PI-CONTROL IN DUAL DUCT SYSTEMS: A STUDY ON MANUAL TUNING AND CONTROL LOOP INTERACTION

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A Thesis

in

The Centre for Building Studies

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

January 1997

0-612-26001-1

ABSTRACT

**PI-Control in Dual Duct Systems:
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Good HVAC control schemes in buildings help reduce energy use and maintain occupant comfort. PID controllers are widely used in commercial buildings to keep variables such as temperature, pressure and humidity at predefined setpoints. These controllers, when they do not include autotuners, must be carefully tuned. Also, one must consider the effect of interactions occurring among the control loops during the fine-tuning stage so one loop response is not improved to the detriment of another.

This research emphasizes PI-control of dual duct systems. Three tuning methods (Ziegler and Nichols' sensitivity method, simplified IMC-PID method, and Bekker et al. method) were selected and tested on a small scale dual duct system with four decentralized control loops (cooling valve, fan speed, damper and heater). Each controller was tuned according to the best performing set of tuning parameters and the system was then operated under design conditions to observe the interactions among the control loops. Since interaction was minimum, a better understanding of interaction was obtained by introducing a sustained oscillation in each loop, one loop at a time. An oversensitive valve was found to have a severe effect on damper control loop and an oversensitive damper was found to interact with fan speed control. Considering these interaction effects, two sets of optimizing curves were developed for the dual duct system. The use of these curves improved some loop responses without rendering other loops unstable.

ACKNOWLEDGEMENTS

I would like to thank my supervisor, Dr. Mohammed Zaheer-uddin, for his guidance, encouragement and financial support during this work. He provided me with the opportunity to study within a field of great interest to me. Thank you.

I would also like to express my gratitude to Dr. Paul Fazio, director of the Centre for Building Studies, for his financial support through the Fonds pour la formation de chercheurs et l'aide à la recherche (FCAR-Centre de recherche).

À Joe Hrib et Jacques Payer, merci les gars pour votre aide si précieuse et votre compréhension lors des moments difficiles.

Pour terminer, j'aimerais remercier ma famille pour tout l'appui et le support moral qu'elle m'a apportés tout au long de mes études.

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NOMENCLATURE

ΔF	= pressure change
C_{ss}	= process steady state value
e	= error signal
K_d	= derivative gain
K_{GR}	= gain reduction factor
K_i	= integral gain
K_p	= proportional gain, or controller gain
K_s	= system gain
K_u	= ultimate proportional gain
L	= dead time, also expressed as θ
M	= process sensitivity
P	= ultimate sensitivity
P_u	= ultimate period of oscillation, also expressed as T
P_v	= velocity pressure
PB	= proportional band
R	= reaction rate
t	= time
T_d	= derivative time or pre-act time, also expressed as τ_d or D
T_i	= integral time (min. per repeat), also expressed as τ_i or I
T_v	= actuator operation time
U	= reset rate (repeats per min.) = 1/integral time (Pessen (7) refers to it as R)
u	= step input
V_o	= offset adjustment parameter
V_p	= controller output

Greek Symbols

μ	= self-regulation index of process = RL/C_{ss}
ν_1	= integral parameter
ν_2	= sensitivity parameter
ν_3	= derivative parameter
τ	= time constant, also expressed as Z

TABLE OF CONVERSIONS

Measure	SI Units	Imperial Units
Length	25.4 mm	1 in.
Temperature	x °C	32+(x°C)*1.8 °F
Pressure	248.8 Pa	1 in.WG
Flow	1 L/s	15.8 gpm
Flow	1 L/s	2.11 cfm
Power	1 Watt	3.412 BH
Power	3.52 kW	1 Ton of refrigeration = 12000 BH

1.0 INTRODUCTION

The goal of HVAC design is to provide comfort to the occupants of a building. Most of the time, an HVAC system is not used to its full capacity, therefore it must be complemented with a good control scheme to maintain comfort under any conditions. Good control will also reduce energy use by keeping the process variables (temperature, pressure, relative humidity, etc.) to their setpoint efficiently. This is a very important factor considering air handling units can use up to 30% of electrical energy in buildings with electrical cooling and fuel fired heating (1). The efficiency of a control scheme to maintain comfort depends on the selection of appropriate setpoints and proper controller tuning. Tuning can be performed manually following one of the several tuning rules available, automatically with autotuners which have a special feature to be turned on and off by an operator, or automatically with adaptive controllers which are self-tuning and do not need an operator at all. Currently, most buildings are equipped with controllers which are tuned manually or with autotuners. In these cases, interactions between the different control loops found in the system can occur. This results in one loop being tuned to the detriment of another. Some researchers have already noted that problem, however no generalized steps were defined to minimize interaction.

Until now, tuning rules were developed by testing them through several simulations and/or on one specific control loop. Some researchers compared their new set of rules with older ones to show their improved responses and robustness. However, their comparisons were based on one control loop only. One software, CAT (2), allows the comparison of four tuning methods but requires a complete mathematical model of the system to be tuned. Furthermore, the interaction occurring between the numerous control loops of HVAC systems is rarely discussed. One article by Krakow et al. (3) was found to directly assess a problem of interaction between compressor and evaporator fan speed loops during temperature and relative humidity control.

1.1 Objectives of the Research

This research will focus on PID controllers used in commercial/institutional buildings which may still be manually tuned because they are infrequently subjected to large changes in loads or setpoint.

The main objectives are:

1. To review existing tuning methods applicable to HVAC systems;
2. To apply three tuning methods on four control loops and compare the results;
3. To evaluate the interaction between four control loops of a dual-duct HVAC system;
4. To develop fine-tuning curves for the system under study.

To reach these four objectives, a laboratory size dual-duct system was built. It includes four control loops activated by analog PID controllers which maintain air temperature or air flow rate in the system. The study investigates a small scale system instead of full scale because it will be easier to analyze the effect one parameter may have on the rest of the system. In full scale systems this is very complicated, if not impossible, to study. The work will consist of finding which tuning rule, out of the three tested, provide the best response for each loop under PI-control. With each controller properly tuned individually, the system will be operated under design conditions to determine between which loops interaction occurs. Then, simulation of poorly tuned controllers emphasizing interaction among the loops will be analyzed so fine-tuning curves can be developed based on interaction effect.

Therefore, this work will document interaction occurring among four of the control loops of a dual-duct system. It may help operators identifying the cause of poor loop responses, and offers heuristic fine-tuning curves which will minimize interaction. These curves however will directly apply to the system under study only.

1.2 Uniqueness of this Work

This research differs from others because it tests a complete small scale HVAC system instead of one specific section only. Furthermore, it assesses interaction occurring among four control loops in the system, which has not been performed before, and tries to minimize it through fine-tuning curves.

2.0 LITERATURE REVIEW

In the past, several tuning methods have been developed to improve the tuning process of HVAC systems. The methods were tested through computer simulations or on one control loop of an HVAC system. More recently, research has been mainly oriented towards the development of adaptive controllers including fuzzy logic controllers and neural-network-based controllers. However, most commercial and institutional buildings are equipped with proportional-integral-derivative (PID) controllers which is why such controllers were installed on the laboratory HVAC system. Therefore, this literature review will emphasize PID controllers by covering the basic theory on PID controllers and reviewing existing tuning rules. It also includes an overview of other experimental research performed in the field of HVAC control and an overview of automated controllers. Among the existing tuning rules, three tuning methods will be selected to be tested on the four control loops of the laboratory HVAC system.

2.1 Basic Theory on PID Controllers

The heating and cooling loads of a building vary along with the period of the day and the period of the year. In order to provide comfort, the capacity of the HVAC system equipment must be adjusted according to those loads. Therefore, a control system is required in which a typical control loop consists of a controller, an actuator and one or several sensors. The value of the variable to be controlled such as temperature, pressure or humidity is obtained from a sensor and compared to a setpoint value. If there is a difference between the controlled variable and the setpoint, the controller receives an error signal and adjusts the actuator position to eliminate the error (figure 2.1).

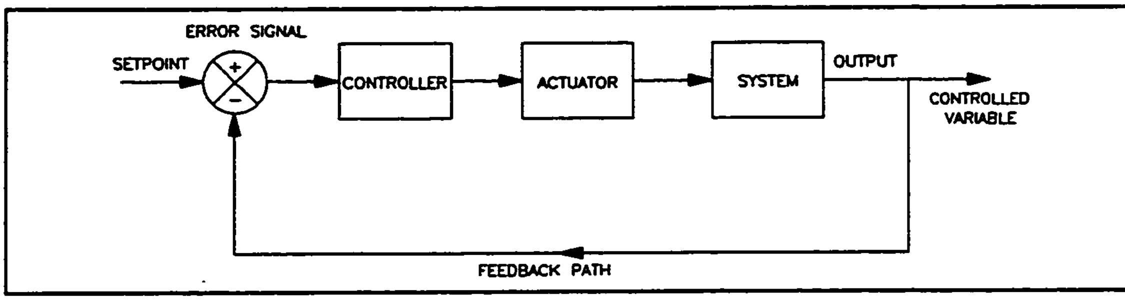

Figure 2.1 Control Loop Schematic

Controllers have three different types of control actions: proportional (P), integral (I), and derivative action (D). Depending on the accuracy and the speed of response required, an operator can configure a controller using one, a combination of two, or the three control actions. In HVAC applications, P and PI controllers are commonly used.

2.1.1 Proportional Control Action

A controller working under proportional action produces a response proportional to the error signal it receives, and is expressed mathematically as follows:

$$V_p = K_p e + V_o \tag{2.1}$$

where V_p = controller output
K_p = proportional gain
e = error signal
V_o = offset adjustment parameter

Proportional action cannot eliminate the error completely which results in an offset from the setpoint. The offset can be reduced by increasing the controller gain but this will also increase the controller instability (4,5).

2.1.2 Integral Control Action

To eliminate the offset produced by proportional action, an integral action must be determined during the controller configuration. The control equation becomes:

(2.2)

where K_i = integral gain
t = time

"The integral term has the effect of continuing to increase the output as long as the error persists, thereby driving the system to eliminate the error (5)." At that time, the integral action is deactivated and the actuator stays in the position required. The value given to the integral gain should be selected with care because a value that is too high may affect the response stability, increase the time required to reach the setpoint, or cause overshoot (4,6).

2.1.3 Derivative Control Action

Derivative control action accelerates the response of the proportional action so a steady-state output can be reached faster. It reduces the overshoot but is unable to eliminate the offset. The derivative term is expressed as $K_d de/dt$ and therefore comes into action when the error varies (4,5). It is added to the proportional and integral terms to obtain a PID controller for which the output equation is:

$$V_p=K_pe+K_i\int edt+K_d\frac{de}{dt}+V_o \tag{2.3}$$

where K_d = derivative gain
de/dt = derivative of the error

In the above equations, the integral gain can be expressed in terms of reset rate or integral time and the derivative gain can be expressed in terms of derivative time. The relations are:

$$K_i=\frac{K_p}{T_i} \qquad \text{or} \qquad K_i=K_pU$$

and $$K_d=K_pT_d$$

where T_i = integral time
U = reset rate
T_d = derivative time

The PID-control equation becomes:

$$V_p=K_p(e+\frac{1}{T_i}\int edt+T_d\frac{de}{dt}) \tag{2.4}$$

To configure a controller, the proportional gain, integral time and derivative time must be determined and there are several different tuning rules to do so.

2.2 Tuning Rules for Controllers

There are three different ways of obtaining the parameters required by the tuning methods to define the dynamic properties of a controlled process:

1. Experimentally, by testing the process in a closed loop;
2. Experimentally, by testing the process in an open loop; or
3. Mathematically.

The tuning methods for which the parameters are obtained experimentally by testing the process in a closed loop require several trials to determine the critical proportional gain generating an amplitude ratio of one. While searching for the critical gain value, "limit cycling" or saturation could be reached and mistaken for the critical gain because they also produce an amplitude ratio of one. The tuning methods for which the parameters are obtained experimentally by testing the process in an open loop do not have those problems. However, the feedback control is not taken into account. A loop which is stable when it is open could become unstable when it is closed. Also, open loops require the analysis of the process-reaction curve to determine the parameters. Any inaccuracy while analyzing the curve can produce a large error. The tuning methods for which the parameters are obtained mathematically are the most accurate. Mathematical models allow the designer to know how the system is going to react before it is installed. However those models are very complex (7).

Tuning based on experimental tests started with Ziegler and Nichols in 1942 (6). This method is still used in the industry and is said to provide a good basis to obtain the optimum settings, although the rules apply only to the specific controller used during the tests. Furthermore, the settings cannot be used for processes with a small ratio of time constant over dead time and for processes with only three time lags (7).

Ziegler and Nichols used the closed-loop method to establish the control settings. They first kept the integral and derivative actions inactive to find that reducing by half the optimum proportional gain, K_u, which is the gain obtained when amplitude ratio equals one, results in a quarter wave decay ratio and provides a good compromise between the offset and the time required to reach steady-state. They then used the "ultimate period of oscillation", P_u, which is encountered at the optimum gain, as an indicator to adjust the reset rate and derivative time. Ziegler and Nichols also determined settings using the open-loop method. They obtained the process-reaction curve for a specific pressure change and used it to establish the dead time, L, and the reaction rate, R (figure 2.2) (6).

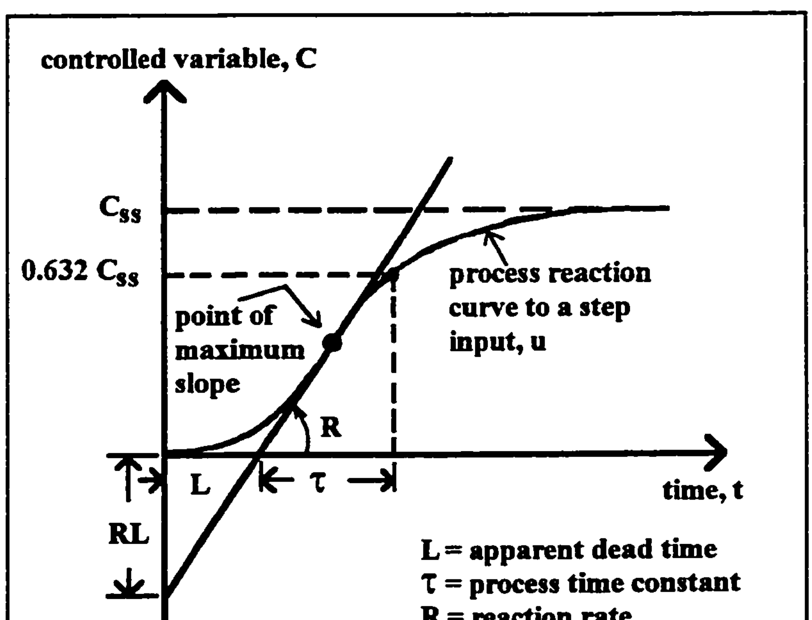

Figure 2.2 Process-Reaction Curve for Dead Time First Order Process

The optimum controller settings obtained from the two methods are expressed as follows:

Controller settings for a P controller:

$K_p = 0.50K_u = \Delta F/RL$

Controller settings for a PI controller:

$K_p = 0.45K_u = 0.9\Delta F/RL$
$U = 1.2/P_u = 0.3/L$

Controller settings for PID controller:

$K_p = 0.60K_u = 2\Delta F/RL$
$U = 2.0/P_u = 0.5/L$
$T_d = P_u/8.0 = 0.5L$

Cohen and Coon (8) developed tuning rules for processes with a dead time period. The relations found depend on dead time, L, reaction rate, R, and process steady-state value, C_{ss}, obtained from the process-reaction curve (figure 2.2). The following linearized equations determine the controller dimensionless parameters which are then used to find the controller settings. Note that the amplitude ratio is obtained from the fundamental harmonic mode and that Cohen and Coon only considered load disturbances, not setpoint changes.

Controller dimensionless parameter for P controllers:

$\nu_2 = 1.03+0.35\mu$

Controller dimensionless parameters for PI controllers:

$\nu_2 = 0.90+0.083\mu$
$\nu_1 = 0.27+0.6\mu$

Controller dimensionless parameters for PID controllers:

$\nu_2 = 1.35+0.25\mu$
$\nu_1 = 0.54+0.33\mu$
$\nu_3 = 0.5$

where ν_1 = integral parameter
ν_2 = sensitivity parameter

ν_3 = derivative parameter
μ = self-regulation index of process = RL/M
M = process sensitivity, in/psi (for Cohen and Coon's test)

When there is no interaction between the integral and derivative actions, it is possible to determine the controller parameters as follows:

$K_p = \nu_2/RL$ = proportional gain
$U = \nu_1/\nu_2 L$ = reset rate
$T_d = \nu_3 L/\nu_2$ = derivative time

But if there is interaction, relationships between the controller dimensionless parameters (ν_1,ν_2,ν_3) and the controller settings (K_p,U,T_d) must be determined.

In 1982, Yuwana and Seborg (9) developed a new on-line method to determine the process parameters needed for tuning. This method requires only one experimental closed-loop test, mainly a step change in the controller setpoint. The process model is reduced to a first-order plus time delay transfer function and combined to a Padé approximation to obtain the equations for the process parameters. These equations are expressed in terms of peaks, minima, steady-state value and period of oscillation found on the step setpoint change response curve, and open-loop gain. Once the process parameters are established, they can be input in any tuning rule to determine the controller settings. The researchers state that their method can be used for control loops which are open-loop unstable; it is not as sensible as the process-reaction curve method to controller calibration error, and it can be applied to higher order processes. They say it provides good initial settings which could then be improved by fine-tuning. However, this method is not valid for processes with long time delays.

In 1984, Pinnella (10) studied one specific control loop in an HVAC system, the supply duct static pressure control loop. His system consisted of a dual duct system with the cold duct sealed to simulate a single duct VAV system with 100% return air. The discharge air temperature control was not used and the zone dampers were controlled manually. He defined a model for each component of the system and then defined a model for the entire system. That model was validated and simplified to facilitate the controller selection and its tuning. Instead of using a P controller or a PI controller to modulate fan speed, he suggested to use an integral only controller. He concluded that it eliminates the offset and is easier to tune than PI controller because it requires the adjustment of only one gain. It does not respond quite as fast as a PI controller but since HVAC systems encounter slow load changes, he did not consider it a disadvantage.

In 1994, Pessen (7) determined new values for the proportional gain, reset rate and derivative time to improve the PID controller-tuning formula found by Ziegler and Nichols. To determine the best combination of reset rate and derivative time, he decided that the time integral of the absolute error should be minimum. The parameters were obtained experimentally by testing the process in a closed loop and the disadvantages were eliminated by using the "damped oscillation method", i.e., by searching for the proportional gain that creates a 0.25 decay ratio instead of the critical proportional gain. Pessen also found new settings for cases where the Ziegler-Nichols settings are not applicable mainly when the dead time is very large compared to the first-order time constant and when the system consists of only three first-order time lags. Computer simulations were used to determine which equation would have the best tuning results.

Controller settings for non-interacting PID controllers:

$K_p = 0.7K_u$
$U = 2.5/P_u$
$T_d = P_u/6.7$

Controller settings for interacting PID controllers:

$K_p = 0.7K_u$
$U = 1.75K_u/P_u$
$T_d = K_uP_u/9.6$

PI controller settings when L/τ is large:

$K_p = 0.25K_u$
$U = 6/P_u$

PID controller settings when L=0 and there are 3 first-order time lags:

$K_p = 2.5K_u$
$U = 4/P_u$
$T_d = P_u/5$

Cohen and Coon and IMC-PID rules (Internal Model Control) (11) also provide better values than Ziegler and Nichols when the dead time is very large compared to the first-order time constant.

The simplified IMC-PID tuning rules developed by Fruehauf et al. (11) are used in the chemical industry. The rules were established for general, flow and level loops. According to Fruehauf et al., these rules benefit from the advantages of the IMC-PID rules but are much simpler to apply. They also provide a good response when applied to nonlinear processes and there is no oscillation or overshoot in the closed-loop response when a load step change occurs. The controller response to changes in the original process parameters showed that the simplified IMC-PID rules are more robust than the Ziegler and Nichols rules, or the Cohen and Coon rules. To apply the simplified IMC-PID rules, an open-loop test must be performed to obtain the process-reaction curve and determine the dead time and reaction rate. The controller settings can then be determined for temperature and pressure control loops using the following relationships:

For PI controllers in general loops:

Controller settings when $\tau/L > 3$: $K_p = 1/2RL$, $T_i = 5L$

Controller settings when $\tau/L < 3$: $K_p = 1/2RL$, $T_i = \tau$

Controller settings when $L < 0.5$: $K_p = 1/R$, $T_i = 4$

For PID controllers, add the following derivative time, T_d, to the above rules: $T_d = 0.5L$.

It is also possible to obtain the controller settings based on equations using the system gain ($K_s = C_{ss}/u$), dead time, time constant and tuning constants. Kaya et al. (12) determined the tuning constants for three PID controls which they call: classical, noninteracting, and industrial. The constants are found for three tuning criteria: the minimum integral of square error (ISE), the minimum integral of absolute error (IAE), and the minimum integral of absolute error multiplied by time (ITAE); and for load disturbance or setpoint change. To determine the tuning constants, they kept the ratio of dead time over the time constant between 0 and 1.

Some tuning rules for PI controllers used with first-order HVAC processes, such as the rules developed by Bekker et al. (13), rely on a more theoretical background. Bekker et al. rules are based on pole-zero cancellation and root locus techniques. The tuning method was verified using computer simulations and a real humidification control system. According to Bekker et al. the following rules should be applied when the loop is equipped with a fast-acting actuator for which the dead time is more than half the system time constant or the actuator operating time is less than 20τ.

Rules for PI controllers linked to a fast-acting actuator:

$$K_p = \frac{\tau}{LK_s} e^{-1}$$

$$K_i = \frac{1}{LK_s} e^{-1}$$

where the time constant, τ, and the system gain, K_s ($K_s = C_{ss}/u$), can be determined using any "system identification technique".

On the other hand, if the control loop is equipped with a slow actuator such as an electrical actuator (also called constant-velocity actuator) for which the dead time is less than half the system time constant, then the following rules should apply:

Rules for PI controllers linked to a slow-acting actuator:

$$K_p = \frac{\tau}{LK_sK_{GR}} e^{-1}$$
$$K_i = \frac{1}{LK_sK_{GR}} e^{-1}$$

where, $$K_{GR} = a_6 X^{[a_1 \ln X + a_3 \ln Y + a_4]} Y^{[a_2 \ln Y + a_5]}$$

a_1 = +0.1092
a_2 = +0.0121
a_3 = -0.0760
a_4 = -0.2685
a_5 = +0.1026
a_6 = +1.1814
X = L/τ
Y = T_v/τ
T_v = actuator operation time
K_{GR} = gain reduction factor

2.3 Experimental Research in HVAC Control

Different types of experimental researches have been conducted in the past. Some experiments were run on full size HVAC systems which tested control strategies or sequences. However they do not discuss controller tuning. For example, Shirey (14) tested two methods of humidity control in an art museum consisting of indoor fan cycling strategies or involving heat pipe heat exchangers. They both provided a more accurate humidity control and reduced the energy consumption. Stotz and Hanson (15) evaluated the use of heat recovery and aquifer wells in a high school to provide an energy efficient HVAC system. They discussed the control sequence and determined the different setpoints required to maintain thermal comfort and save energy.

