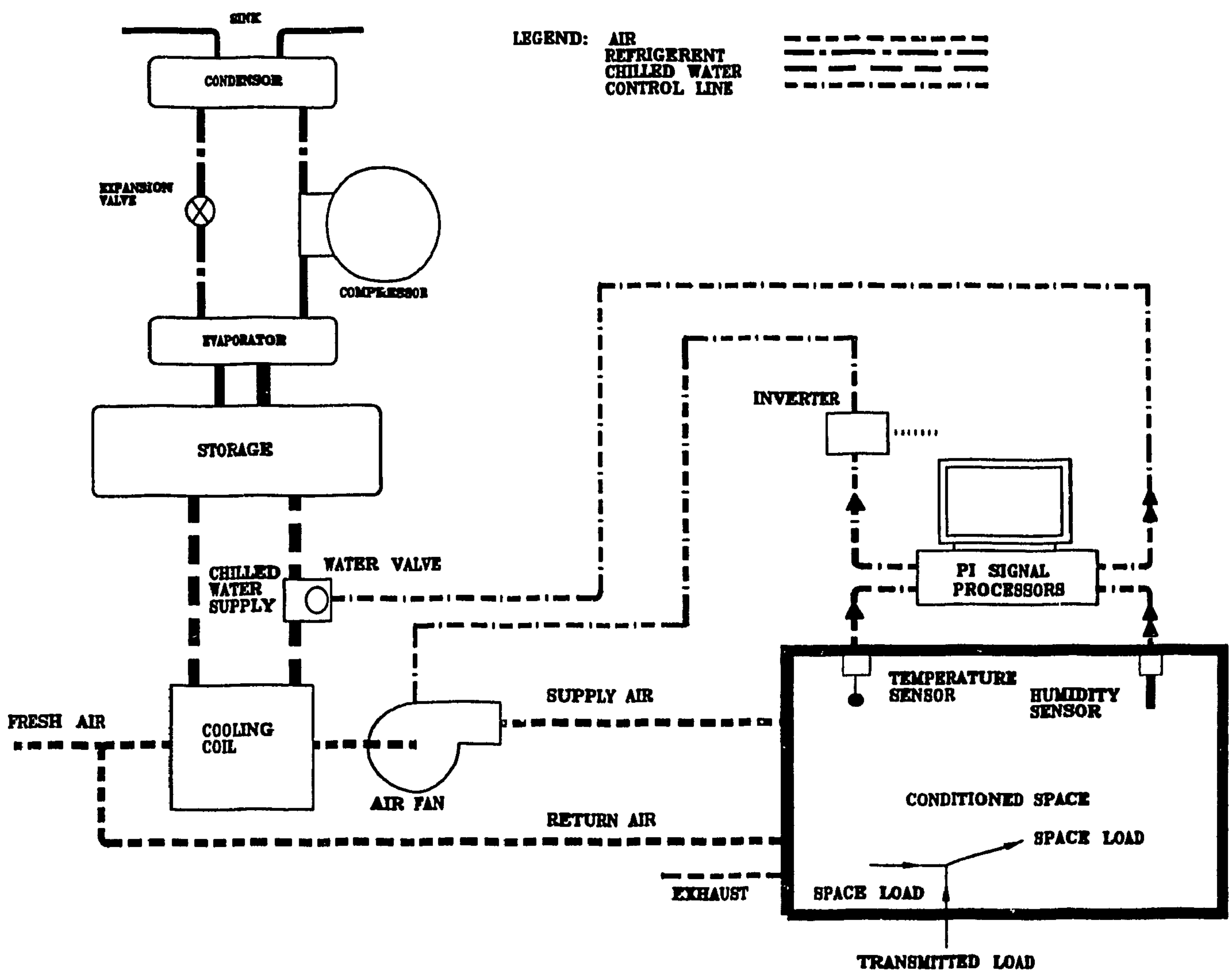

Figure 3.6. Schematic Diagram of the Environmental Control System.

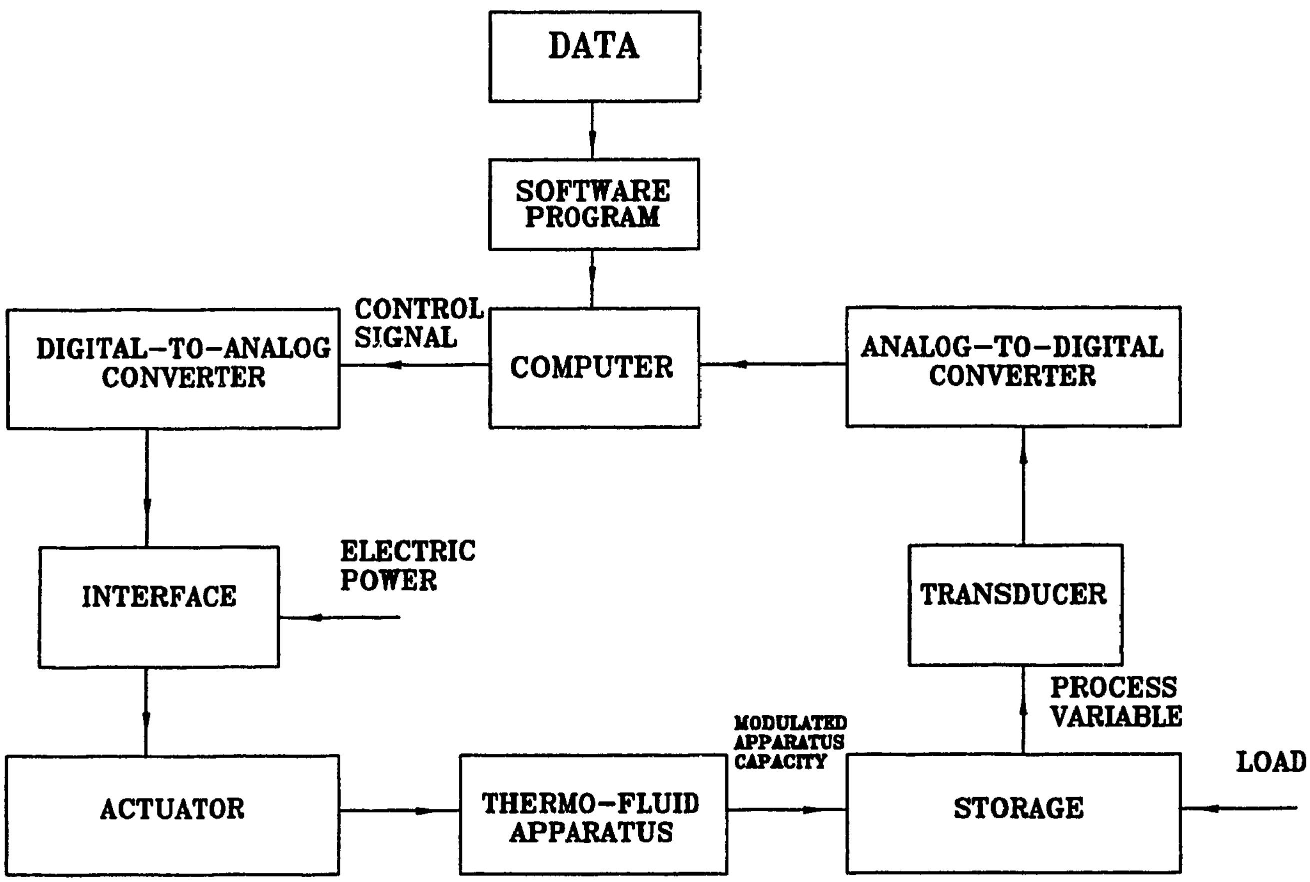

Figure 3.7. Schematic Diagram of a System with a Storage (i.e., a Second Order System).

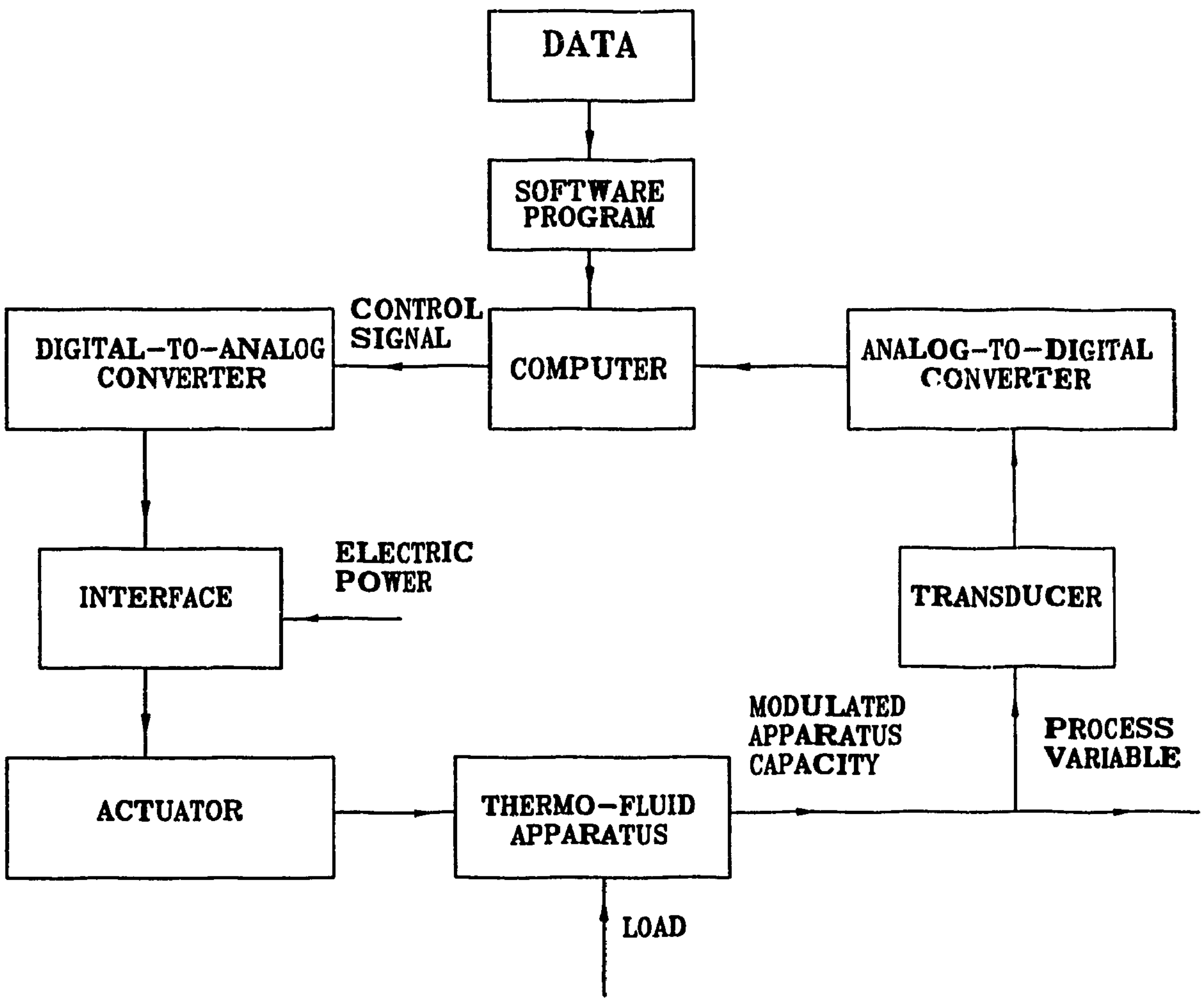

Figure 3.8. Schematic Diagram of a System without a Storage (i.e , a First Order System).

CHAPTER 4

SYSTEM ANALYSIS

4.1 Derivations of System Equations

The PI algorithm presented in Chapter 2, the system conservation of energy, and the system conservation of water vapor lead to first or second order linear differential equations with constant coefficients governing the control systems. Mathematically, systems with storage are classified as second order systems, and systems without storage are classified as first order systems. The terms first and second order refers to the order of the differential equation representing the dynamics of the thermo-fluid apparatus and the PI control system.

A second order system is physically implemented by controlling space temperature and humidity. A first order system is physically implemented by controlling supply air temperature and humidity. Temperature control is related to sensible heat considerations. Specific humidity control is related to water vapor, i.e.. latent heat, considerations.

4.1.1 The PI Equations

A digital PI controller samples the error between a value of a controlled property and its corresponding set point at discrete intervals and outputs a control signal proportional to that error. The control signal, the error and the PI coefficients are related by

$$S = K_p \cdot E + K_I \cdot \sum_{i=1}^{n} E \tag{4.1}$$

where the error E is defined as

$$E = \frac{P_{SP} - P}{P_{MAX} - P_{MIN}} \cdot 100 \tag{4.2}$$

Equation 4.1 is a finite difference equation. To obtain differential equations for control systems, Equation 4.1 is differentiated as follows:

$$\frac{\Delta S}{\Delta t} = K_P \cdot \frac{\Delta E}{\Delta t} + K_I \cdot \frac{E}{\Delta t} \tag{4.3}$$

where

$$\Delta E = E_n - E_{n-1}. \tag{4.4}$$

In differential form, Equation 4.3 is

$$\frac{dS}{dt} = K_P \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E \tag{4.5}$$

The first order differential of the error can be obtained from Equation 4.2 as

$$\frac{dE}{dt} = -\frac{100}{\Delta P} \cdot \frac{dP}{dt} \tag{4.6}$$

while the second order differential of the error is

$$\frac{d^2E}{dt^2} = -\frac{100}{\Delta P} \cdot \frac{d^2P}{dt^2} \tag{4.7}$$

where

$$\Delta P = P_{MAX} - P_{MIN}. \tag{4.8}$$

Equation 4.5 relates the rate of change in the control signal to the PI coefficients and the rate of change in the error. Equation 4.6 relates the rate of change in the error to the rate of change in the controlled property. For second order temperature control system, the controlled property is the temperature of the conditioned space, i.e.,

$$P = T_{SPC}. \tag{4.9}$$

For second order specific humidity control, the controlled property is the specific humidity of the conditioned space, i.e.,

$$P = \omega_{SPC}. \tag{4.10}$$

For first order temperature control system, the controlled property will be the temperature of supply air, i.e.,

$$P = T_{SUP} \tag{4.11}$$

For first order specific humidity control, the controlled property will be the specific humidity of supply air, i.e.,

$$P = \omega_{SUP} \tag{4.12}$$

4.1.2 Mathematical Modelling of Second Order Systems

The temperature and specific humidity in a space depend on the balance between the total load in the space and the chilled water coil capacity. The total load in the space is due to the internal space load, the transmitted load, and the fresh air load. The sensible component of the space load, due to lighting, people, machines and appliances, is time dependent due to cooling load factors and occupancy schedules. The latent component of the space load, due to people, machines, is time dependent due to occupancy schedules. The transmitted load, sensible heat due to conduction and solar insolation, is time dependent due to ambient condition variations. When ventilation is only by infiltration, sensible and latent components due to fresh air are included in the internal space load. Because the system load is time dependent, the temperature and humidity of the conditioned space will be time dependent unless the chilled water capacity and the system load are balanced.

4.1.2.1 Space Temperature Control Model

The variations of the space air temperature as a function of time are determined by applying the laws of conservation of energy to the system air. The law of conservation of energy applied to the space air, is represented by

$$\frac{dU}{dt} = Q_{LD,S} + Q_{CWC,S} \tag{4.13}$$

where

$$Q_{LD,S} = Q_{LD,S,SPC} + Q_{LD,S,TR} + Q_{LD,S,FA} \tag{4.14}$$

The internal energy U can be expressed as

$$U = M_{DA} \cdot c_{V,MA} \cdot T_{SPC} \tag{4.15}$$

where the mass of dry air is

$$M_{DA} = \frac{V}{v_{DA,SPC}} \tag{4.16}$$

and the specific heat at constant volume of the moisture air is

$$c_{V,MA} = c_{V,DA} + \omega \cdot c_{V,WV} \cdot \tag{4.17}$$

Differentiating Equation 4.15 with respect to time, gives

$$\frac{dU}{dt} = M_{DA} \cdot c_{V,MA} \cdot \frac{dT_{SPC}}{dt} \tag{4.18}$$

Then

$$\frac{dT_{SPC}}{dt} = \frac{1}{M_{DA} \cdot c_{V,MA}} \cdot \frac{dU}{dt} \tag{4.19}$$

Substituting Equations 4.13 and 4.16 into Equation 4.19, gives

$$\frac{dT_{SPC}}{dt} = \frac{(Q_{LD,S} + Q_{CWC,S}) \cdot v_{DA,SPC}}{V \cdot c_{V,MA}}. \tag{4.20}$$

The sensible heat capacity of the space, $C_{SPC,S}$, is defined as

$$C_{SPC,S} = c_{V,MA} \cdot \frac{V}{v_{DA,SPC}} \tag{4.21}$$

Substituting the expression for $C_{SPC,S}$ in Equation 4.20, yields

$$\frac{dT_{SPC}}{dt} = \frac{1}{C_{SPC,S}} \cdot \left(Q_{LD,S} + Q_{CWC,S} \right) \tag{4.22}$$

Cooling capacities, i.e., heat transfer rates from the conditioned space to the chilled water, are considered negative. Cooling loads, i.e., heat transfer rates to the conditioned space, are considered positive. Differentiating Equation 4.22 yields

$$\frac{d^2T_{SPC}}{dt^2} = \frac{1}{C_{SPC,S}} \cdot \left(\frac{dQ_{LD,S}}{dt} + \frac{dQ_{CWC,S}}{dt} \right) \tag{4.23}$$

The sensible component of the space load, $Q_{LD,S}$, is a function of time. Then its time rate of change will also be a function of time, i.e.,

$$\frac{dQ_{LD,S}}{dt} = f_{LD,S}(t). \tag{4.24}$$

Analysis in Chapter 3 shows that the sensible component of a chilled water coil capacity is basically dependent on five independent variables: the temperature and specific humidity of the inlet (space) air, the flow rate of air through the coil, the flow rate of the chilled water through the cooling coil, and the temperature of the chilled water, i.e.,

$$Q_{CWC,S} = f_{CWC,S}(T_{SPC}, \omega_{SPC}, \dot{V}_{AIR}, T_{CW}, \dot{m}_{CW}). \tag{4.25}$$

The sensible capacity component is proportional to the temperature of the air in space and the temperature of the chilled water flowing in the cooling coil. The sensible capacity component is a strong function of the state of the space air and the air flow rate. The space temperature may be controlled by varying the air flow rate. In this case, the rate is proportional to the control signal from the PI controller. The signal modulates the fan motor speed hence varying the air flow rate to control the space temperature. The sensible component of the chilled water coil capacity may be represented by

$$Q_{CWC,S} = Q_{CWC,S,o} + D_S \cdot S \tag{4.26}$$

The ratio of the capacity differential to the signal differential represents a characteristic of the thermo-fluid apparatus, i.e., fan and chilled water coil. This ratio is termed the sensible capacity sensitivity and defined by

$$D_S = \frac{dQ_{CWC,S}}{dS}. \tag{4.27}$$

The sensible capacity sensitivity is assumed to be constant. The time rate of change of the reference sensible capacity component is assumed to be independent of time, i.e.,

$$\frac{dQ_{CWC,S\,o}}{dt} = 0 \tag{4.28}$$

The time rate of change of the sensible component of the chilled water coil may therefore be obtained from Equation 4.26 and represented by

$$\frac{dQ_{CWC,S}}{dt} = D_S \cdot \frac{dS}{dt}. \tag{4.29}$$

Combining Equations 4.23, 4.24 and 4.29 yields

$$\frac{d^2T_{SPC}}{dt^2} = \frac{1}{C_{SPC,S}} \cdot \left(f_{LD,S}(t) + D_S \cdot \frac{dS}{dt} \right) \tag{4.30}$$

Combining Equations 4.7 and 4.30 yields

$$\frac{d^2T_{SPC}}{dt^2} = f_{SYS,S}(t) + \frac{D_S}{C_{SPC,S}} \cdot \frac{dS}{dt} = -\frac{\Delta T}{100} \cdot \frac{d^2E}{dt^2} \tag{4.31}$$

where

$$f_{SYS,S}(t) = \frac{1}{C_{SPC,S}} \cdot f_{LD,S}(t) \qquad (4.32)$$

The value for the time rate of change of the control signal as a function of the error is obtained by combining Equations 4.31 and 4.32, therefore

$$\frac{dS}{dt} = -\frac{\Delta T}{100} \cdot \frac{C_{SPC,S}}{D_S} \cdot \frac{d^2E}{dt^2} - \frac{C_{SPC,S}}{D_S} \cdot f_{SYS,S}(t) \qquad (4.33)$$

A second order process sensitivity is defined as the differential of the time rate of change of the process variable with respect to control signal. Differentiating Equation 4.22 with respect to signal, for constant load, yields

$$\sigma_S = \frac{d}{dS}\left(\frac{dT_{SPC}}{dt}\right) = \frac{1}{C_{SPC,S}} \cdot \left(\frac{dQ_{CWC,S}}{dS}\right) = \frac{D_S}{C_{SPC,S}} \qquad (4.34)$$

Combining Equations 4.5, 4.33 and 4.34 yields

$$\frac{\Delta T}{100} \cdot \frac{1}{\sigma_S} \cdot \frac{d^2E}{dt^2} + K_P \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = -\frac{1}{\sigma_S} f_{SYS,S}(t) \qquad (4.35)$$

To simplify notations, a generic load function is defined as follows:

$$g_{SYS,S}(t) = -\frac{1}{\sigma_S} f_{SYS,S}(t) \qquad (4.36)$$

By defining the normalized second order process sensitivity as

$$\sigma_{N,S} = \sigma_S \cdot \frac{100}{\Delta T} \qquad (4.37)$$

Equation 4.35 may be rewritten as

$$\frac{1}{\sigma_{N,S}} \cdot \frac{d^2E}{dt^2} + K_P \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = g_{SYS,S}(t) \tag{4.38}$$

The exact format of the generic load function will not be required to determine the PI coefficients. The normalized second order process sensitivity has the dimension of the reciprocal of time, i.e., sec^{-1}. Equation 4.38 can be written as

$$C_{2,T} \cdot \frac{d^2E}{dt^2} + C_{1,T} \cdot \frac{dE}{dt} + C_{0,T} \cdot E = g_{SYS,S}(t) \tag{4.39}$$

where

$$C_{2,T} = \frac{1}{\sigma_{N,S}} \tag{4.40}$$

$$C_{1,T} = K_P \tag{4.41}$$

$$C_{0,T} = \frac{K_I}{\Delta t}. \tag{4.42}$$

Equation 4.39 is the required second-order differential equation representing the PI temperature control system.