Other researches are more similar to the current research work and use a laboratory scale system to run tests on control loops. Nesler and Stoecker (16) analyzed the effect of proportional and integral gains and non-linearity on a heating coil discharge air-temperature loop and provided recommendations related to tuning PI digital controllers. Green (17) worked on a new hydronic system in which the usual control valves were replaced by variable-speed pumps. His objective was to facilitate design and commissioning and reduce energy consumption at a competitive price. He used Ziegler and Nichols open-loop tuning method and Chien et al. method (18) to define two sets of parameters for PI-control. He selected the tuning parameters obtained from the Chien et al. method to tune his controller but did not provide any comparison between the two tuning methods in his paper.

Finally, experimental systems can be built to validate numerical models. For example, Krakow and Lin (19) developed a numerical model to determine the PI gains for digital controllers which would maintain a constant discharge pressure at different loads. They used an experimental system similar to the modeled system for validation. Both systems provided similar results with some differences due to unaccounted inertial effects in the numerical model.

2.4 Automation of Tuning Procedures

Poorly tuned systems result in higher energy consumption and reduced comfort, so precise tuning is very important. If tuning is performed manually, the controller parameters must be selected properly. This is not easy and requires a well trained operator. It is also very time-consuming, therefore costly. The cost of manual tuning increases even more when retuning is required periodically to account for important load changes caused by different factors such as weather conditions or change in occupancy. The controller could be tuned conservatively at commissioning to avoid retuning, however a sluggish response would be obtained most of the time (20,21,22).

The costs of tuning were lowered by automating the tuning procedure. This reduced commissioning time and allowed system operation to be performed by less skilled operators. There are three different levels of automation:

1. Computer-assisted controller tuning: Tuning is simplified by the use of a software which guides the operator through all the steps of tuning (20). One example of computer-assisted controller tuning would be the use of CAT (computer-aided tuning) software which allow to test and compare four different tuning methods as long as a mathematical model of the system to be tuned is available (2).
2. Automatic tuning: Autotuners are controllers similar to conventional controllers but with an extra feature which allows them to define the controller parameters when the feature is turned on by the operator. When this feature is off, the controller works in normal operation and the controller parameters are kept constant (20,23).
3. Adaptive controller: Adaptive controllers continuously adjust their parameters during operation to accommodate for load or setpoint changes. They do not

require any intervention from the operator unless malfunction occurs (20,23).

The automatic tuning and adaptive controller technologies apply to PI as well as PID controllers. Autotuners and adaptive controllers must perform at least as well if not better than conventional controllers (manually tuned controllers).

Adaptive controllers effectively control processes for which it is very difficult to determine the controller parameters and systems with frequent load changes. They enter the "learn mode" (tuning mode) at commissioning, and subsequently, every time the control response goes below an acceptable limit. In the "learn mode", a system identifier constructs a mathematical model of the process. This model is used to determine the new parameters needed to provide the desired response. Once the new parameters are determined and the controller provides proper response, the controller returns to the "normal mode" (24).

Adaptive controllers require constant excitation to perform properly; consequently, the control algorithm must be protected by a software which will, as stated by Brandt, "detect and correct specific control conditions" such as lack of excitation (25).

2.5 Conclusions

Derivative control action can significantly improve control loops requiring a very accurate response. However, HVAC systems such as the one under study have to cope with slow load changes and few setpoint changes so that PI controllers can perform the control task properly. Actually, a simple proportional-only controller may perform well if the offset is not important. For example, an offset of 0.25°C for a temperature control loop will not affect comfort and should not considerably affect the energy consumption either. Therefore, only tuning rules providing settings for PI controllers will be considered in this research. From the literature review, three rules were chosen: the following two recent set of rules,

- Simplified IMC-PID tuning rules (11), 1994
- Bekker et al. tuning rules (13), 1991

and the first experimentally based set of rules,

- Ziegler and Nichols' tuning rules (6), 1942.

The process parameters required in those tuning rules will be determined using the process-reaction curve method or the sensitivity method.

3.0 SYSTEM DESCRIPTION

In order to compare different tuning rules and study the interaction occurring between the different control loops found in HVAC systems, an air handling unit (AHU) with four decentralized control loops was built in the laboratory. The system represents the main section of a multizone AHU, that is, the supply fan, the two ducts - hot and cold -, the mixing box and the mixed air duct. Five sensing stations were installed to measure flow and temperature. Two of those stations are also equipped with a humidity sensor. Humidity control is not part of the present research but could be included in future research. The four control loops of interest are:

1. Fan speed control
2. Damper control
3. Cooling valve control
4. Heater control

This chapter provides detailed information on the system components, the control scheme and the controllers' configuration. It also includes the instruments' calibration required to obtain accurate readings.

3.1 Air Handling Unit Components

The test facility built in the laboratory is the main section of a multizone air handling unit. It includes a supply fan, two main ducts -hot and cold-, a mixing box, and a mixed air duct (figures 3.1 and 3.2). This type of system was selected because it can create a load on the setpoint without having to link to the zone or having to know the zone conditions. All ducts have a constant cross-section area of 1ft x 1ft. The cold duct is equipped with a chilled water cooling and dehumidifying coil and two dampers to regulate the air flow. One damper is located in the duct itself while the other is located in a branch to exhaust air out of the system. The hot duct is equipped with an electrical heater and two dampers also. Just like in the cold duct, one damper is located in the duct itself and the other damper is located in a branch to exhaust air out of the system. Both hot and cold ducts are also insulated up to the mixing box (not shown on figure 3.2) to avoid heat losses/gains. Room air is supplied to the fan and flows into the hot and cold ducts. The warm and cool air streams then mix in the mixing box and return to the room through the mixed air duct. The hot and cold ducts are both equipped with two sensing stations for flow and temperature measurements and there is a fifth sensing station in the mixed air duct. Finally, the control panel consists of four analog controllers, one flow indicator with a channel selector, a potentiometer for manual fan speed control and a fan speed indicator.

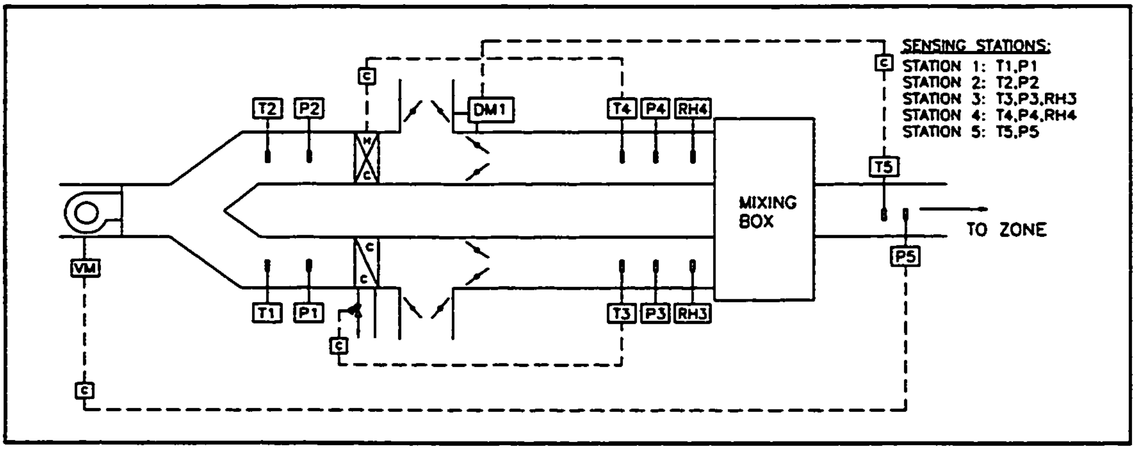

Figure 3.1 Control Diagram of HVAC System

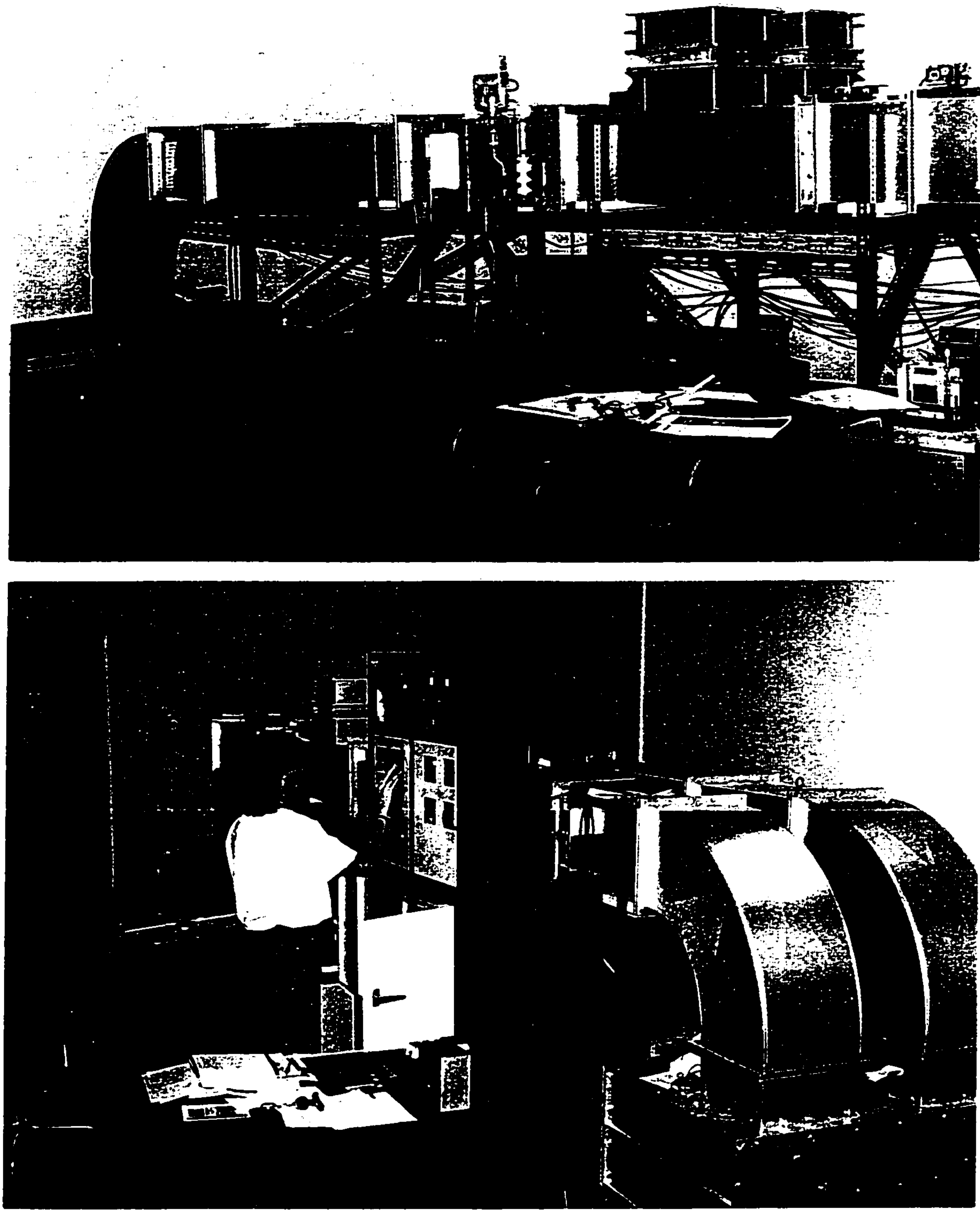

Figure 3.2 Laboratory Test Facility

3.1.1 Fan Speed Control Loop

Controlling the fan static pressure is a very important issue. As stated by Warren and Norford (26), one of the major energy use in commercial buildings is the supply air fan power. Several papers (26,27,28,29) discuss this subject for VAV systems and three important selections were identified as affecting the energy consumption of a fan:

1. Selection of the static pressure sensor(s) location;
2. Selection of the controller type to be used (P, PI, PID);
3. Selection on how the flow rate is to be controlled.

The location of the pressure sensor which was linked to the fan speed controller can be selected in different ways. In general, designers locate it two thirds length away from the supply fan to benefit from some fan energy savings. However, there are instruments such as pressure independent type VAV box (27) and strategies such as the static pressure reset strategy requiring direct digital controls (26) which allow reduction of the fan power to the lowest level possible without affecting comfort. Finding a proper location for the sensor which controls the fan speed was not a complicated issue in this research. Because the laboratory AHU does not include all the branching a real commercial system would, the best place to locate the sensor was at station 5, in the mixed air duct (figure 3.1).

Haines (28) explained in one of his papers that P-control causes a higher fan energy consumption than PI-control, because of the offset which is always present. At full load, it is possible to eliminate that difference in consumption by lowering the setpoint when

P-control is used. This way, the offset merely brings back the setpoint to the required value (figure 3.3a). However, as the load decreases, the offset increases, increasing the fan energy consumption compared to PI-control (figure 3.3b).

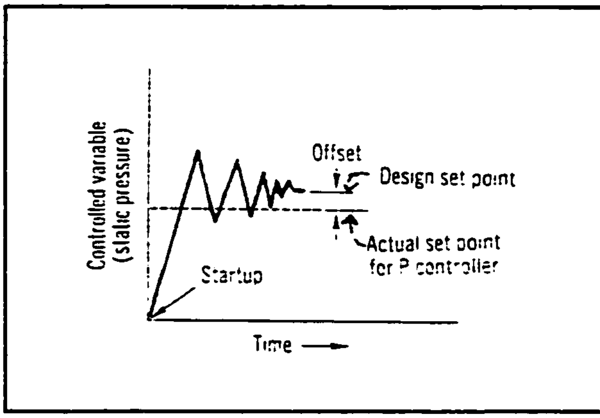

Figure 3.3a Adjustment of Setpoint for P-control (28)

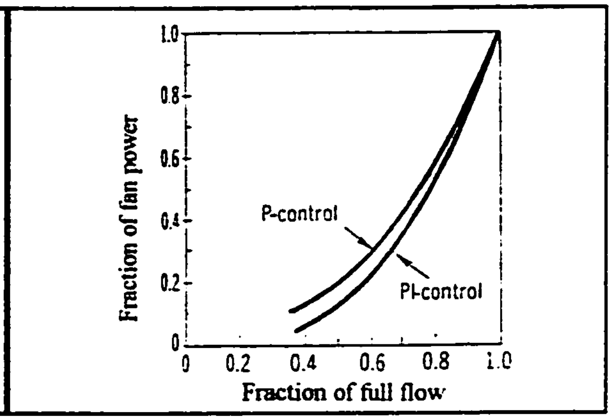

Figure 3.3b Fan Power for P and PI-Control (28)

Brothers and Warren (29) determined the effect of different flow-control techniques, control algorithms, and fan types on fan energy consumption. To analyze the fan energy consumption, they simulated a 10,000ft^2 commercial building using DOE-2.1. Three flow-control techniques were simulated (system dampers, inlet vane control, and variable speed control) as well as two control algorithms (P-control and PI-control), and two fan types (backward-inclined centrifugal fan and vane-axial fan). Having studied fan consumption for five locations, they came to the conclusion that a centrifugal fan consumes less energy than a vane-axial fan; that the use of variable speed control provides significant savings compared to inlet vane control and system damper control; and that PI-control reduces the fan energy use when compared to P-control.

These conclusions support the decisions taken regarding the selection of the fan speed control loop components such as the supply fan and the controller.

a) Supply Fan

The selection of a centrifugal fan with backwardly inclined blades and equipped with a variable speed motor to supply air to the laboratory system (figure 3.4) was justified, energy wise, based on Brothers and Warren's conclusions. The fan speed varies between 360 RPM and 1600 RPM producing an average air flow of 860 cfm. Appendix A provides more details.

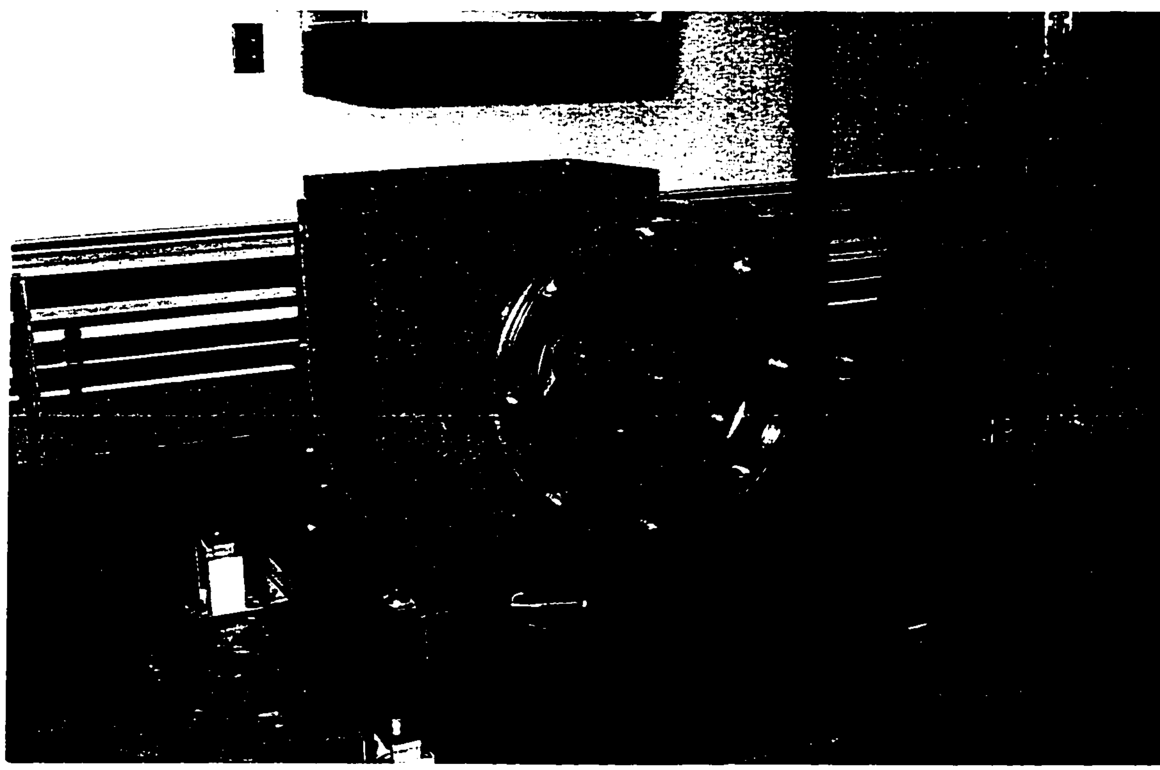

Figure 3.4 Centrifugal Fan

b) Fan Speed Controller

The purpose of the fan speed controller is to vary the fan speed, through a control device (or actuator), in order to maintain a constant velocity or static pressure in the mixed air duct (figure 3.5). For example, when the process variable (in this case, velocity pressure in the mixed air duct) is above the setpoint, the fan speed must be reduced. The controller sends a current signal to the actuator which in turn sends a voltage signal to the fan to reduce its speed. The fan speed controller chosen is a Eurotherm 847 which can be connected to the computer. The following control algorithms are available: a PID algorithm, a PID algorithm with a starting ramp or an ON/OFF control scheme. Appendix A contains more information on the controller and the fan speed control device, while the controller's configuration record sheet is included in Appendix B.

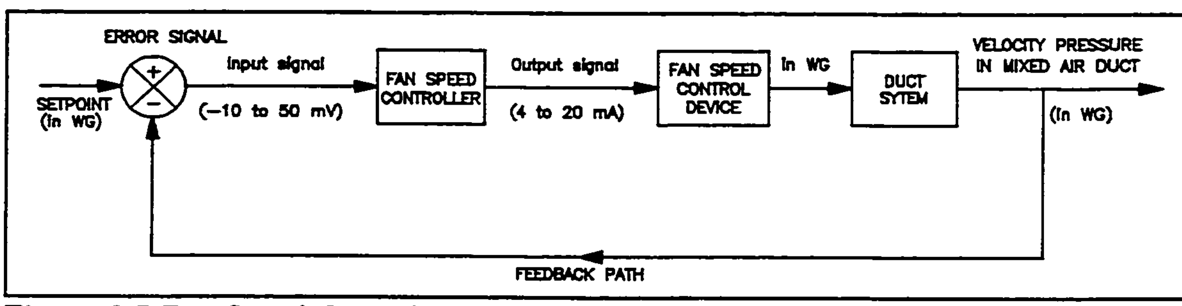

Figure 3.5 Fan Speed Control Loop Schematic

3.1.2 Cooling Valve Control Loop

The cooling valve control loop consists of a cooling coil, a valve which modulates the cold water flow into the coil, a refrigeration system and a controller which automatically sets the valve position.

a) Cooling Coil

The cooling coil selected has a capacity of 12,000 BH (3517 W). Chilled water runs through the coil in countercross flow. This type of design provides a higher coil efficiency than parallel-flow design because of a higher overall temperature difference between air and water throughout the coil. Furthermore, the coil is equipped with two headers which reduces temperature stratification from the bottom to the top of the coil (figure 3.6).

b) Valve

The valve is a Honeywell two-way valve with ½in. diameter pipe connections. It has a flow coefficient of $C_v = 4$ gpm/(psi)$^{1/2}$, and its actuator provides linear movement to the valve plug. Figure 3.6 shows how the valve is connected to the cooling coil. The circulating pump is too powerful for the water flow required by the system so a bypass from the main pipe to the tank was installed. This also provides good water mixing in the tank thereby avoiding temperature stratification.

c) Refrigeration System

The refrigeration system is a hot gas bypass system built in the laboratory (figure 3.7). It can provide 2 tons of refrigeration and uses freon-22 as refrigerant. Pressure in the suction line was set to 87 psig in order to obtain a water temperature of 6.5°C during design conditions. When the HVAC system operates under conditions different from design, there is no automatic control to keep water at a constant temperature so control is performed through manual ON/OFF. Antifreeze was added to the water to prevent deterioration of the evaporator and the cooling coil.