4.1.2.2 Space Humidity control Model

The time rate of change of space specific humidity is dependent on the latent component of the cooling load, the latent component of the capacity of the chilled water coil, and the volume of the conditioned space. Applying the law of conservation of mass to the space water vapor, yields

$$\frac{dM_{WV}}{dt} = \dot{m}_{CWC,WV} + \dot{m}_{LD,WV} \tag{4.43}$$

The latent load is related to the moisture flow rate by

$$Q_L = h_o \cdot \dot{m}_{WV} \tag{4.44}$$

where h_o = 1061 BTU/lbm is a reference enthalpy [25]. Therefore

$$Q_{CWC,L} = h_o \cdot \dot{m}_{CWC,WV} \tag{4.45}$$

and

$$Q_{LD,L} = h_o \cdot \dot{m}_{LD,WV} \tag{4.46}$$

Multiply Equation 4.43 by h_o and substitute for Equations 4.45 and 4.46 yields

$$M_{DA} \cdot h_o \cdot \frac{d\omega_{SPC}}{dt} = \left(Q_{LD,L} + Q_{CWC,L}\right) \tag{4.47}$$

Differentiating Equation 4.47 and rearrange yields

$$\frac{d^2\omega_{SPC}}{dt^2} = \frac{1}{M_{DA} \cdot h_o} \cdot \left(\frac{dQ_{LD,L}}{dt} + \frac{dQ_{CWC,L}}{dt} \right) \tag{4.48}$$

Whereas the latent component of the load is time dependent, its time rate of change is a function of time, i.e.,

$$\frac{dQ_{LD,L}}{dt} = f_{LD,L}(t). \tag{4.49}$$

The analysis in Chapter 3 shows that the latent component of a chilled water coil capacity is basically dependent on five independent variables: the temperature and specific humidity of the space (inlet) air, the flow rate of air through the cooling coil, the flow rate of the chilled water through the cooling coil, and the temperature of the chilled water, i.e.,

$$Q_{CWC,L} = f_{CWC,L}(T_{SPC}, \omega_{SPC}, \dot{V}_{AIR}, T_{CW}, \dot{m}_{CW}). \tag{4.50}$$

The latent capacity component is proportional to the state of the air entering the coil and the state of the chilled water flowing in the cooling coil. For a throttling valve, the chilled water supply temperature is constant. Therefore, the latent capacity component will be a strong function of the chilled water flow rate. In this case, the rate is proportional to the control signal from the PI controller. The signal modulates the valve position hence varying the water flow rate to control the space humidity. For a mixing valve, the mixed chilled water temperature is a function of the recirculation ratio. The latent capacity component will be a function of the mixed water temperature and the recirculation ratio. The signal is proportional to the mixed water temperature. The signal modulates the valve position hence varying the recirculation ratio to control the space humidity. In both cases, the latent component of the chilled water capacity may be assumed to be represented by

$$Q_{CWC,L} = Q_{CWC,L,o} + D_L \cdot S \tag{4.51}$$

The ratio of the capacity differential to the signal differential represents a characteristic of the thermo-fluid apparatus (fan and coil). This ratio is termed the latent capacity sensitivity and defined by

$$D_L = \frac{dQ_{CWC,L}}{dS}. \tag{4.52}$$

The latent capacity sensitivity is assumed to be constant at a mean, although it varies over operating range. The time rate of change of the reference latent capacity component is assumed to be independent of time, i.e.,

$$\frac{dQ_{CWC,L,o}}{dt} = 0 \tag{4.53}$$

The time rate of change of the latent component of the chilled water system may therefore be represented by

$$\frac{dQ_{CWC,L}}{dt} = D_L \cdot \frac{dS}{dt}. \tag{4.54}$$

The system latent heat capacity is defined as

$$C_{SPC,L} = m_{DA} \cdot h_o \tag{4.55}$$

Combining Equations 4.48, 4.49, 4.54 and 4.55 yields

$$\frac{d^2\omega_{SPC}}{dt^2} = \frac{1}{C_{SPC,L}} \cdot \left(f_{LD,L}(t) + D_L \cdot \frac{dS}{dt} \right) \tag{4.56}$$

Combining Equations 4.7 and 4.56 yields

$$\frac{d^2\omega_{SPC}}{dt^2} = f_{SYS,L}(t) + \frac{D_L}{C_{SPC,L}} \cdot \frac{dS}{dt} = -\frac{\Delta\omega}{100} \cdot \frac{d^2E}{dt^2} \tag{4.57}$$

where

$$f_{SYS,L}(t) = \frac{1}{C_{SPC,L}} \cdot f_{LD,L}(t) \tag{4.58}$$

The value for the time rate of change of the control signal as a function of the error is obtained by combining Equations 4.57 and 4.58, therefore

$$\frac{dS}{dt} = -\frac{\Delta\omega}{100} \cdot \frac{C_{SPC,L}}{D_L} \frac{d^2E}{dt^2} - \frac{C_{SPC,L}}{D_L} \cdot f_{SYS,L}(t) \tag{4.59}$$

A second order process sensitivity is defined as the differential of the time rate of change of the process variable with respect to control signal. Differentiating Equation 4.47 with respect to signal yields

$$\sigma_L = \frac{d}{dS}\left(\frac{d\omega_{SPC}}{dt}\right) = \frac{1}{C_{SPC,L}} \cdot \left(\frac{dQ_{CWC,L}}{dS}\right) = \frac{D_L}{C_{SPC,L}} \tag{4.60}$$

Combining Equations 4.5, 4.59 and 4.60 yields

$$\frac{\Delta\omega}{100} \cdot \frac{1}{\sigma_L} \cdot \frac{d^2E}{dt^2} + K_P \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = -\frac{1}{\sigma_L} f_{SYS,L}(t) \tag{4.61}$$

To simplify notations, a generic load function is defined as follows:

$$g_{SYS,L}(t) = -\frac{1}{\sigma_L} f_{SYS,L}(t) \tag{4.62}$$

By defining the normalized second order process sensitivity as

$$\sigma_{N,L} = \sigma_L \cdot \frac{100}{\Delta\omega} \tag{4.63}$$

Equation 4.61 may be rewritten as

$$\frac{1}{\sigma_{N,L}} \cdot \frac{d^2E}{dt^2} + K_P \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = g_{SYS,L}(t) \tag{4.64}$$

The exact format of the generic load function will not be required to determine the PI coefficients. The normalized second order latent process sensitivity has the dimension of the reciprocal of time, i.e., sec^{-1}. Equation 4.64 can be written as

$$C_{2,\omega} \cdot \frac{d^2E}{dt^2} + C_{1,\omega} \cdot \frac{dE}{dt} + C_{0,\omega} \cdot E = g_{SYS,L}(t) \tag{4.65}$$

where

$$C_{2,\omega} = \frac{1}{\sigma_{N,L}} \tag{4.66}$$

$$C_{1,\omega} = K_P \tag{4.67}$$

$$C_{0,\omega} = \frac{K_I}{\Delta t}. \tag{4.68}$$

Equation 4.65 is the required second-order differential equation representing the PI humidity control system.

4.1.2.3 Solution of Differential Equations for PI control

The solutions of Equations 4.39 and 4.65 each have a complementary component and a particular component. The PI coefficients are determined from the complementary component only. The exact formats of the functions of their particular component are not required. Parameters of system characteristics are determined from the roots of the complementary component only. The complementary component is the solution of the homogenous component of Equations 4.39 and 4.65. For temperature control, the homogenous equation is

$$C_{2,T} \cdot \frac{d^2E}{dt^2} + C_{1,T} \cdot \frac{dE}{dt} + C_{0,T} \cdot E = 0 \tag{4.69}$$

The homogenous equation for humidity control will be

$$C_{2,\omega} \cdot \frac{d^2E}{dt^2} + C_{1,\omega} \cdot \frac{dE}{dt} + C_{0,\omega} \cdot E = 0 \tag{4.70}$$

Parameters of system characteristics are determined from the roots of the above two auxiliary equations.

4.1.2.4 Relations of PI coefficients to System Characteristic Parameters

A system response characteristic, i.e., error as a function of time, may be underdamped, overdamped, or critically damped. The response characteristic may therefore be described in terms of certain parameters, critical damping, damping ratio, period, and amplitude ratios. These parameters are analogous to those of a displacement response characteristic for a mass-damper-spring system, or any system governed by a second order differential equation. These properties may be determined by analogy to those for a mass-damper-spring system as described in vibration textbooks [26]. PI coefficients may be selected to yield desired values for the parameters.

Underdamped, critically damped, and overdamped response characteristics correspond to damping ratios less than one, equal to one, and greater than one, respectively. Theoretically, the process variable will oscillate about the set point value for underdamped response characteristics, and will not attain the set point value for critically damped and overdamped response characteristics. The target response should therefore be an underdamped response. The solutions of Equations 4.69 and 4.70 would be expressed in terms of damping ratios and periods [22].

4.1.2.4.1 Damping

The critical damping coefficient for the systems governed by Equations 4.69 and 4.70 is

$$C_{c,CR} = 2 \cdot \sqrt{C_2 \cdot C_0}. \tag{4.71}$$

The damping ratio is

$$\xi = \frac{C_c}{C_{c,CR}} = \frac{C_1}{2 \cdot \sqrt{C_2 \cdot C_0}}. \tag{4.72}$$

Combining Equations 4.40 through 4.42 with Equation 4.72 yields a relation between K_P and K_I for temperature control systems as

$$K_P = \pm 2 \cdot \xi \cdot \sqrt{\frac{K_I}{\Delta t \cdot \sigma_{N,S}}} \tag{4.73}$$

The sign of K_P, K_I and the sensible system sensitivity must be the same.

Combining Equations 4.66 through 4.68 with Equation 4.72 yields a relation between K_P and K_I for specific humidity control systems as

$$K_P = \pm 2 \cdot \xi \cdot \sqrt{\frac{K_I}{\Delta t \cdot \sigma_{N,L}}} \tag{4.74}$$

The sign of K_P, K_I and the latent system sensitivity must be the same.

4.1.2.4.2 Period

The period is defined as the interval for one complete oscillation or cycle and is represented by

$$\tau = \frac{2 \cdot \pi}{\sqrt{1 - \xi^2}} \cdot \sqrt{\frac{C_2}{C_0}}. \tag{4.75}$$

Combining Equations 4.40 through 4.42 with Equation 4.75 yields a relation between K_I the period of the response characteristic for temperature control systems as

$$\tau = 2\cdot\pi\cdot\sqrt{\frac{\Delta t}{K_I\cdot(1-\xi^2)\cdot\sigma_{N,S}}}. \tag{4.76}$$

Combining Equations 4.66 through 4.68 with Equation 4.75 yields a relation between K_I and the period of the response characteristic for specific humidity control system as

$$\tau = 2\cdot\pi\cdot\sqrt{\frac{\Delta t}{K_I\cdot(1-\xi^2)\cdot\sigma_{N,L}}}. \tag{4.77}$$

4.1.2.4.3 Amplitude Ratios

Two amplitude ratios may be considered. The amplitude ratio of two consecutive peaks with the same signs (i.e., + + or - -) separated by one period, in terms of the logarithmic decrement, is

$$A_\tau = \frac{1}{e^\delta} \tag{4.78}$$

where

$$\delta = \frac{2\cdot\pi\cdot\xi}{\sqrt{1-\xi^2}}. \tag{4.79}$$

The amplitude ratio of two consecutive peaks with opposite signs (i.e., + and - or - and +) separated by one-half period is

$$A_{\tau/2} = \frac{1}{e^{\delta/2}}. \tag{4.80}$$

Figure 4.1 shows the values of amplitude ratios as functions of the damping ratio.

4.1.2.5 Formulae for PI coefficients

The normalized process sensitivity, the sampling interval, the period and damping ratio of the target system response characteristic are required to determine PI coefficients. The normalized process sensitivity is a system characteristic. The integral coefficient may be determined from Equations 4.76 and 4.77, i.e.,

$$K_I = \frac{4 \cdot \pi^2 \cdot \Delta t}{\tau^2 \cdot (1 - \xi^2) \cdot \sigma_N}. \tag{4.81}$$

The proportional coefficient may be determined from Equations 4.73 and 4.74, i.e.,

$$K_P = \pm\, 2 \cdot \xi \cdot \sqrt{\frac{K_I}{\Delta t} \cdot \frac{1}{\sigma_N}}. \tag{4.82}$$

For temperature control the sensible process sensitivity is used. For humidity control, the latent process sensitivity is used. The sign of the proportional coefficient must be the same as the sign of the process sensitivity. The signs of both PI coefficients must be the same.

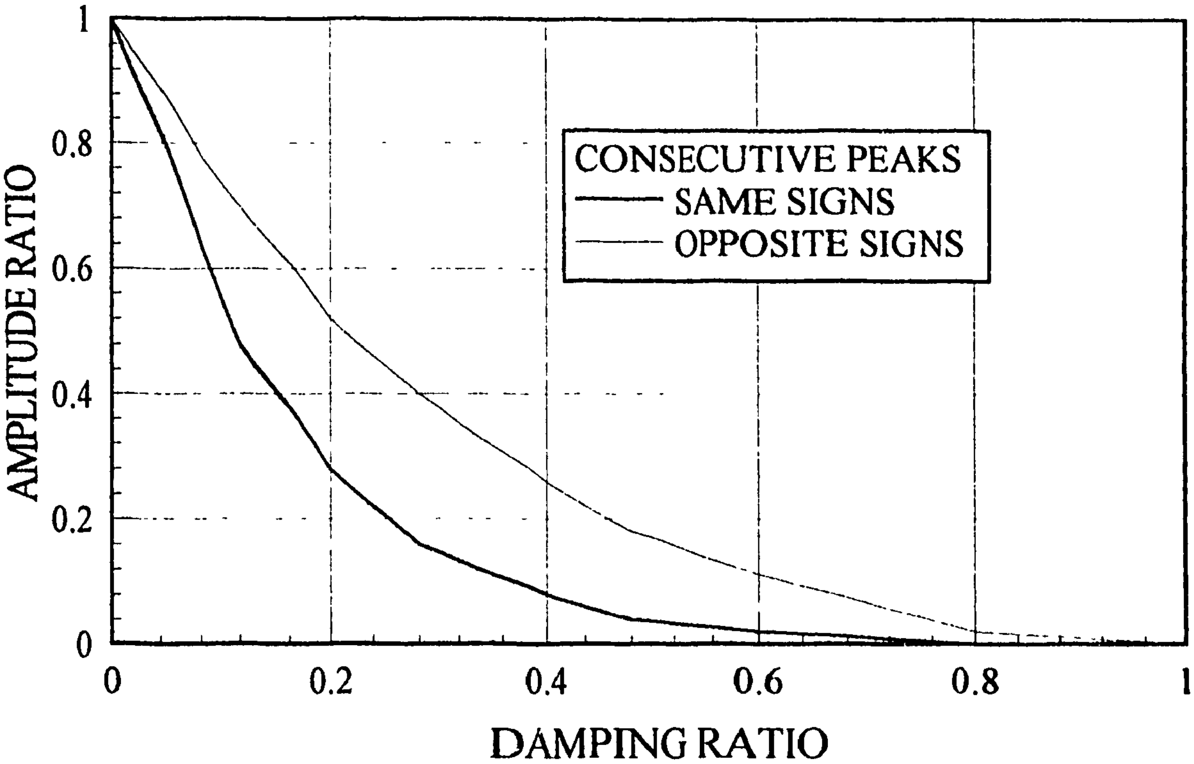

Figure 4.1. Amplitude Ratios as Functions of Damping Ratios

4.1.3 Mathematical Modelling of First Order Systems

The temperature and specific humidity of air leaving a cooling coil (supply air) may be controlled by varying the coil heat transfer rate. The heat transfer rate may be varied by varying the flow rate of the conditioned air, or the flow rate or temperature of the chilled water. Supply air temperature is controlled by varying the air flow rate by the fan motor speed modulation. Supply air specific humidity is controlled by varying the chilled water flow rate or temperature into the coil by modulating the water valve position. Such systems are considered as first order systems because no storage is involved. The temperature and specific humidity of air leaving a thermo-fluid apparatus (fan and coil unit) are the controlled properties.

4.1.3.1 Supply Air Temperature Control Model

The energy conservation equation for air flowing through a chilled water cooling coil relating inlet temperature, supply air temperature, and sensible heat transfer rate to or from the air is

$$Q_{CWC,s} + \dot{c}_T \cdot T_{IN} = \dot{c}_T \cdot T \tag{4.83}$$

where

$$\dot{c}_T = c_p \cdot \dot{m}. \tag{4.84}$$

The process variable is the supply air temperature. The load is the inlet air temperature which is the independent variable that is time dependent. The modulated capacity, which is a function of the signal, is the heat transfer rate to or from the air. For this system, the modulated capacity $Q_{CWC,s}$ is dependent on the load. Therefore, the process variable may be considered as a function of the control signal and time, i.e.,

$$T = T(S,t). \tag{4.85}$$

The differential of the control variable with respect to time is

$$\frac{dT}{dt} = \left(\frac{\partial T}{\partial S}\right)_t \cdot \frac{dS}{dt} + \left(\frac{\partial T}{\partial t}\right)_S . \tag{4.86}$$

Whereas the load is time dependent and is the independent variable of the system, constant time is equivalent to constant load, i.e.,

$$\left(\frac{\partial T}{\partial S}\right)_t = \left(\frac{\partial T}{\partial S}\right)_{T_{IN}} . \tag{4.87}$$

A first order sensible process sensitivity relating the process variable differential to the signal differential is defined by

$$\sigma_s \triangleq \left(\frac{\partial T}{\partial S}\right)_{T_{IN}} . \tag{4.88}$$

If the air temperature is a linear function of the heat transfer rate, the process sensitivity may also be represented in terms of a sensible capacity sensitivity D_s by differentiating Equation 4.83 with respect to signal as

$$\sigma_s = \frac{dT}{dS} = \frac{1}{\dot{c}_T} \cdot \frac{dQ_{CWC,s}}{dS} = \frac{D_s}{\dot{c}_T}. \tag{4.89}$$

The load is the independent variable, the partial differential of the process variable with respect to time is proportional to the differential of the load with respect to time, i.e.,

$$\left(\frac{\partial T}{\partial t} \right)_S = b \cdot \frac{dT_{IN}(t)}{dt}. \tag{4.90}$$

To simplify notations, a generic load function is defined as

$$l(t) \triangleq -\frac{b}{\sigma_s} \cdot \frac{dT_{IN}(t)}{dt} = -\frac{1}{\sigma_s} \left(\frac{\partial T}{\partial t} \right)_S. \tag{4.91}$$

The exact format of the generic load function will not be required to determine the PI coefficients. Combining Equations 4.6, 4.11 and 4.86 through 4.91 yields the following equation, relating the time rate of change of the control signal to the time rate of change of the error:

$$\frac{dS}{dt} = -\frac{1}{\sigma_s} \cdot \frac{\Delta T}{100} \cdot \frac{dE}{dt} + l(t). \tag{4.92}$$

By defining the normalized first order sensible process sensitivity as

$$\sigma_{N,s} = \sigma_s \cdot \frac{100}{\Delta T} = \frac{dT}{dS} \cdot \frac{100}{\Delta T} \tag{4.93}$$

Equation 4.92 may be rewritten as

$$\frac{dS}{dt} = -\frac{1}{\sigma_{N,s}} \cdot \frac{dE}{dt} + l(t) . \qquad (4.94)$$

The normalized first order process sensitivity is dimensionless. Combining Equations 4.5 and 4.94 yields the following equation relating the error, the PI coefficients, the sampling interval, the normalized sensible sensitivity, and the load:

$$\left(K_P + \frac{1}{\sigma_{N,s}} \right) \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = l(t) . \qquad (4.95)$$

Equation 4.95 is the required first order differential equation representing the PI temperature control system.

4.1.3.2 Supply Air Humidity Control Model

The mass conservation equation for water vapor through a chilled water cooling coil relating inlet specific humidity, supply specific humidity, the mass of dry air and the mass of the water condensate is

$$\dot{m}_{CWC} + \dot{m}_{DA} \cdot \omega = \dot{m}_{DA} \cdot \omega_{IN} \qquad (4.96)$$

Multiply Equation 4.96 by the reference enthalpy, h_o yields

$$Q_{CWC,L} + \dot{c}_{\omega} \cdot \omega = \dot{c}_{\omega} \cdot \omega_{IN} \qquad (4.97)$$

where

$$\dot{c}_{\omega} = h_o \cdot \dot{m}_{DA} . \qquad (4.98)$$

The process variable is the supply specific humidity. The load is the inlet specific humidity which is time dependent. The modulated capacity, which is a function of the signal, is the latent heat transfer rate to or from the air. For this system, the modulated capacity $Q_{CWC,L}$ is dependent on the load. Therefore, the process variable may be considered as a function of the control signal and time, i.e.,

$$\omega = \omega(S,t). \tag{4.99}$$

The differential of the control variable with respect to time is

$$\frac{d\omega}{dt} = \left(\frac{\partial\omega}{\partial S}\right)_t \cdot \frac{dS}{dt} + \left(\frac{\partial\omega}{\partial t}\right)_S . \tag{4.100}$$

Whereas the load is time dependent and is the independent variable of the system, constant time is equivalent to constant load, i.e.,

$$\left(\frac{\partial\omega}{\partial S}\right)_t = \left(\frac{\partial\omega}{\partial S}\right)_{\omega_{IN}} . \tag{4.101}$$

A first order latent process sensitivity relating the process variable differential to the signal differential is defined by

$$\sigma_L \triangleq \left(\frac{\partial\omega}{\partial S}\right)_{\omega_{IN}} . \tag{4.102}$$

If the air humidity is a linear function of the latent heat transfer rate the process sensitivity may also be represented in terms of a latent capacity sensitivity, D_L, as

$$\sigma_L = \frac{d\omega}{dS} = \frac{1}{\dot{c}_\omega}\cdot\left(\frac{dQ_{CWC,L}}{dS}\right) = \frac{D_L}{\dot{c}_\omega}. \tag{4.103}$$

The load is the independent variable, the partial differential of the process variable with respect to time is proportional to the differential of the load with respect to time, i.e.,

$$\left(\frac{\partial\omega}{\partial t}\right)_S = b\cdot\frac{d\omega_{IN}(t)}{dt}. \tag{4.104}$$

To simplify notations, a generic load function is defined as

$$l(t) \triangleq -\frac{b}{\sigma_L}\cdot\frac{d\omega_{IN}(t)}{dt} = -\frac{1}{\sigma_L}\left(\frac{\partial\omega}{\partial t}\right)_S. \tag{4.105}$$

The exact format of the generic load function will not be required to determine the PI coefficients. Combining Equations 4.6, 4.12 and 4.100 through 4.105 yields the following equation, relating the time rate of change of the control signal to the time rate of change of the error:

$$\frac{dS}{dt} = -\frac{1}{\sigma_L}\cdot\frac{\Delta\omega}{100}\cdot\frac{dE}{dt} + l(t). \tag{4.106}$$

By defining the normalized first order latent process sensitivity as

$$\sigma_{N,L} = \sigma_L\cdot\frac{100}{\Delta\omega} = \frac{d\omega}{dS}\cdot\frac{100}{\Delta\omega} \tag{4.107}$$

Equation 4.106 may be rewritten as

$$\frac{dS}{dt} = -\frac{1}{\sigma_{N,L}} \cdot \frac{dE}{dt} + l(t). \tag{4.108}$$

The normalized first order process sensitivity is dimensionless. Combining Equations 4.5 and 4.108 yields the following equation relating the error, the PI coefficients, the sampling interval, the normalized sensitivity, and the load:

$$\left(K_P + \frac{1}{\sigma_{N,L}}\right) \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = l(t). \tag{4.109}$$

Equation 4.109 is the required first order differential equation representing the PI specific humidity control system.

4.1.3.3 Solution of Differential Equations for PI Control

The solutions of Equations 4.95 and 4.109 each have a complementary component and a particular component. The PI coefficients are determined from the complementary component only. The exact formats of the functions of their particular component are not required. Parameters of system characteristics are determined from the roots of the complementary component only. The complementary component is the solution of the homogenous component of Equations 4.95 and 4.109. For temperature control, the homogenous equation is

$$\left(K_P + \frac{1}{\sigma_{N,s}}\right) \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = 0. \tag{4.110}$$

The homogenous equation for humidity control will be

$$\left(K_P + \frac{1}{\sigma_{N,L}}\right) \cdot \frac{dE}{dt} + \frac{K_I}{\Delta t} \cdot E = 0. \qquad (4.111)$$

Parameters of system characteristics are determined from the roots of the above two auxiliary equations.

4.1.3.4 Formulae for PI coefficients

The PI coefficients are determined from the solutions of Equations 4.110 and 4.111, i.e.,

$$\Gamma \cdot \frac{dE}{dt} + E = 0 \qquad (4.112)$$

where the response characteristic time is defined as

$$\Gamma \triangleq \frac{K_P + \frac{1}{\sigma_N}}{K_I} \cdot \Delta t. \qquad (4.113)$$

For temperature and humidity control use the sensible and latent normalized sensitivities respectively. The solution of Equation 4.112 is

$$E = E_0 \cdot e^{-\frac{t}{\Gamma}}. \qquad (4.114)$$

Equation 4.114 represents an exponential error attenuation. The attenuation factor per sampling interval is defined as

$$a_1 \triangleq 1 - \frac{E_n}{E_{n-1}} = 1 - e^{-\epsilon} \qquad (4.115)$$

where the attenuation exponent is

$$\epsilon = \frac{\Delta t}{\Gamma} = \frac{K_I}{K_p + \frac{1}{\sigma_N}}. \tag{4.116}$$

It is evident from Equation 4.115, that for small values of an attenuation factor per sampling interval

$$\epsilon \approx a_1 \tag{4.117}$$

and

$$\frac{\Delta t}{\Gamma} \approx a_1. \tag{4.118}$$

The attenuation factor at the time equal to the response characteristic time, is

$$a_\Gamma = 1 - e^{-1} = 0.632 = 63.2\%. \tag{4.119}$$

It is preferable that the sign of the proportional coefficient is the same as the sign of the sensitivity. If the sign of the proportional coefficient is the opposite of the sign of the sensitivity, the control will be satisfactory but it may require a greater time to attain the set point value. The attenuation exponent must be positive.

The normalized process sensitivity, the sampling interval, and the attenuation factor of the target system response characteristic are required to determine PI coefficients. The normalized process sensitivity is a system characteristic. The attenuation exponent is determined, using Equation 4.117 or 4.118, from the attenuation factor. The sampling interval, response characteristic time, and attenuation are related by Equation 4.118. The integral coefficient is then determined, using Equation 4.116, from the attenuation exponent, the proportional coefficient, and the normalized process sensitivity.

4.1.3.5 Limitation on magnitudes of PI coefficients

Instabilities may occur if either the magnitude of the proportional coefficient or the attenuation factor is too large. Equation 4.4 may be rewritten as

$$\Delta S = K_P \cdot \Delta E + K_I \cdot E \tag{4.120}$$

where

$$\Delta E = E_n - E_{n-1} . \tag{4.121}$$

If the error is equal, or approximately equal, to zero, a maximum value of the proportional coefficient may be related to the maximum values of the control signal differential and the error differential, as per Equation 4.120, i.e.,

$$|K_{P,MAX}| \le \frac{\Delta S_{MAX}}{|\Delta E_{MAX}|} = \frac{100}{|\Delta E_{MAX}|} \tag{4.122}$$

The maximum error differential is 100, therefore, the maximum value of the proportional coefficient is one, i.e.,

$$K_{P,MAX} \le 1 . \tag{4.123}$$

For attenuation to occur,

$$\epsilon = \frac{K_I}{K_P + \frac{1}{\sigma_N}} \geq 0. \tag{4.124}$$

4.2 Implementation Considerations

Calculations of proportional and integral coefficients are based on the following parameters:

- maximum and minimum process variable values
- the sampling interval, and
- the process sensitivity,
- the period and damping ratio for a second order target response, or
- the time response characteristic, or the attenuation factor for a first order target response.

Equations relating the PI coefficients to these parameters are detailed in previous sections. Considerations required for the selection of these parameters are detailed in the following sections

4.2.1 Selection of Maximum and Minimum Process Variable Values

The maximum and minimum process variable values are required to normalize the error so that its value will be within 0 and 100 percent under all operating conditions. The maximum and minimum process variable values should be selected such that they exceed the anticipated values of the process variable during system operation as well as during system shut down thereby ensuring that the error will not exceed its limits. The values may be such that their difference is a round number, for example, 100. These values are also used to normalize the process sensitivity.