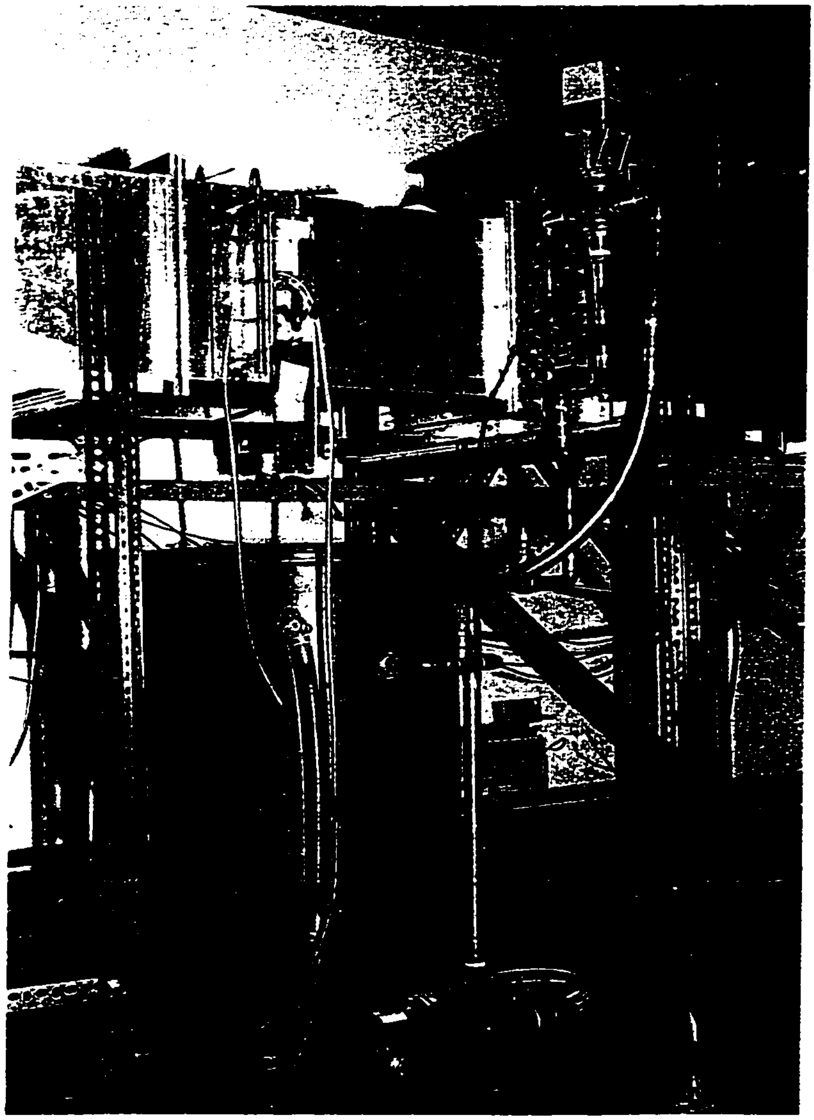

Figure 3.6 Cooling Coil and Valve Installation

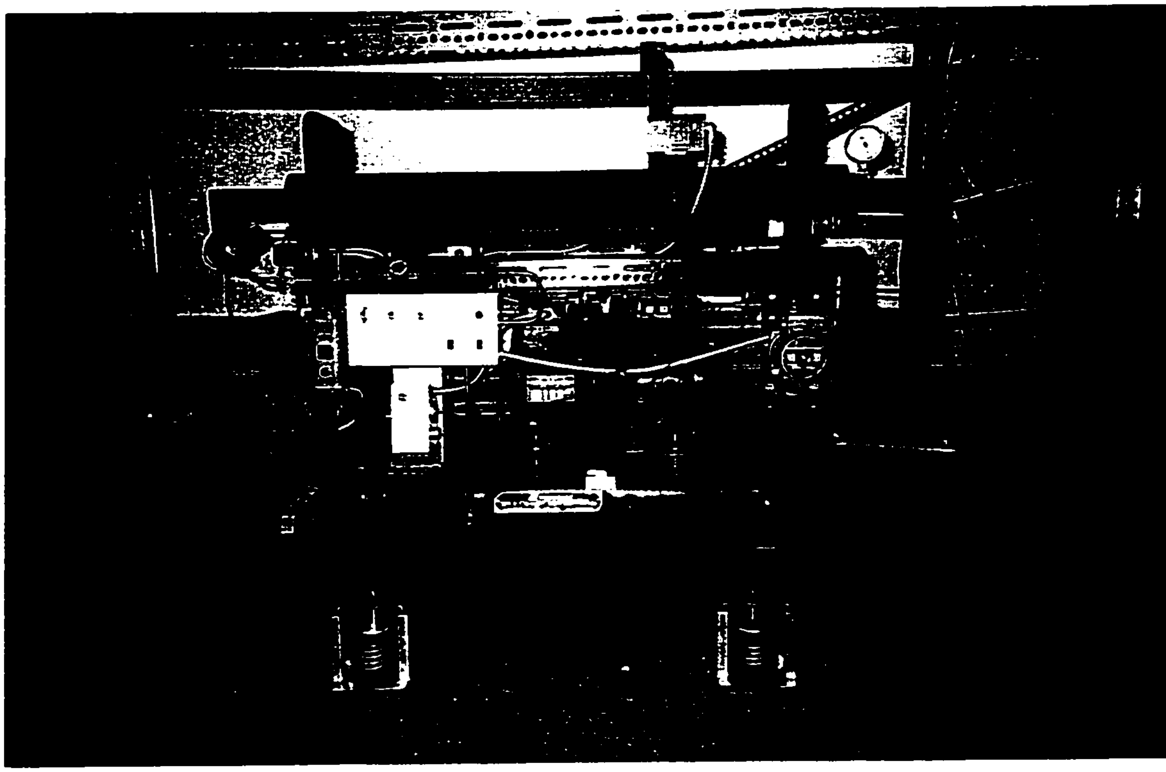
Figure 3.7 Refrigeration System

d) Valve Controller

The purpose of the cooling coil valve controller is to vary the water flow rate in the coil, through a valve actuator, in order to maintain the air in the cold duct at a constant temperature (figure 3.8). For example, when the process variable (air temperature at station 3 in cold duct) is above the setpoint, the valve plug must move upward to allow a higher water flow rate in the coil. The controller receives the temperature signal in voltage, determines the amplitude of the error and sends a current signal to the actuator so the latter can move the valve plug. The valve controller chosen is a Honeywell UDC 2000 Mini-Pro. The following algorithms can be used for control: a PID algorithm, a PD algorithm with manual reset, a three position step control scheme or an ON/OFF control scheme. Appendix A contains more information on the controller and the valve actuator. Furthermore, the configuration record sheet is included in Appendix B.

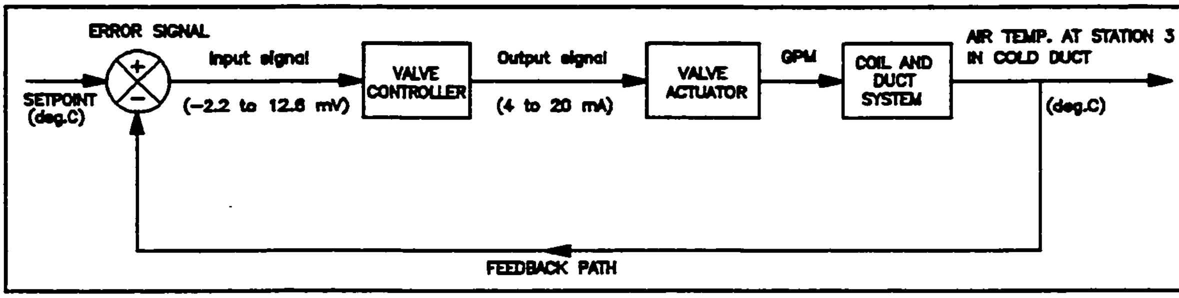

Figure 3.8 Cooling Valve Control Loop Schematic

3.1.3 Heater Control Loop

The heater control loop consists of an electrical heating coil and a controller to determine the amount of heat required according to a selected setpoint.

a) Electrical Heating Coil

Heating is performed by an electrical coil having a maximum capacity of 4800W at 120V input. The coil consists of two double rack heaters and each rack has an element of 1200W. The control mode of the heating coil is the ON/OFF type. Safety features were installed to prevent overheating when the fan is off. See Appendix A for more details.

b) Heater Controller

The purpose of the heater controller is to vary the ON/OFF time of the electrical heating coil, through a control device, in order to maintain the air temperature in the hot duct at a desired temperature (figure 3.9). For example, when the process variable (air temperature in hot duct) is above the setpoint, the heater must temporarily be switched off to reduce the air temperature in the hot duct. The controller receives the temperature signal in voltage, determines the amplitude of the error and sends a current signal to the actuator which in turn controls the voltage signal to the heating coil. The heater controller is a Honeywell UDC 2000 Mini-Pro, just like the valve controller. Appendix A contains more information on the controller and the heater control device. Furthermore, the configuration record sheet is included in Appendix B.

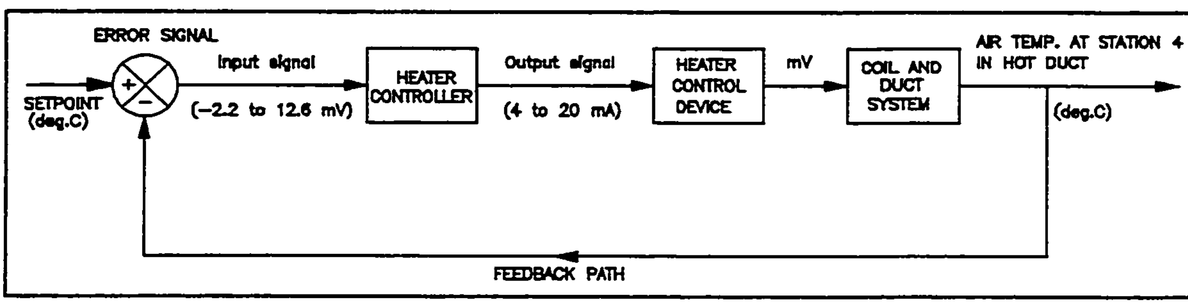

Figure 3.9 Heater Control Loop Schematic

3.1.4 Damper Control Loop

Only the dampers in the hot duct were automated by a controller. The exhaust damper in the cold duct was kept fully closed and the cold duct damper was manually controlled.

a) Dampers

The dimensions of the dampers are 12in. x 12in.. They have opposed blade position which distributes air more uniformly than parallel blade position (figure 3.10). During full operation of the system, the hot duct damper is linked by an arm to the exhaust damper of the same duct in order to open the exhaust damper when the hot duct damper closes. When the hot duct damper is fully closed the exhaust damper is ¼ open to avoid overheating.

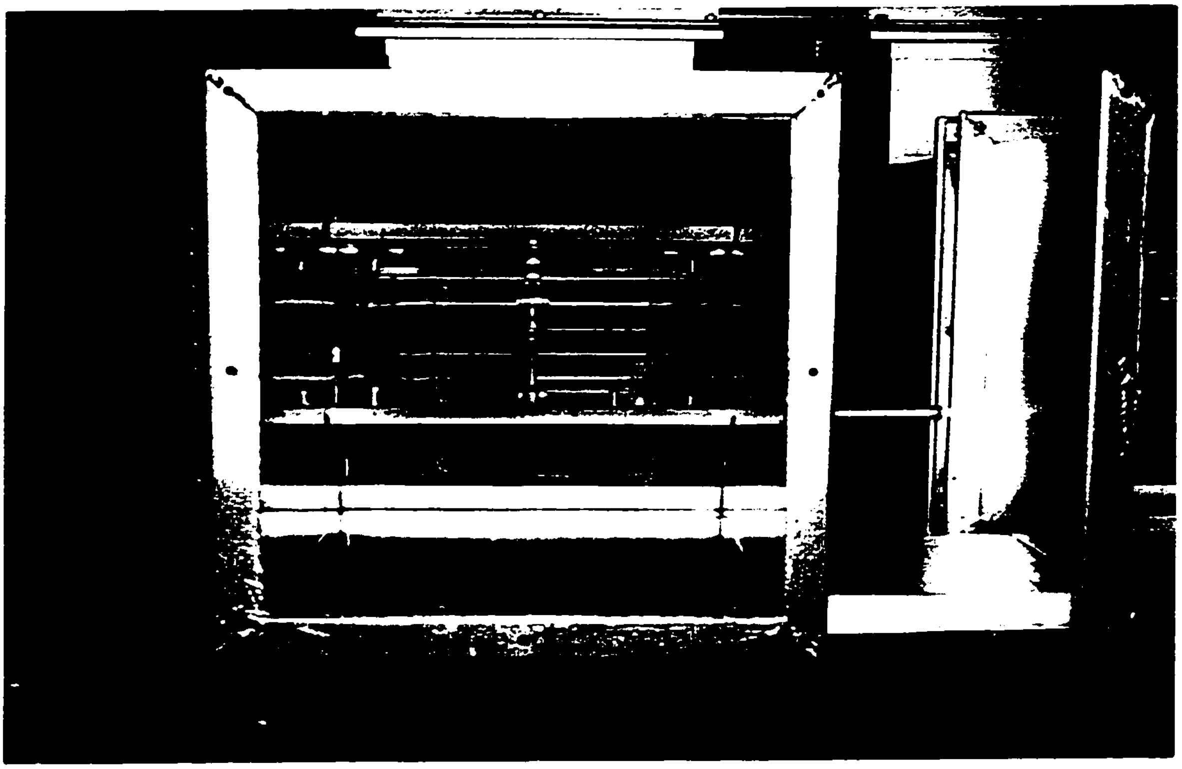

Figure 3.10 Damper and Cooling Coil

b) Damper Controller

The purpose of the damper controller is to vary the damper position in the hot duct, through a damper actuator, in order to maintain the air in the mixed air duct at a constant temperature (figure 3.11). For example, when the process variable (air temperature in mixed air duct) is above the setpoint, the damper must close to reduce the amount of warm air reaching the mixed air duct. The controller receives the temperature signal in voltage, determines the amplitude of the error and sends a current signal to the actuator so the latter can move the damper. The damper controller is a Honeywell UDC 2000 Mini-Pro, just like the valve controller. Appendix A contains more information on the controller and the damper actuator. Furthermore, the configuration record sheet is included in Appendix B.

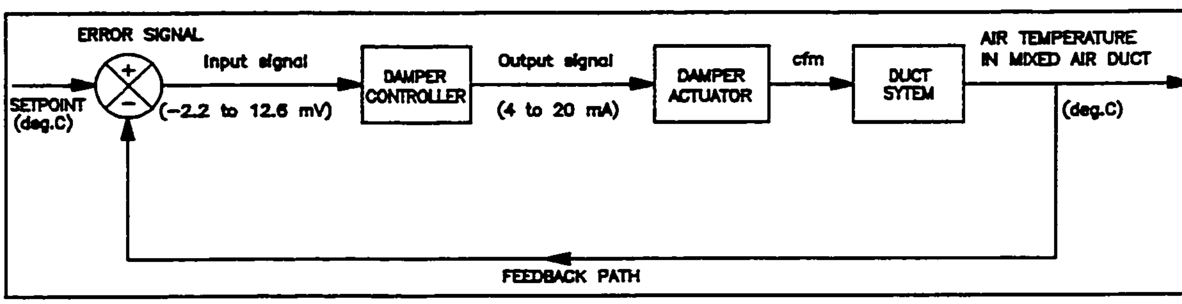

Figure 3.11 Damper Control Loop Schematic

3.1.5 Sensors

Thermocouples and flow sensors were installed at the different sensing stations to perform temperature and velocity pressure measurements.

a) Thermocouples

Thermocouples are used to measure air temperature at the sensing stations in the ducts, and also to measure temperature of water in the tank and going in and out of the cooling coil. Each sensing station in the ducts has 25 thermocouples, spread across the flow sensor grid (figure 3.12). The values are averaged and a signal in millivolts is sent either to the controllers, the data acquisition system, or both depending on the station. The two sensing stations in the water pipes each have three thermocouples inserted in wells; again, the values are averaged and one signal is sent to the data acquisition system. To measure temperature of cool water, a single thermocouple was inserted in the tank. T type, extension grade thermocouples were installed. They can read temperature from -62.2°C to 260°C and have an accuracy of ±1.0°C or 0.75% above 0°C (whichever is greater).

b) Flow Sensors

The flow sensors read two pressures: total pressure and static pressure. Copper tubes forming a grid and with holes facing the air flow read total pressure while a pitot tube reads static pressure (figure 3.12). Both sensing instruments were designed in the laboratory following ASHRAE standards described in the Fundamental Handbook (30). A pressure transducer is connected to the two sensors and subtracts the static pressure from total pressure to obtain velocity pressure.

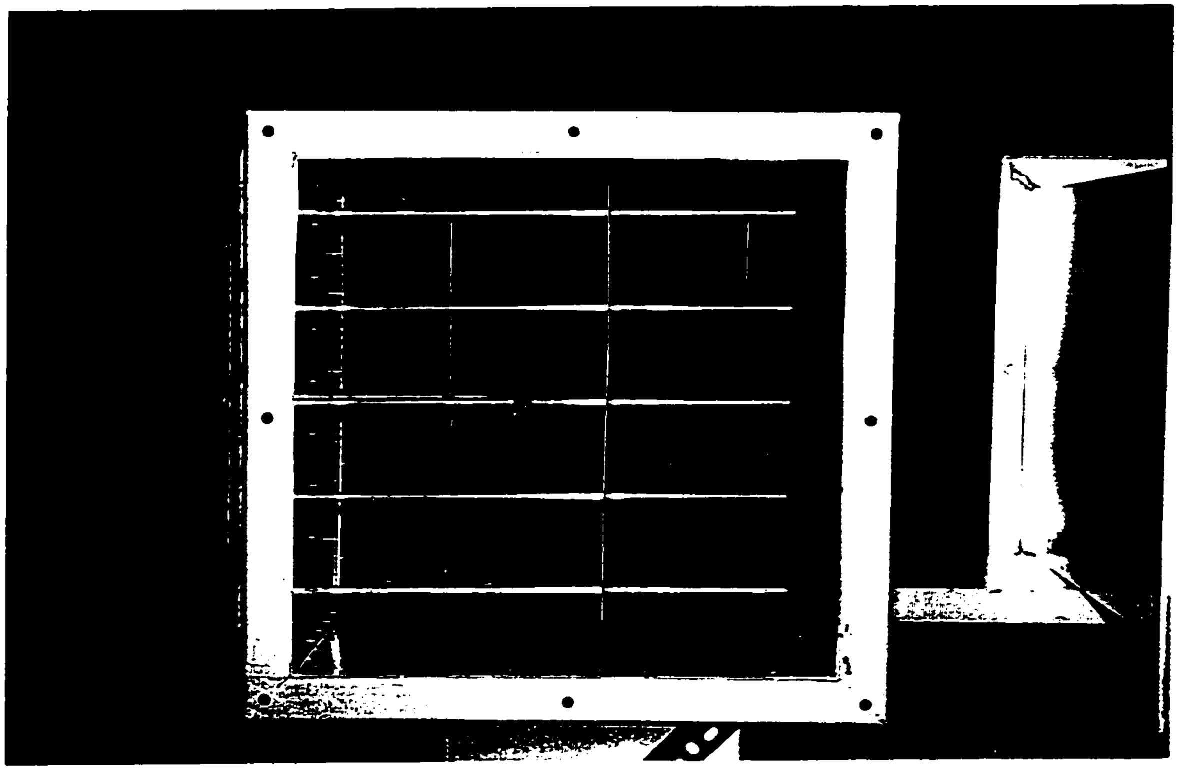

Figure 3.12 Sensing Station

3.1.6 Monitoring Equipment

There are two instruments used for monitoring: an air flow indicator and a data acquisition system.

a) Air Flow Indicator

In order to keep track of the air flow in the different ducts, an indicator was added to the control panel. Through a channel selector, it is possible to read the air flow at any sensing station. The indicator receives a voltage signal from the transducer, converts the signal to cfm and displays the obtained value. This indicator is a Eurotherm 842 and its complete configuration is presented in Appendix B.

b) Data Acquisition System

The data acquisition system has two purposes in this research. First, it provides temperature and pressure readings of the sensors connected to it. Second, it stores that information so the system response to changes can be analyzed. Howell-Mayhew Engineering's software "Copilot" (31) is used with a Labmate Series 7000, a data acquisition system from Scientific Instruments, to manage the information.

3.2 Instruments Calibration

The accuracy of the controllers and transducers was verified prior to their use. If the readings were not within 3% of the reference values, then calibration was performed. Detailed information on the calibration instruments is provided in Appendix C.

3.2.1 Controllers Calibration

a) Honeywell Controllers Calibration

Both input and output signals of the Honeywell controllers were calibrated. The input signal calibration required a thermocouple calibrator, a voltmeter and an ice bath, all connected as shown in figure 3.13. The temperature limits of T type, extension grade thermocouples are -62.2°C and 260°C; this represents -2.225 and 12.572 mV. The thermocouple calibrator was used to send the minimum and maximum limit voltage signals to the controller input terminals. Because the calibrator was not quite accurate, a voltmeter was connected in parallel to ensure the precision of the signal sent. To confirm that calibration was successful, one end of a thermocouple was placed in the ice bath while the other end was connected to the controller input terminals. All Honeywell controllers read 0.1°C ±0.1°C, which is within the accuracy of the ice bath and the thermocouple.

The calibration of the output signal only required a milliammeter. The milliammeter was used to ensure that the zero value and span value of the controller were reading 4mA and 20mA respectively. See figure 3.14 for the connections between the milliammeter and the controller.

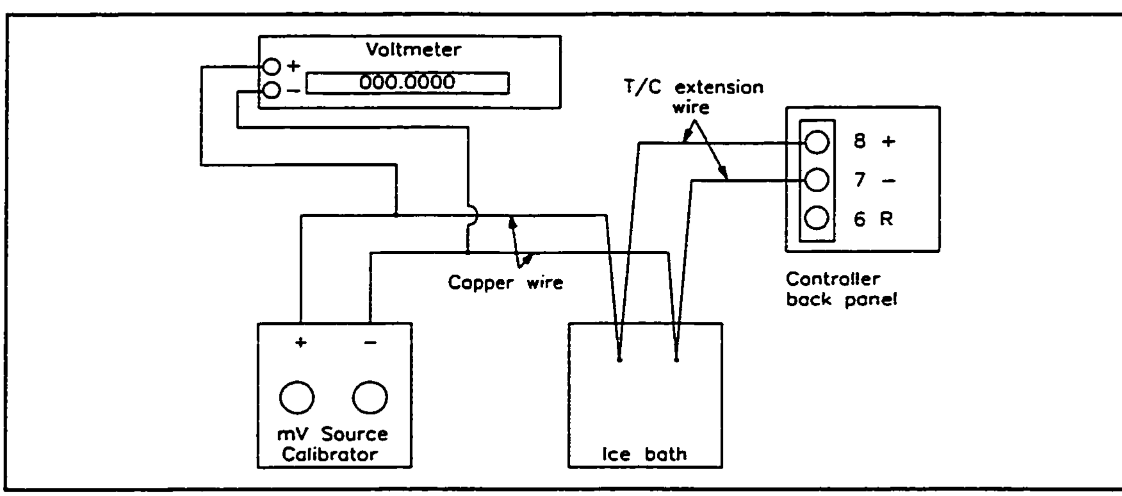

Figure 3.13 Calibration Connections for Honeywell Controller Input

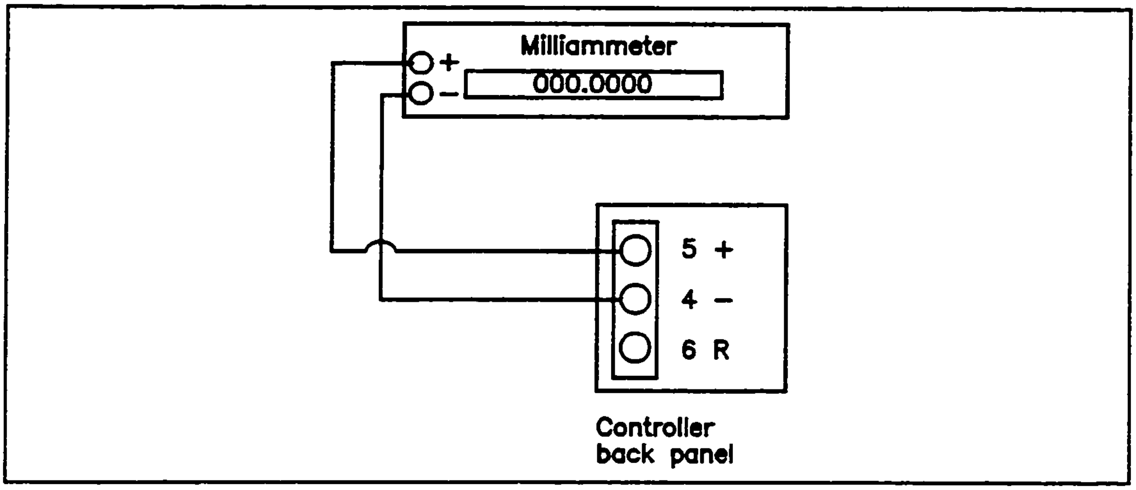

Figure 3.14 Calibration Connections for Honeywell Controller Output

b) Eurotherm Controller Calibration

The Eurotherm controller receives its input signal from a portable transducer with digital display. Since readings from the controller and transducer were equal, the controller was not recalibrated.

3.2.2 Transducers Calibration

The transducers are used to translate the pneumatic signal obtained from the flow sensors to an electrical signal sent to the fan speed controller, the flow indicator or the data acquisition system. The accuracy of the transducer readings was verified using a micromanometer. Pressure was applied to both micromanometer and transducer using a syringe and the instruments' readings were compared. The micromanometer has an accuracy of ±0.00025 in.WG. As table 3.1 shows, transducer #1 and the micromanometer have identical readings up to 0.1 in.WG, for higher pressures, the readings diverge more and more as the pressure increases. For transducer #2, pressure reaches 0.20 in.WG before the readings start to diverge. However, since the velocity pressure in the ducts is well within the 0 to 0.1 in.WG range and because the largest difference is only 2%, the transducers were not calibrated.

Table 3.1 Verification of the Accuracy of Transducers

Transducer #1
(Used for control of fan speed at station 5)

Transducer reading in. WG	Micromanometer reading in. WG
0.00	0.000
0.01	0.010
0.02	0.020
0.03	0.030
0.04	0.040
0.06	0.060
0.08	0.080
0.10	0.100
0.15	0.152
0.20	0.203
0.25	0.254
0.30	0.305
0.40	0.408
0.50	0.510

Transducer #2
(Used for monitoring purpose only)

Transducer reading in. WG	Micromanometer reading in. WG
0.00	0.000
0.01	0.010
0.02	0.020
0.03	0.030
0.04	0.040
0.06	0.060
0.08	0.080
0.10	0.100
0.15	0.150
0.20	0.200
0.25	0.252
0.30	0.304
0.40	0.408
0.50	0.510

4.0 OPEN-LOOP ANALYSES OF THE DUAL-DUCT SYSTEM

In order to properly tune the controllers of an HVAC system, a good understanding of the system dynamics is required. After having identified the existing tuning rules and calibrated the sensors, pressure transducers and controllers, it is important to study how the different variables such as pressure and temperature react to changes in fan speed, damper position or valve position. Some of the curves found will be used during the tuning phase to determine time constant, dead time and other parameters required by the tuning methods. Studying the system dynamics is also a good way to insure that the system equipment performs properly in open loop. This way, if any problems occur during the closed-loop tuning phase it will not be due to improper functioning of the equipment.

For this research, five variables were observed: the velocity pressure in the mixed air duct, the air flow in the cold and hot ducts, the water flow through the cooling coil and the air temperature in the cold duct. The following analyses were performed:

- Velocity pressure changes due to variation of the fan speed were studied;
- Air flow due to variation in damper position was studied;
- The water flow rate through the cooling coil at different valve position was determined;
- Air temperature changes in the cold duct due to a step change in valve position (water flow rate) were studied;

- Air temperature changes in the cold duct due to variation in damper position were studied;
- Air temperature changes in the cold duct due to variation of the fan speed were studied.

Table 4.1 explains clearly which parameters were varied and which parameters were kept constant for each analysis.

Table 4.1 Open-Loop Analysis

Analysis	Variable studied	Variable parameters	Constant parameters
#1	Velocity pressure in mixed air duct	Fan speed	Damper position Valve position
#2	Velocity pressure in mixed air duct	Damper position	Fan speed Valve position
#3	Water flow through the cooling coil	Valve position	Fan speed Damper position
#4	Air temperature in cold deck	Valve position	Fan speed Damper position
#5	Air temperature in cool duct	Damper position	Valve position Fan speed
#6	Air temperature in cool duct	Fan speed	Valve position Damper position

It can be noticed that the heater in not included in the above analyses. The reason is simple: the heater cannot be operated in open loop because the control device does not include a manual potentiometer.

4.1 Analysis of the Air Distribution System

The dynamic of the air distribution system was defined by conducting three different experiments. The first analysis allowed to determine the transient response of the system to step changes in fan speed; the second analysis allowed to determine the steady-state relationship between the fan speed and the velocity pressure in the mixed air duct; and the last analysis allowed to determine the effect of the hot and cold duct dampers' position on the air flow in each duct. Therefore, the variable parameters were the fan speed or the dampers' position. There was no cooling nor heating from the system's coils during these experiments.

4.1.1 Transient Analysis

In order to determine the response of the air distribution system to changes in fan speed, a step change was applied to the system and the velocity pressure was monitored until it reached steady-state. Three different step changes were applied to simulate a small, medium, and large load and each test was repeated three times to insure the accuracy of the results.

The dampers' position was not varied in these tests; they were all kept fully closed except the cold duct damper which was fully open.

a) Effect of a Small Step Change in Fan Speed on Velocity Pressure

The small step change in fan speed was performed by manually turning the potentiometer from the lowest fan speed possible (about 440 RPM with the potentiometer knob turned completely to the left) to a fan speed of about 800 RPM. The potentiometer does not offer much precision and it was very difficult to turn it exactly to the same location several times. This explains the differences between the final RPM values of the three tests seen in figure 4.1. The differences between the starting RPM values is due to the fan blade effect at this low speed. The pressure sensor is sensitive enough to feel the blade push air through the duct as a pulse. From the figure it can be concluded that the system requires three seconds to reach steady-state after a small step change in fan speed.

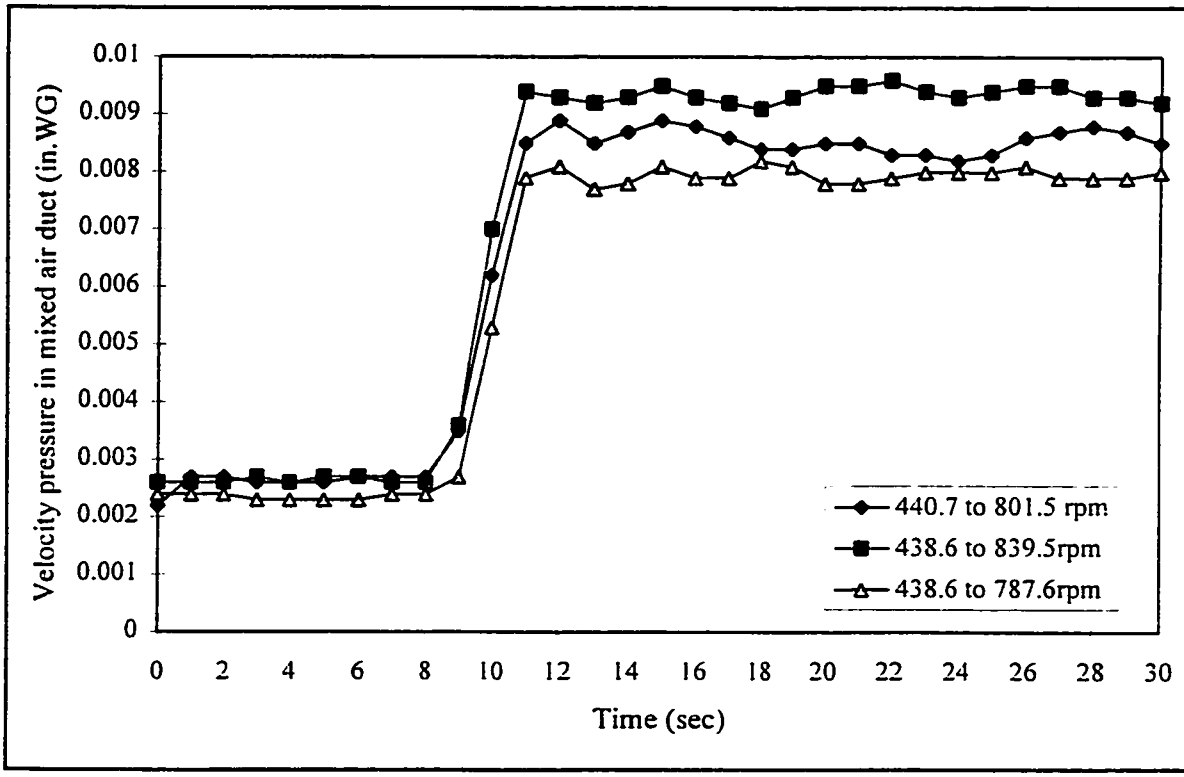

Figure 4.1 Variation in Velocity Pressure Due to a Fan Speed Change at Low Load

b) Effect of a Medium Step Change in Fan Speed on Velocity Pressure

The medium step change in fan speed was performed exactly like the small step change except that the final RPM value to reach was about 1200 RPM. From figure 4.2 it can be concluded that it takes four seconds for the system to reach steady-state after a medium step change in fan speed.

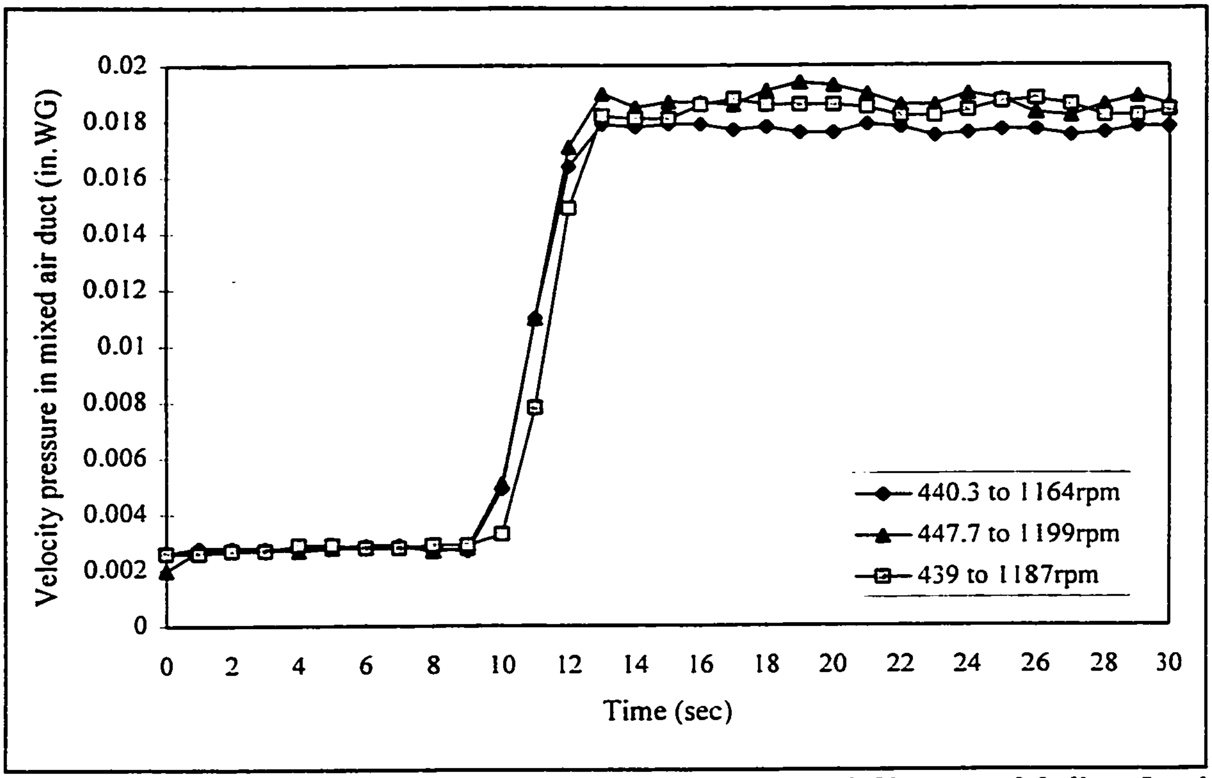

Figure 4.2 Variation in Velocity Pressure Due to a Fan Speed Change at Medium Load

c) Effect of a Large Step Change in Fan Speed on Velocity Pressure

The maximum fan speed is limited to 1200 RPM with manual control, but the actual maximum speed is 1700 RPM with the controller; therefore, the controller was used to perform the step change in this test. This was accomplished by increasing the controller setpoint to a high value so the fan speed had to rise to the maximum capacity to try to cope with the demand. The minimum fan speed is however 100 RPM higher than with manual control which means that the minimum velocity pressure to be maintained in the mixed air duct with the controller will be 0.003 in.WG. From figure 4.3 it can be concluded that it takes five seconds for the system to reach steady-state after a large step change in fan speed. The graph also shows that accuracy in repeating the same test several times is improved by using the controller instead of manual control. However, this test cannot be considered purely open loop because the controller parameters affect the process variable response.

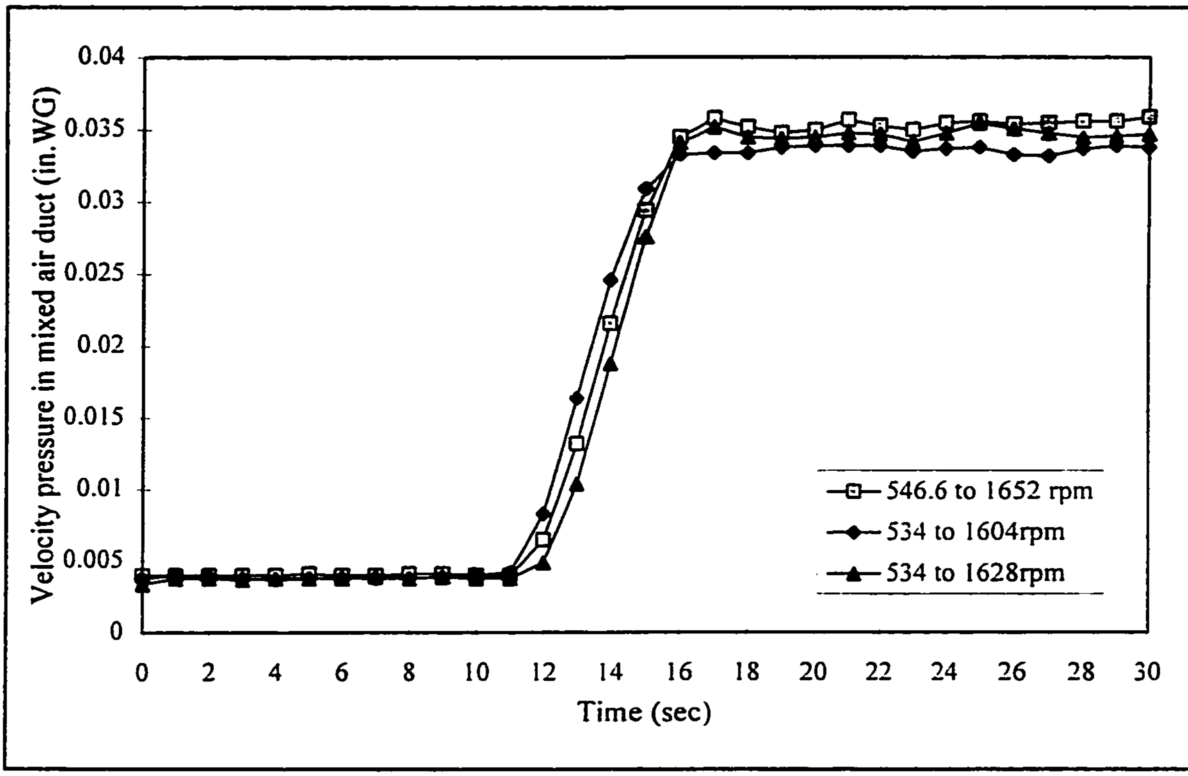

Figure 4.3 Variation in Velocity Pressure Due to a Fan Speed Change at Full Load

d) Observations

The air distribution system has a fast response to fan speed changes with a maximum steady-state response time of five seconds for the test performed at full capacity. The fan speed can vary from 440 to 1600 RPM, which represents a range of 220 to 900 cfm. The system responded as expected to fan speed step changes; in each of the three different tests, the curves show an exponential response. These curves were used later on to tune the fan speed controller.

4.1.2 Steady-State Analysis

Two tests were conducted to analyze the air distribution system under steady-state conditions. The first one consisted of varying the fan speed and reading the velocity pressure once it had stabilized. This was performed with different sets of position for the cold and hot duct dampers. The second test consisted of varying the cold and hot duct dampers' position simultaneously and reading the air flow in each duct once it had stabilized.

a) Velocity Pressure Readings in Mixed Air Duct at Different Fan Speeds

From monitoring the velocity pressure in the mixed air duct at different fan speeds, it was possible to obtain the relationship between the velocity pressure and the fan speed (figure 4.4). This relationship was determined for different positions of the cold and hot duct dampers. The position of the dampers was set manually. Lines were located approximately on the exterior of the dampers to identify whether the dampers were 1/4, 1/2 or 3/4 open. A line was also drawn on the controlling rod of the damper, and the dampers were positioned by aligning the lines. Three sets of position were chosen for the test: the cold duct damper is fully open while the hot duct damper is fully closed, both dampers are half open, and the cold duct damper is fully closed while the hot duct damper is fully open. The last two sets of dampers' position provided very similar results and are shown in figure 4.4 by one curve. All the results follow a quadratic trend; however, the curve obtained with the cold duct damper fully open is a little lower than the other curve which indicates that the flow resistance of the cooling coil is higher than the flow resistance of the heating coil. This is caused by the more complex geometry of the cooling coil.

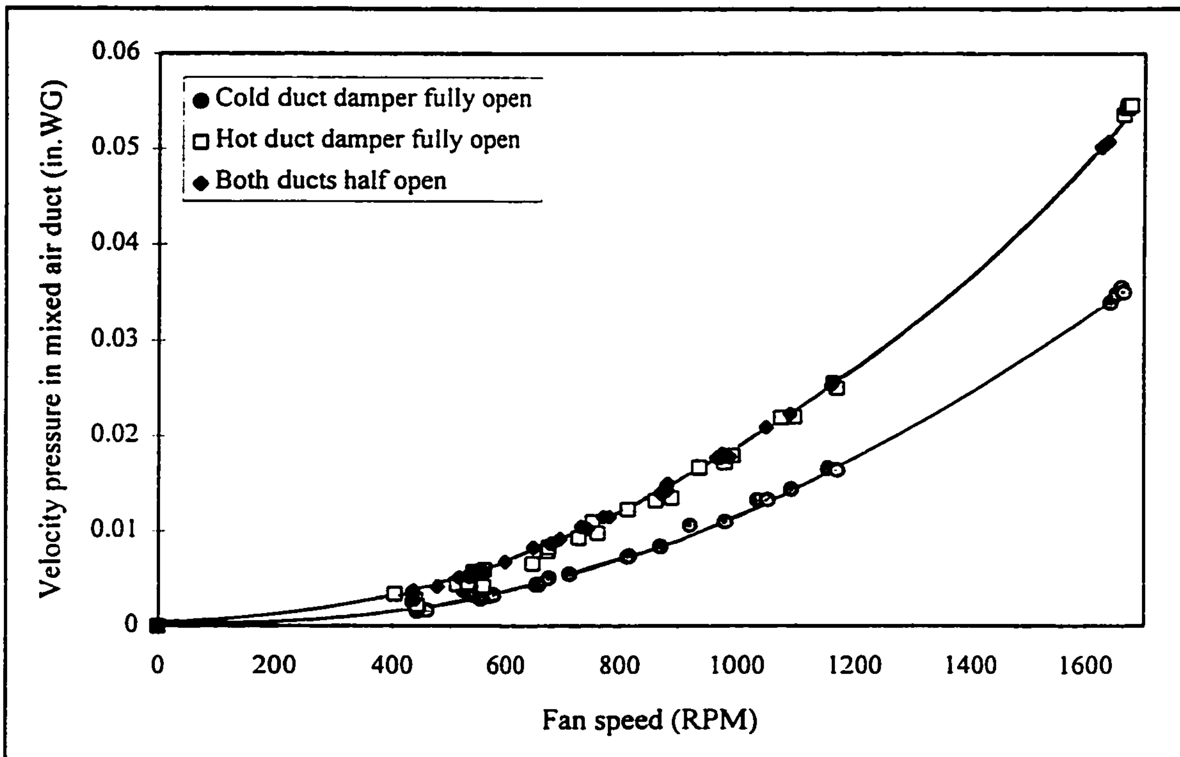

Figure 4.4 Velocity Pressure at Different Fan Speed and Damper Positions

b) Air Flow Readings in Both Ducts for Different Damper Positions

Air flow readings were taken in the hot and cold ducts for different damper positions in order to determine the characteristic of the dampers. The dampers' position was varied inversely from the cold duct fully open and the hot duct fully closed, to the cold duct fully closed and the hot duct fully open (figure 4.5). The test was conducted at constant fan speed and was repeated twice to determine the sensitivity of the system to manual damper control inaccuracy. A speed of 1000 RPM was selected because at that level, manipulation of the fan potentiometer allowed to maintain the speed constant from one set of damper position to another. The test was tried at a fan speed of 1600 RPM, reached by using the controller, but it was impossible to keep the speed constant for all the sets of damper positions. The curves show that the system is not sensitive to very small variations in damper position, and therefore it is possible to obtain very similar results with the manual control described in section 4.1.2a. But most important, the curves also show that the dampers almost have a linear effect on the air flow.