4.2.2 Selection of Sampling Interval

The selection of the sampling interval is related to the response of the modulated apparatus capacity to a change in control signal and to the time required for system fluids to circulate through the thermo-fluid apparatus. In a digitally-controlled system, the response curve is composed of a series of consecutive sampling interval responses. Each sampling interval response corresponds to a step change in the control signal. The sampling interval should be sufficiently long to enable the response to stabilize with respect to time. The time for the response to stabilize is considered the response time. A sampling interval greater than the response time should be used. Long sampling intervals enable a complete response of the system to a change in the control signal. However, the sampling interval should not be excessively long. Shorter sampling intervals track load variations more accurately than longer ones. For temperature control, the control signal modulates a rotational motor that drives the fan. For humidity control, the control signal modulates a positional motor that drives the chilled water valve.

4.2.2.1 Rotational Motors - Fan Motor

When a rotational motor is modulated, the response time of the motor is related to its acceleration capability. The response time of a motor is the time required for the motor to accelerate in response to a step change in its control signal, while running. It was observed that the response time of a nominal one horsepower motor-driven forward-curved fan responding to a 50% change in control signal (rotational speed) was approximately 10 seconds. It was also observed that the parameters associated with system power, i.e., acceleration, responded within a similar interval as the motor power. The heat transfer rate and the air flow rate of a fan-coil unit containing the aforementioned fan responded rapidly as predicted by system power equilibrium considerations. The sampling interval for systems involving rotational motors should exceed the response time of the motor. It may therefore be concluded that the sampling interval for systems involving rotational motors should be of the order of 10 to 15 seconds. At start-up, a motor requires a greater time than the response time to achieve a steady-state condition. Control implementation algorithms should allow for start-up stabilization [22].

4.2.2.2 Positional motors - Valve Motor

When the modulated motor is positional, the time required for the valve to respond to its maximum position differential may be of the order of 60 to 90 seconds. The sampling interval for systems involving positional motors should exceed the response time of the motor. A sampling interval of 60 sec. for the humidity control system is satisfactory [22].

4.2.2.3 Circulation

The time required by some system fluids to circulate through a thermo-fluid apparatus is defined as the circulation interval. It is the interval between the time an element of fluid enters and leaves the apparatus. The interval for an element of fluid to pass through a heat exchanger, and the interval for an element of fluid to pass through a duct system are examples of circulation intervals. The circulation interval is a function of the fluid velocity and the apparatus geometry. The sampling interval should exceed the circulation interval.

4.2.2.4 Additional considerations

Response times of rotational motors and circulation intervals may vary over the operating range of a system. They are not readily accurately determinable. Sampling interval should therefore be sized conservatively. In selecting the PI coefficients to yield a target response characteristic, random noise effects on the transducer output must be considered. Thermocouples are more susceptible to random noise than thermistors and resistance temperature detectors (RTDs). If a random noise is present, the value of the proportional coefficient should be kept as small as practical. The resolution of the transducer should also be considered in selecting the sampling interval. The sampling interval should be sufficiently large so that a change in transducer output is significant.

4.2.3 Selection of Target Response

The target response of second order temperature and humidity systems is specified by a selected period and damping ratio. The target response of first order temperature and humidity systems is specified by a selected time response characteristic or attenuation factor. The target response is related to the response of the process variable to a change in control signal. The criteria for the selection of the target response are different for second order and first order systems.

4.2.3.1 Second order systems

Controlled response characteristics of a second order system may be underdamped, critically damped, or overdamped. Underdamped responses yield a process variable value which oscillates about the set point value. The magnitude and period of the oscillation may be selected, i.e., targeted. Critically damped responses may exist at one, and only one, operating condition because sensitivities generally vary over the operating range of a system. Critically damped responses cannot be described by a time constant because the error attenuation rate is time dependent. It is suggested, from practical considerations, that a target response be an underdamped response with a damping ratio close to one rather than a critically damped response. Overdamped responses never attain a zero error condition and are therefore undesirable. The current investigations found that damping ratios of the order of 0.8 yield satisfactory control.

4.2.3.2 First order systems

The selection of the time constant for a target response for a first order system is related to the attenuation factor and the sampling interval as represented by Equation 4.118. The sampling interval is selected such that it is greater than the motor response time. The maximum attenuation factor is limited by stability considerations. A maximum attenuation factor per sampling interval of the order of 0.05 was found to yield satisfactory responses. The time response characteristic of a target response should therefore exceed the sampling interval divided by the maximum attenuation factor, i.e.,

$$\Gamma > \frac{\Delta t}{a_{1,MAX}}. \tag{4.125}$$

4.2.3.3 Deviation from Target Response

The system response characteristic will deviate from the target system response characteristic due to the following factors:

- the variation of process sensitivity over operating range,
- the load is a weak function of process variable,
- the apparatus operates at lower or higher control signal limits (thermo-fluid apparatus capacity too high or too low).

Deviation from target response is acceptable as far as the control signal follows the load.

4.2.4 Sensitivity Evaluation

The critical parameter required for the analytical determination of the PI coefficients is the process sensitivity. The process sensitivity is a characteristic of the thermo-fluid components and storage, if present. The process sensitivity can be determined experimentally or analytically. Experimental determination of the sensitivity is based on open loop system considerations. The sensitivity may be obtained graphically from the response of the process variable due to variations in the modulated variable. For example, for temperature second order control system, the sensitivity can be determined graphically from the response of the space temperature due to an increase in the air flow rate. Chapter 5 illustrates the experimental determination of the process sensitivity. Analytically, the process sensitivity could be determined from Equations 4.34 and 4.60 for a second order system based on the chilled water coil characteristics presented in Chapter 3. The validity of the analytical sensitivity depends on the accuracy of the coil modelling. For first order systems, Equations 4.89 and 4.103 will determine the sensitivities.

CHAPTER 5

EXPERIMENTAL INVESTIGATIONS

5.1 Experimental Facilities

Experimental testing of temperature and humidity control systems and control strategies were performed at Concordia University's Loyola Campus Laboratories. An environmental control system similar, but not identical to that shown schematically in Figure 3.6 was tested. One of the rooms at the laboratories, containing 1.5 tons chiller system equipment, was tested. A plate finned tube cooling coil was employed. Coolant consisted of water and glycol to avoid freezing. Room temperature and humidity control represents a second order system. Temperature and humidity control of air leaving the cooling coil, i.e., supply air, represents a first order system. Implementation of control strategies was performed with a controller consisting of a data logger and a computer.

5.1.1 The Room

The room used in experimental implementation was a closed 2640 cubic foot environment. Fresh air was provided by infiltiation through windows. An air handling unit consisting of a fan, a 1hp motor, and a plenum was designed to circulate the air in the room at a rates of 400 to 800 cfm. A variable frequency inverter, connected to the fan motor, modulated the fan speed hence the air flow rate. The coolant system and a 30 USG storage coolant tank were installed inside the room. Details are not relevant to the current objectives. The chiller provides a constant coolant supply temperature of 37 °F. Three electric heaters were used to heat the room and to vary the load. A steam injection humidifier was used as the moisture source throughout the experiments. Two small fans were installed at one of the room walls and used to vary the space humidity by supplying additional dry air. Two nominal 1/2 in. water valves were used: a throttling valve and a mixing valve. The controller supplied the control signal to the motor of the valve modulating the valve position hence the coolant supply flow rate, or temperature, to the coil.

5.1.2 The Cooling Coil

The cooling coil used in conducting experiments was a plate finned tube coil with 10 inch height and 4.33 inch depth. It consists of 8 face tubes, 4 rows deep, 10 fines per/inch with 18 inches long, and half inch diameter tubes. The simulated coil characteristics, with pure water as coolant, were shown in Figures 3.4 and 3.5 for operation with throttling and mixing water valves respectively.

5.1.3 The Sensors

Although several types of water temperature sensors were installed and tested at the initiation of the project, only one type was used in testing the control schemes. RTD sensors proved to be less sensitive to electrical noise hence they were used for water temperature measurements. Capacitance type humidity sensors were employed. Air temperature and humidity sensors were placed at the inlet and outlet of the cooling coil. The inlet air sensors measure the space temperature and humidity. The outlet air sensors measure the coil leaving air temperature and humidity. Implementing second order systems was done by control system activation by space air sensors. Implementing first order systems was done by control system activation by supply air sensors.

5.1.4 The Controller

The PI algorithm presented in preceding chapters was implemented by a computer program. The computer hardware was connected to the system transducers through a data logger. The data logger converts the sensor analogue signal to a digital signal which is supplied to the computer. The computer calculates the required control signal according to the software program, specified data input, and input from the data logger. The computer outputs a digital signal to the data logger that converts the signal to an analogue one supplied to the actuators.

The software program contains the PI algorithm relating the control signal to the process variable, i.e., relating the computer output and input signals. The program also contains the windup prevention algorithm developed in Chapter 2. The specified data input consists of the following essential parameters:

- set point values for temperature and specific humidity
- minimum and maximum values of temperature and humidity
- sampling intervals for temperature and humidity
- the PI coefficients determined for temperature and humidity control.

The program is shown in the Appendix. The same program used for space temperature and humidity control was modified for supply air temperature and humidity control.

5.2 Experimental Testing

The theory developed in Chapter 4 was implemented by performing four major experiments. One experiment was the simultaneous control of space temperature and humidity using a throttling water valve. The same experiment was repeated using a mixing water valve. A third experiment was the simultaneous control of temperature and humidity of the air leaving the cooling coil (supply air) using a throttling water valve. The same experiment was repeated using a mixing water valve. The first two experiments implement second order systems. And the next two experiments implement first order systems. The sensitivities used to determine the PI coefficients were determined experimentally. The experimental system employed a mixture of water and glycol as coolant instead of pure water.

5.2.1 Space Temperature and Humidity Control Experiments using a Throttling Water Valve

A schematic of the physical system that implements the control strategy for space temperature and humidity is shown in Figure 5.1. The temperature sensor is connected to the PI controller that modulates the fan motor through a variable frequency inverter. The humidity sensor is connected to the PI controller that modulates the valve motor. The chilled water supply was maintained at constant temperature by modulating the compressor speed. The theory developed in Chapter 4, as well as the implementation considerations presented, were used to choose the system characteristic parameters leading to the determination of PI coefficients.

5.2.1.1 Temperature Control

To control the space temperature, the sensible sensitivity is required. Once this sensitivity is determined, the PI coefficients can be evaluated using Equations 4.81 and 4.82. The determinations of the PI coefficients also require the identification of the target system response parameters by choosing the appropriate damping ratio and period (Section 4.2).

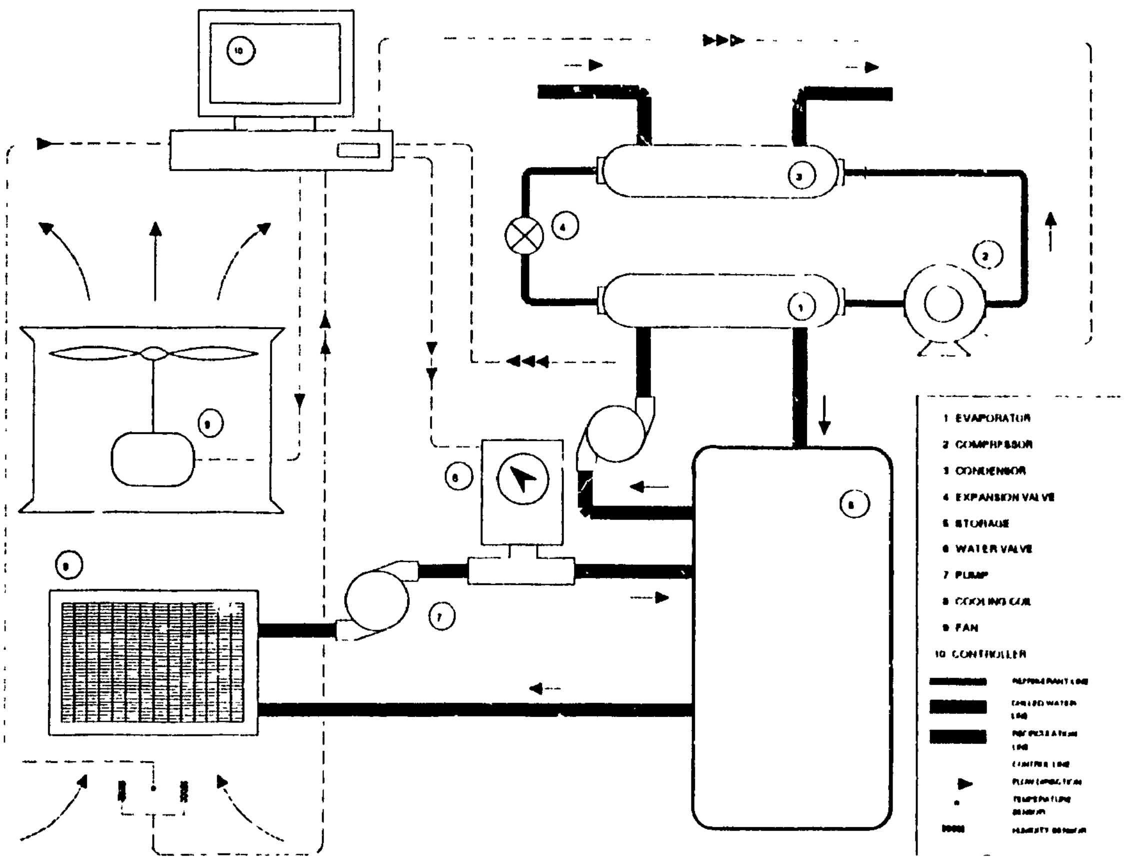

Figure 5.1 Schematic representation of the system used for Space Temperature and Humidity Control using a Throttling Water Valve.

5.2.1.1.1 Sensible Sensitivity Evaluation

Equation 4.34 defines the system sensible sensitivity as the ratio of the capacity sensitivity to the space sensible heat capacity. The sensible heat capacity of the room can be evaluated using Equation 4.21 and the air specific volume at the set point temperature, yields

$$C_{SPC,S} = c_{v,MA} \cdot \frac{V}{v_{DA,SPC}} = \left(0.17 \text{ BTU/lbm °F}\right) \cdot \frac{2640 \text{ ft}^3}{13.7 \text{ ft}^3\text{/lbm}} \tag{5.1}$$

$$\Rightarrow C_{SPC,S} = 32.76 \text{ BTU/°F} \tag{5.2}$$

The sensible capacity sensitivity D_s was not determined theoretically from the simulated cooling coil characteristics presented in Figure 3.4. These characteristics were derived for water while the experimental system uses 30% glycol. When glycol is mixed with the water mass, the heat transfer rate of the cooling coil is reduced affecting the simulated coil characteristics. Determination of the capacity sensitivity was performed experimentally.

Equation 4.27 implies that the capacity sensitivity is the change of the sensible capacity of the cooling coil per unit change in the control signal. A zero signal to the fan is equivalent to 35 Hz and a 100% signal is equivalent to 75 Hz. For this experiment, the valve signal was set to 50%. For that signal, the valve provides 2550 lbm/h of coolant. The system was started. It stabilized at equilibrium conditions at a 40 Hz fan signal. Then the signal was increased manually to 70 Hz. The results are shown in Figure 5.2.

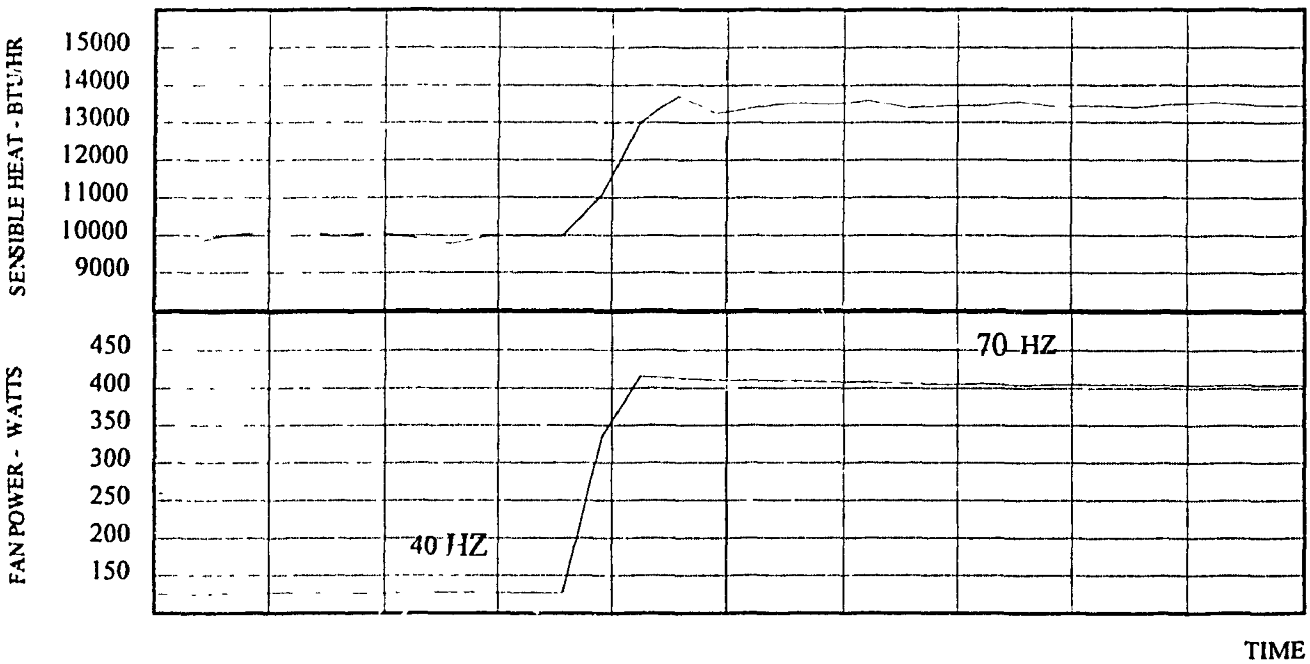

Figure 5.2. Sensible Heat Response to a 75% Change in Fan Control Signal; the Water Valve Signal was set at 50%.

The resulting change in the sensible heat is 3500 BTU/h. Then the capacity sensitivity is

$$D_S = \frac{3500\ \text{BTU/h}}{75\ \%\text{signal}} = 46.67\ \frac{\text{BTU/h}}{\%\text{signal}} \tag{5.3}$$

When the signal increases, the sensible capacity increases therefore the room temperature decreases. Hence D_S is positive and the sensible process sensitivity is negative, and is given by

$$\sigma_S = -\frac{46.67\ \text{BTU/h}\cdot\%\text{signal}}{32.76\ \text{BTU/}^{\circ}\text{F}}\ \frac{\text{h}}{3600\text{sec}} = -3.957\cdot 10^{-4}\ {}^{\circ}\text{F/sec}\cdot\%\text{signal} \tag{5.4}$$

The normalized sensible sensitivity given by Equation 4.37 is

$$\sigma_{N,S} = -\left(3.957\cdot 10^{-4}\ {}^{\circ}\text{F/sec}\cdot\%\text{signal}\right)\cdot\frac{100\ \%\text{signal}}{100\ {}^{\circ}\text{F}} = -0.00039\ \text{sec}^{-1} \tag{5.5}$$

A minimum to maximum temperature difference of 100 °F was selected based on Section 4.2.2. The normalized sensitivity is the parameter required to evaluate the PI coefficients.

5.2.1.1.2 Determination of PI Coefficients

Based on the implementation considerations presented in Chapter 4, a sampling interval of 15 sec. was selected for the fan rotational motor. A damping ratio of 0.8 and a period of 1200 sec. were chosen for the temperature control target response [22]. Using these parameters with Equations 4.81 and 4.82 yields the PI coefficients,

$$K_I = - \frac{4 \cdot \pi^2 \cdot 15 \text{ sec.}}{1200^2 \text{ sec}^2 \cdot (1 - 0.8^2) \cdot 0.00039 \text{ sec}^{-1}} = - 2.93 \tag{5.6}$$

And the corresponding proportional coefficient is

$$K_P = - 2 \cdot (0.8) \cdot \sqrt{\frac{2.93}{15 \text{ sec} \cdot (0.00039 \text{ sec}^{-1})}} = - 35.81 \tag{5.7}$$

The proportional and integral coefficients were inputted to the basic program for space temperature control.

5.2.1.2 Specific Humidity Control

To control the space specific humidity, the latent sensitivity is required. Once this sensitivity is determined, the PI coefficients can be evaluated using Equations 4.81 and 4.82. The determination of the PI coefficients also requires the identification of the target system response parameters by choosing the appropriate damping ratio and period (Section 4.2).

5.2.1.2.1 Latent Sensitivity Evaluation

Equation 4.60 defines the system latent sensitivity as the ratio of the capacity sensitivity to the space latent heat capacity. The latent heat capacity of the room can be evaluated using Equation 4.55 and the air specific volume at set point operating conditions, yielding

$$C_{SPC,L} = m_{DA} \cdot h_o = \frac{V}{v_{DA}} \cdot h_o = \frac{2640 \ ft^3}{13.7 \ ft^3/lbm} \cdot 1061 \ BTU/lbm \tag{5.8}$$

then

$$C_{SPC,L} = 204455.5 \ BTU \tag{5.9}$$

Equation 4.52 implies that the latent capacity sensitivity is the change of the latent capacity of the cooling coil per change in the control signal. A zero signal to the valve provides a chilled water flow rate of 2100 lbm/h while a 100% signal provides a flow rate of 3000 lbm/h. The system was started. It was stabilized at equilibrium conditions with a chilled water flow rate of 2500 lbm/h for 45% signal. Then the signal was increased manually to provide 3000 lbm/h of water at 100% signal. The fan was set at 55 Hz, i.e., 50% signal. The results are shown in Figure 5.3. The resulting change in the sensible heat is 500 BTU/h. Then the latent capacity sensitivity is

$$D_L = \frac{500 \ BTU/h}{55 \ \%signal} = 9.1 \ \frac{BTU/h}{\%signal} \tag{5.10}$$

When the signal increases, the latent capacity increases while the room specific humidity decreases. Therefore D_L is positive and the system latent process sensitivity is negative given by

$$\sigma_L = - \frac{9.1 \ BTU/h \cdot \%signal}{204455.5 \ BTU} \cdot \frac{h}{3600 \ sec.} = - 1.23 \cdot 10^{8} \ 1/sec \cdot \%signal \tag{5.11}$$

The normalized latent sensitivity given by Equation 4.63 is

$$\sigma_{N,L} = -\left(1.23 \cdot 10^{-8}\ 1/\mathrm{sec} \cdot \%\mathrm{signal}\right) \cdot \frac{100\ \%\mathrm{signal}}{0.023} = -\ 5.35 \cdot 10^{\ 5}\ \mathrm{sec}^{-1} \qquad (5.12)$$

A minimum to maximum specific humidity difference of 0.023 was selected based on Section 4.2.2. The normalized sensitivity is the parameter required to determine the PI coefficients.

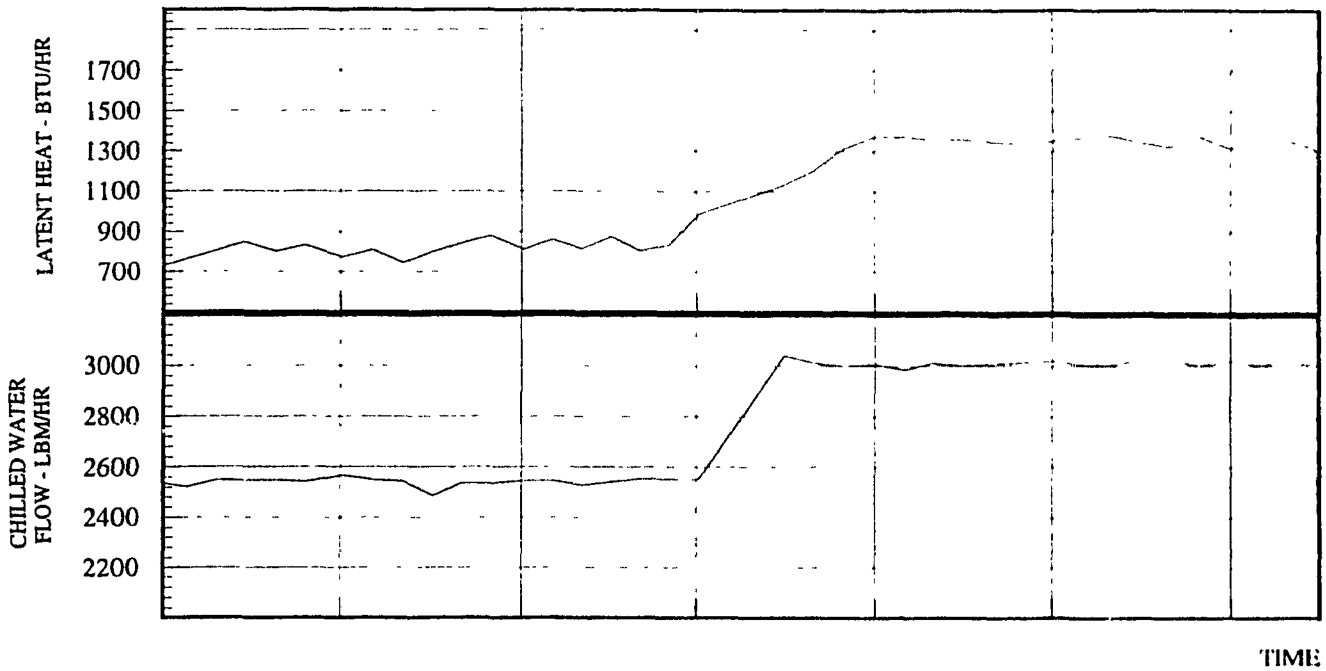

Figure 5.3. Latent Heat Response to a 55% Change in Valve Control Signal, the Fan Signal was set at 50%.

5.2.1.2.2 Determination of PI Coefficients

Based on the implementation considerations presented in Chapter 4, a sampling interval of 60 sec. was selected for the valve positional motor. A damping ratio of 0.5 and a period of 4200 sec. were chosen for the humidity control target response [22]. Using these data with Equations 4.81 and 4.82 gives the PI coefficients,

$$K_I = -\frac{4\cdot\pi^2\cdot 60\ \text{sec.}}{4200^2\ \text{sec}^2\cdot(1-0.5^2)\cdot 5.35\cdot 10^{-5}\ \text{sec}^{-1}} = -3.37 \tag{5.13}$$

The corresponding proportional coefficient is

$$K_P = -2\cdot(0.5)\cdot\sqrt{\frac{3.37}{60\ \text{sec}\cdot(5.35\cdot 10^{-5}\ \text{sec}^{-1})}} = -32.40 \tag{5.14}$$

The proportional and integral coefficients were inputted to the basic program for space humidity control.

5.2.1.3 Simultaneous Control of Space Temperature and Specific Humidity

The PI coefficients for space temperature and specific humidity control determined in Section 5.2.1.2 were inputted to the basic program. A temperature set point of 72 °F and a specific humidity set point of 0.009 were also inputted to the basic program. The system was run for five hours. During the running time, variations in the load were performed by turning on/off a 1500 watts room heater and by turning on/off the wall fans. Figure 5.4 shows the resulting response of the system.

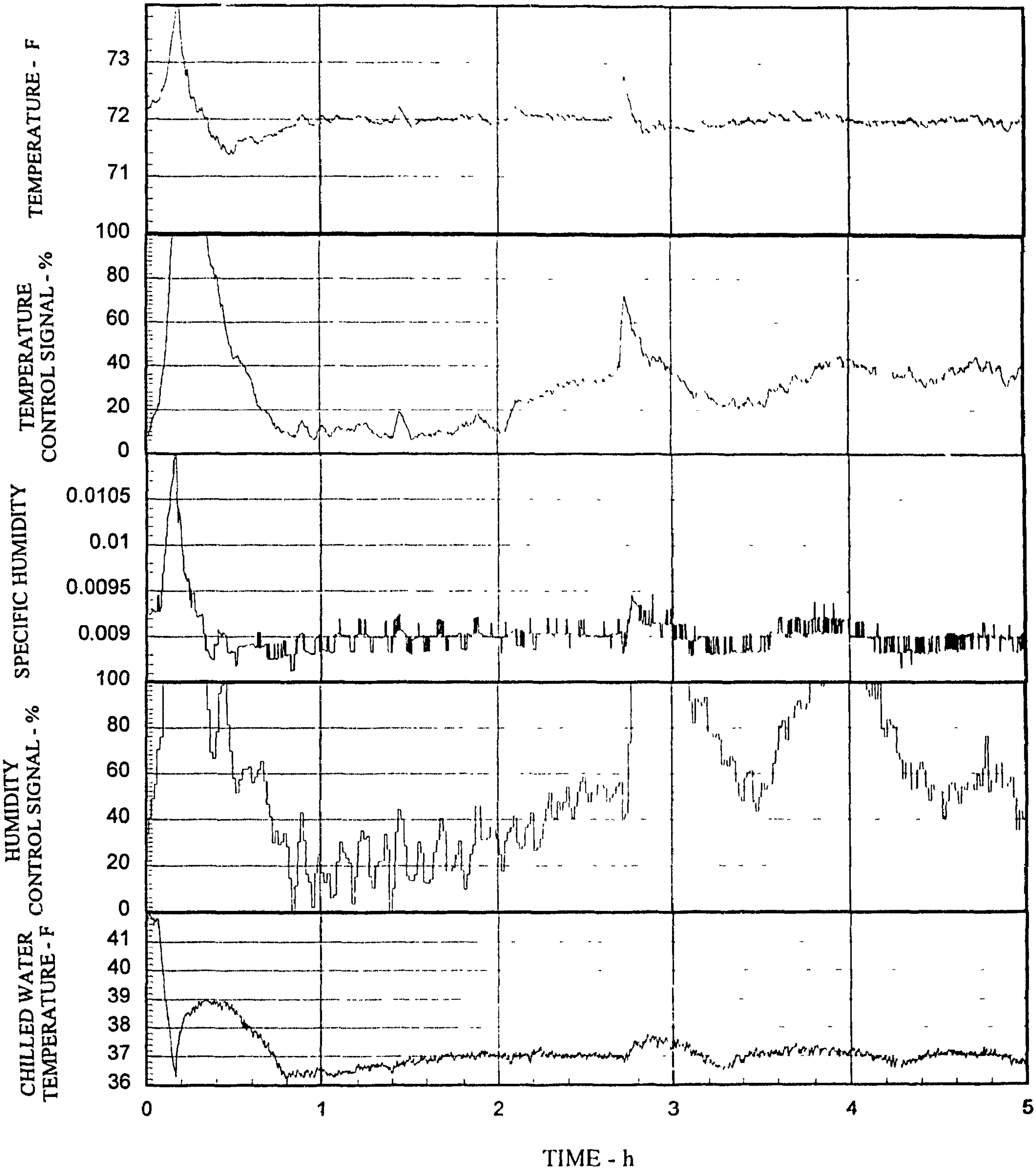

Figure 5.4. Space Temperature and Humidity Control at 72°F and 0.009 using a Throttling Water Valve.

The peaks in the temperature and humidity responses imply a change in the load. Both humidity and temperature responses were underdamped with a slight undershoot. The set points were attained within forty minutes. The control signals follow the load accordingly and the set points were maintained. The chilled water temperature was about 37 °F throughout the experiment. Stepping in the humidity control signal was a result of the stepped valve motor response. The valve motor was insensitive to small changes in current inputs. Electrical noise was significant in the humidity reading. The humidity control signal was sensitive to the noise. Temperature control was between +/- 0.2 °F. Humidity control was between +/- 0.0002.

5.2.2 Space Temperature and Humidity Control Experiments with mixing water valve

A schematic of the physical system that implements the control strategy of space temperature and humidity using a mixing water valve is shown in Figure 5.5. The mixed water temperature varies depending on the position of the valve. Similar procedure to those followed with the throttling valve experiments was used for this experiment. The sensible and latent capacity sensitivities were determined from the open loop test data shown in Figures 5.6 and 5.7.

5.2.2.1 Determination of PI Coefficients

The sensible capacity sensitivity obtained from Figure 5.6 was 32 BTU/h.%signal. The experiment was conducted at 50% valve signal. The fan signal was increased from 50

Hz to 70 Hz, i.e., increased by 50%. The system sensible sensitivity was calculated to be $-2.713 \cdot 10^{-4}$ °F/sec.%signal. And the normalized sensitivity was obtained as $-2.713 \cdot 10^{-4}$ sec.$^{-1}$. A damping ratio of 0.6 and a period of 1200 sec. were selected. The integral coefficient, K_I, was calculated from theses parameters according to Equation 4.81.

$$K_I = -\frac{4 \cdot \pi^2 \cdot 15 \text{ sec.}}{1200^2 \text{ sec}^2 \cdot (1 - 0.6^2) \cdot 0.0002713 \text{ sec}^{-1}} = -2.37 \tag{5.15}$$

The corresponding proportional controller gain, K_P, was calculated using Equation 4.82 as

$$K_P = -2 \cdot (0.6) \cdot \sqrt{\frac{2.37}{15 \text{ sec} \cdot (0.0002713 \text{ sec}^{-1})}} = -29 \tag{5.16}$$

The latent capacity sensitivity obtained from figure 5.7 was 10 BTU/h.%signal. The experiment was conducted at 50% fan signal. The mixing valve signal was decreased from 100% to 25%, i.e., decreased by 75%. As the signal decreases, the valve closes decreasing the supply and increasing the return. The recirculation coefficient is minimum when the valve is fully open. The system latent sensitivity was calculated to be $-1.36 \cdot 10^{-8}$ (sec.%signal)$^{-1}$. The normalized sensitivity was obtained to be $5.92 \cdot 10^{-5}$ sec.$^{-1}$. A damping ratio of 0.5 and a period of 4200 sec. were selected. The integral coefficient, K_I, was calculated from theses parameters according to Equation 4.81.

$$K_I = -\frac{4 \cdot \pi^2 \cdot 60 \text{ sec.}}{4200^2 \text{ sec}^2 \cdot (1 - 0.5^2) \cdot 5.92 \cdot 10^{-5} \text{ sec}^{-1}} = -3.02 \tag{5.17}$$

The proportional coefficient, K_p, was calculated using Equation 4.82 as

$$K_p = -2\cdot(0.5)\cdot\sqrt{\frac{3.02}{60\ \text{sec}\cdot(5.92\cdot 10^{-5}\ \text{sec}^{-1})}} = -29.16 \tag{5.18}$$

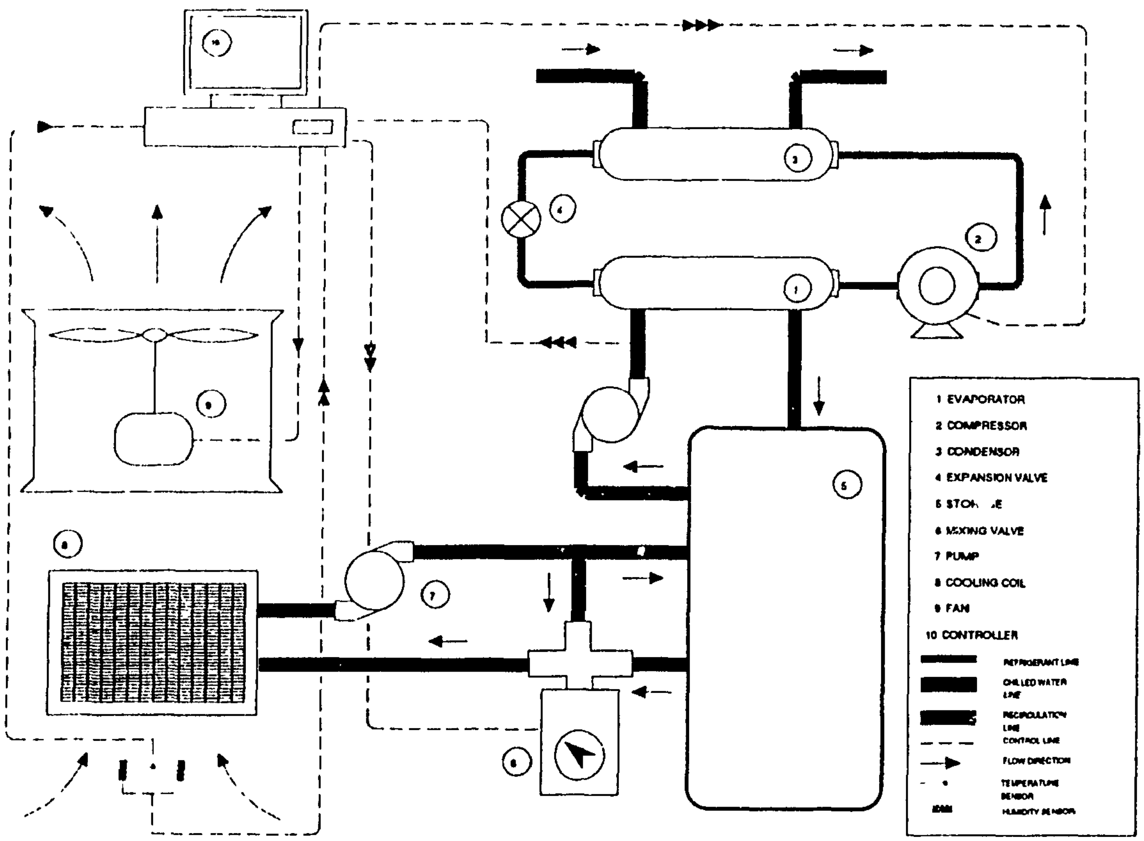

Figure 5.5 Schematic Diagram of the Physical Control System with a Mixing Water Valve.

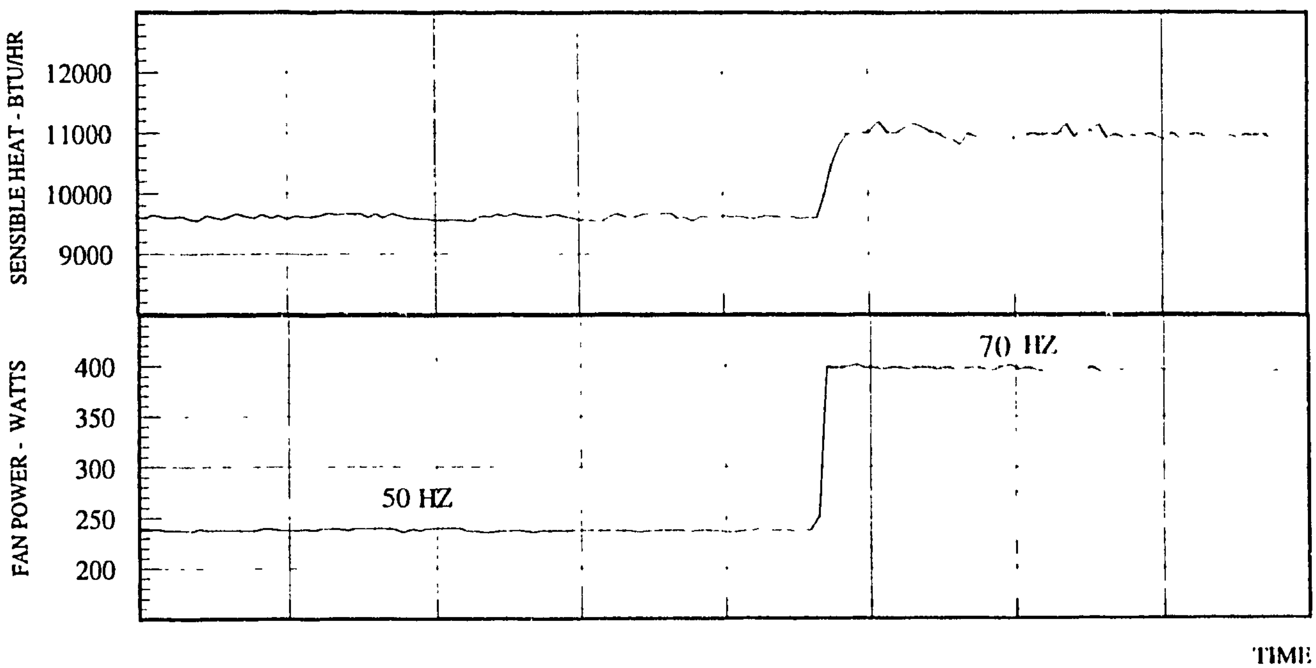

Figure 5.6. Sensible Heat Response to a 50% Change in Fan Control Signal, the Valve Control Signal was set at 50%

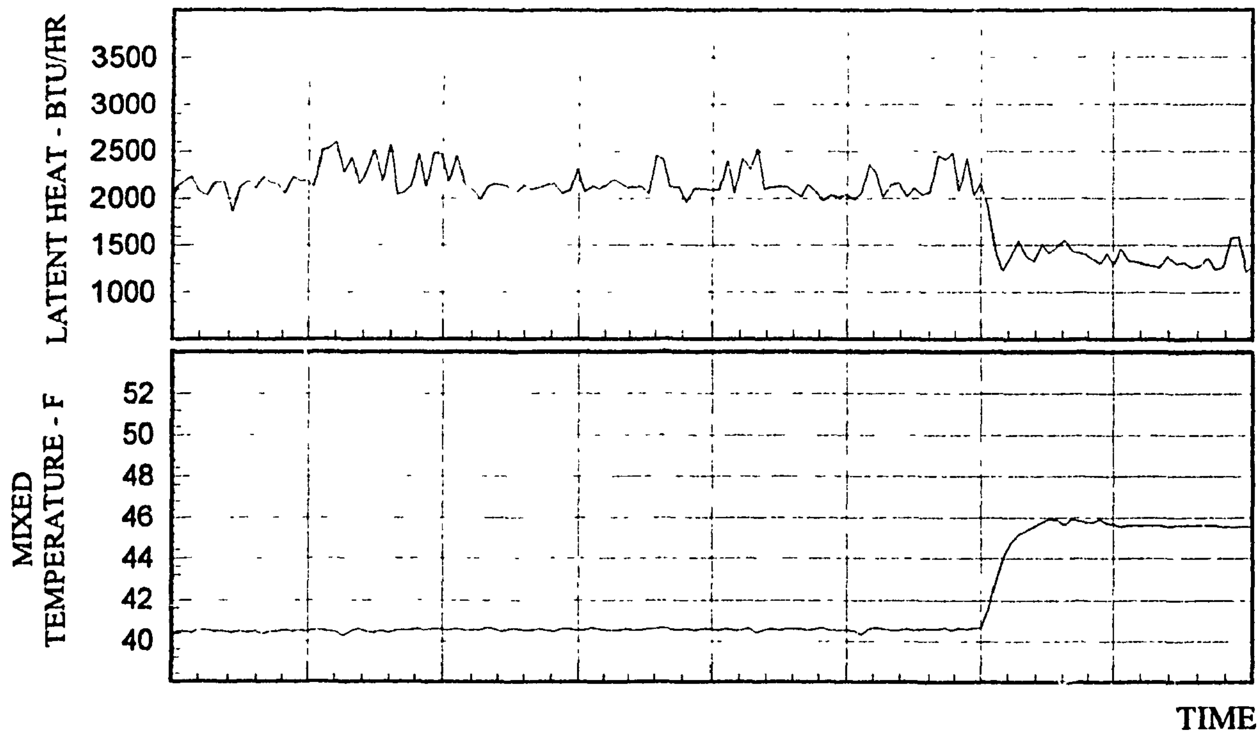

Figure 5.7. Latent Heat Response to a 75% Decrease in Valve Control Signal; the Fan Signal was set at 50%.

5.2.2.2 Simultaneous Control of Space Temperature and Specific Humidity

The PI coefficients for space temperature and specific humidity control determined in the above section were inputted to the basic program. A temperature set point of 72 °F and a specific humidity set point of 0.009 were also inputted to the basic program. Similar conditions to those for the experiment with the throttling water valve were maintained. Figure 5.8 shows the resulting response of the system. Both humidity and temperature responses were underdamped with a slight undershoot. The set points were attained within less than half hour. The control signals follow the load continuously and the set points were maintained. The mixed water temperature was varying accordingly with the valve control signal. Stepping in the humidity control signal was a result of the stepped valve motor response. Step response is also significant in the mixed water temperature response. Electrical noise was significant in the humidity reading. The humidity control signal was sensitive to the noise. To reduce electrical noise, numerical filtering was added to the control program. Temperature control was between +/- 0.1 °F. Humidity control was between +/- 0.0002. The supply water temperature was constant at 37 °F throughout the experiment.

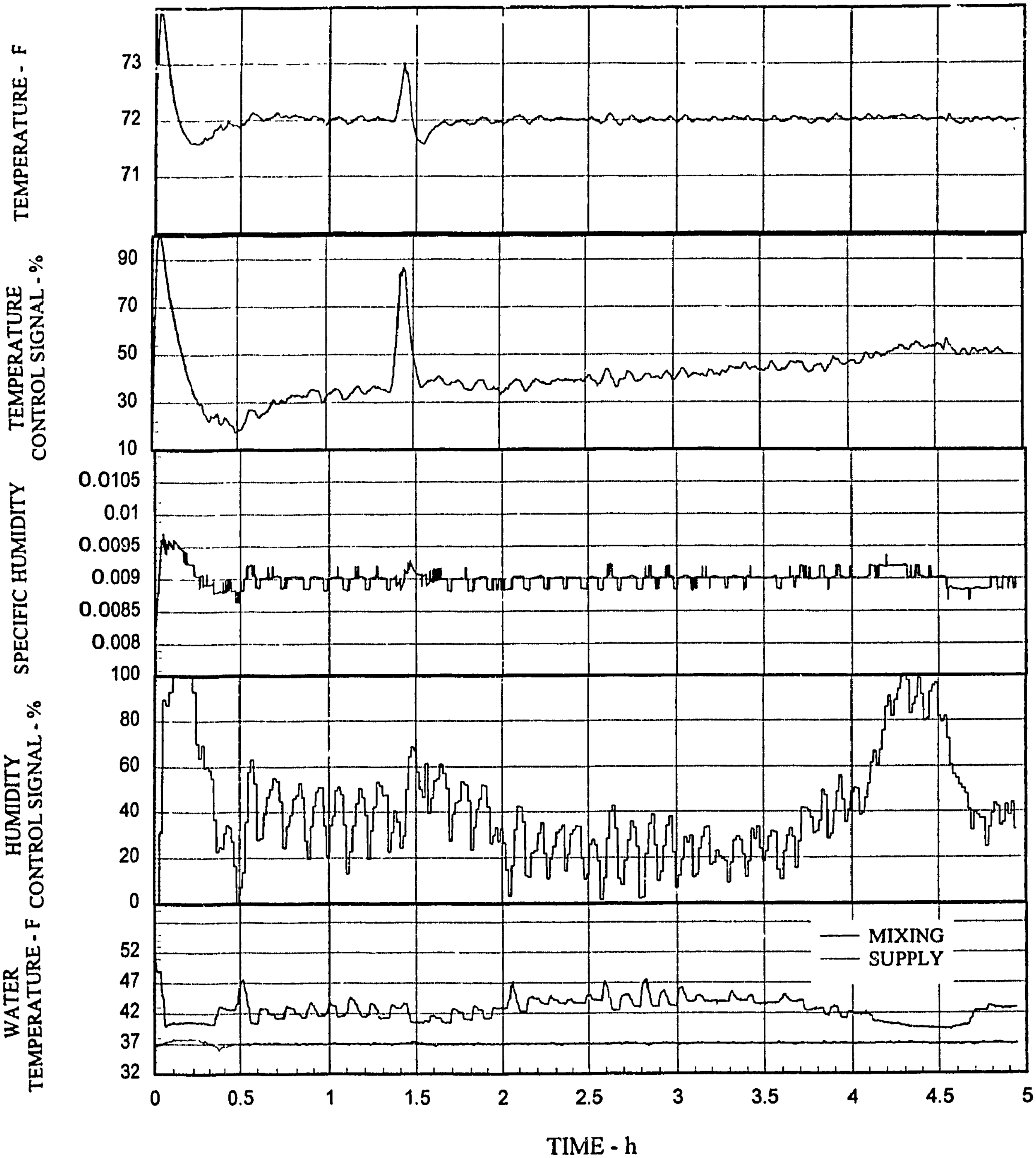

Figure 5.8. Space Temperature and Humidity Control at 72°F and 0.009 using a Mixing Water Valve.

5.2.3 Supply Air Temperature and Humidity Control using a Throttling Water Valve

A schematic of the physical system to implement the control strategy of supply air temperature and humidity is shown in Figure 5.9. The temperature sensor at the outlet of the cooling coil is connected to the PI controller that modulates the fan motor through a variable frequency inverter. The humidity sensor at the outlet of the coil is connected to the PI controller that modulates the throttling valve motor. The theory developed in Chapter 4 for first order systems, as well as the implementation considerations presented were used to select the system characteristic parameters leading to the determination of PI coefficients.

5.2.3.1 Temperature Control

To control the supply air temperature, the sensible sensitivity is required. Once this sensitivity is determined, the PI coefficients can be evaluated using Equation 4.116. The determination of the PI coefficients also requires the selection of the target system response parameters by choosing the appropriate attenuation factor (Section 4.2.3.2).

5.2.3.1.1 Sensible Sensitivity Evaluation

Equation 4.89 implies that the system sensible sensitivity is the change of the air temperature leaving the cooling coil per change in the control signal. The system was started. It was stabilized at equilibrium conditions at 70 Hz fan signal. Then the signal was decreased manually to 50 Hz. The valve signal was set to 50%.