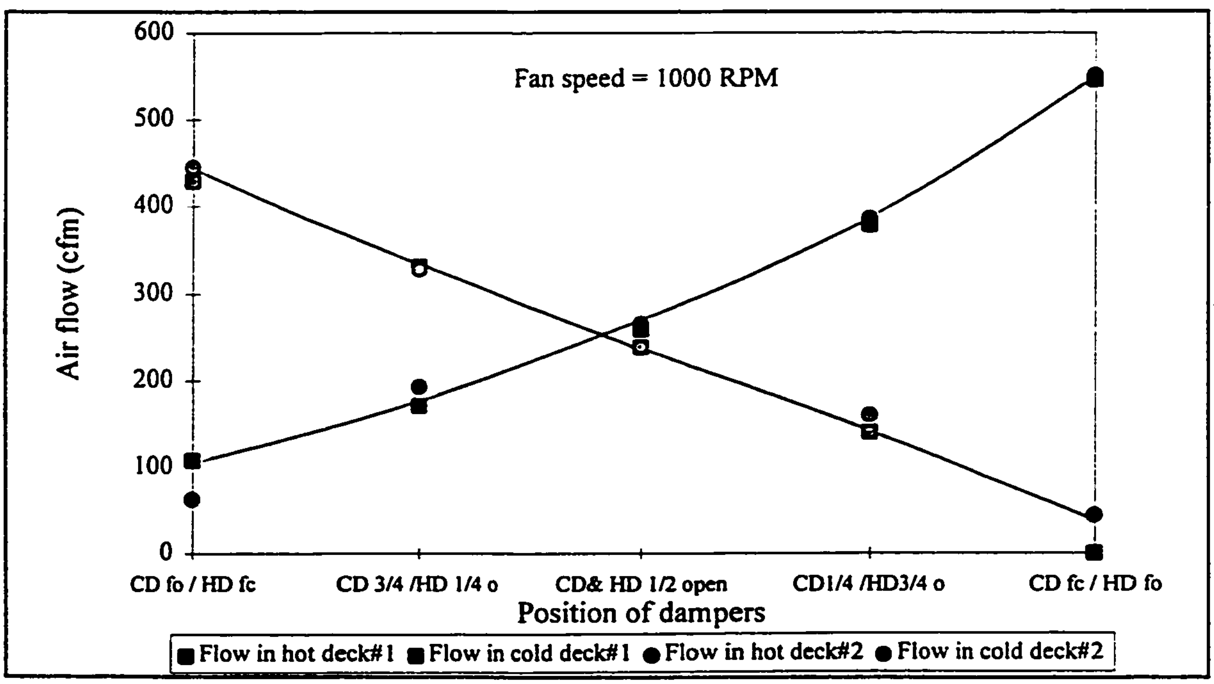

Figure 4.5 Effect of Dampers' Position on Air Flow in both Ducts

c) Observations

To determine if the curves of the two steady-state analyses performed provide accurate results, they were compared with previous measurements. Those previous measurements consist of velocity pressure at sensing stations 3, 4 and 5 taken simultaneously with the hot and cold duct dampers half open (exhaust dampers fully closed) and the fan speed constant. The values are recalled in table 4.2. Table 4.2 also includes the air flow in each duct which was calculated using the following equations:

$$V=\sqrt{p_v}\times 4005 \tag{4.1}$$

Since the area of the duct is $1 \mathrm{ft}^2$, then the air velocity is numerically equal to the air flow:

$$V=CFM$$

The sum of the hot and cold duct air flows should equal the air flow in the mixed air duct. There is an error of 3.4% which is probably due to the accuracy of the pressure transducers. From figure 4.4 it is possible to obtain the air flow in the mixed air duct with the dampers half open and the fan speed at 1000 RPM. Data points were available for 986 RPM and 1049 RPM; therefore, a simple linear interpolation could be performed to obtain the flow at 1000 RPM. From figure 4.5 it is possible to obtain the air flow in the cold and hot ducts with the dampers half open (the fan speed was constant at 1000 RPM throughout the test). The values are reported in table 4.3. The error has now increased to 9% but this is not very significant.

There is a discrepancy when comparing the flows in the hot and cold ducts of table 4.2

and table 4.3. Table 4.2 shows a higher flow in the cold duct while table 4.3 shows a higher flow in the hot duct. However, the difference is so small between the air flows in the cold and hot ducts (about 20 cfm) that it can be assumed in both cases that the air flow in both ducts are equal when the dampers are half open. Note that it is impossible to individually compare the air flows because the fan speed was different for the test reported in table 4.2.

Table 4.2 Velocity Pressure and Air Flow Repartition in the System (20/02/95)

	Mixed air duct	Hot duct	Cold duct
velocity pressure (in.WG)	0.028	0.007	0.008
air flow (cfm)	670	335	358
% error	$\frac{(335+358)-670}{670} = 0.034 \Rightarrow$ **3.4%**		

Table 4.3 Velocity Pressure and Air Flow in the System Obtained During Different Tests

	TEST #1	TEST #2	
	Mixed air duct	Hot duct	Cold duct
velocity pressure (in.WG)	0.0185	0.004	0.0035
air flow (cfm)	545	258	238
% error	$\frac{(258+238)-545}{545} = 0.09 \Rightarrow$ **9.0%**		

4.2 Analysis of the Water Distribution System

The dynamic of the water distribution system was defined by conducting four different experiments. The first analysis allowed to determine how the valve position affects water flow; the second analysis allowed to determine the transient performance of the system to step changes in valve position; the third analysis allowed to determine how the system responds to changes in cold duct damper position in the cooling mode; and the last analysis allowed to determine how the system responds to changes in fan speed while in the cooling mode. Therefore, the varying parameters were the valve position, the cold duct damper position or the fan speed. The hot duct dampers were closed throughout the open-loop tests.

Note that the water distribution system is used for cooling only. Under full load conditions, the valve is fully open, the cold duct damper is open and the fan speed is about 1200 RPM providing a water temperature of 6.5°C in the tank. The water temperature is allowed to fluctuate between 6.3 and 6.7°C.

4.2.1 Valve Characteristic

By finding the relationship between the water flow and the position of the two-way valve, it is possible to determine if the valve has quick-opening, linear or equal percentage characteristic. Modulating valves used in control must have linear or equal percentage characteristic to perform adequately.

a) Effect of the Valve Position on Water Flow

Measurements of the water flow were taken for five different valve positions: fully closed,

¼ open, ½ open, ¾ open and fully open. Figure 4.6 shows that the valve follows the equal percentage flow characteristic.

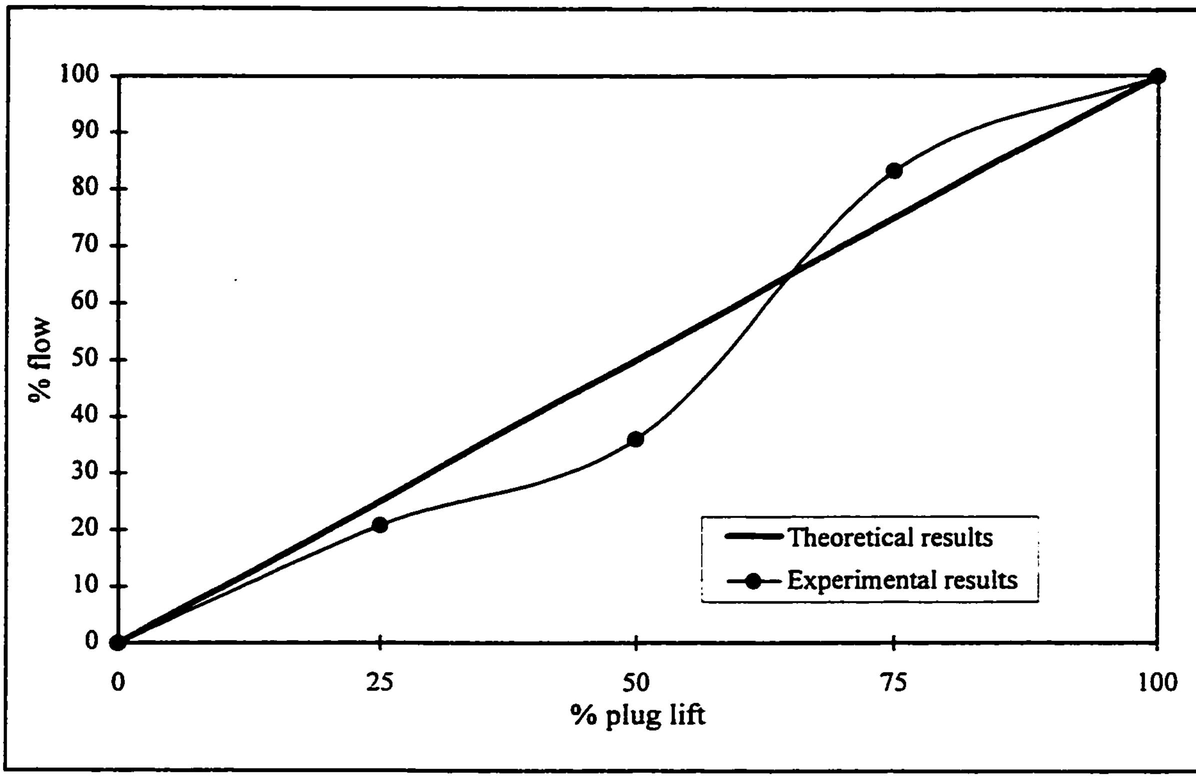

Figure 4.6 Variation in Water Flow Due to Changes in Valve Position

4.2.2 Transient Analysis

In order to determine the response of the water distribution system to changes in valve position, four different step changes were imposed on the system and the air temperature in the cold duct was monitored until it reached steady-state. The cold duct damper was fully open throughout the test and the fan speed was constant at 1200 RPM. Valve movement was performed manually using a rest block with different heights to obtain the desired valve position. Each transient test started once the water temperature in the tank had stabilized at 6.5°C with the valve closed. Therefore, each test started with the air in the cold duct at room temperature. The air temperature readings in the cold duct were obtained from a thermocouple inserted in the duct through the exhaust damper which is near the cooling coil. For each valve displacement, the experiment was repeated three times to insure accuracy of the results.

a) Effect of ¼ Valve Displacement on Air Temperature in the Cold Duct

A valve displacement from fully closed to ¼ open causes an air temperature drop of 4.5°C in the cold duct (figure 4.7). The first 2.8°C ($0.632C_{ss}$) temperature drop occurs in 36 seconds but steady-state (C_{ss}) is reached only after 110 seconds. The three trials shown in figure 4.7 provide almost identical results which means that valve manual control can be accurate for specific displacements.

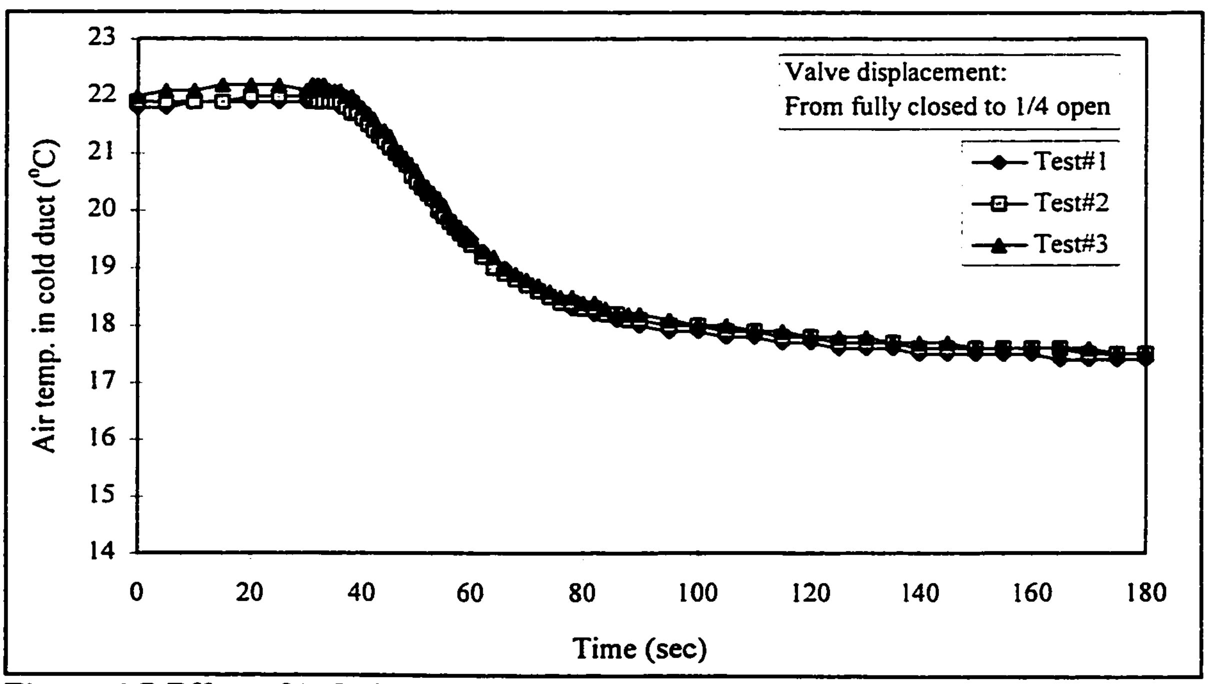

Figure 4.7 Effect of ¼ Valve Displacement on Air Temperature in Cold Duct

b) Effect of ½ Valve Displacement on Air Temperature in the Cold Duct

A valve displacement from fully closed to ½ open causes an air temperature drop of 6.2°C (figure 4.8). The first 3.9°C (0.632C_{ss}) temperature drop occurs in 24 seconds but steady-state (C_{ss}) is reached only after 110 seconds. Therefore, the process variable requires the same amount of time to reach steady-state for both ¼ and ½ displacements but 0.632C_{ss} is reached 12 seconds faster for the ½ valve displacement. Again, the three trials provide almost identical results.

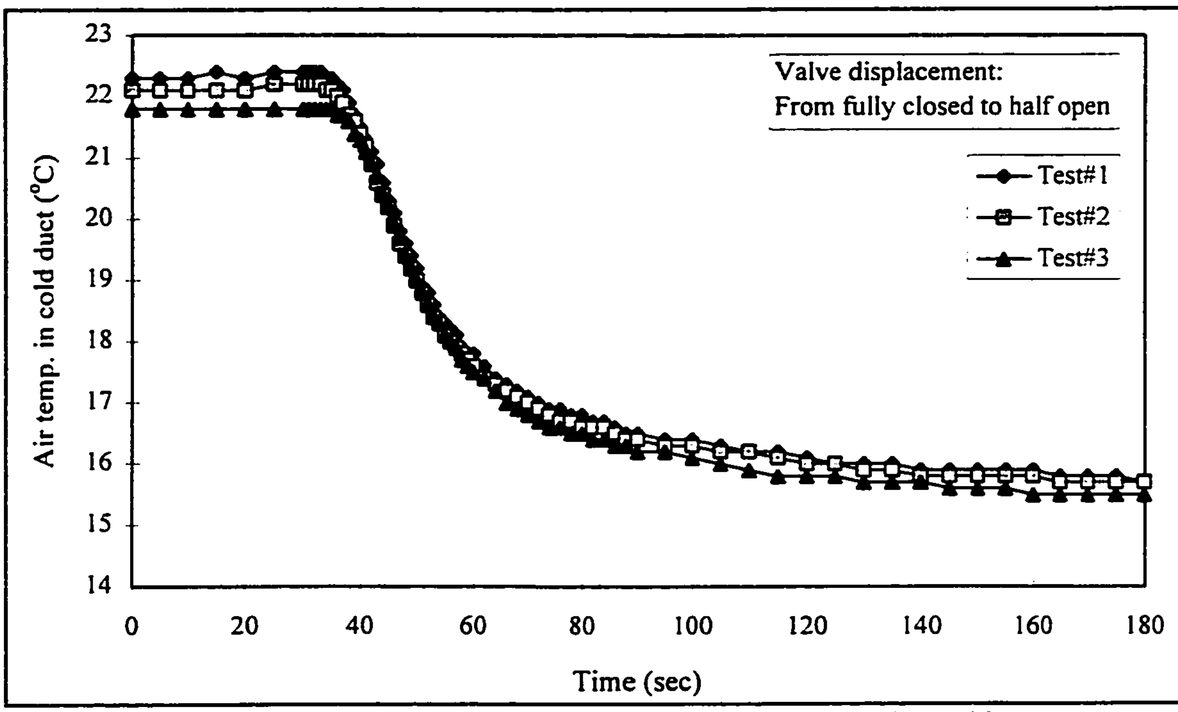

Figure 4.8 Effect of ½ Valve Displacement on Air Temperature in Cold Duct

c) Effect of ¾ Valve Displacement on Air Temperature in the Cold Duct

A valve displacement from fully closed to ¾ open causes an air temperature drop of 8.2°C (figure 4.9) with the first 5.2°C (0.632C_{ss}) temperature drop occurring in 25 seconds. Similar to the two previous valve displacements, steady-state (C_{ss}) is reached after 110 seconds but the process variable required one extra second to reach $0.632C_{ss}$ compared to ½ valve displacement. Finally, the three trials were not started at exactly the same temperature and therefore the steady-state values after the step change in valve position differ, mainly for test#2. However, the three trials show similar air temperature decrease slopes.

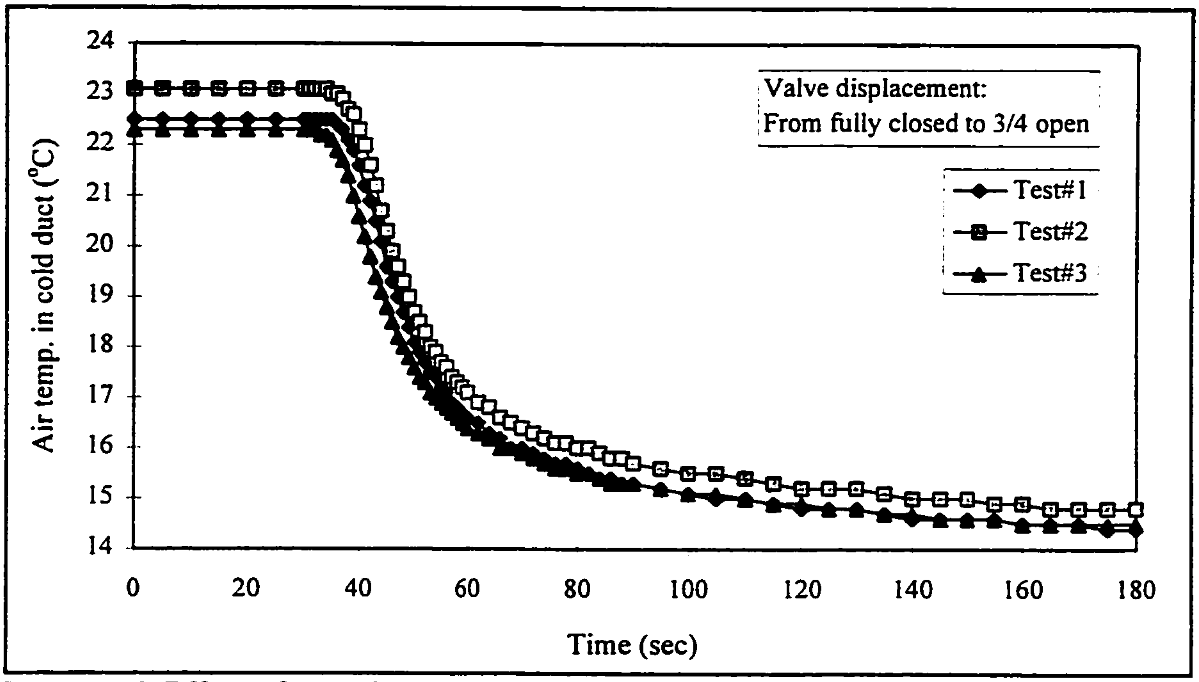

Figure 4.9 Effect of ¾ Valve Displacement on Air Temperature in Cold Duct

d) Effect of Full Valve Displacement on Air Temperature in the Cold Duct

A valve displacement from fully closed to fully open causes an air temperature drop of 8.3°C (figure 4.10). This is only 0.1°C more than a ¾ valve displacement. However, the first 5.2°C ($0.632C_{ss}$) temperature drop is reached seven seconds faster, that is after 18 seconds, and the steady-state (C_{SS}) is reached 10 seconds faster, that is after 100 seconds. As for the ¾ valve displacement, the three trials were not started at exactly the same temperature and therefore the steady-state values after the step change in valve position differ slightly. However, the three trials show similar air temperature decrease slopes.

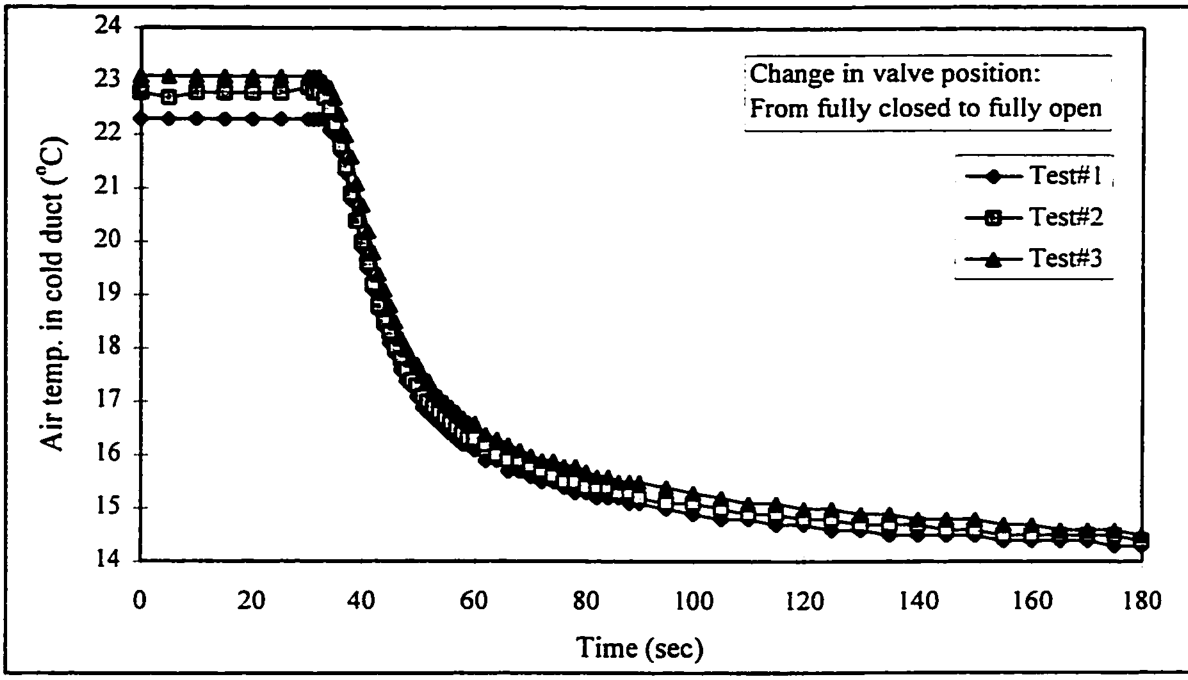

Figure 4.10 Effect of Valve Full Displacement on Air Temperature in Cold Duct

e) Water Temperature in the Coil During Valve Displacement

Figure 4.11 shows how the temperature of the water going in and coming out of the coil varies during a valve step change from fully closed to fully open. Just after the valve is open, the large temperature difference shown between the two water streams clearly indicates that the heat exchange from air to water is much higher at that time than after steady-state is reached. While the valve is still closed, the air temperature is the same

(22°C) on both sides of the coil, so when cold water starts running in the coil, the water must cool the air down before and after the coil. The air entering the coil is always at 22°C, however, after a few seconds, the air after the coil starts to cool down (see figure 4.10) so heat exchange from air to water starts to decrease until it reaches a steady-state. Steady-state occurs when the temperature difference of the two water streams reaches a constant value, which is 2.2°C in this case.