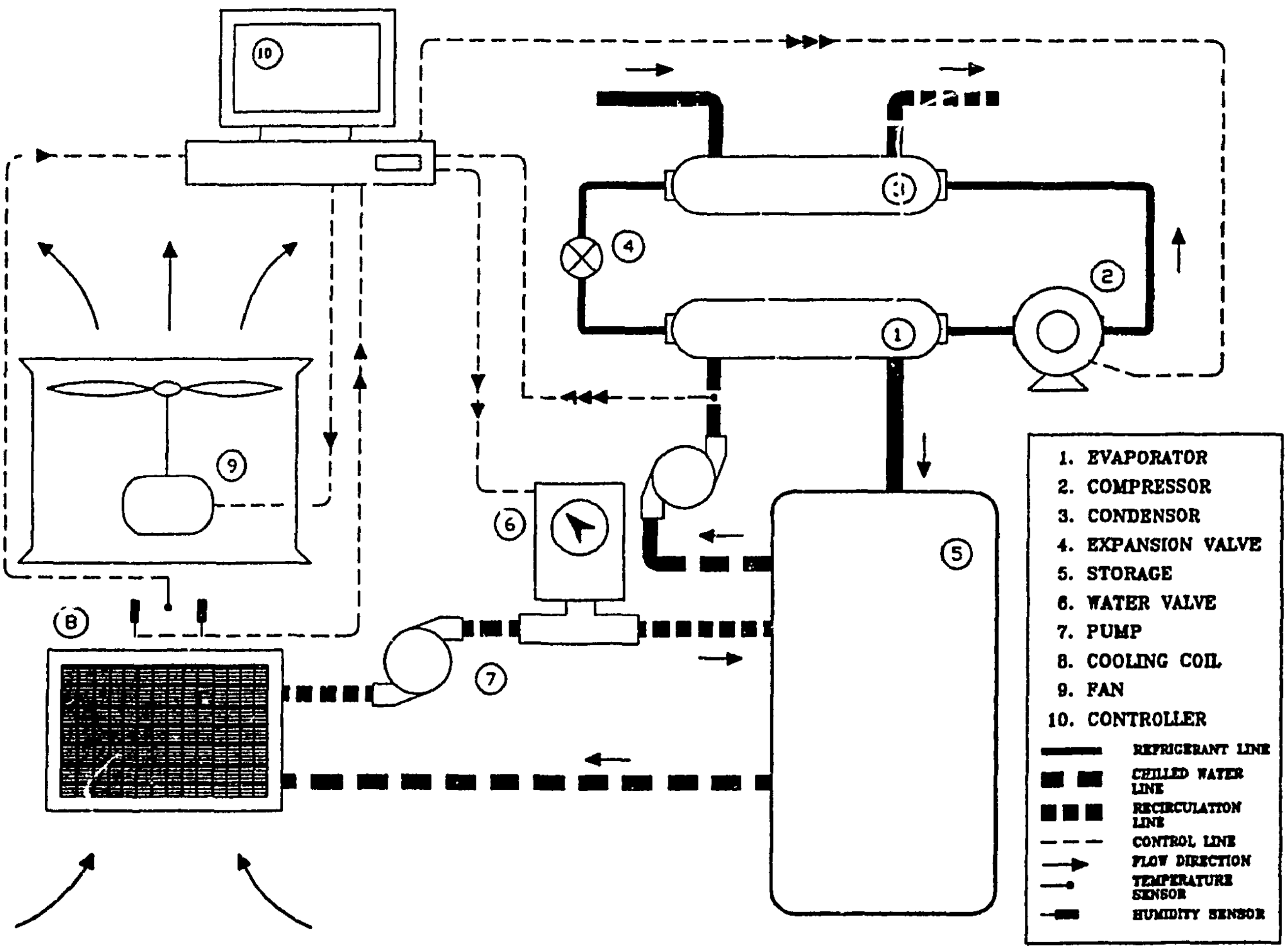

Figure 5.9 Schematic representation of the system used for Supply Air Temperature and Humidity Control using a Throttling Water Valve.

The results are shown in Figure 5.10. The resulting change in the air temperature was 3.9 °F for a 50% change in the control signal. Then the sensible sensitivity is

$$\sigma_S = \frac{3.9\ ^{\circ}\mathrm{F}}{50\ \%\mathrm{signal}} = 0.078\ \ ^{\circ}\mathrm{F}/\%\mathrm{signal} \qquad (5.19)$$

When the signal decreases, the air temperature decreases. Therefore, the sensible sensitivity is positive. The normalized sensible sensitivity is also positive given by Equation 4.93 as

$$\sigma_{N,S} = (0.078\ \ ^{\circ}\mathrm{F}/\%\mathrm{signal}) \cdot \frac{100\ \ \%\mathrm{signal}}{100\ ^{\circ}\mathrm{F}} = 0.078 \qquad (5.20)$$

The normalized sensitivity is the dimensionless parameter required to evaluate the PI coefficients.

5.2.3.1.2 Determination of PI Coefficients

Based on the implementation considerations presented in Chapter 4, an attenuation factor of 0.05 was selected. A proportional coefficient of 0.9 was chosen for the temperature control model. Using these parameters with Equations 4.116 and 5.20 gives the integral coefficient,

$$K_I = 0.05 \cdot (0.9 + \frac{1}{0.078}) = 0.69 \qquad (5.21)$$

The proportional and integral coefficients were inputted to the basic program for supply air temperature control.

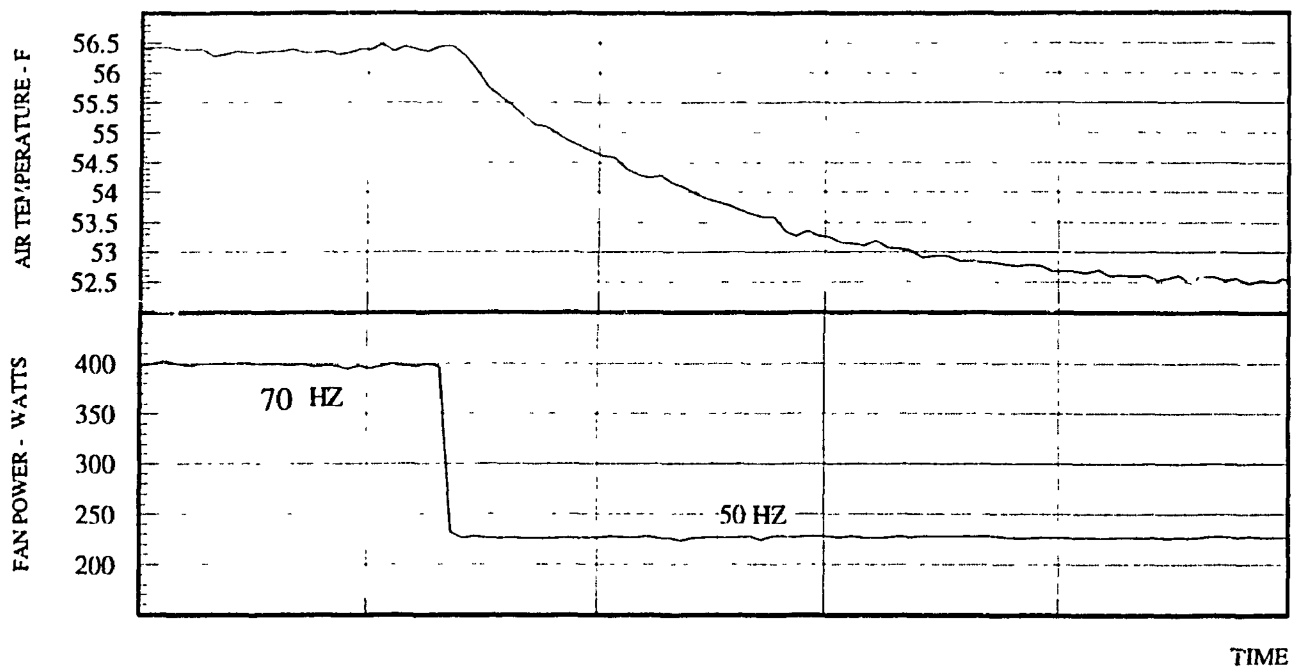

Figure 5.10. Supply Air Temperature Response to a 50% Change in Fan Control Signal; the Valve Signal was set at 50%.

5.2.3.2 Specific Humidity Control

To control the supply air specific humidity, the latent sensitivity is required. Once this sensitivity is determined, the PI coefficients can be evaluated using Equation 4.116. The determination of the PI coefficients also requires the identification of the target system response parameters by choosing the appropriate attenuation factor (Section 4.2).

5.2.3.2.1 Latent Sensitivity Evaluation

Equation 4.103 implies that the system latent sensitivity is the change of the supply air specific humidity per change in the control signal. The system was started. It was stabilized at certain equilibrium conditions with a chilled water flow rate of 2650 lbm/h. Then the signal was increased manually to provide 2850 lbm/h of water. The fan was operating at 50% signal. The results are shown in Figure 5.11.The change in the air specific humidity was 0.00015 for a 22% signal change. Then the latent sensitivity is

$$\sigma_L = -\frac{0.00015}{22\ \%\text{signal}} = -6.8\cdot 10^{-6}\ \%\text{signal}^{-1} \tag{5.22}$$

When the signal increases, the specific humidity decreases. Therefore, the latent sensitivity is negative. The normalized latent sensitivity is also negative and given by Equation 4.107 as

$$\sigma_{N,L} = -\ (6.8\cdot 10^{-6}\ 1/\%\text{signal})\cdot\frac{100\ \%\text{signal}}{0.023} = -\ 0.03 \tag{5.23}$$

The normalized sensitivity is the dimensionless parameter required to determine the PI coefficients.

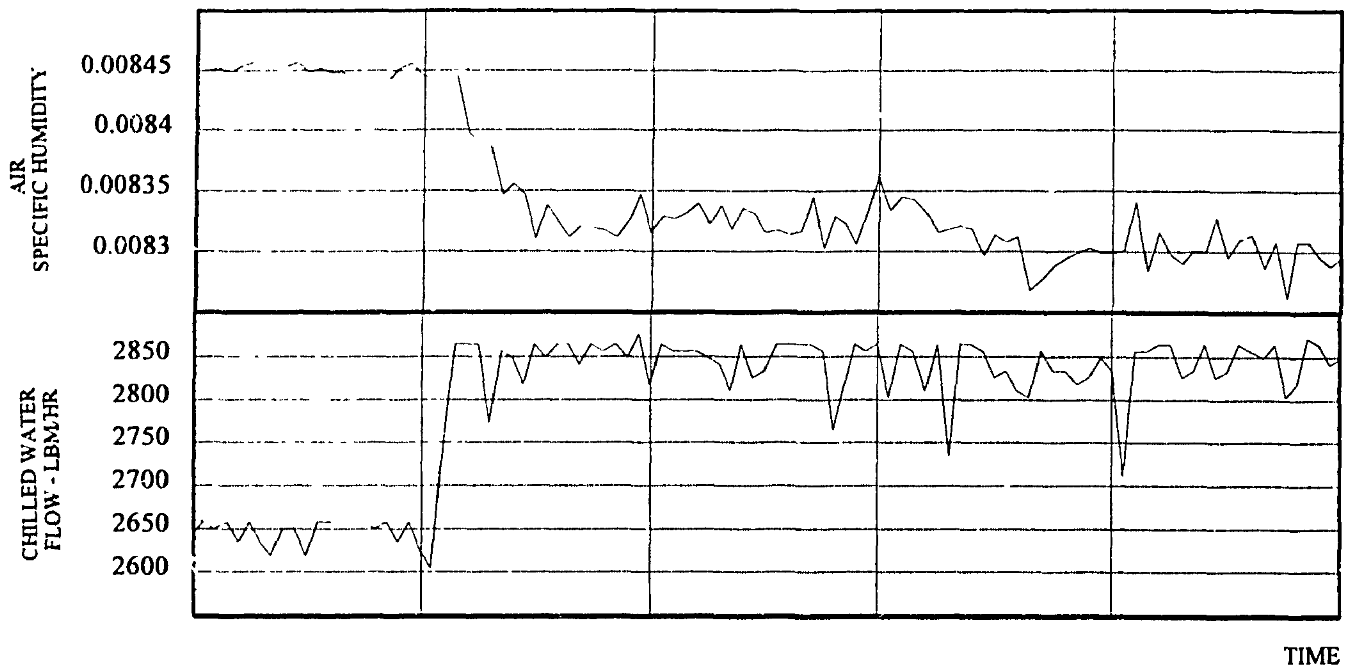

Figure 5.11. Supply Air Specific Humidity Response to a 22% Change in Valve Control Signal; the Fan Control Signal was set at 50%.

5.2.3.2.2 Determination of PI Coefficients

Based on the implementation considerations presented in Chapter 4, an attenuation factor of 0.05 was selected. A proportional coefficient of - 0.9 was chosen for the specific humidity control model. Using these parameters with Equation 4.116 gives the integral coefficient,

$$K_I = 0.05 \cdot \left(-0.9 + \frac{1}{-0.03} \right) = -1.71 \qquad (5.24)$$

The proportional and integral coefficients were inputted to the basic program for air specific humidity control.

5.2.3.3 Simultaneous Control of Supply Air Temperature and Specific Humidity

The PI coefficients for air temperature and specific humidity control, determined by Equations 5.21 and 5.24, were inputted to the control program. A temperature set point of 57 °F and a specific humidity set point of 0.009 were also inputted to the control program. The system was run for one and half hour. During the running time, variations in the load were performed by turning on/off a 1500 watts room heater and by turning on/off the wall fans. Figure 5.12 shows the resulting response of the system. The set points were attained within short time. The control signal followed the load continuously and the set points were perfectly maintained. The chilled water temperature was about 37 °F throughout the experiment.

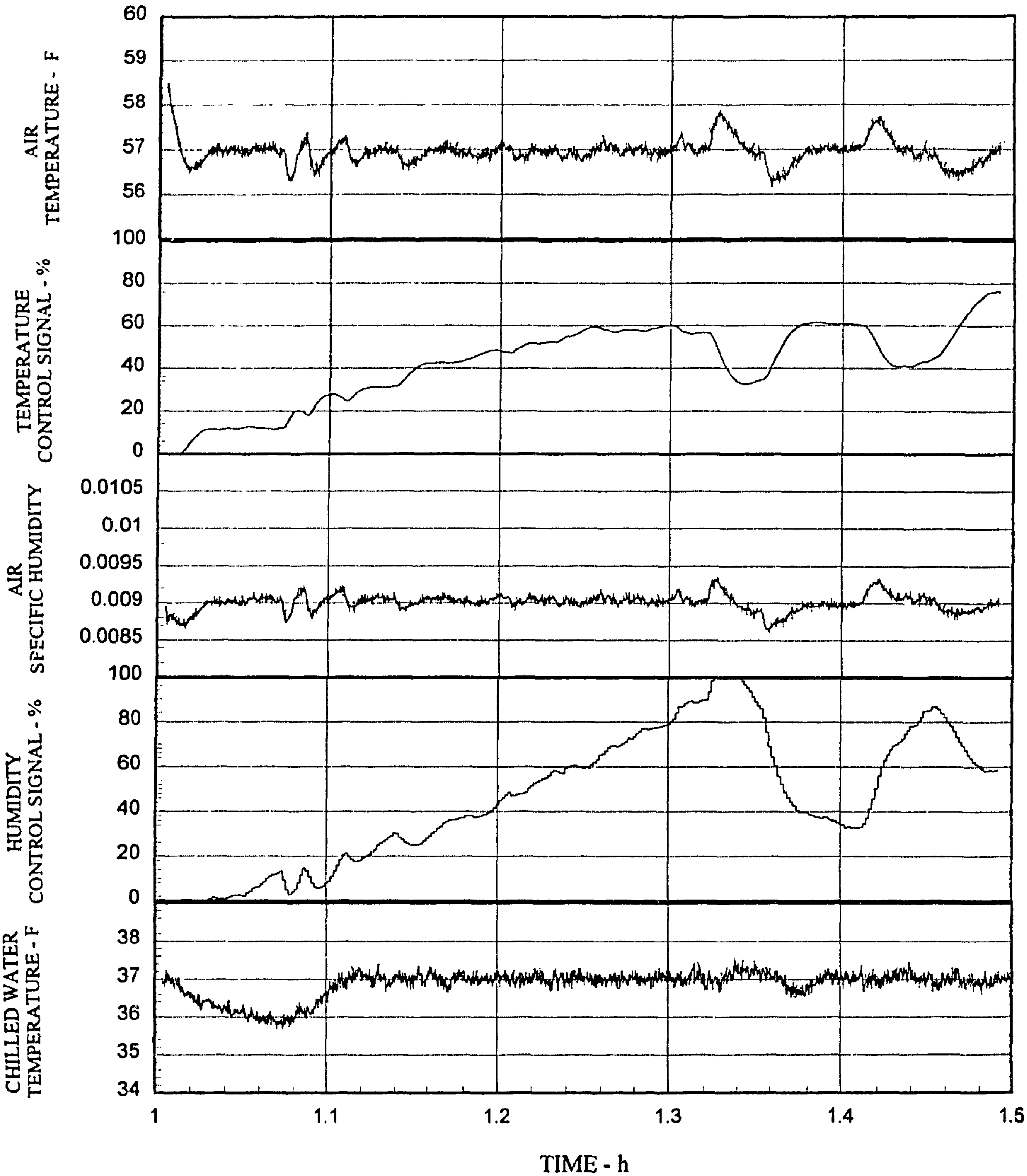

Figure 5.12. Supply Air Temperature and Humidity Control at 57°F and 0.009 using a Throttling Water Valve.

5.2.4 Supply Air Temperature and Humidity Control using a Mixing Water Valve.

A schematic of the physical system that implements the control strategy of supply air temperature and humidity using mixing water valve is shown in Figure 5.13. The mixed water temperature varied depending on the position of the valve. Similar procedure to that followed with the throttling valve experiments was used for this experiment. The sensible and latent system sensitivities were determined from the open loop test data shown in Figures 5.14 and 5.15.

5.2.4.1 Determination of PI Coefficients

The sensible sensitivity obtained from Figure 5.14 was 0.054 °F/%signal. The system normalized sensitivity was obtained as 0.054. An attenuation factor of 0.05 was chosen. For a proportional controller of 0.9, the integral controller gain, K_I, was calculated from theses parameters based on Equation 4.116 as

$$K_I = 0.05 \cdot (0.9 + \frac{1}{0.054}) = 0.97 \tag{5.25}$$

The latent system sensitivity obtained from Figure 5.15 was $-8.5 \cdot 10^{-6}$ $(\%signal)^{1}$. The system normalized sensitivity was obtained as -0.037. For a proportional controller gain of -0.9 and an attenuation factor of 0.05, the integral gain K_I was calculated based on Equation 4.116

$$K_I = 0.05 \cdot \left(-0.9 + \frac{1}{-0.037} \right) = -1.39 \tag{5.26}$$

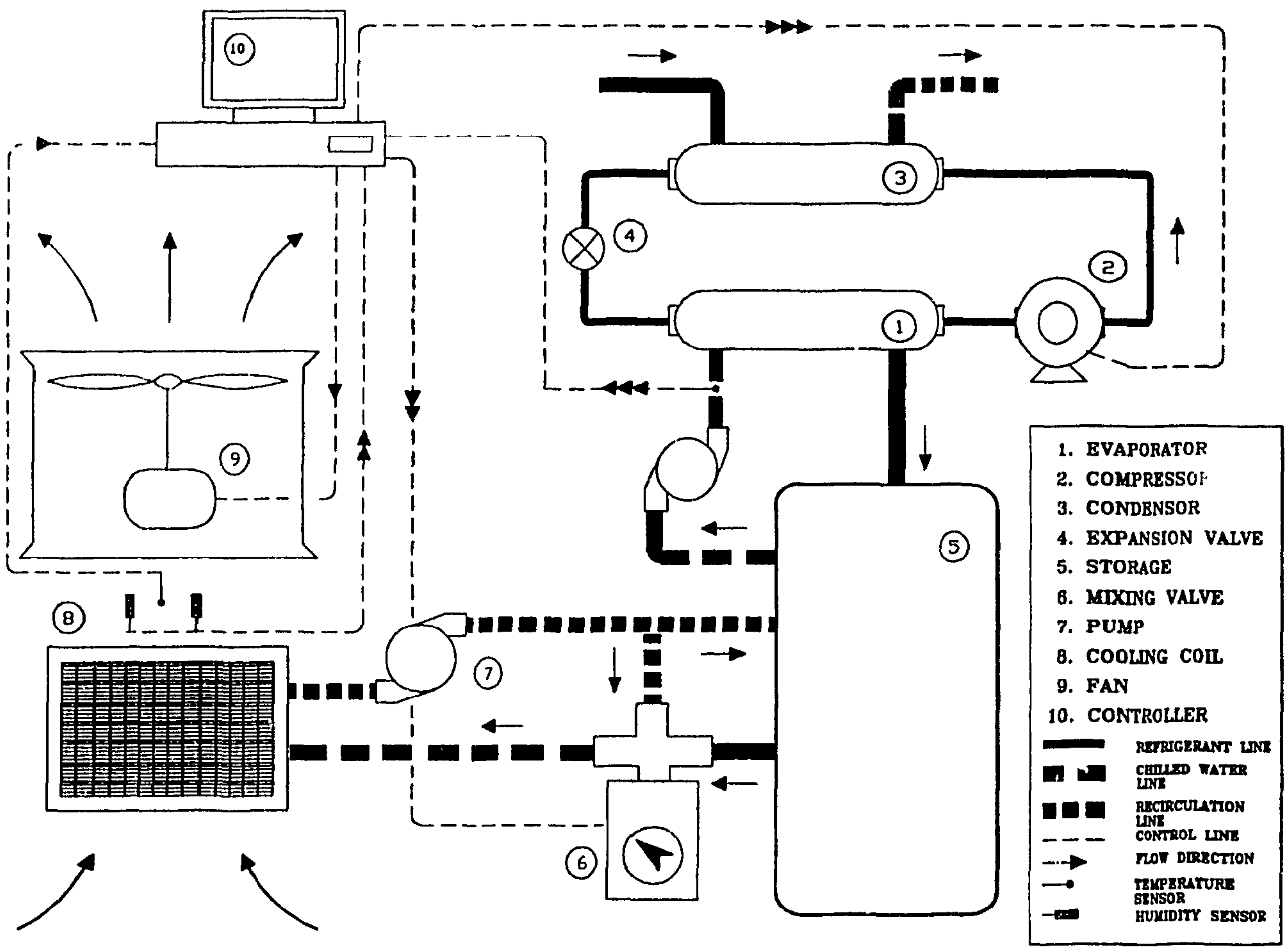

Figure 5.13. Schematic Diagram of the Physical Control System using Mixing Water Valve.

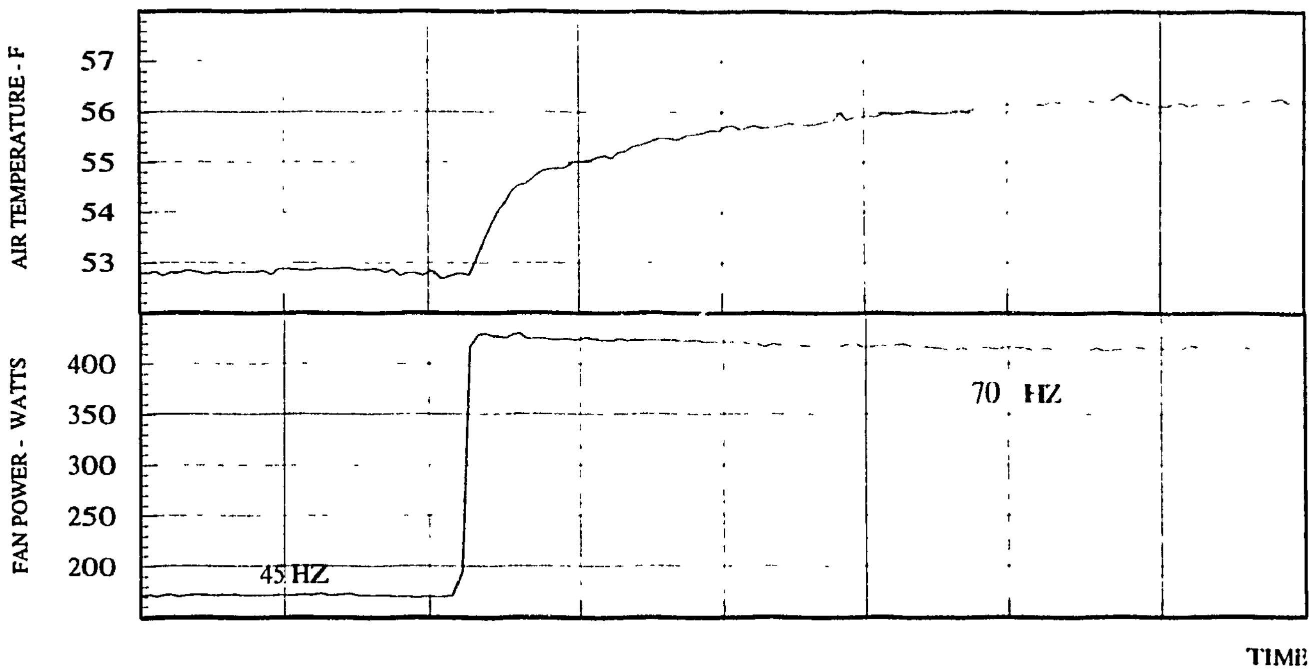

Figure 5.14. Supply Air Temperature Response to a 60% Change in Fan Control Signal, the Valve Control Signal was set at 50%

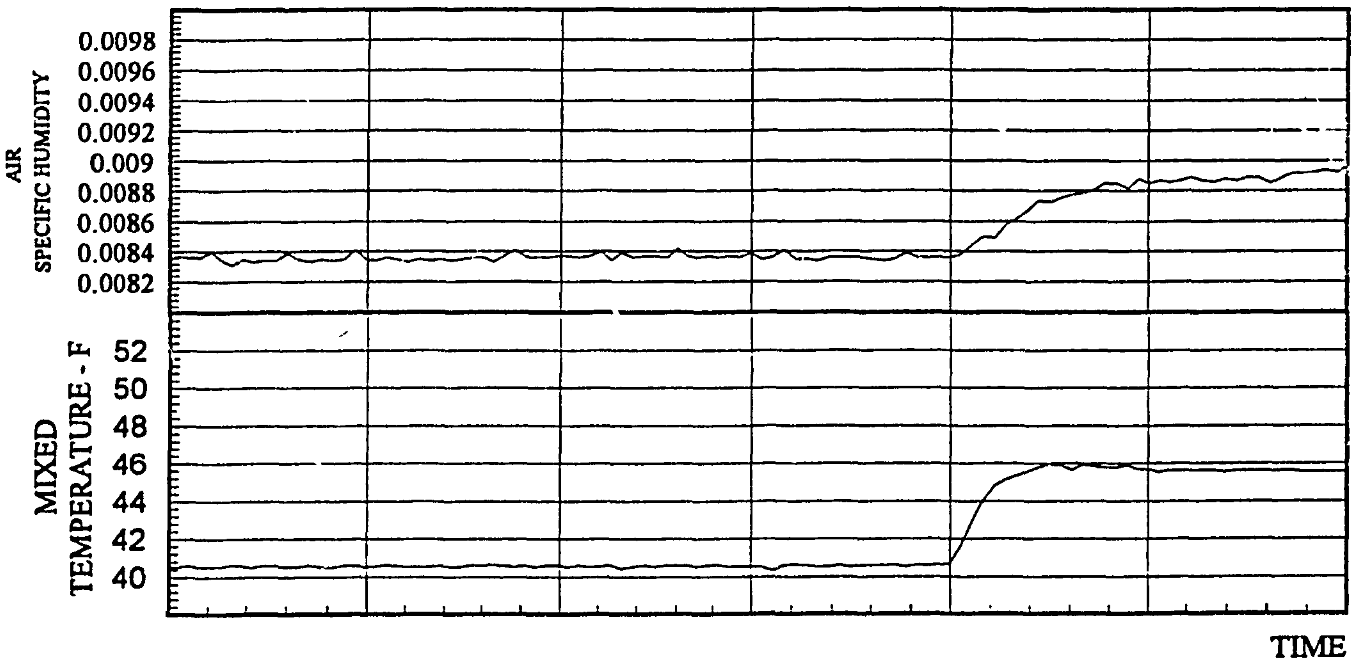

Figure 5.15 Supply Air Specific Humidity Response to a 75% Change in Valve Control Signal; the Fan Signal was set at 50%.

5.2.4.2 Simultaneous Control of Air Temperature and Specific Humidity

The PI coefficients for supply air temperature and specific humidity control determined in the preceding section were inputted to the control program. A temperature set point of 57 °F and a specific humidity set point of 0.009 were also inputted to the control program. The system was run for five hours. During the running time, variations in the load were performed by turning on/off a 1500 watts room heater and by turning on/off the wall fans. Figure 5.16 shows the resulting response of the system. The set points were attained within short time. The control signals follow the load continuously and the set points were maintained. The chilled water temperature was about 37 °F throughout the experiment. The mixed water temperature decreases accordingly to compensate for the increase in humidity. Stepping in the mixed water response is due to the stepped response of the valve motor. Air temperature was controlled within +/- 0.2 °F. Air humidity was controlled within +/- 0.0001.

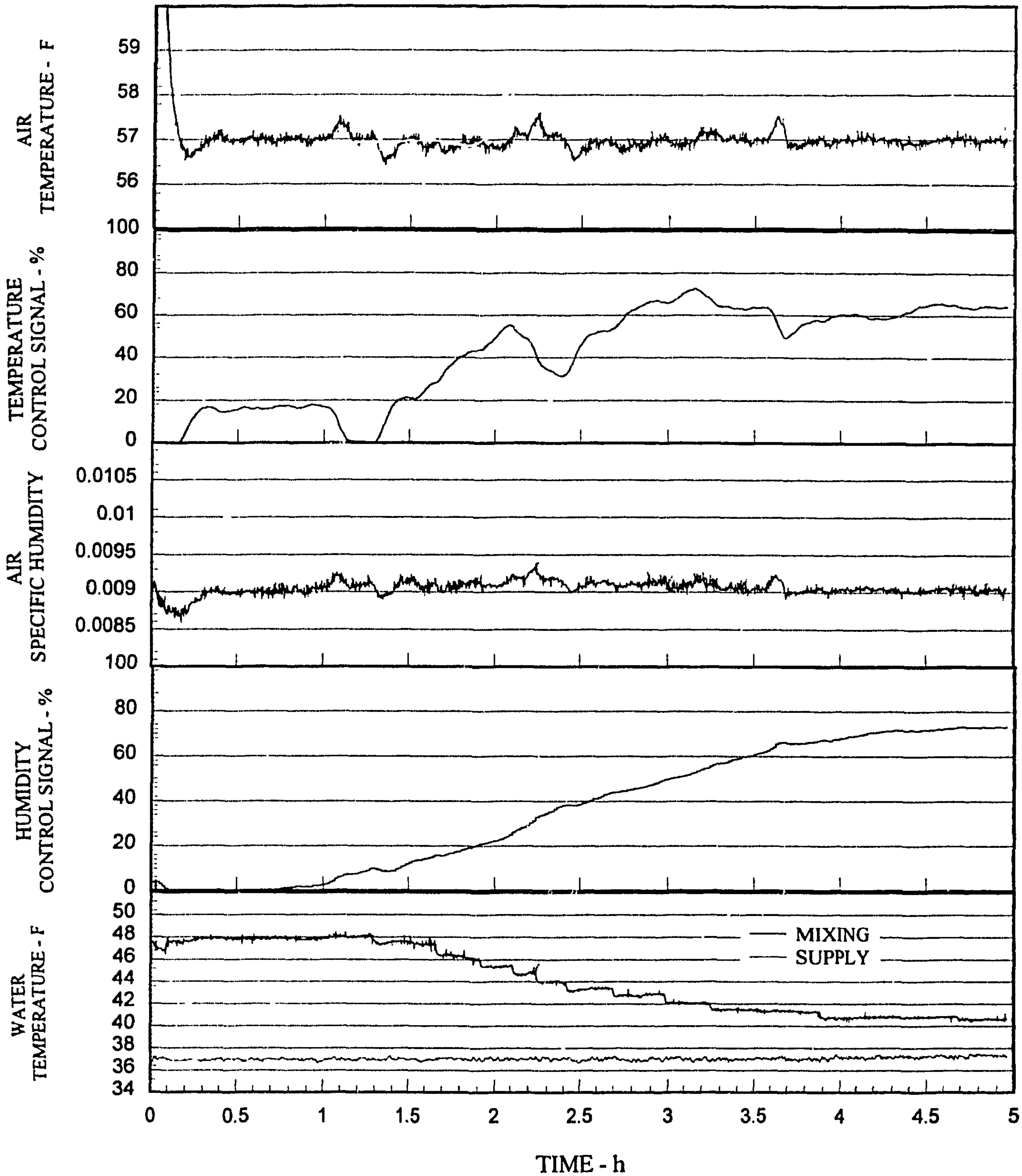

Figure 5.16. Supply Air Temperature and Humidity Control at 57°F and 0.009 using a Mixing Water Valve.

5.3 Control Techniques with Variable Set Point

Comfort level in environmental spaces depends on the temperature, the humidity, the air circulation, and the radiation conditions. A cooling coil is generally designed to provide a space temperature of 78°F and a relative humidity of 50%. At these operating conditions, human comfort levels are satisfied. When the space condition becomes too dry, i.e., the relative humidity decreases such as 40%, an increase in the room temperature to 82°F for example will also satisfy comfort levels. Another way to improve the performance of HVAC system is the use of optimal control by varying the set point maintaining satisfactory comfort conditions. Variable set point control will assume a set point as a function of the relative humidity, the cfm and the radiation conditions.

Some experiments were performed for the purpose of investigating comfort control strategies. The PI coefficients determined in Section 5.2.2.1 were used for implementation and only the temperature was controlled. The basic control program was modified to operate the system based on variable set point. The control signal modulating the fan motor speed was a function of the error between the set point temperature and the measured space temperature.

Two control strategies were used. The first strategy considers the set point as a function of a reference set point value and the cfm. The cfm is a function of the control signal hence the set point was considered as a function of the control signal. Because the control signal varies between 0 and 100%, the variable set point is expressed as

$$T_{SP} = T_{SPo} + 0.04 \cdot (SG) \tag{5.27}$$

At 0% control signal, the set point temperature is equal to the reference set point value. At 100% control signal, the set point temperature will be equivalent to the reference set point value plus four degrees. The experimental performance of this control strategy (Figure 5.17) results in a unstable system response. The set point being a function of the signal oscillates tremendously between higher and lower values. The error follows as well as the control signal. The control strategy fails.

The second strategy expresses the set point temperature as a function of a reference set point and the relative humidity such as

$$T_{SP} = T_{SPo} - 15 \cdot (\phi) \tag{5.28}$$

The reference set point temperature was selected to be 80 °F. The space relative humidity varied between 40 and 65%. The system was started. It was stabilized at 50% fixed water valve position. The space relative humidity was varied by turning off/on a room humidifier. The system response is shown in Figure 5.18. At constant room humidity, the room temperature followed the set point continuously. As the humidity decreased, the set point increased to peak at 73°F and the room temperature followed. When the humidity oscillated, the set point oscillated and so did the room temperature. The control strategy of variable set point temperature versus relative humidity represents a curve on the psychometric chart. The curve can be regarded as a comfort curve along which the room maintains acceptable comfort levels. Increase in temperature for a decrease in humidity should result in energy conservation. It is not the objective of this paper to discuss energy conservation.

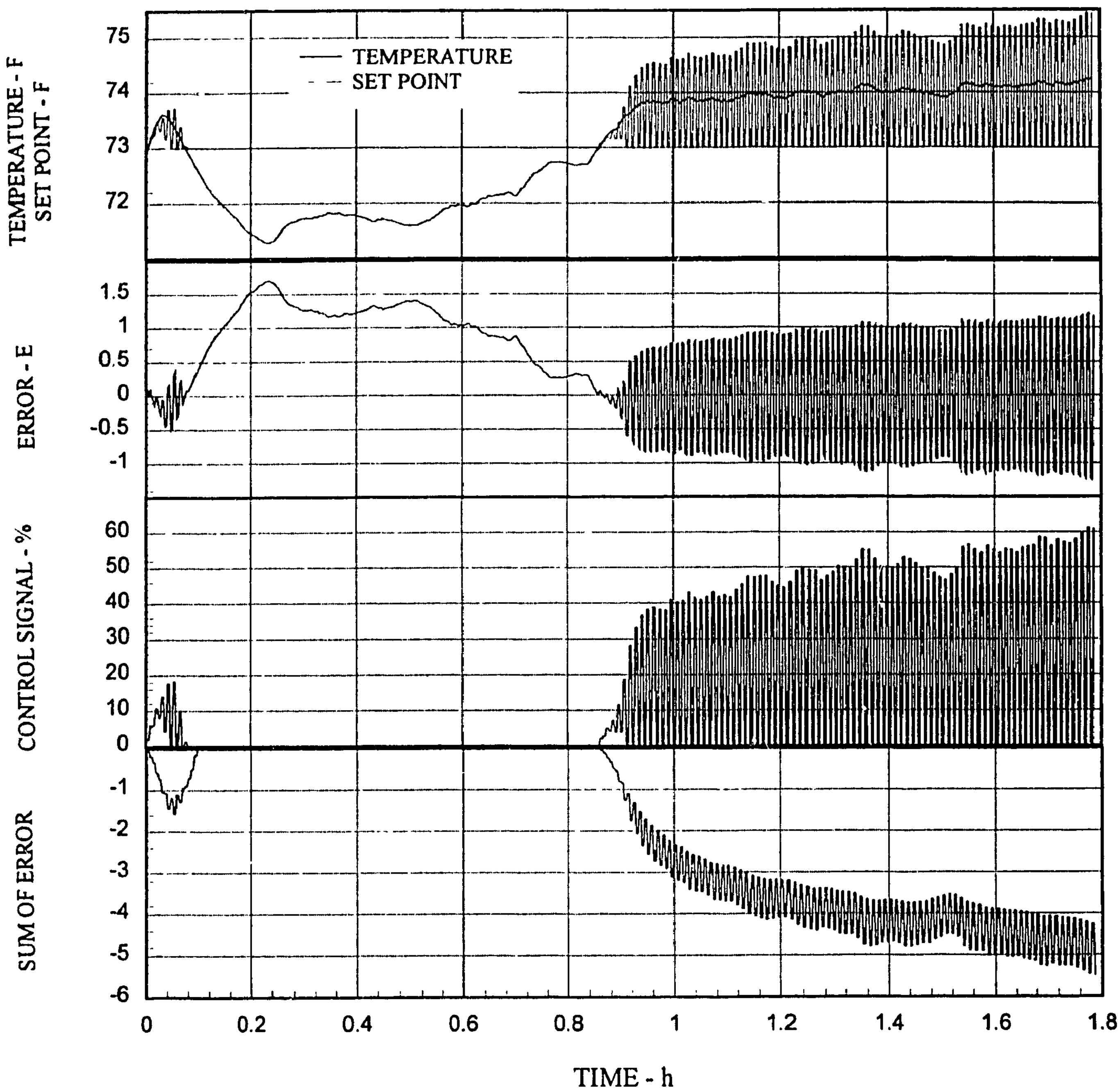

Figure 5.17. Response of the System for Variable Set Point Control when the Set Point is Expressed as Function of the cfm.

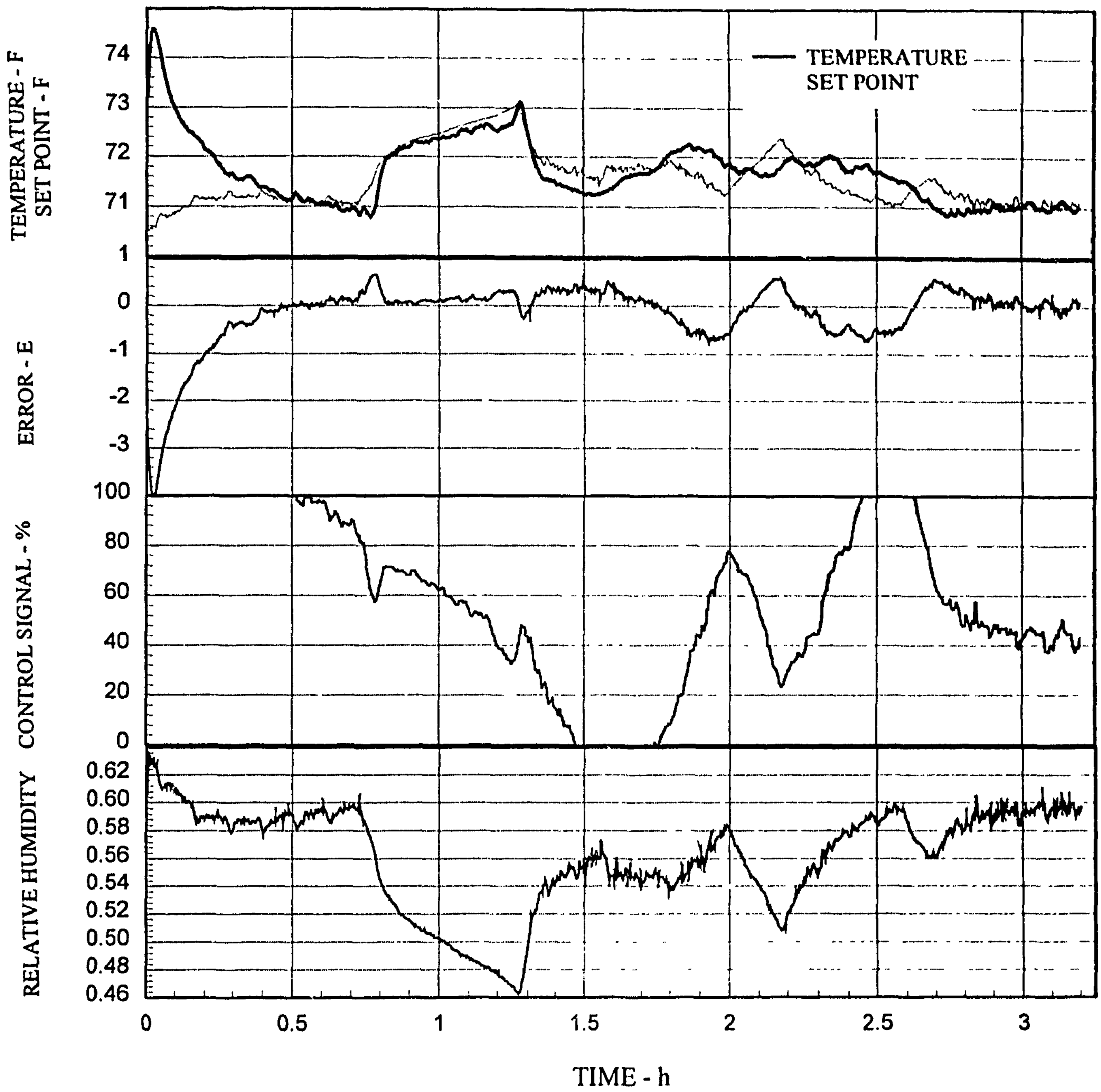

Figure 5.18 Response of the System for Variable Set Point Control when the Set Point is Expressed as Function of the Relative Humidity.

5.4 Conclusions

The following conclusions summarize the results of experimental testing of the control strategies:

- Once the sensitivity is determined, the PI coefficients can be easily calculated
- The PI coefficients determined and used in the presented control strategies led to satisfactory system response.
- No interference between temperature and humidity control systems. They both resulted in a stable system response.
- Experiments with throttling valve were more acceptable than those carried out with mixing valve. Varying the mixing valve position, varied the mixed water flow rate. A mixing valve should only vary the mixed water temperature.
- Second order systems are practical systems for space temperature and humidity control. The first order systems are practical for process control.
- Fluctuations in humidity readings were a result of the stepped response of the valve motor. For less than 6% variations in control signals, the valve motor does not respond.

CHAPTER 6

SENSITIVITY CONSIDERATIONS

The determination of PI coefficients used in the control strategies presented in Chapter 5 depends on the evaluation of the thermo-fluid system characteristic parameter defined as sensitivity. The second order system implemented by the space temperature and humidity control strategy identifies two system sensitivities. The sensible sensitivity related to temperature control and the latent sensitivity related to humidity control. Same discussion applies for the first order system implemented by the supply air temperature and humidity control. This chapter compares the analytical and experimental sensitivities and analyses the effects of variations on system response characteristics.

6.1 Space Temperature and Humidity Control:Analytical versus Experimental Sensitivities

As presented in Chapter 3, the performance of both control strategies depends on the cooling coil capacity. Cooling coil capacity can be determined either analytically or experimentally. Analytical evaluation of system sensitivities is based on the cooling coil characteristic models developed and presented in Figures 3.4 and 3.5. The analytical coil maps were derived for pure water. Experimentally, the system sensitivities' evaluations are presented in the procedure followed in Chapter 5. The experimental work was performed based on the experimental sensitivities' evaluation because glycol was mixed with water to prevent system freezing. The use of glycol is expected to decrease the heat transfer through the coil therefore affect the coil characteristics. The effect of glycol addition to the system water is investigated. A comparison between experimental and analytical sensitivities for the different control strategies is presented.

6.1.1 System with Throttling Valve

The system capacity sensitivity for space temperature model can be determined, analytically, from Figure 3.4. The vertically inclined lines are constant chilled water lines and the horizontally inclined lines are constant cfm lines. For temperature control, the water valve signal was set to 50% while the fan was modulated between minimum and maximum limits. On the coil map, the process is simulated by the central line. Projections from the intersections of this line, with the higher and lower cfm lines to the vertical sensible heat axis, give the change in sensible heat capacity for a 100% change in the fan control signal at constant water flow rates. The analytical sensible capacity sensitivity is obtained as

$$D_S = \frac{dQ_S}{dS} = \frac{(15400 - 10200)\ \text{BTU/h}}{100\ \%\text{signal}} = 52\ \text{BTU/h.\%signal} \tag{6.1}$$

The analytical system sensible sensitivity is obtained by application of Equations 4.34, 5.2 and 6.1 as

$$\sigma_S = \frac{1}{C_{SPC,S}} \cdot \frac{dQ_S}{dS} = -\frac{52\ \text{BTU/h.\%signal}}{32.76\ \text{BTU/}^{\circ}\text{F}} \cdot \frac{\text{h}}{3600\ \text{sec.}} = -4.410 \cdot 10^{-4}\ ^{\circ}\text{F/sec.\%signal} \tag{6.2}$$

The sensible sensitivity is negative because the room temperature decreases for an increase in the fan signal. This analytical sensitivity obtained in Equation 6.2 may be compared to the experimental sensitivity obtained from Equation 5.4, $-3.957 \cdot 10^{-4}$ °F/sec.%signal. The percent difference between the two values is 10%.

For specific humidity control, the air fan was operated at 50% signal while the throttling valve was modulated between minimum and maximum limits. On the coil map, the process is simulated by the central line. Projections from the intersections of this line, with the higher and lower ppm lines to the horizontal latent heat axis, give the change in latent heat capacity for a 100% change in the valve control signal at constant fan operating condition. The latent capacity sensitivity is obtained as

$$D_L = \frac{dQ_L}{dS} = \frac{(5000 - 2800)\ \text{BTU/h}}{100\ \%\text{signal}} = 22\ \text{BTU/h.\%signal} \qquad (6.3)$$

The analytical system latent sensitivity is obtained by application of Equations 4.60, 5.10 and 6.3 as

$$\sigma_L = \frac{1}{C_{SPC,L}} \cdot \frac{dQ_L}{dS} = -\frac{22\ \text{BTU/h.\%signal}}{204455.5\ \text{BTU}} \cdot \frac{\text{h}}{3600\text{sec}} = -2.9 \cdot 10^{8}\ \text{1/sec.\%signal} \qquad (6.4)$$

The latent sensitivity is negative because the room specific humidity decreases for an increase in the valve control signal. This analytical latent sensitivity obtained in Equation 6.4 may be compared to the experimental latent sensitivity obtained from Equation 5.11, $-1.22 \cdot 10^{-8}$ 1/sec.%signal. The percent difference between the two values is 59%.

6.1.2 System with Mixing Valve

Using Figure 3.5, the system capacity sensitivity for space temperature model with a mixing valve can be determined. The vertical lines are constant recirculation coefficient lines and the horizontal lines are constant cfm lines. For temperature control, the mixing water valve signal was set to 50% while the fan was modulated between minimum and maximum limits. On the coil map, the process is simulated by the central line. Projections from the intersections of this line, with the higher and lower cfm lines to the vertical sensible heat axis, give the change in sensible heat capacity for a 100% change in the fan control signal at constant water flow rates. The sensible capacity sensitivity is obtained as

$$D_S = \frac{dQ_S}{dS} = \frac{(15500 - 10100)\ \text{BTU/h}}{100\ \%\text{signal}} = 54\ \text{BTU/h.\%signal} \tag{6.5}$$

The analytical system sensible sensitivity is obtained by application of Equations 4.34, 5.2 and 6.5 as

$$\sigma_S = \frac{1}{C_{SPC,S}} \cdot \frac{dQ_S}{dS} = -\frac{54\ \text{BTU/h.\%signal}}{32.76\ \text{BTU/}^\circ\text{F}} \cdot \frac{\text{h}}{3600\ \text{sec.}} = -4.579 \cdot 10^{-4}\ {}^\circ\text{F/sec.\%signal} \tag{6.6}$$

The sensible sensitivity is negative because the space temperature decreases for an increase in the fan control signal. The analytical sensitivity obtained in Equation 6.6 may be compared to the experimental sensitivity obtained from Section 5.2.2.1, - $2.713 \cdot 10^{-4}$ °F/sec.%signal. The percent difference between the two values is 41%.

For specific humidity control with mixing valve, the fan was operated at 50% signal while the mixing valve was modulated between minimum and maximum limits. On the coil map, the process is simulated by the central line. Projections from the intersections of this line, with the higher and lower CRR lines to the horizontal latent heat axis, gives the change in latent heat capacity for a 100% change in the mixing valve control signal at constant fan operating conditions. The latent capacity sensitivity is obtained as

$$D_L = \frac{dQ_L}{dS} = \frac{(5000 - 2950)\ \text{BTU/h}}{100\ \%\text{signal}} = 20.5\ \text{BTU/h.\%signal} \qquad (6.7)$$

The analytical system latent sensitivity is obtained by application of Equations 4.60, 5.9 and 6.7 as

$$\sigma_L = \frac{1}{C_{SPC,L}} \cdot \frac{dQ_L}{dS} = -\frac{20.5\ \text{BTU/h.\%signal}}{204455.5\ \text{BTU}} \cdot \frac{h}{3600}\ \text{sec.} = -2.785 \cdot 10^{-8}\ \text{1/sec.\%signal} \qquad (6.8)$$

The latent sensitivity is negative because the space specific humidity decreases for an increase in th᷈ valve control signal. The analytical latent sensitivity obtained in Equation 6.8 may be compared to the experimental latent sensitivity obtained from Section 5.2.2, $-1.361 \cdot 10^{-8}$ 1/sec.%signal. The percent difference between the two values is 51%.

6.2 Supply Air Temperature and Humidity Control:Analytical versus Experimental Sensitivities

The analytical determination of the first order systems sensitivities can be performed using the analytical sensitivities of the second order systems. The sensible heat transfer rate of the cooling coil expressed in Equation 4.83 can be written as

$$Q_{CWC,s} = \dot{c}_T \cdot \left(T_{AIR,IN} - T_{AIR,SUP} \right) \tag{6.9}$$

Differentiating Equation 6.9 with respect to the control signal assuming constant air inlet temperature yields

$$\frac{dQ_{CWC,s}}{dS} = - \dot{c}_T \cdot \frac{dT_{AIR,SUP}}{dS} \tag{6.10}$$

Rewriting Equation 6.10 yields

$$\frac{dT_{AIR,SUP}}{dS} = - \frac{1}{\dot{c}_T} \cdot \frac{dQ_{CWC,s}}{dS} \tag{6.11}$$

Using Equations 4.27 and 4.89, Equation 6.11 can be expressed as

$$\left(\sigma_s\right)_{FO} = - \frac{1}{\dot{c}_T} \cdot \left(D_s\right)_{SO} \tag{6.12}$$

The latent heat transfer rate of the cooling coil expressed by Equation 4.97 can be written as

$$Q_{CWC,L} = \dot{c}_\omega \cdot \left(\omega_{AIR,IN} - \omega_{AIR,SUP} \right) \tag{6.13}$$

Differentiating Equation 6.13 with respect to the control signal assuming constant air inlet specific humidity yields

$$\frac{dQ_{CWC,L}}{dS} = - \dot{c}_{\omega} \cdot \frac{d\omega_{AIR,SUP}}{dS} \tag{6.14}$$

Rewriting Equation 6.14 yields

$$\frac{d\omega_{AIR,SUP}}{dS} = - \frac{1}{\dot{c}_{\omega}} \cdot \frac{dQ_{CWC,L}}{dS} \tag{6.15}$$

Using Equations 4.52 and 4.103, Equation 6.15 can be expressed as

$$\left(\sigma_L\right)_{F\,O} = - \frac{1}{\dot{c}_{\omega}} \cdot \left(D_L\right)_{S\,O} \tag{6.16}$$

Equations 6.12 and 6.16 are required to evaluate the analytical sensible and latent sensitivities for the first order system.