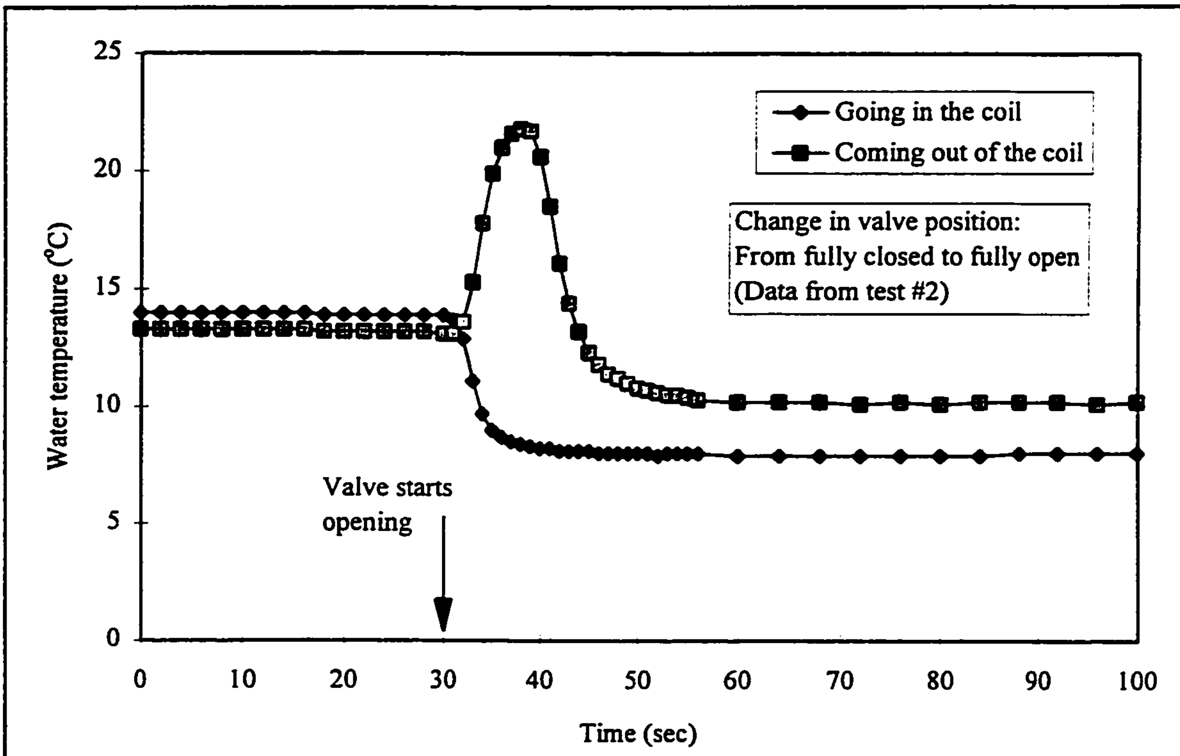

Figure 4.11 Cooling Coil Water Temperature Change Due to Valve Movement

f) Observations

The water distribution system has a slow response to changes in valve position and requires about 100 seconds to reach steady-state in all cases. With a fan speed of 1200

RPM, the highest cold air temperature is 17.6°C for ¼ valve displacement and the lowest temperature is 14.6°C for full valve displacement. The system responded as expected to valve displacement step changes; in each test, the curves show an exponential response. These curves were used later on to tune the valve controller.

4.2.3 Steady-State Analysis

Two different tests were performed to determine how changes in air flow could affect the cooling capacity of the system. The first test consisted of varying the cold duct damper position while the second test consisted of varying the fan speed. Readings of the air temperature in the cold duct were taken once the system had reached steady-state after each change with the cooling valve fully open.

a) Effect of Cold Duct Damper Position on Air Temperature in the Duct

Table 4.4 shows the air temperature fluctuation in the cold duct due to changes in the damper position. The damper position actually does not have a large effect on the air temperature in the cold duct during cooling. The air temperature falls by only 1°C when the damper is moved from fully open to a quarter open. The table also indicates that reducing the air flow to 335 cfm will reduce the air temperature to 13.1°C. Therefore, the minimum cold air temperature at part load is approximately 13°C.

Table 4.4 Effect of Damper Position on Temperature and Air Flow in Cold Duct

	Test #1		Test #2		Test #3	
	Temp.(°C)	cfm	Temp.(°C)	cfm	Temp.(°C)	cfm
fully open	14.2	537	14.3	552	14.3	552
3/4 open	14.0	507	14.2	507	14.3	507
1/2 open	13.7	457	13.9	439	13.9	439
1/4 open	13.1	335	13.2	335	13.3	335
fully closed	19.7	127	21.4	0	20.2	0

b) Effect of Fan Speed on Air Temperature in the Cold Duct

Variation of fan speed, which is represented by air flow changes in figure 4.12, has a larger effect on air temperature in the cold duct than changes in damper position. Figure 4.12 shows a temperature rise of about 3°C over the total range of achievable air flows. Air flows from 180 to 520 cfm were obtained by varying the fan speed manually and the maximum air flow, 750 cfm, was obtained by increasing the fan speed to its maximum closed-loop value with the controller (see section 4.1.1c).

4.3 Observations

During these experiments, open-loop testing was facilitated because the HVAC system under consideration was installed within one room. The fan speed was easily varied manually from the control panel with the potentiometer. However, for the valve and damper, it was necessary to be physically near them in order to manually move the valve plug or the damper arm to the desired positions. For a real HVAC system, this is time consuming because it involves walking around in the building to move valves and dampers.

The transient and steady-state analyses performed showed that the air and water distribution systems are stable under open-loop conditions. From the transient analyses it can also be concluded that the system has a much faster response to a step change in fan speed than to a step change in valve position. The fan speed control loop response time is actually 20 times faster than the valve control loop response time.

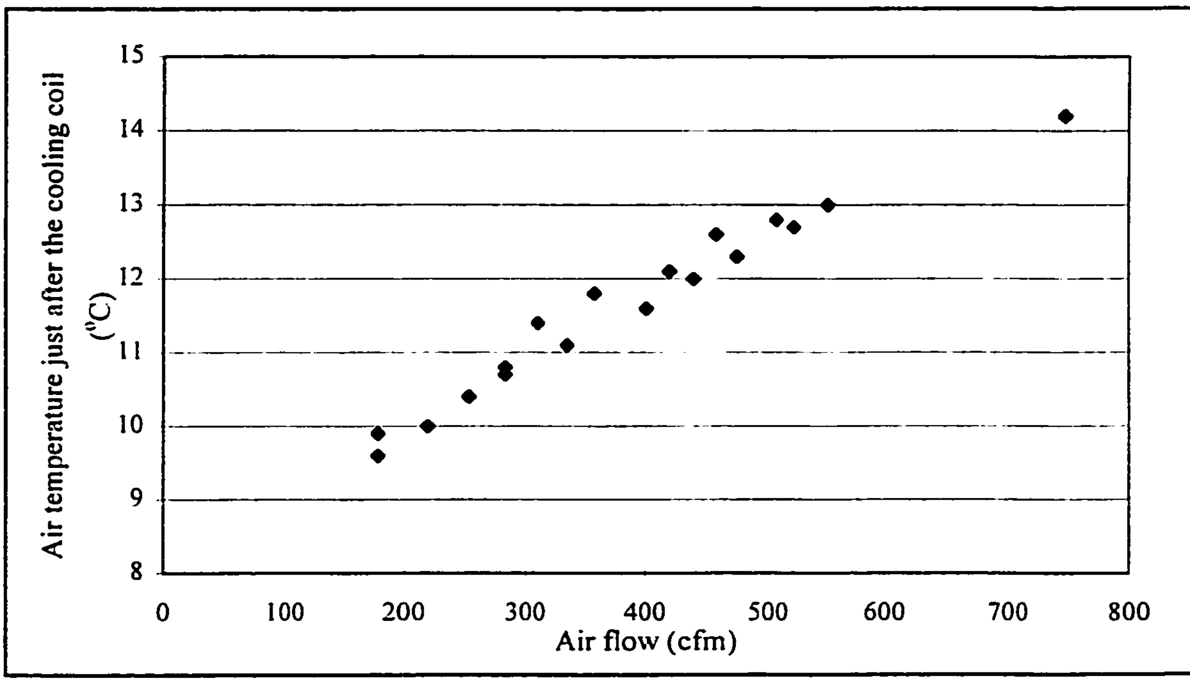

Figure 4.12 Air Temperature in Cold Duct for Different Air Flow

5.0 MANUAL TUNING OF THE CONTROLLERS

Before any interaction analysis can be conducted, each control loop must at least perform properly individually. All four controllers were tuned according to the three tuning methods selected:

1. Ziegler and Nichols' tuning rules (closed-loop method only) (6);
2. Simplified IMC-PID tuning rules (11); and
3. Bekker et al. tuning rules (13).

Then, for each controller, the selection of the most reliable set of tuning parameters depended on how well its control loop responded to step changes in setpoint or load. The purpose of these tests was not to determine the optimum settings, but merely to find tuning parameters that would provide an acceptable response for each individual loop.

In this research, all controllers were tuned with a PI algorithm; therefore, the proportional and integral terms had to be defined for each of them. Depending on the company which designed the controller, this meant defining the proportional gain (K_p) or the proportional band (PB) and defining the integral time (T_i) or the reset rate (U). The relationship between the proportional gain and the proportional band is defined in the Honeywell UDC 2000 Mini-Pro product manual as (32):

$$K_p = \frac{100\%}{PB\%}$$

The relationship between the integral time and the reset rate is as shown in the literature review:

$$T_i = \frac{1}{U}$$

5.1 Fan Speed Controller Tuning

The fan speed controller requires the input of the proportional band and the integral time. The proportional band can be displayed in linear input units or in percent. Since the simplified IMC-PID and Bekker et al. tuning rules provide values of PB in percent, the percent display was chosen. For the three tuning methods, two different types of tests were conducted to evaluate the fan speed loop response under PI-control: startup tests and load disturbance tests. At startup, the tests were performed at two setpoints: 0.013 in.WG and 0.028 in.WG. Furthermore, all dampers were kept at a fixed position, that is, the cold duct damper fully open, the hot duct damper half open and both exhaust dampers fully closed. In the case of load disturbance tests, only one setpoint was used, 0.028 in.WG; disturbances were induced by varying the hot duct damper position, keeping the position of the other dampers unchanged.

5.1.1 Ziegler and Nichols' Tuning Rules

To determine the tuning parameters according to Ziegler and Nichols' tuning rules, the ultimate period of oscillation and the ultimate proportional band must be established. This can require several closed-loop tests.

a) Setting the Controller Gains

The installation and operation manual of the Eurotherm controller (33) includes a section on manual tuning following Ziegler and Nichols' method. The step by step procedure was followed to obtain the ultimate period of oscillation (T) and the ultimate sensitivity (P). It simply required to monitor the variation in velocity pressure in the mixed air duct at different values of proportional band with the integral time off. The setpoint was chosen to be 0.028 in.WG. The ultimate period of oscillation was measured when the process variable showed large sustained oscillation about the setpoint, while the ultimate sensitivity was recorded at the onset of oscillation. From figures 5.1 and 5.2,

$$P = 60\% \text{ and } T = 3.5 \text{ sec.}$$

Therefore, following Ziegler and Nichols' PI-control settings, the proportional band and the integral time to be input in the controller were:

$$PB = 2.2P = 132\%$$
$$T_i = 0.8T = 3 \text{ seconds}$$

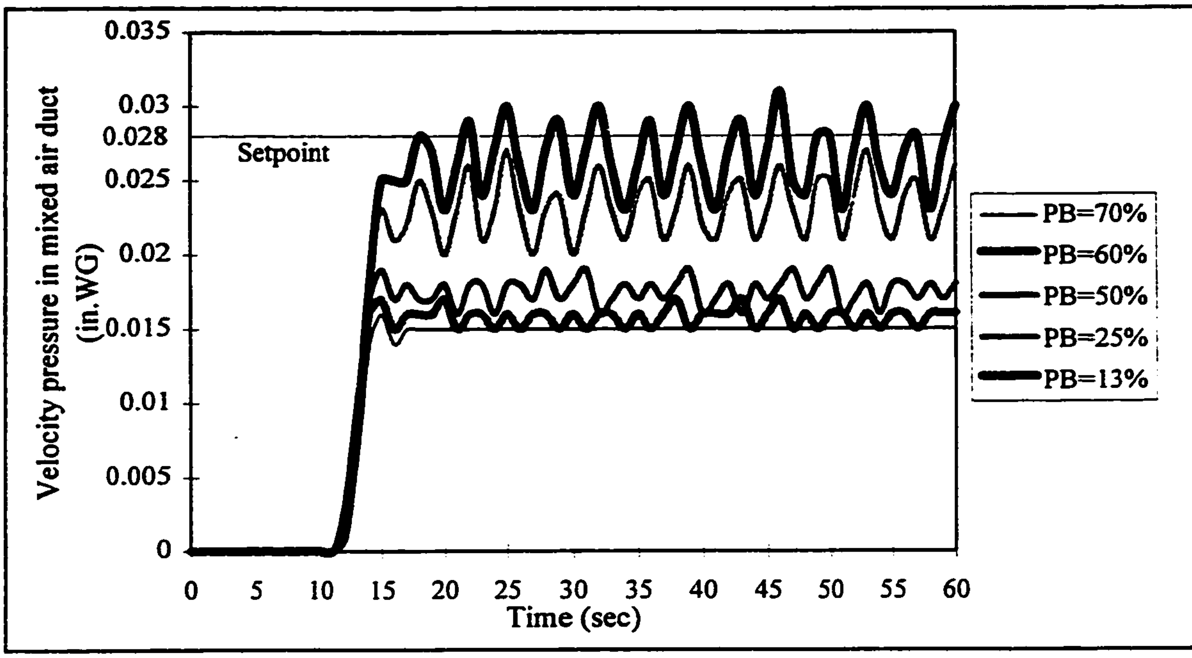

Figure 5.1 Closed-Loop Tests on Fan Speed at Startup

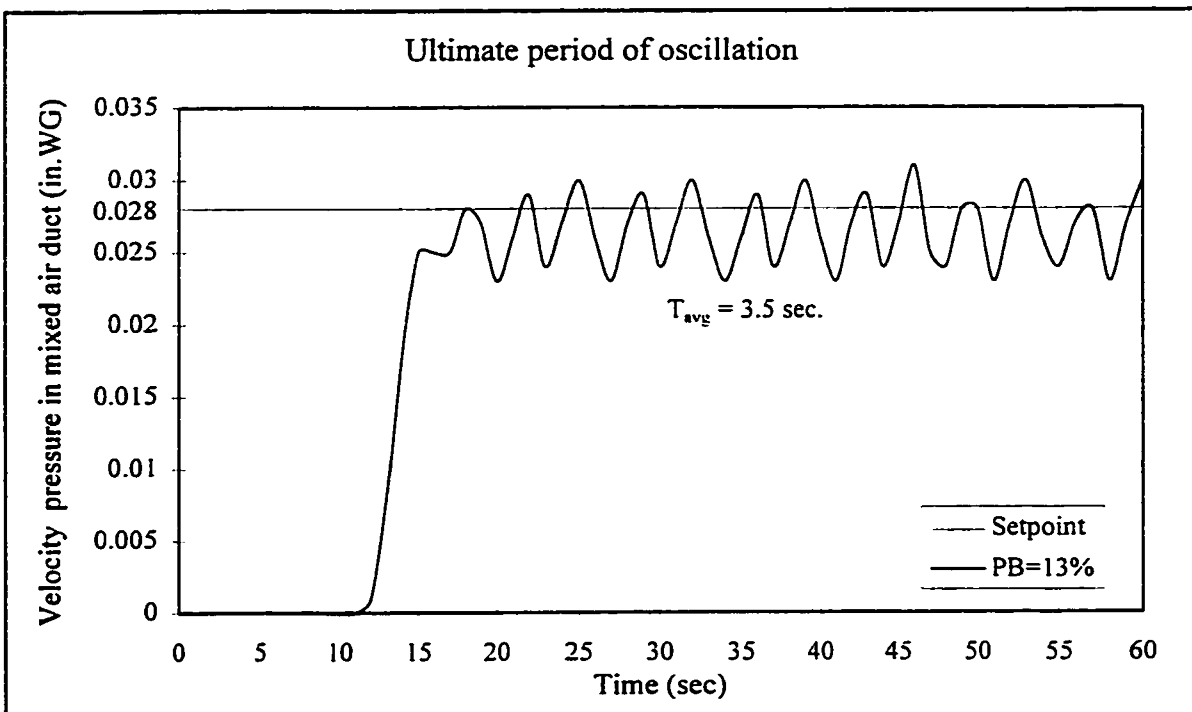

Figure 5.2 Closed-Loop Test on Fan Speed at Startup (Ultimate Period of Oscillation)

b) Results

When the fan speed controller is tuned with Ziegler and Nichols' parameters, there is a large overshoot at startup (figure 5.3). Actually, the fan speed reaches its maximum level (100% power) for one second before it decreases so velocity pressure can reach its setpoint in the mixed air duct. For a setpoint value identical to the one used while defining the tuning parameters (0.028 in.WG), startup causes an overshoot of 0.024 in.WG (86%) and the steady-state value is reached in 43 seconds. For a lower setpoint (0.013 in.WG), startup causes a higher overshoot, 0.036 in.WG (277%), and it takes 47 seconds to reach steady-state. This tends to indicate that the loop response efficiency decreases when the operating conditions are different from the tuning conditions. However, further investigation on part-load performance is required to confirm this tendency. Such analysis is found in section 5.1.4b.

Figure 5.4 shows how the fan speed loop responds to load disturbances induced by changes in the hot duct damper's position. For small damper movements (1/4 of full displacement), the loop responds rapidly and with little effect on velocity pressure. The displacements produce a maximum pressure fluctuation of 0.002 in.WG (7%) and the process variable requires up to 25 seconds to return to its setpoint. There is one exception when the damper is moved from a quarter open to a fully closed position; in that case the velocity pressure peaks 14% below the setpoint. For damper movements equal to half the total displacement, the percentage difference between peaks and setpoint varies between 11% and 25%. However, the highest peak is reached when the damper travels from fully closed to fully open; the percentage difference between peak and setpoint is then 43%. When the damper travels at least half of the total displacement the steady-state response time varies between 20 and 30 seconds. Note that these results would be different if another setpoint had been chosen.

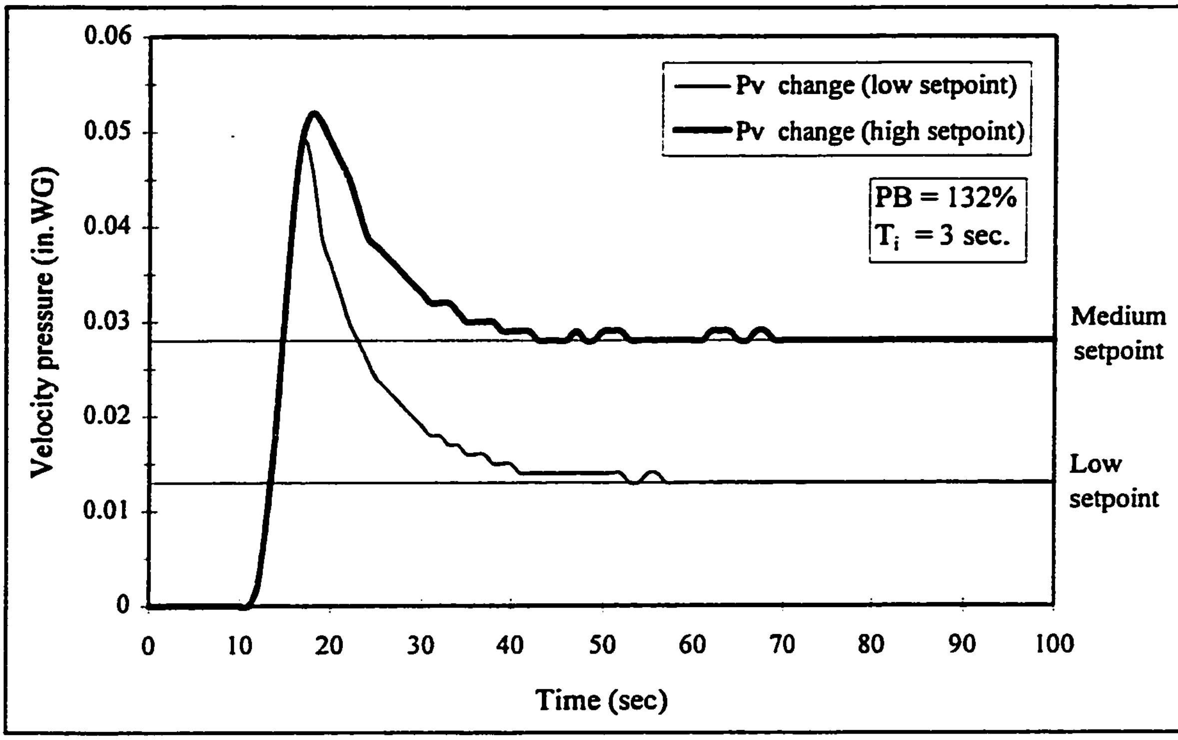

Figure 5.3 PI-Control of Fan Speed at Startup (Ziegler & Nichols' Tuning Rules)

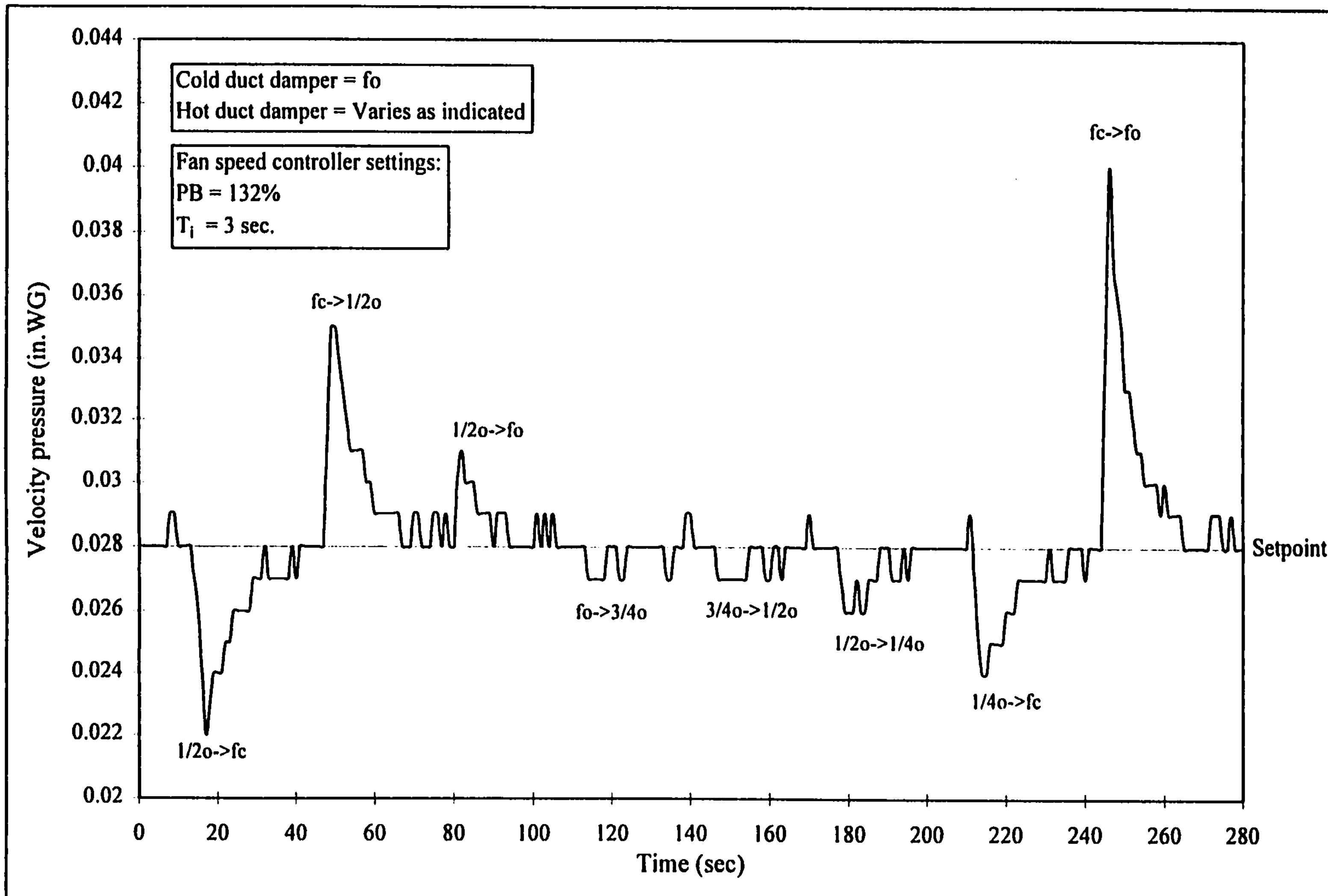

Figure 5.4 Velocity Pressure Fluctuations During Load Disturbances (Ziegler & Nichols' Tuning Rules)

5.1.2 Simplified IMC-PID Tuning Rules

To tune a controller with the simplified IMC-PID tuning rules, an open-loop test must first be performed to determine the dead time, time constant and reaction rate of the process. These parameters can then be used in the rules to calculate the proportional gain and integral time.

a) Setting the controller gains

The open-loop test was performed to simulate a medium load disturbance. The fan speed was kept at 441.7 RPM for 10 seconds and then increased to 1166 RPM; in pressure, this represents a step from 0.004 to 0.027 in.WG. Figure 5.5 shows how the velocity pressure in the mixed air duct varied with time due to the step change in fan speed. A line through the maximum slope was drawn to define the dead time (L), the time constant (τ) and the reaction rate (R). The ratio of τ/L determines the set of simplified IMC-PID rules to be used. In this case, with:

$$\frac{\tau}{L}=\frac{1.04}{2.26}=0.46<3$$

the proportional gain and the integral time are given by:

$$K_p=\frac{1}{2RL}=\frac{1}{2\times0.606\times2.26}=0.365 \Rightarrow PB=\frac{100}{0.365}=274\%$$

where, $$R=\frac{\left(\frac{0.0185-0.004}{13.3-12.26}\right)}{(0.027-0.004)}=0.606\text{sec.}^{-1}$$

and $T_i=\tau=1\text{sec.}$

There is another simplified IMC-PID set of rules which can be applied because the dead time is less than 0.5 minute. Following this constraint, the proportional gain and the time constant would be equal to:

$$K_p = \frac{1}{R} = \frac{1}{0.606} = 1.65\text{sec.}$$

and $$T_i = 4\text{min} = 240\text{sec.}$$

However, the proportional gain is not dimensionless anymore and has units of time. The proportional band cannot therefore be expressed in percent as required by the controller and this set of rules was disregarded.