6.2.1 System with Throttling Valve

The second order sensible capacity sensitivity evaluated by Equation 6.1 is 52 BTU/h.%signal. Using Equation 6.12 yields

$$\sigma_{s,F.O} = \frac{1}{(600 \ ft^3/min) \cdot (0.24 \ BTU/lbm \ {}^oR)} \cdot \frac{52BTU}{h.\%signal} \cdot 13.7lbm/ft^3 \cdot \frac{h}{60min} \tag{6.17}$$

A temperature difference expressed in degrees Rankine is equivalent to temperature difference expressed in degrees Fahrenheit. Then the analytical sensible process sensitivity is

$$\left(\sigma_s\right)_{F.O} = 0.0824\ {}^{\circ}F/\%signal \tag{6.18}$$

The sensible first order sensitivity is positive because the air temperature increases for an increase in the fan control signal. The analytical sensitivity obtained in Equation 6.18 may be compared to the experimental sensitivity obtained from Equation 5.19 as 0.078 °F/%signal. The percent difference between the two values is 5%.

The second order latent capacity sensitivity evaluated by Equation 6.3 is 22 BTU/h.%signal. Equation 6.16 yields

$$\sigma_{L,F.O} = -\frac{1}{(600\ ft^3/min)\cdot(1061\ BTU/lbm)}\cdot\frac{22BTU}{h.\%signal}\cdot 13.7ft^3/lbm\cdot\frac{h}{60min} \tag{6.19}$$

Then the analytical latent sensitivity is

$$\left(\sigma_L\right)_{F.O} = -\ 7.876\cdot 10^{-6}\ (\%signal)^{-1} \tag{6.20}$$

The latent first order sensitivity is negative because the air specific humidity decreases for an increase in the throttling valve control signal. The analytical sensitivity obtained in Equation 6.20 is compared to the experimental sensitivity obtained from Equation 5.22 as $-6.8\cdot10^{-6}$ $\%signal^{-1}$. The percent difference between the two values is 13.6%.

6.2.2 System with Mixing Valve

The second order sensible capacity sensitivity evaluated by Equation 6.5 is 54 BTU/h.%signal. Equation 6.12 yields

$$\left(\sigma_s\right)_{F.O} = \frac{1}{(600\ \mathrm{ft}^3/\mathrm{min})\cdot(0.24\ \mathrm{BTU/lbm}\ {}^{\circ}\mathrm{R})} \cdot \frac{54\mathrm{BTU}}{\mathrm{h.\%signal}} \cdot 13.7\mathrm{lbm/ft}^3 \cdot \frac{\mathrm{h}}{60\mathrm{min}} \quad (6.21)$$

Then the sensible analytical sensitivity is

$$\left(\sigma_s\right)_{F.O} = 0.0856\ {}^{\circ}\mathrm{F}/\%\mathrm{signal} \quad (6.22)$$

The sensible first order sensitivity is positive because the supply air temperature increases for an increase in the fan control signal. The analytical sensitivity obtained in Equation 6.22 is compared to the experimental sensitivity obtained from Section 5.2.4 as 0.054 °F/%signal. The percent difference between the two values is 37%.

The second order latent capacity sensitivity evaluated by Equation 6.7 is 20.5 BTU/h.%signal. Equation 6.16 yields

$$\left(\sigma_L\right)_{F.O} = -\frac{1}{(600\ \mathrm{ft}^3/\mathrm{min})\cdot(1061\ \mathrm{BTU/lbm})} \cdot \frac{20.5\mathrm{BTU}}{\mathrm{h.\%signal}} \cdot 13.7\mathrm{ft}^3/\mathrm{lbm} \cdot \frac{\mathrm{h}}{60\mathrm{min}} \quad (6.23)$$

Then the latent analytical sensitivity is

$$\left(\sigma_L\right)_{F.O} = -\ 7.347\cdot 10^{-6}\ (\%\mathrm{signal})^{-1} \quad (6.24)$$

The latent first order sensitivity is negative because the supply air specific humidity decreases for an increase in the throttling valve control signal. The analytical sensitivity obtained in Equation 6.24 is compared to the experimental sensitivity obtained from Section 5.2.4 as $-8.5 \cdot 10^{-6}$ %signal^{-1}. The percent difference between the two values is 16%.

6.3 Effects of Sensitivity Variations on Response Characteristics

The results of the current calculations are summarized in Table 6.1. Eight different sensitivities were obtained analytically and experimentally. The percentage difference between analytical and experimental values ranges between low acceptable values such as 5% and high values such as 59%. Differences until 20% were expected because of the use of glycol in the coolant system. The analytical sensitivities evaluated for the different systems are higher than the experimental sensitivities. The use of glycol decreases the heat transfer capacity of the coil by an average of 15%. This reduction in cooling capacity should be accounted when sizing a cooling coil operating with water-antifreeze mixture.

The high percentage of error for sensitivities evaluated with mixing valve can be explained by the variation of coolant mass flow rate in addition to temperature and by errors induced in latent heat calculations. It was clear during the experimental work that a change in the mixed water flow rate occurred as the mixing valve signal varies. Ideally, the mixing valve should only vary the mixed coolant temperature as it opens or closes. Calculations of latent heat loads were based on the monitored relative humidity. Humidity sensors were sensitive to electrical noise, hence the monitored values were subject to errors.

Table 6.1 Experimental versus Analytical Process Sensitivities

Sensitivity	Valve Type	Experimental Sensitivity	Analytical Sensitivity	Units	% Difference
First Order Sensible	Throttling	0.078	0.0824	°F/%signal	5
First Order Latent	Throttling	$-6.8 \cdot 10^{-6}$	$-7.876 \cdot 10^{6}$	$(\%signal)^{-1}$	14
First Order Sensible	Mixing	0.054	0.0856	°F/%signal	37
First Oruer Latent	Mixing	$-8.5 \cdot 10^{-6}$	$-7.34 \cdot 10^{6}$	$(\%signal)^{-1}$	16
Second Order Sensible	Throttling	$-3.957 \cdot 10^{-4}$	$-4.410 \cdot 10^{4}$	°F/sec.%signal	10
Second Order Latent	Throttling	$-1.23 \cdot 10^{-8}$	$-2.90 \cdot 10^{-8}$	$(sec.\%signal)^{1}$	59
Second Order Sensible	Mixing	$-2.713 \cdot 10^{-4}$	$-4.579 \cdot 10^{-4}$	°F/sec.%signal	41
Second Order Latent	Mixing	$-1.36 \cdot 10^{-8}$	$-2.785 \cdot 10^{8}$	$(sec.\%signal)^{1}$	51

The comparisons of analytical and experimental results show that the cooling coil capacity varies with the type of coolant flowing into the cooling coil. The theory developed in Chapter 4 relates the PI coefficients to the sensible and latent sensitivities obtained for different control strategies. It also relates the target system parameters to the PI coefficients. As the sensitivities vary, the target system parameters will vary. It is interesting to investigate the variations in the target system parameters due to variations in process sensitivities.

6.3.1 Second Order Systems

For second order systems, the damping ratio is related to the PI coefficients and the normalized sensitivity by Equation 4.82 as

$$\xi = \frac{K_P}{2 \cdot \sqrt{\dfrac{K_I}{\Delta t \cdot \sigma_N}}} \tag{6.25}$$

Differentiate Equation 6.25 with respect to sensitivity, simplifying and rearranging yields

$$\frac{\Delta\xi}{\xi} = \frac{1}{2 \cdot K_I} \cdot \frac{\Delta\sigma_N}{\sigma_N} \tag{6.26}$$

As a numerical example, consider the temperature control model of the second order system with throttling valve. The K_I obtained in Section 5.2.1 was -2.93 for damping ratio of 0.8. Applying Equation 6.26 using the aforementioned values with the data shown in the fifth row of Table 6.1 yields a deviation in the damping ratio by 7%. Therefore, a small increase in the sensitivity leads to a small increase in the damping ratio.

The period is related to the system parameters by Equation 4.81. Combining Equation 4.82 and Equation 4.81, simplifying and rearranging terms, yields

$$\tau = 4\pi \cdot \sqrt{\frac{K_I \cdot \Delta t}{(4K_I - K^2_P \, \Delta t) \cdot \sigma_N}} \tag{6.27}$$

Differentiating Equation 6.27, simplifying and dividing by τ yields

$$\frac{\Delta\tau}{\tau} = -\frac{1}{2}\cdot\frac{\Delta\sigma_N}{\sigma_N} \tag{6.28}$$

Numerically, consider the temperature control model of the second order system with throttling valve. The period used in Section 5.2.1 was 1200 sec. Applying Equation 6.28 using the aforementioned value with the data shown in the fifth row of Table 6.1 yields a deviation in the period by 22%. Therefore, an increase in the sensitivity leads to a small increase in the period of the system response.

6.3.2 First Order Systems

The attenuation factor is related to the system normalized sensitivity by Equation 4.116. Differentiating Equation 4.116 with respect to sensitivity yields

$$\frac{da_1}{d\sigma_N} = \frac{\dfrac{K_I}{\sigma_N^{\,2}}}{\left(K_P + \dfrac{1}{\sigma_N}\right)^2} \tag{6.29}$$

Rewriting Equation 6.29 in terms of deviations and rearranging terms yields

$$\Delta a_1 = \Delta\sigma_N \cdot \frac{\dfrac{K_I}{\sigma_N^{\,2}}}{\left(K_P + \dfrac{1}{\sigma_N}\right)^2} \tag{6.30}$$

The percentage change in attenuation factor for change in sensitivity is obtained by dividing

Equation 6.30 by a_I and simplifying with Equation 4.116, yielding

$$\frac{\Delta a_I}{a_I^2} = \frac{1}{K_I} \cdot \frac{\Delta \sigma_N}{\sigma_N^2} \tag{6.31}$$

Equation 6.31 relates the change in the attenuation factor to the change in the normalized sensitivity. The sensitivity may deviate from the assumed value due to variations over operating range or inaccuracies in its determination. The deviation in the attenuation factor is proportional to the sensitivity deviation by the inverse of the integral coefficient K_I. As a numerical example, consider the temperature control model of the first order system with throttling valve. The K_I obtained in Section 5.2.3 was 0.69 for an attenuation factor of 0.05. Applying Equation 6.31 using the aforementioned values with the data shown in the first row of Table 6.1 yields a deviation in the attenuation factor by 4%. Therefore, a small increase in the sensitivity leads to a small increase in the attenuation factor hence a slightly slower response.

The deviations in the normalized process sensitivity affect the system response parameters. The resulting response may not have the same target response characteristics. However, the response will be satisfactory.

CHAPTER 7

EXTENSIONS TO MULTIZONE SYSTEMS

The PI control theory of space temperature and specific humidity with chilled water system developed in Chapters 2 through 6 worked well for the experimental system existing in the laboratory, but can it be applied to other HVAC systems as well? The answer to this question depends on whether the system sensitivity can be determined and whether the physical system can be thermally modeled. Because thermal equations are not difficult to generate and system sensitivity can be obtained experimentally or analytically, the theory can be applied to multizone HVAC systems similar to those shown in Figures 7.1 and 7.2.

For these systems, each zone has its own coil, valve and damper. One chiller supplies the chilled water to the system coils. The space temperature can be monitored in the return duct then used to modulate a damper motor through a PI controller. The damper will open and close proportional to the signal varying the flow rate of air and modulating the space temperature. The space humidity can be monitored to modulate a valve motor through a PI controller. The valve will modulate the chilled water flow rate or temperature controlling the humidity.

Tuning the controllers will be based on the theory developed in Chapter 4. The only equations to be used are Equations 4.81 and 4.82. Sensitivities' evaluation can be performed either based on the coil characteristics as in Chapter 3 or on open loop experiments as in Chapter 5. If glycol is mixed with water, a reduction of 10-20% on the analytical sensitivities is required. Different zones may have different sensitivities. The load criterion is only required to size the coils. It is not required to tune the controllers.

If multizone systems similar to those showed in Figures 7.1 and 7.2 are implemented, an additional control loop must be considered. The control loop has to control the static pressure after the supply fan by modulating the fan motor speed. Controlling the fan discharge static pressure will compensate for pressure variations in the supply duct due to opening or closing of the dampers.

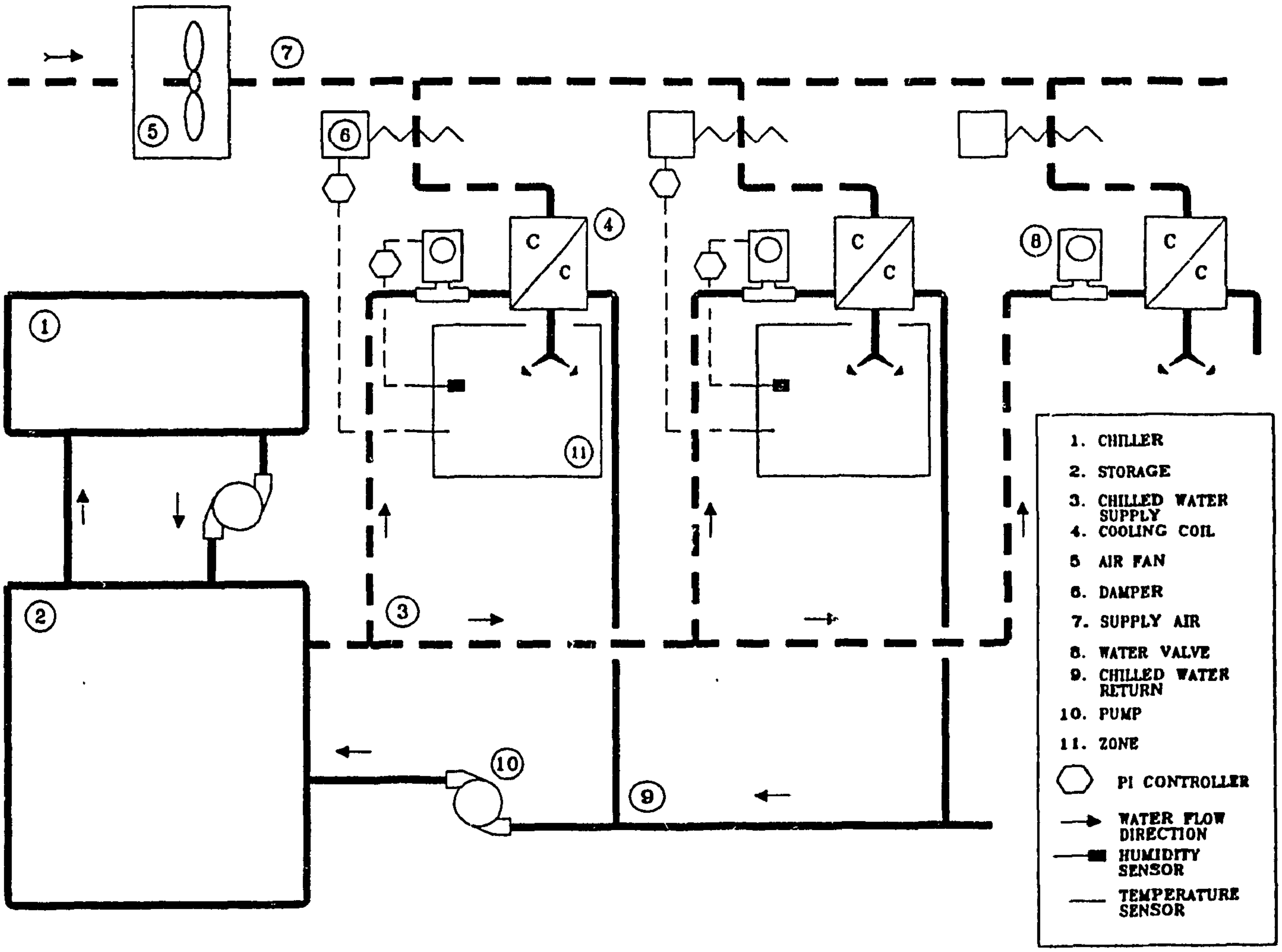

Figure 7.1 A Multizone HVAC System using a Throttling Water Valve.

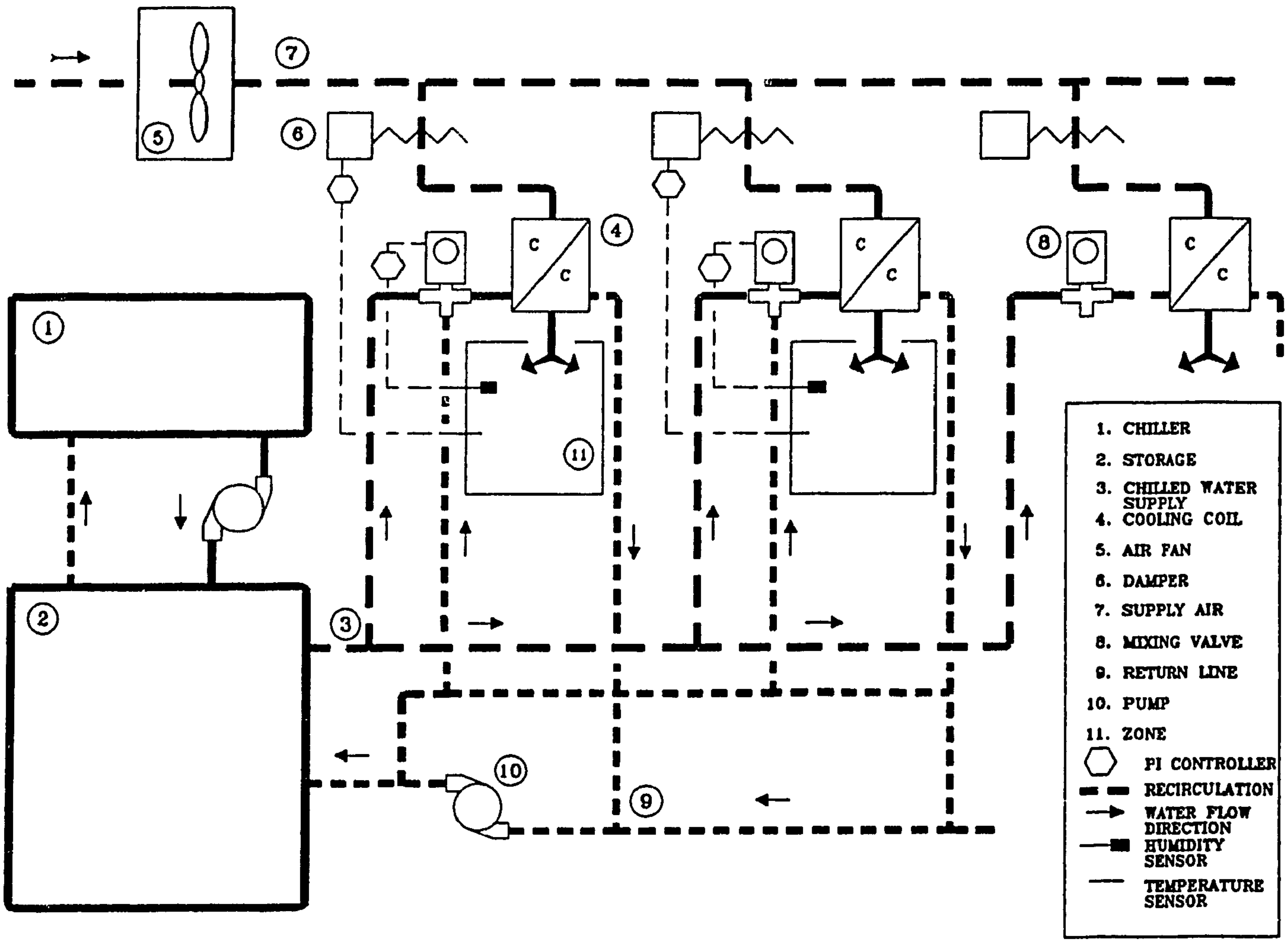

Figure 7.2 A Multizone HVAC System using a Mixing Water Valve.

CHAPTER 8

CONCLUSION

8.1 Summary

This research had as its objective using standard, industrial cooling coil and equipment to achieve quick, stable control of space temperature and humidity as an HVAC process. For this purpose, the cooling coil and thermal systems were to be modeled, control strategies were tested and evaluated, and tuning guide lines were developed.

Two thermal systems were employed to perform experimentation. They include 2640 cubic feet enclosed space, a chiller, a plate finned tube cooling coil, a variable speed fan, and two variable position valves: a throttling valve and a mixing valve. One system considers the properties of the space. It was classified (with the controller) as second order system. The other considers the properties of supply air. It was classified (with the controller) as first order system. Modelling the two systems was based on the thermo-fluid equations. Modelling the cooling coil was based on computer programs.

A proportional-plus-integral (PI) controller was selected as the process controller because of its simplicity, ability to eliminate steady state error, and flexibility of implementation with a computer program. To improve the performance of the controller, a windup prevention algorithm was developed. The anti-windup algorithm, based on limiting the sum of error in the PI algorithm, is efficient in use, simple to implement, and does not exist in current literature. For second order systems, an underdamped target response was selected because of its fast and stable steady state response. The characteristic parameters of such response are the damping ratio and the period. For first order response, the characteristic parameters are the attenuation factor or the response time characteristic.

A tuning method was developed for use with PI controlled, space and supply air systems The method identifies and defines process sensitivities. The process sensitivities are related to the system thermal characteristics. They can be evaluated either experimentally or analytically. Tunning the controller uses process sensitivities and a sampling intervals to obtain a desired response characteristic. The method is developed from a thermo-fluid engineering perspective rather than from a control engineering perspective. The method is relatively unaffected by differences between experimental and analytical sensitivities.

Two control strategies were used. One with the throttling valve and the other with the mixing valve. Both valves show a stepped response of the valve motor. This response affects the humidity control in a minor way. Experiments indicate that the throttling valve to be superior to the mixing valve. The mixing valve showed variations in the water flow rate due to variations in the control signal. Both strategies show stable responses maintaining the set points.

Specific tests were performed to verify the strategy of using variable set point control. These tests represent comfort control techniques. Comfort control was not a primary issue in this investigation. It was only used to implement the control strategy by variable set point control technique. Additional tests were performed to test the sensitivity of the temperature and humidity sensors to electrical noise. These tests showed the necessity of inciuding a signal conditioner in the system electronic apparatus.

The theoretical modelling of the control process, verified by experiments, leads to the application of the presented theory to multizone systems. Systems whereby temperature is controlled by varying the air flow rate and humidity is controlled by varying the chilled water supply flow rate or temperature, are variable volume - variable temperature (VVVT) systems. The simplicity of the tuning method allows its application to small and large systems.

In modelling the cooling coil and the thermal systems, developing a tuning method, defining and evaluating processes sensitivities, relating PI coefficients to thermal system parameters, the research has achieved its objectives.

8.2 Recommendations for Further Research

The work performed in this thesis suggests several areas for further research. These include extensions to self tuning algorithms, energy analysis, relation to conventional control theory for stability analysis, and application to indoor air quality problems.

The theory presented along this paper can be generalized to be applicable to any thermal system. Generalizing the theory may include some modifications to the analysis by considering a perturbation term with the PI coefficients' equations. Perturbation is included for systems in which the capacity is affected by the process variable. Some work had been done, by the author, in this area on the control temperature and humidity by outside ventilation.

The sensitivity analysis opens the door for further work in the presented theory. The work may consider variable system sensitivity therefore variable coefficients for a process. Variations in system sensitivity proved to affect the system target response parameters. The technique may assume control with variable sensitivity to improve the resulting response of the different control strategies.

The simultaneous control of temperature and humidity and the control technique with variable set points may lead to energy conservation for some operating conditions. An energy analysis should be conducted on proposed systems in further investigation.

Humidity control was performed by varying a valve position. Water valves are less linear than centrifugal pumps. A valve can be replaced with a variable speed water pump. Good control characteristics at all loads are expected with simplicity in system design.

Some researchers may not be comfortable with the absence of Laplace transforms and transfer functions in this paper. The perspective followed in the presented work is analogues to the transfer functions' perspective. Relating the proposed theory to the conventional control system theory is interesting but not required for the current research objectives.

Buildings occupants are becoming accustomed to more effective control of HVAC and are becoming more aware of indoor air quality. The proposed theory can be applied to control the outdoor air supply by monitoring the carbon dioxide concentration. Some theoretical work has been done by the author in this area. The theory has been extended to consider such system. Numerical simulations proved the feasibility of such control. Experimental work is suggested for future applications [27].

REFERENCES

[1] 1977. *ASHRAE Handbook of Fundamentals.* American Society of Heating, Refrigeration, and Air Conditioning Engineers. New York.

[2] Sterling, E.M., Arundel, A. ,and Sterling T.D. 1985. "Criteria for Human Exposure to Humidity in Occupied Buildings." *ASHRAE Transactions.*

[3] Cole, R.J. 1983. "Energy Conscious Design, the Factors Influencing the Thermal Performance and Energy Requirements of Buildings." School of Architecture, University of British Columbia. Vancouver, B.C.

[4] BringMann, A. And Meuter, R. 1985. "Clean Room Technology for Production Processes." *Sulzer Technical Review*, vol 65.

[5] Benson, F.B.;Henderson, J.J.;and Caldwell, D.E. 1972. "Indoor-Outdoor air Pollution Relationships: a Literature Review." U.S. E.P.A. Publication #AP-112. U.S. Government Printing Office. Washington D.C.

[6] 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, Pt. 2.

[7] Stoecker, W.F., and Stoecker, P.A. 1989. *Microcomputer Control of Thermal and Mechanical Systems.* Van Nostrand Reinhold. New York, NY.

[8] Deshpande, P.B., and Ash, R.H. 1981. *Microcomputer Control of Thermal and Mechanical Systems.* Van Nostrand reinhold. New York, NY.

[9] Ziegler, J.G., and Nichols, N.B. 1942. "Optimum Settings for Automatic Controllers." *ASME Transactions.*

[10] Pinella, M.J., Hittle, D.C., Wechselberger, E. And Pedersen, C.O. 1986. "Self-Tuning Digital Integral Control." *ASHRAE Transactions.*

[11] MacArthur, J.W., Grald, E.W. and Konar, A.F. 1989. An Effective Approach for Dynamically Compensated Adaptive Control." *ASHRAE Transactions.*

[12] Nesler, C.G. 1985. "Experiences in Applying Adaptive control to Thermal processes in Buildings." Proceedings of the 1985 American Control conference, Boston, MA, June 19-21.

[13] Donoghue, J.F. 1977. "Review of Control design Approaches for Transport Delay processes," *ISA Transactions.*

[14] Pinella, M.J. 1985. "Modelling, Tuning, and Experimental Verification of a Fan static pressure Control System." Master's Thesis. University of Illinois. Urbana-Champaign.

[15] Bekker, J.E., Meckel, P.H., and Hittle, D.C. 1991. "A Tuning Method for First-Order Processes with PI Controllers." *ASHRAE Transactions.*

[16] Bekker, J.E. 1990. "Humidity Control Systems." Master's Thesis. School of Nechanical Engineering, Purdue University, West Lafayette, IN.

[17] Hussein, G.M. 1995. "A Tuning Method for First Order Processes." Course Project. Concordia University. Montreal, QC.

[18] Krakow, K.I., Lin, S., and Zeng, Z.S. 1995. "Temperature and Humidity Control during Cooling and Dehumidifying by Compressor and Evaporator fan Speed Variation." *ASHRAE Transactions,* vol. 101., Pt. 1.

[19] Krakow, K.I., Lin, S., and Zeng, Z.S. 1995. "Analytical Determination of PID Coefficients for Temperature and Humidity Control During Cooling and Dehumidifying by Compressor and Evaporator Fan Speed Variation." *ASHRAE Transactions,* vol. 101, Pt. 1.

[20] Astrom K.J., and Hagglund, T. 1989. *Automatic Tuning of PID Controllers.* Instrument Society of America. New Jersy. Prentice hall Inc.

[21] Kuo, B.C. 1987. Automatic Control Systems. Englewood Cliffs, New Jersy.: Prentice-Hall Inc.

[22] Krakow K.I, Hussein G., and Lin S. 1995 "A Generalized Theory of PI Control of HVAC Systems. Part I-II. Derivations - Implementations." Concordia University. Montreal, QC.

[23] Mirth, D.R., and Ramadhyani, S. 1991 "Comparison of Methods of Modelling the Air-Side Heat and Mass Transfer in Chilled-Water Cooling Coils." *ASHRAE Transactions.*

[24] 1993. *A Toolkit for Secondary HVAC System Energy Calculations*. American Society of Heating, Refrigeration, and Air Conditioning Engineers. New York.

[25] 1989. *ASHRAE Handbook of Fundamentals*. American Society of Heating, Refrigeration, and Air Conditioning Engineers. New York.

[26] Thomson, W.T. 1993. *Theory of Vibration with Applications*. 4th Edition. Prentice Hall, N.J.

[27] Hussein, G.M. 1995. "PI Controls of Dampers to Improve IAQ by Monitoring the Space Carbon Dioxide Concentration." Course Project. Concordia University. Montreal, QC.

APPENDIX

A PROGRAM USED FOR AIR TEMPERATURE AND HUMIDITY CONTROL

```
COMMON SHARED MAXTABSIZE     ' Model 236 for High Speed
DECLARE FUNCTION TCS! (TPV!, TINI%)
DECLARE FUNCTION HCS! (HPV!, HINI%)
DECLARE FUNCTION SH! (T!, RH!)
DECLARE FUNCTION PVAP! (T!)
DECLARE FUNCTION CCS! (CPV!, CINI%)
DECLARE FUNCTION TDEGF! (TC!)

DECLARE SUB CallTest (badd%, IdByte%)
DECLARE SUB CheckID (badd%, IdByte%)
DECLARE SUB TestAll ()
DECLARE SUB GetInteger (prompt$, x%)
DECLARE SUB TEST210 (badd%)
DECLARE SUB TEST220 (badd%)
DECLARE SUB TEST236 (badd%)

DECLARE SUB SetThmst (minthmst#(), maxthmst#(), thmsta#())
DECLARE SUB SetTcpNBS (mintcp#(), maxtcp#(), tcpa#())
DECLARE SUB TEST236 (badd%)
DECLARE SUB Delay236 (badd%, range%, ER%)
DECLARE SUB VoltsHLG236 (badd%, chan%, range%, HorL%, AZflag%, volts!,ER%)
DECLARE SUB Interthermistor236 (badd%, temp!, ER%)
DECLARE SUB Errors236 (ermes$, ER%)
DECLARE SUB Volts236 (badd%, chan%, range%, AZflag%, volts!, ER%)
DECLARE SUB Ohms3w236 (badd%, chan%, range%, Ohms3!, ER%)
DECLARE SUB AtoD236 (badd%, chan%, range%, func1%, func2%, HorL%, AZflag%,
count%, ER%)
DECLARE SUB Thermistor236 (badd%, chan%, range%, typ$, temp!, ER%)
DECLARE SUB Thermocouple236 (badd%, chan%, range%, ReftempChan%, typ$,
NBS%, temp!, ER%)
DECLARE SUB Init236 (badd%, ER%)
DECLARE SUB FindAtoD236 (badd%, bhold%, count%, ER%)
DECLARE SUB Checkbaddid (badd%, ModelNumber%, ER%)
DECLARE SUB Errors200 (ermes$, ER%)
DECLARE SUB PressAnyKey ()
DECLARE SUB PrintError (ermes$, ER%)
DECLARE SUB Thermistor (Ohms!, ty$, T!, ER%)
```

```
DECLARE SUB ThermocoupleNBS (vd!, rf!, typ$, T!, ER%)
DECLARE SUB ThermocoupleSI (vd!, rf!, typ$, T!, ER%)
DECLARE SUB ReadDin210 (badd%, chan%, state%, ER%)
DECLARE SUB ReadDout210 (badd%, chan%, state%, ER%)
DECLARE SUB WriteDout210 (badd%, chan%, state%, ER%)
DECLARE SUB ReadDigOut210 (badd%, chan%, state%, ER%)
DECLARE SUB OutUnits210 (badd%, chan%, range%, value!, ER%)
DECLARE SUB WriteDigOuts210 (badd%, chan%, state%, ER%)
DECLARE SUB TEST210 (badd%)
DECLARE SUB Errors210 (ermes$, ER%)
DECLARE SUB OutDtoA210 (badd%, chan%, dtoacount%, ER%)
DECLARE SUB WriteDigiOuts210 (badd%, chan%, state%, ER%)
DECLARE SUB ReadDigIn210 (badd%, chan%, state%, ER%)
DECLARE SUB ReadDigiOut210 (badd%, chan%, state%, ER%)
DECLARE SUB TEST220 (badd%)
DECLARE SUB TimeDelay (secs!)
DECLARE SUB Errors220 (ermes$, ER%)
DECLARE SUB Readrelay220 (badd%, chan%, state%, ER%)
DECLARE SUB Resetallrelays220 (badd%, ER%)
DECLARE SUB Writerelay220 (badd%, chan%, state%, ER%)

' DUMMY SUBROUTINES
DECLARE SUB TEST202 (badd%)
DECLARE SUB TEST203 (badd%)
DECLARE SUB TEST204 (badd%)
DECLARE SUB TEST222 (badd%)
DECLARE SUB TEST231 (badd%)
DECLARE SUB TEST232 (badd%)
DECLARE SUB TEST233 (badd%)
DECLARE SUB TEST234 (badd%)
DECLARE SUB TEST240 (badd%)
DECLARE SUB TEST241 (badd%)
DECLARE SUB TEST242 (badd%)
DECLARE SUB TEST243 (badd%, setup212%, setup243B&)

DIM CV(0 TO 23), SUM(0 TO 23), AV(0 TO 23)

' Roundoff functions for 236.bas
 DEF fna (x) = INT(x + .5)
 DEF fnb (x) = INT(x * 10 + .5) / 10
 DEF fnc (x) = INT(x * 100 + .5) / 100
 DEF fnd (x) = INT(x * 1000 + .5) / 1000
 DEF fne (x) = INT(x * 10000 + .5) / 10000
```

```
  DEF fnf (x) = INT(x * 100000 + .5) / 100000
  DEF fng (x) = INT(x * 1000000 + .5) / 1000000

' SYSTEM TESTS
CLS
PRINT "RUN TESTS"
PRINT "RUN TestAll: 0 = NO  1 = GO"
INPUT IOPT%
IF IOPT = 1 THEN CALL TestAll

PRINT "RUN TEST236: 0 = NO  1 = GO"
INPUT IOPT%
IF IOPT% = 1 THEN CALL TEST236(672)

PRINT "RUN TEST210: 0 = NO  1 = GO"
INPUT IOPT%
IF IOPT% = 1 THEN CALL TEST210(656)

PRINT "RUN TEST220: 0 = NO  1 = GO"
INPUT IOPT%
IF IOPT% = 1 THEN CALL TEST220(640)

PRINT "INITIALIZE"

PRINT "OUTPUT FILE1 NAME ?  "
INPUT OFILE1$
PRINT "OUTPUT FILE2 NAME ?  "
INPUT OFILE2$
OPEN OFILE1$ FOR OUTPUT AS #1
PRINT #1, "\", OFILE1$, DATE$, TIME$
PRINT #1,
PRINT #1, "TIME     FPWR    DP   VAIR    RH2B   TA2B    RH2A   TA2A
RHROOM  TROOM  CPWR    PPHCC  PPHEVP    TEVP2   TCC2    TCC1
TEVP1  TSTR    TMIX    TRTN    TSE    TSG    CSE    CSG    HSE    HSG
TSP   TRIN    TROUT "
CLOSE #1
OPEN OFILE2$ FOR OUTPUT AS #2
PRINT #2, "\", OFILE2$, DATE$, TIME$
PRINT #2,
PRINT #2, "TIME    TROOM    RHROOM    TEVP1    TSG    HSG    CSG
SHUM    TMIX    TRTN"
CLOSE #2
NCHAN% = 23
```

```
' INPUT DATA
    TSCAN = 15
    HSCAN = 60
    CSCAN = 10
    DSCAN = 10
    DLOG = 10

    TSP = 72
    TSPO = TSP
    TKP = -33
    TKI = -3
    TIMIN = 0
    TIMAX = 100
    TOMIN = 819
    TOMAX = 4095
    DTII = TIMAX - TIMIN

    HSP = .009
    HKP = -33
    HKI = -3.4
    HIMIN = 0
    HIMAX = .023
    HOMIN = 1800
    HOMAX = 3200
    DHII = HIMAX - HIMIN

    CSP = 37
    CKP = -54
    CKI = -4.18
    CIMIN = 0
    CIMAX = 100
    COMIN = 1000
    COMAX = 4095
    DCII = CIMAX - CIMIN

    WTFILTR% = 100
    TFILTR% = 100
    HFILTR% = 400
    TSG = 0
    HSG = 0
    CSG = 0
    TSEMAX = 100 / TKI
    CSEMAX = 100 / CKI
```

```
    HSEMAX = 100 / HKI
    TSEMIN = 0
    CSEMIN = 0
    HSEMIN = 0


' INITIALIZE
 CALL Init236(672, ER%)
 IF ER% THEN PRINT " ERROR 236 INITIALIZATION: "; ER%
' FAN
 count% = 2000
 CALL OutDtoA210(656, 0, count%, ER%)
' VALVE
 count% = 2000
 CALL OutDtoA210(656, 2, count%, ER%)
' COMPRESSOR
 count% = 2000
 CALL OutDtoA210(656, 1, count%, ER%)
 PRINT "INITIALIZATION OK"
 KDATA = 0
 TSE = 0
 HSE = 0
 CSE = 0
 FOR i% = 1 TO NCHAN%
 SUM(i%) = 0
 NEXT i%

'LOOP
 IGO = 0
 IF IGO = 0 THEN TZERO = TIMER
 DSGOTIME = 0
 TGOTIME = 0
 HGOTIME = 0
 CGOTIME = 0
 DLGOTIME = 0
 RHAPRS = .5
 RHBPRS = .5
 PREHUM = HSP
 HFR = 1
 KSH% = 0
 ODATE$ = DATE$
 PS = 100
 PE = 100
```

```
DO

 SELECT CASE INKEY$
  CASE CHR$(32)
   TSP = TSP - 1
'   TSE = TSEMAX
   PRINT "SET POINT CHANGED TO", USING "##.##"; TSP
  CASE CHR$(13)
   TSP = TSP + 1
'   TSE = TSEMIN
   PRINT "SET POINT CHANGED TO", USING "##.##"; TSP
  CASE CHR$(8)
   CLS
  CASE ELSE
   GOTO 10
10 END SELECT

 IGO = IGO + 1

 CTIME = TIMER
 IF DATE$ <> ODATE$ THEN CTIME = TIMER + 86400
 XTIME = CTIME - TZERO

  ' DATA SCAN LOOP
  IF CTIME >= DSGOTIME THEN
       DSGOTIME = CTIME + DSCAN
       LOCATE 1, 1
       PRINT "LOG  "; TIME$
       CALL Thermocouple236(672, 0, 3, -1, "T", 0, TSTR, ER%)
       IF ER% THEN PRINT " ERROR TSTR: "; 0, ER%
       TSTR = TDEGF(TSTR)
       CV(21) = TSTR
'     PRINT "TSTR  DEGF  ", USING "##.##  "; TSTR; AV(21)
       CALL Thermocouple236(672, 20, 3, -1, "T", 0, TROUT, ER%)
       IF ER% THEN PRINT " ERROR TROUT: "; 1, ER%
       TROUT = TDEGF(TROUT)
       CV(20) = TROUT
       PRINT "TROUT  DEGF  ", USING "##.##  "; TROUT; AV(20)
       CALL Thermocouple236(672, 21, 3, -1, "T", 0, TRIN, ER%)
       IF ER% THEN PRINT " ERROR TRIN: "; 13, ER%
       TRIN = TDEGF(TRIN)
       CV(15) = TRIN
```

```
PRINT "TRIN  DEGF  ", USING "##.##  "; TRIN; AV(15)
CALL Volts236(672, 16, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR PPHEVP "; 16, ER%
PPHEVP = -1250 + 3125 * volts
CV(14) = PPHEVP
PRINT "PPHEVP PPH  ", USING "#####  "; PPHEVP; AV(14)
CALL Volts236(672, 17, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR PPHCC "; 17, ER%
PPHCC = -1250 + 3125 * volts
CV(13) = PPHCC
PRINT "PPHCC  PPH  ", USING "#####  "; PPHCC; AV(13)
CALL Volts236(672, 18, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR CPWR "; 18, ER%
CPWR = 0 + 1000 * volts
CV(12) = CPWR
PRINT "CPWR  WATTS ", USING "#####  "; CPWR; AV(12)
CALL VoltsHLG236(672, 26, 0, 1, 0, volts, ER%)
IF ER% THEN PRINT " ERROR TA2A "; 26, ER%
TA2A = -22 + 23.4 * volts
CV(7) = TA2A
'    PRINT "TA2A  DEGF  ", USING "##.##  "; TA2A; AV(7)
CALL VoltsHLG236(672, 26, 0, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR RH2A "; 26, ER%
RH2A = 0 - .2 * volts
CV(6) = RH2A
'    PRINT "RH2A  RH  ", USING "##.###  "; RH2A; AV(6)
CALL VoltsHLG236(672, 27, 0, 1, 0, volts, ER%)
IF ER% THEN PRINT " ERROR TA2B "; 27, ER%
TA2B = -22 + 23.4 * volts
CV(5) = TA2B
'    PRINT "TA2B  DEGF  ", USING "##.##  "; TA2B; AV(5)
CALL VoltsHLG236(672, 27, 0, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR RH2B "; 27, ER%
RH2B = 0 - .2 * volts
CV(4) = RH2B
'    PRINT "RH2B  RH  ", USING "##.###  "; RH2B; AV(4)
CALL Volts236(672, 28, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR VAIR "; 28, ER%
VAIR = 56.962 - 608.682 * volts + 4406.97 * volts ^ 2
CV(3) = VAIR
PRINT "VAIR  FPM   ", USING "####  "; VAIR; AV(3)
CALL Volts236(672, 29, 0, 0, volts, ER%)
IF ER% THEN PRINT " ERROR DP "; 29, ER%
```

```
  DP = volts
  CV(2) = DP
  CALL Volts236(672, 30, 1, 0, volts, ER%)
  IF ER% THEN PRINT " ERROR FPWR "; 30, ER%
  FPWR = 5000 * volts
  CV(1) = FPWR
  PRINT "FPWR  WATTS ", USING "####  "; FPWR; AV(1)
  CALL Thermistor236(672, 19, 2, "2", TMIX, ER%)
  IF ER% THEN PRINT " ERROR TMIX: "; 19, ER%
  TMIX = TDEGF(TMIX)
  CV(22) = TMIX
'  PRINT "TMIX  DEGF ", USING "##.##  "; TMIX; AV(22)
  CALL Thermistor236(672, 23, 2, "2", TRTN, ER%)
  IF ER% THEN PRINT " ERROR TCC2: "; 23, ER%
  TRTN = TDEGF(TRTN)
  CV(23) = TRTN
'  PRINT "TRTN  DEGF ", USING "##.##  "; TRTN; AV(23)
  CALL Thermistor236(672, 11, 2, "2", TCC2, ER%)
  IF ER% THEN PRINT " ERROR TCC2: "; 11, ER%
  TCC2 = TDEGF(TCC2)
  CV(17) = TCC2
  PRINT "TCC2  DEGF ", USING "##.##  "; TCC2; AV(17)
  CALL Thermistor236(672, 7, 2, "2", TCC1, ER%)
  IF ER% THEN PRINT " ERROR TCC1: "; 7, ER%
  TCC1 = TDEGF(TCC1)
  CV(18) = TCC1
  PRINT "TCC1  DEGF ", USING "##.##  "; TCC1; AV(18)
  CALL Thermistor236(672, 12, 2, "2", TEVP2, ER%)
  IF ER% THEN PRINT " ERROR TEVP2: "; 12, ER%
  TEVP2 = TDEGF(TEVP2)
  CV(16) = TEVP2
  PRINT "TEVP2 DEGF ", USING "##.##  "; TEVP2; AV(16)

  FOR K% = 0 TO WTFILTR%
  CALL Thermistor236(672, 5, 2, "2", TEVAP1, ER%)
  IF ER% THEN PRINT " ERROR TEVP1: "; 5, ER%
  CTEVP1 = TDEGF(TEVAP1)
  TEVP1 = (TEVP1 * K% + CTEVP1) / (K% + 1)
  NEXT K%
  CV(19) = TEVP1
  PRINT "TEVP1 DEGF ", USING "##.##  "; TEVP1; AV(19)

  FOR K% = 0 TO TFILTR%
```

```
    CALL VoltsHLG236(672, 24, 0, 1, 0, volts, ER%)
    IF ER% THEN PRINT " ERROR TA1A "; 24, ER%
    CTA1A = -22 + 23.4 * volts
    TA1A = (TA1A * K% + CTA1A) / (K% + 1)
    CALL VoltsHLG236(672, 25, 0, 1, 0, volts, ER%)
    IF ER% THEN PRINT " ERROR TA1B "; 25, ER%
    CTA1B = -22 + 23.4 * volts
    TA1B = (TA1B * K% + CTA1B) / (K% + 1)
    NEXT K%

    CALL VoltsHLG236(672, 24, 0, 0, 0, volts, ER%)
    IF ER% THEN PRINT " ERROR RH1A "; 24, ER%
    RH1A = 0 - .2 * volts
    IF RH1A < .01 OR RH1A > .99 THEN
        RH1A = RHAPRS
        RHAPRS = RH1A
      END IF
'   RH1A = (RH1A * K% + CRH1A) / (K% + 1)
    CV(10) = RH1A

    FOR K% = 0 TO HFILTR%
    CALL VoltsHLG236(672, 25, 0, 0, 0, volts, ER%)
    IF ER% THEN PRINT " ERROR RH1B "; 25, ER%
    CRH1B = 0 - .2 * volts
    IF CRH1B < .01 OR CRH1B > .99 THEN
        CRH1B = RHBPRS
        RHBPRS = CRH1B
    END IF
    RH1B = (RH1B * K% + CRH1B) / (K% + 1)
    NEXT K%

    CV(11) = TA1A
    CV(9) = TA1B
    TROOM = (CV(11) + CV(9)) / 2
    TAV = (AV(9) + AV(11)) / 2
    CV(8) = RH1B
    RHROOM = RH1B
    RAV = AV(8)

    SHUM = SH(TROOM, RHROOM)
    HUM = (HUM * KSH% + SHUM) / (KSH% + 1)
    KSH% = KSH% + 1
'   HUM = HFR * SHUM + (1 - HFR) * PREHUM
```

```
'     PREHUM = SHUM

        PRINT "TROOM DEGF ", USING "##.##  "; TROOM; TAV
        PRINT "RHROOM RH ", USING "##.###  "; RHROOM; RAV
        PRINT "SHUM W    ", USING "#.#####  "; HUM
        PRINT "TSG TE TSE  ", USING "###.##  ###.####  ###.####"; TSG; TE; TSE
        PRINT "HSG HE HSE  ", USING "###.##  ###.####  ###.####"; HSG; HE; HSE
        PRINT "CSG CE CSE  ", USING "###.##  ###.####  ###.####"; CSG; CE; CSE
'     TSP = TSPO + .04 * TSG
        TSP = TSPO - 15 * RHROOM
        PRINT "TSP DEGF ", USING "##.##  "; TSP
        DE = TE - TEP
        TEP = TE
        PRINT "TE   ", USING "##.##     "; TE;
        PRINT "DE   ", USING "##.##     "; DE
        DTSG = TSG - TSGP
        TSGP = TSG
        PRINT "TSG   ", USING "###.##    "; TSG;
        PRINT "DTSG    ", USING "###.##  "; DTSG

  END IF

  NDATA% = NDATA% + 1
  FOR i% = 1 TO NCHAN%
    SUM(i%) = SUM(i%) + CV(i%)
  NEXT i%

' TEMPERATURE CONTROL LOOP (FAN)
  IF CTIME >= TGOTIME AND TSCAN > 0 THEN
        TGOTIME = CTIME + TSCAN
        count% = TCS(TROOM, TINI%)
        CALL OutDtoA210(656, 0, count%, ER%)
        IF ER% THEN PRINT " ERROR OutDtoA FAN "; 0, ER%
  END IF



' HUMIDITY CONTROL LOOP (VALVE)
  IF CTIME >= HGOTIME AND HSCAN > 0 THEN
        HGOTIME = CTIME + HSCAN
        count% = HCS(HUM, HINI%)
        CALL OutDtoA210(656, 2, count%, ER%)
        IF ER% THEN PRINT " ERROR OutDtoA VALVE"; 2, ER%
```

```
        KSH% = 0
    END IF

' COMPRESSOR CONTROL LOOP
    IF CTIME >= CGOTIME AND CSCAN > 0 THEN
        CGOTIME = CTIME + CSCAN
        count% = CCS(TEVP1, CINI%)
        CALL OutDtoA210(656, 1, count%, ER%)
        IF ER% THEN PRINT " ERROR OutDtoA COMPRESSOR"; 1, ER%
    END IF

' DATA LOG LOOP
    IF CTIME >= DLGOTIME AND DLOG <> 0 THEN
        DLGOTIME = CTIME + DLOG
        FOR i% = 1 TO NCHAN%
          AV(i%) = SUM(i%) / NDATA%
          SUM(i%) = 0
        NEXT i%
        NDATA% = 0
        CV(0) = DLGOTIME
        OPEN OFILE1$ FOR APPEND AS #1
        PRINT #1,
        PRINT #1, USING "######.##"; CV(0); AV(1); AV(2); AV(3); AV(4); AV(5);
        AV(6); AV(7); RAV; TAV; AV(12); AV(13); AV(14); AV(16); AV(17);
        AV(18); AV(19); AV(21); AV(22); AV(23); TSE; TSG; CSE; CSG; HSE; HSG;
        TSP; TRIN; TROUT;
        CLOSE #1
        OPEN OFILE2$ FOR APPEND AS #2
        PRINT #2,
        PRINT #2, USING " #####.##"; CV(0); TROOM; RHROOM; CV(19); TSG;
        HSG; CSG; HUM; TMIX; TRTN;
        CLOSE #2
    END IF
LOOP UNTIL INKEY$ = CHR$(27)
END

SUBROUTINES
......................
...................... Subroutines called in the program are manufacturer proprietary
                       information
```