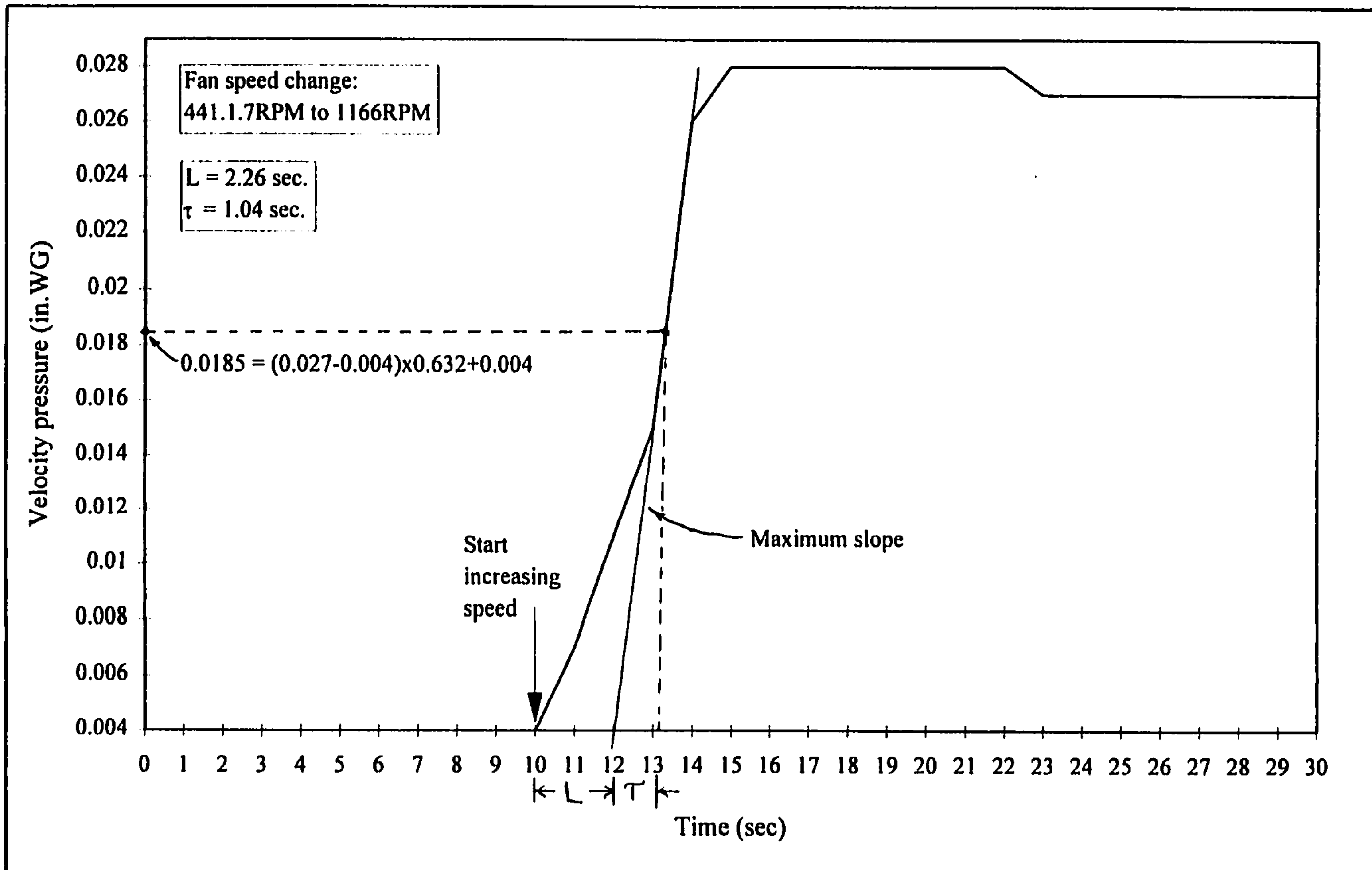

Figure 5.5 Effect of a Medium Fan Speed Step Change on Velocity Pressure

b) Results

The tuning parameters found with the simplified IMC-PID rules (10) for $\tau/L<3$ produce a large overshoot at startup but the velocity pressure returns rapidly to the setpoint (figure 5.6). The control loop responses compare with the startup tests performed using Ziegler and Nichols' tuning parameters. For a setpoint of 0.028 in.WG, the fan speed also reaches its maximum level (100% power) for one second before it decreases allowing the process variable to settle at its setpoint. Therefore, it produces the same overshoot, however, the steady-state response time is shorter, setpoint being reached within 26 seconds instead of 43 seconds. For a lower setpoint (0.013 in.WG), startup causes an overshoot of 0.037 in.WG (285%), and steady-state is also reached within 26 seconds.

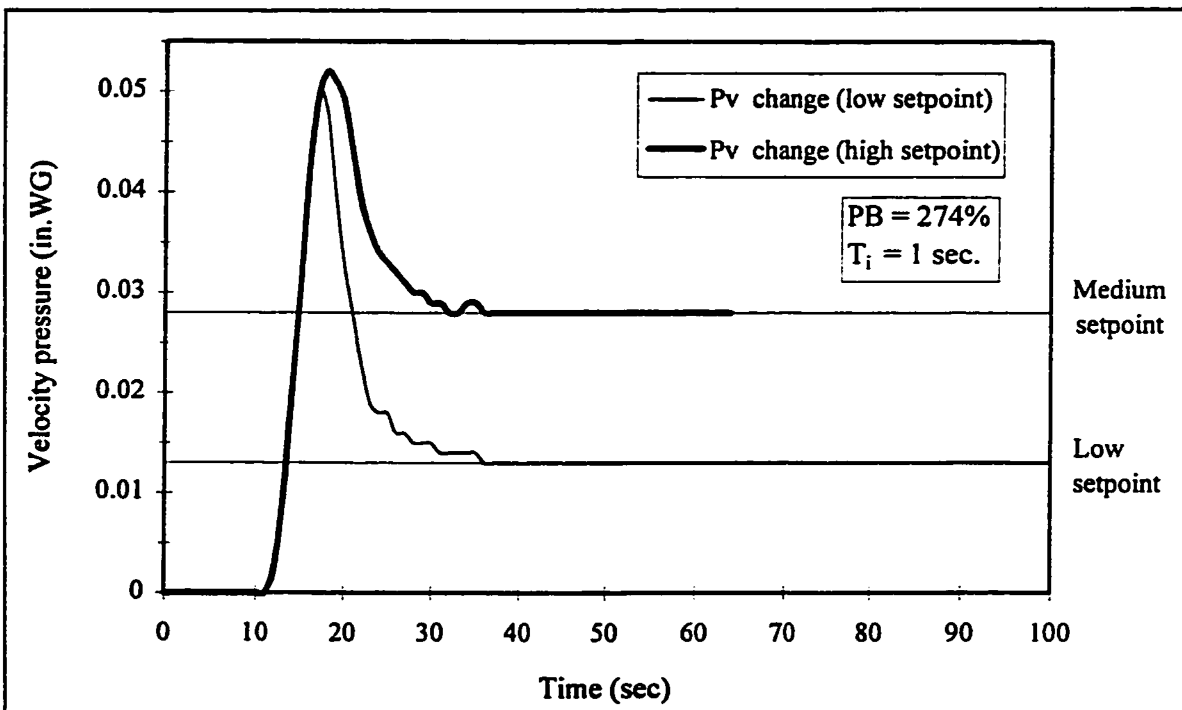

Figure 5.6 PI-Control of Fan Speed at Startup with $\tau/L<3$(Simp. IMC-PID Tuning Rules)

Figure 5.7 shows the fan speed response to load disturbances when the controller is tuned with the simplified IMC-PID parameters. It is very similar to the response obtained with Ziegler and Nichols' tuning parameters, however the loop reacts faster to the disturbances. Small hot duct damper movements (1/4 of full displacement) cause maximum pressure fluctuations of 0.002 in.WG (7%) which are dissipated within 15 seconds. There is one exception when the damper is moved from a quarter open to a fully closed position; in that case the velocity pressure peaks 18% below the setpoint. For damper movements equal to half the total displacement, the percentage difference between peaks and setpoint varies between 11% and 29% but the highest peak is reached when the damper travels from fully closed to fully open; the percentage difference between peak and setpoint is then 43%. When the damper travels at least half of the total displacement, the loop requires between 10 and 20 seconds to return to the setpoint. These results are specific to the setpoint selected and would be different if another setpoint had been chosen.

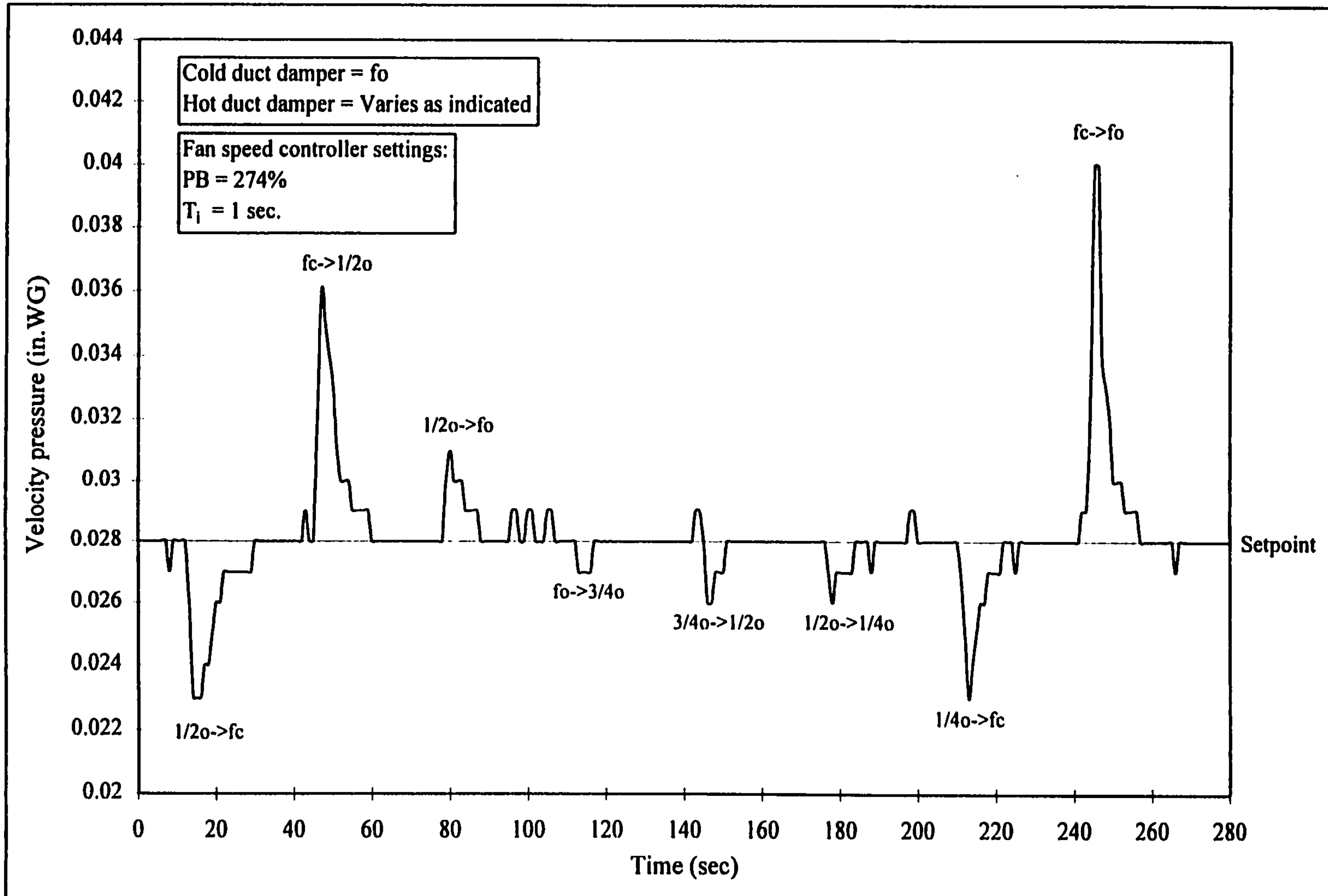

Figure 5.7 Velocity Pressure Fluctuations During Load Disturbances (Simplified IMC-PID Tuning Rules)

5.1.3 Bekker et al. Tuning Rules

The last tuning method to be tested on the fan speed controller was from Bekker et al.. To calculate the tuning parameters, the process dead time and time constant were required. These values have already been obtained in the open-loop test needed by the simplified IMC-PID tuning rules (figure 5.5).

a) Setting the Controller Gains

Bekker et al. tuning rules take into account the actuator speed of travel. Because the fan speed actuator is fast-acting, the tuning parameters were set to:

$$K_p=\frac{\tau}{LK_s}e^{-1}=\frac{1.04\times e^{-1}}{2.26\times 1.17}=0.144 \Rightarrow PB=\frac{100}{0.144}=694\%$$

where $$K_s=\frac{c_{ss}}{u}=\frac{0.027}{0.027-0.004}=1.17$$

with C_{ss} = process steady-state value
u = step input

and $$K_i=\frac{1}{LK_s}e^{-1}=\frac{e^{-1}}{2.26\times 1.17}=0.139\text{sec.}^{-1}$$

so $$T_i=\frac{K_p}{K_i}=\frac{0.133}{0.139}=1\text{sec.}$$

b) Results

The results obtained with Bekker et al. tuning parameters, which are shown in figures 5.8 and 5.9, differ from the other rules. The overshoot at startup reaches its maximum level

(100% power) independently of the setpoint value and the process variable stays at that level longer: three seconds for a setpoint at 0.013 in.WG and seven seconds for a setpoint at 0.028 in.WG. It also takes more time to reach steady-state: 68 seconds for a setpoint at 0.013 in.WG and 70 seconds for a setpoint at 0.028 in.WG. The difference between the two response times is very small; however, the low setpoint startup test has a much larger overshoot than the medium setpoint startup test, so it still seems that the loop response efficiency decreases when the operating conditions differ from the tuning conditions.

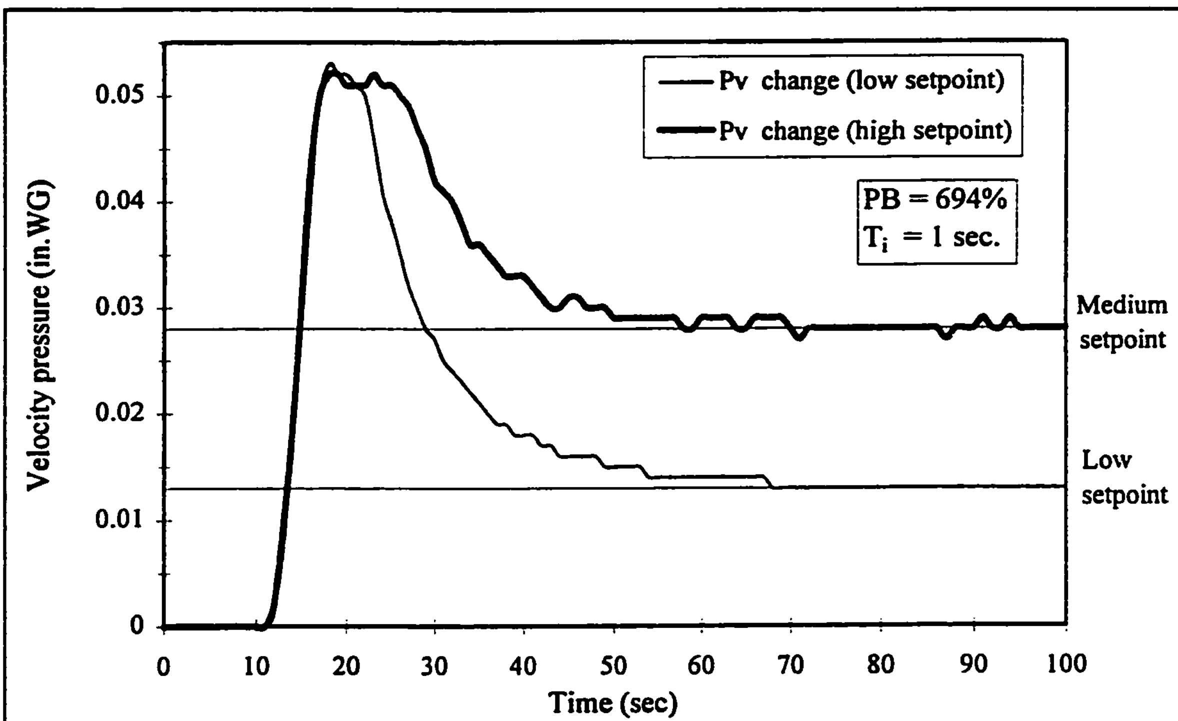

Figure 5.8 PI-Control of Fan Speed at Startup (Bekker et al. Tuning Rules)

Figure 5.9 shows that the system is also slower to react to load disturbances. Furthermore, the changes in damper position create higher velocity pressure peaks. The fan speed loop requires up to 35 seconds to return to the setpoint when disturbances are induced by small hot duct damper movements (1/4 of full displacement). The small displacements cause a maximum pressure fluctuation of 0.002 in.WG (7%) except when the damper is moved from a quarter open to a fully closed position; in that case the velocity pressure peaks 21% below the setpoint. For damper movements equal to half the total displacement, the percentage difference between peaks and setpoint varies between 11% and 36%. The highest peak is reached when the damper travels from fully closed to fully open; the percentage difference between peak and setpoint is then 57%. When the damper travels at least half the total displacement, the loop requires between 35 and 45 seconds to return to the setpoint. Again, these results are specific to the setpoint selected and would be different if another setpoint had been chosen.

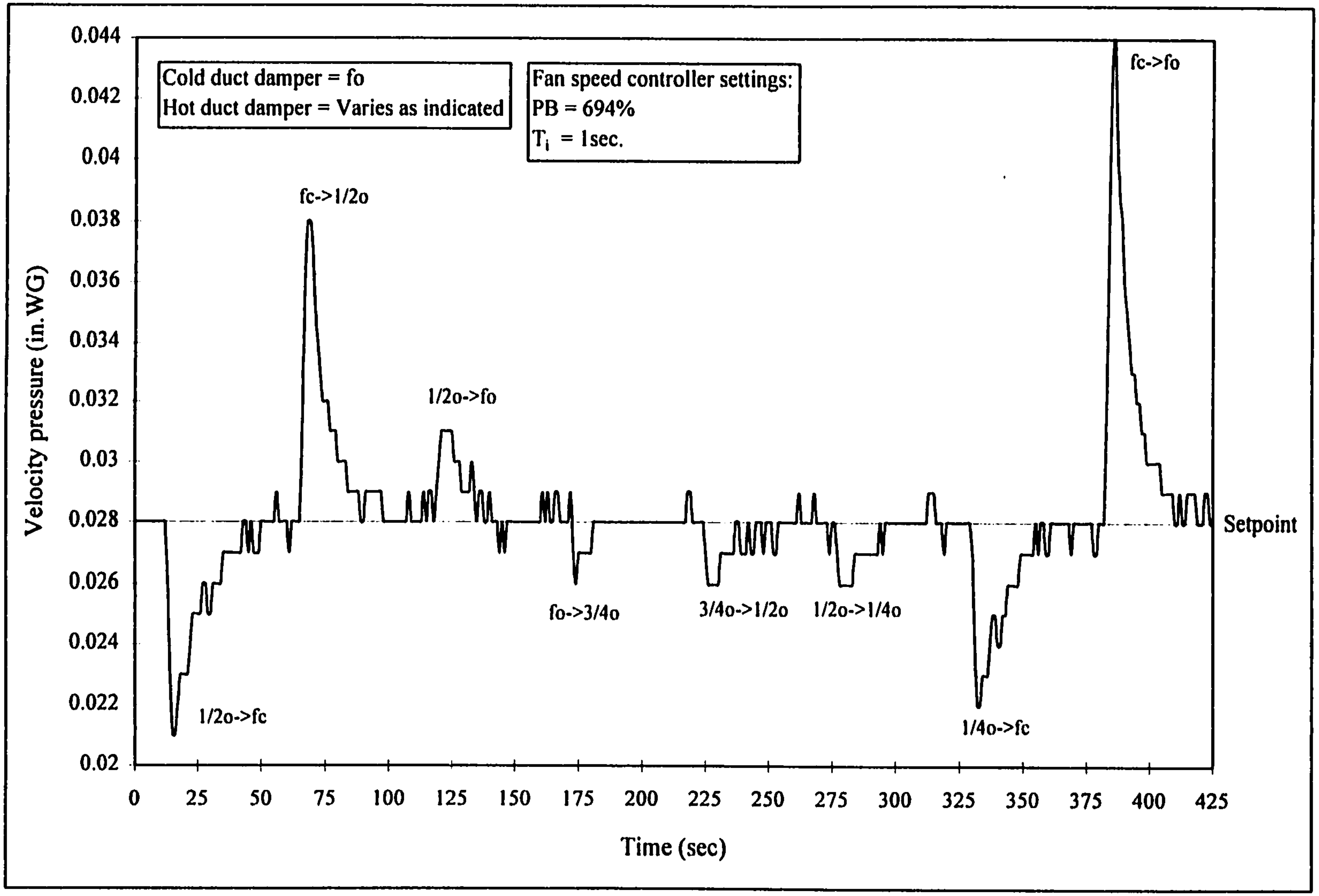

Figure 5.9 Velocity Pressure Fluctuations During Load Disturbances (Bekker et al. Tuning Rules)

5.1.4 Observations

Before comparing the different tuning rules to determine the most efficient method to maintain velocity pressure at its setpoint, two important observations will be made.

First, tuning the fan speed controller with three different tuning rules showed that it will be extremely difficult to eliminate the overshoot at startup; all three settings create an important overshoot which is higher than any peaks encountered during the load disturbance tests. One way of reducing the startup overshoot would be to adjust the cutback parameters of the controller; cutback operation takes action when the power is at 0 or 100% and the controller is saturated.

Second, the effect of different damper movements on fan speed control showed that large damper displacements should be avoided because they cause high velocity pressure peaks. Therefore, the damper controller settings shall be selected in order to obtain several small displacements rather than one large displacement. This will reduce the interaction between the damper control loop and fan speed control loop.

a) Comparison Between the Different Tuning Methods

The three tuning rules investigated provide different settings for fan speed control; they are summarized in table 5.1. It can be observed that the difference between the simplified IMC-PID tuning parameters (τ/L constraint) and Bekker et al. parameters lies only in the proportional band: $PB_{\text{Bekker et al.}}$ is about twice the value of $PB_{\text{simplified IMC-PID}}$. This explains why the response under Bekker et al. rules is slower. Ziegler and Nichols' tuning rules define a lower proportional band but a higher integral time which ends up providing results very similar to the simplified IMC-PID rules.

Table 5.1 Summary of Tuning Parameters for the Fan Speed Controller

	Ziegler and Nichols	Simplified IMC-PID	Bekker et al.
Proportional term (PB)	132%	274%	694%
Integral term (T_i)	3 seconds	1 second	1 second

Comparing the results obtained at fan startup, it can be seen in figure 5.10 that for both setpoints, the simplified IMC-PID tuning rules provide the fastest steady-state response time with 26 seconds. The fan speed control loop would require at least 33% more time to reach setpoint with Ziegler and Nichols' parameters. Bekker et al. tuning method (13) comes last, providing the worst response time which is 60±2 seconds for both setpoints. Figure 5.10 also shows that Ziegler and Nichols' and the simplified IMC-PID settings produce similar overshoots within ±0.001 in.WG, while Bekker et al. settings produce a larger overshoot for the small setpoint and keep the process variable at its peak value longer.

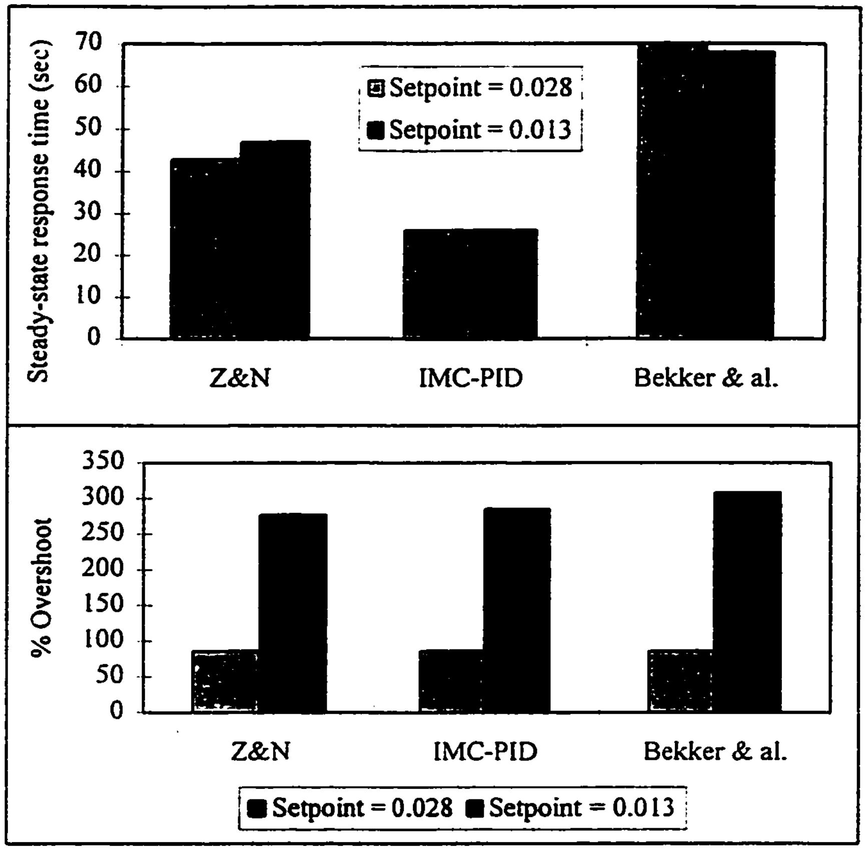

Figure 5.10 Summary of Results for Fan Startup Tests

Similar conclusions as the ones obtained with the startup tests can be made by observing how the fan speed control loop responded to load disturbances induced in the system. Again, the simplified IMC-PID tuning rules provide the fastest steady-state response time; Ziegler and Nichols' response time comes second, so Bekker et al. response time comes last. The velocity pressure peaks for Ziegler and Nichols and the simplified IMC-PID tuning settings are very similar, while Bekker et al. settings usually cause larger peaks. For the three tuning methods, it is impossible to relate damper displacements with the peak values; however, it is noted that for the same travelling distance, the fan speed control loop is affected more by a damper displacement starting at the fully closed position than by a displacement ending at the fully closed position or starting at the fully open position. Therefore, opening the damper from shut position has a larger effect on air flow than closing it. Furthermore, the largest peak always occurs when the damper travels from a fully closed to a fully open position. Such large displacements are not expected to occur during the full operation of the system so these high peaks should not be encountered. Table 5.2 summarizes the results obtained during the load disturbance tests.

Table 5.2 Summary of Results for Load Disturbance Tests on Fan Speed Control

Damper movement	1/4 move			1/2 + move		
Tuning rules	Ziegler & Nichols	Simp. IMC-PID	Bekker et al.	Ziegler & Nichols	Simp. IMC-PID	Bekker et al.
P_v peak value	0.001 - 0.004in.WG	0.001 - 0.005in.WG	0.002 - 0.006in.WG	0.003 - 0.12in.WG	0.003 - 0.012in.WG	0.003 - 0.016in.WG
Steady-state response time	11-25 sec.	5-12 sec.	9-32 sec.	20-30 sec.	10-20 sec.	38-42 sec.

Therefore, according to startup tests and load disturbance tests, the simplified IMC-PID tuning method with the constraint $\tau/L<3$ provides the best response for the fan speed controller.

b) Part-Load Fan Operation

Controllers are said to provide a proper response when the operating conditions are identical to the conditions used to determine the tuning parameters. However, if the operating conditions differ from the design conditions, then the response may become sluggish or oscillatory. In order to verify this statement, tuning parameters were determined for a low and medium load and then used at startup to reach a low, medium and high setpoint. The loop response to setpoint changes was also analyzed. In both cases, the simplified IMC-PID tuning method was used.

Tuning parameters at medium load have already been defined but a second open-loop test was required to determine the tuning parameters at low load. The fan speed was kept at 455.7 RPM for 10 seconds and then increased to 804 RPM; in pressure, this represents a step from 0.004 to 0.012 in.WG. Figure 5.11 shows how the velocity pressure in the mixed air duct varied with time due to the step change in fan speed. The line through the maximum slope defines the dead time (L), the time constant (τ) and the reaction rate (R). The ratio of τ/L determines the set of simplified IMC-PID rules to be used. In this case, with:

$$\frac{\tau}{L}=\frac{1.6}{1}=1.6<3$$

The proportional gain and the time constant are given by:

$$K_p=\frac{1}{2RL}=\frac{1}{2\times0.395\times1}=1.27=>PB=\frac{100}{1.27}=79\%$$

where, $$R=\frac{\left(\frac{0.0091-0.004}{12.6-11}\right)}{(0.012-0.004)}=0.395\text{sec.}^{-1}$$

and $$T_i=\tau=1.6\text{sec.}$$

Table 5.3 displays the tuning parameters found at low and medium loads. Testing both sets of tuning parameters for fan startup with a setpoint of 0.013 in. WG (figure 5.12) shows that both responses reach setpoint at the same time, but the overshoot is 85% lower for the parameters determined at low load. When the setpoint is equal to 0.028 in.WG and the controller is tuned with the low load parameters (figure 5.14), the loop response oscillates around the setpoint with ±0.002 in.WG. The response obtained using the medium load parameters has an 11% higher overshoot but it settles down to the setpoint in 26 seconds without oscillation. Therefore, for both setpoints, the best performing response is obtained when operating conditions are identical to design conditions. For the last startup test with a setpoint of 0.045 in.WG (figure 13), the loop response requires 23% more time to return to the setpoint with the medium load settings than with the low load settings. In both cases, the response fluctuates around the setpoint with ±0.001 in.WG. In conclusion, if the tuning parameters are not to be changed during the operation of the system, it seems in this case, to be more appropriate to tune the loop at medium capacity. The medium load tuning parameters, have the major advantage of never inducing a permanent oscillation in the loop response, even if they create a larger overshoot for a few seconds or require a longer steady-state response time.

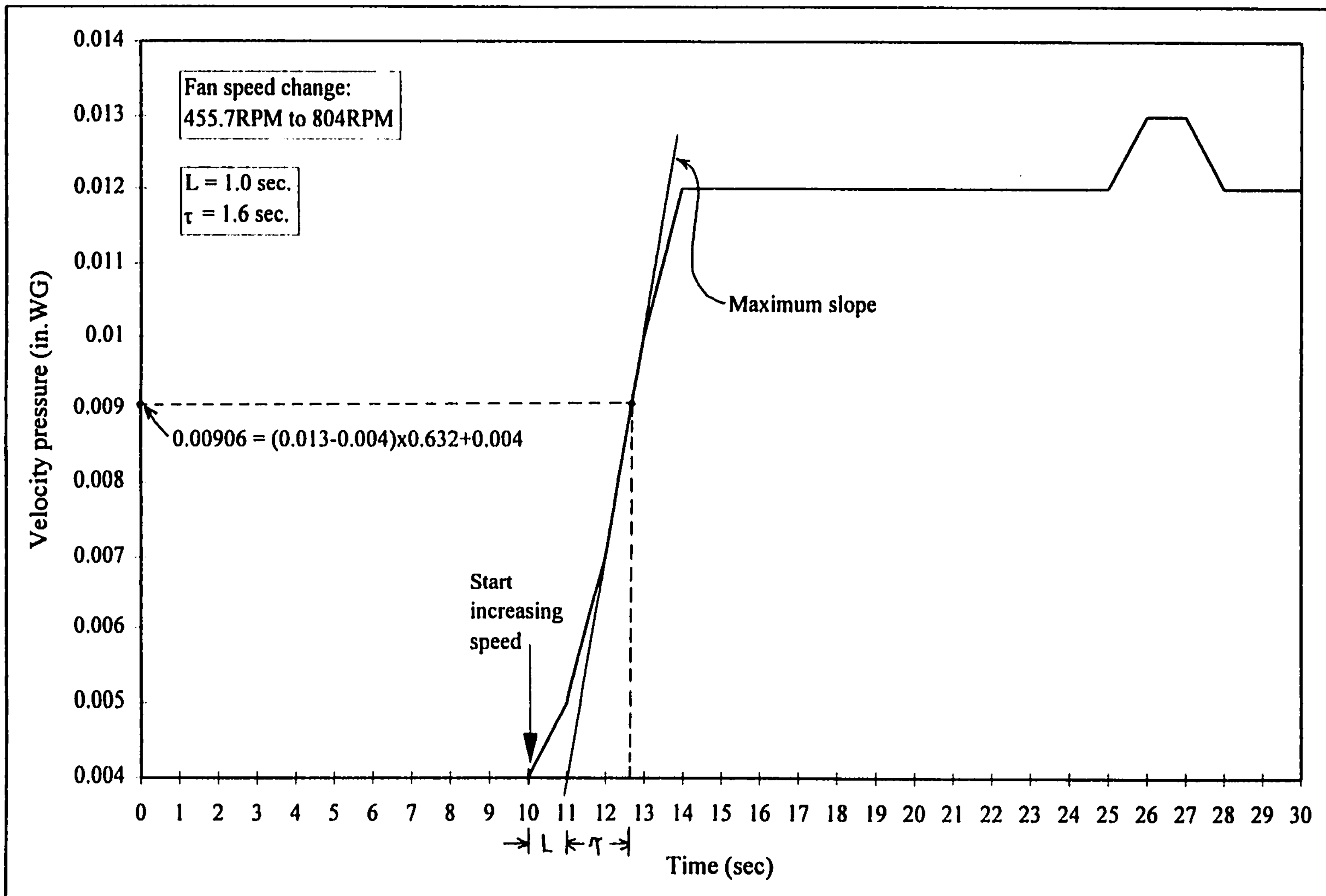

Figure 5.11 Effect of Small Fan Speed Step Change on Velocity Pressure

Table 5.3 Simplified IMC-PID Settings for Fan Speed Control

	Low load settings	Medium load settings
PB	79%	274%
T_i	2 sec.	1 sec.

As a last observation, in the case of the fan speed controller, the simplified IMC-PID tuning method defines parameters which provide good response to setpoint changes (figure 5.15). Within 30 seconds the process variable has settled to a new setpoint without creating oscillation, whether the setpoint was above or below the tuning design conditions. Note that the setpoint step change represented a change of 180 cfm (from 0.028 to 0.045 in.WG) and 213 cfm (from 0.028 to 0.013 in.WG).

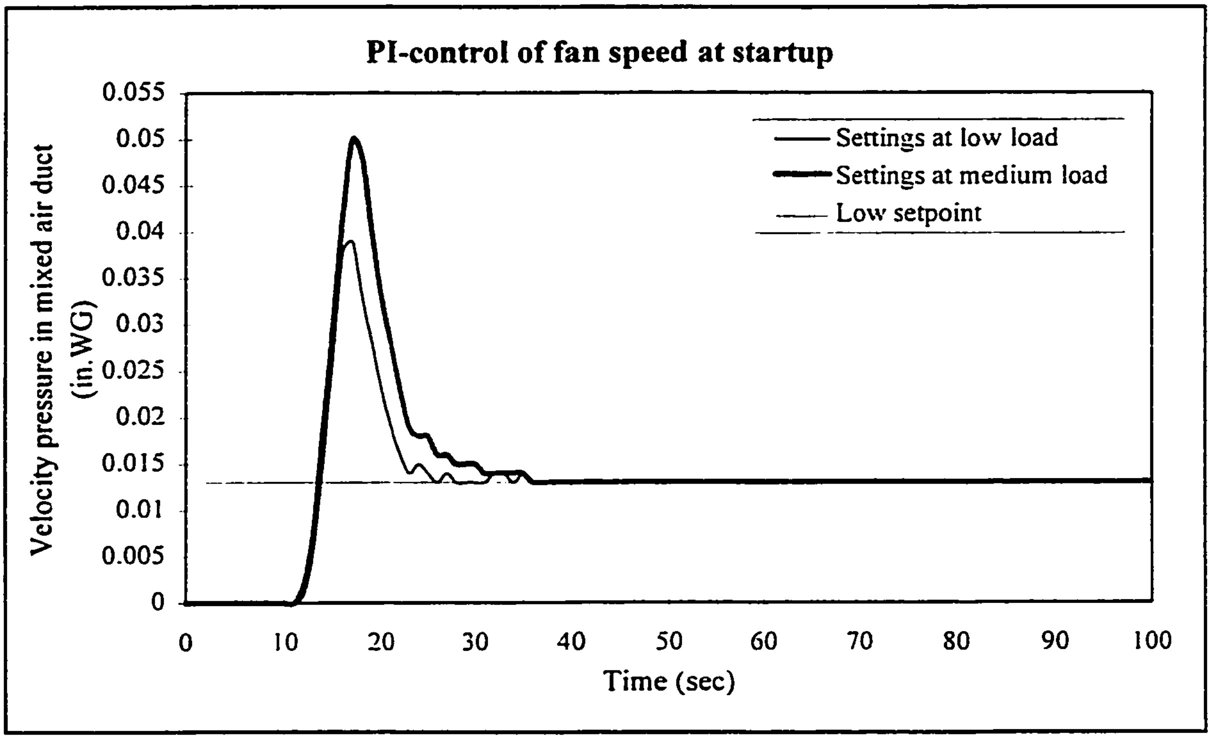

Figure 5.12 System Response to Different Settings (Low Setpoint)

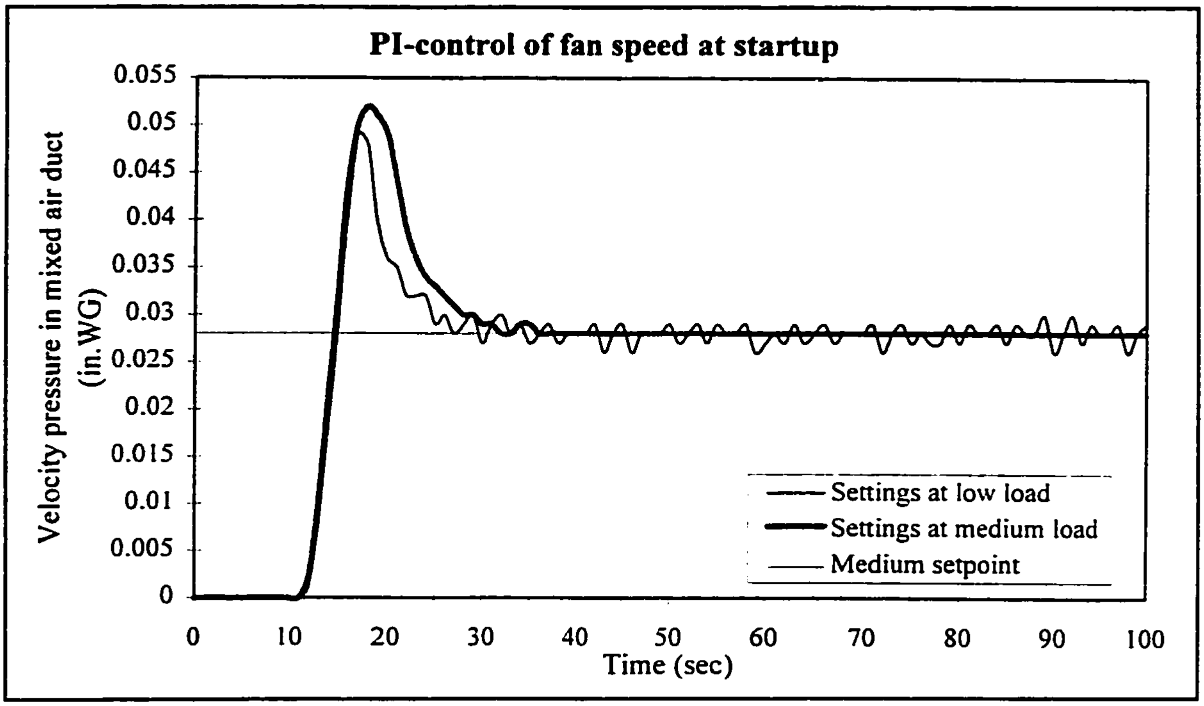

Figure 5.13 System Response to Different Settings (Medium Setpoint)

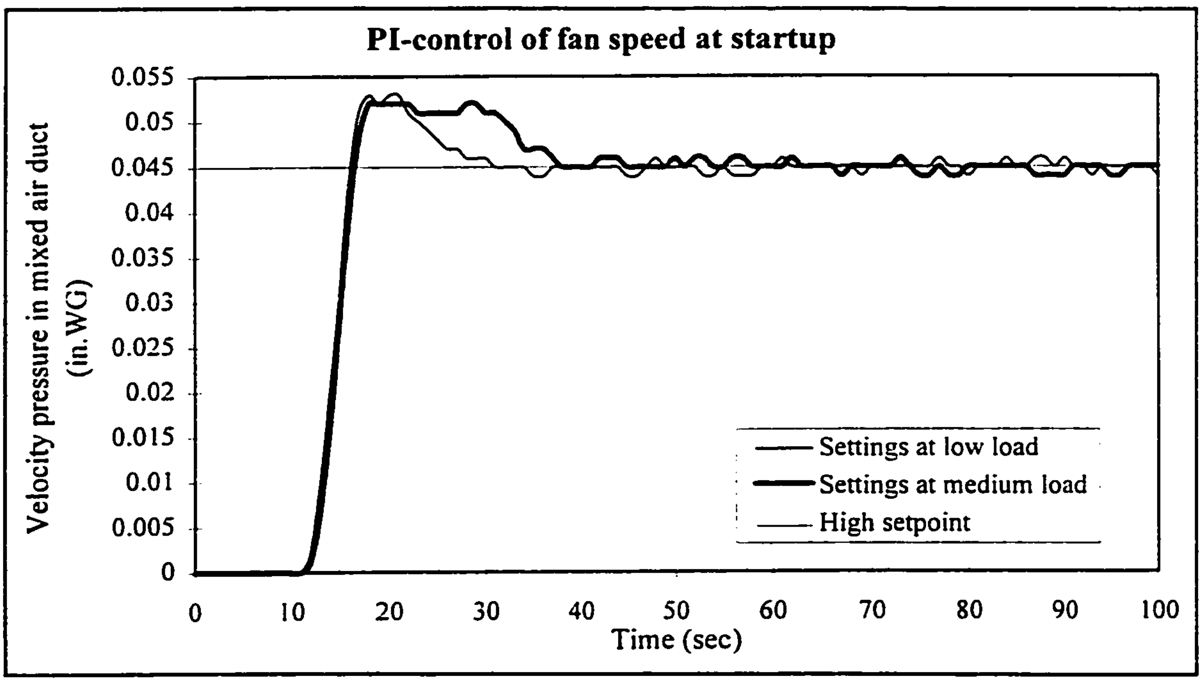

Figure 5.14 System Response to Different Settings (High Setpoint)

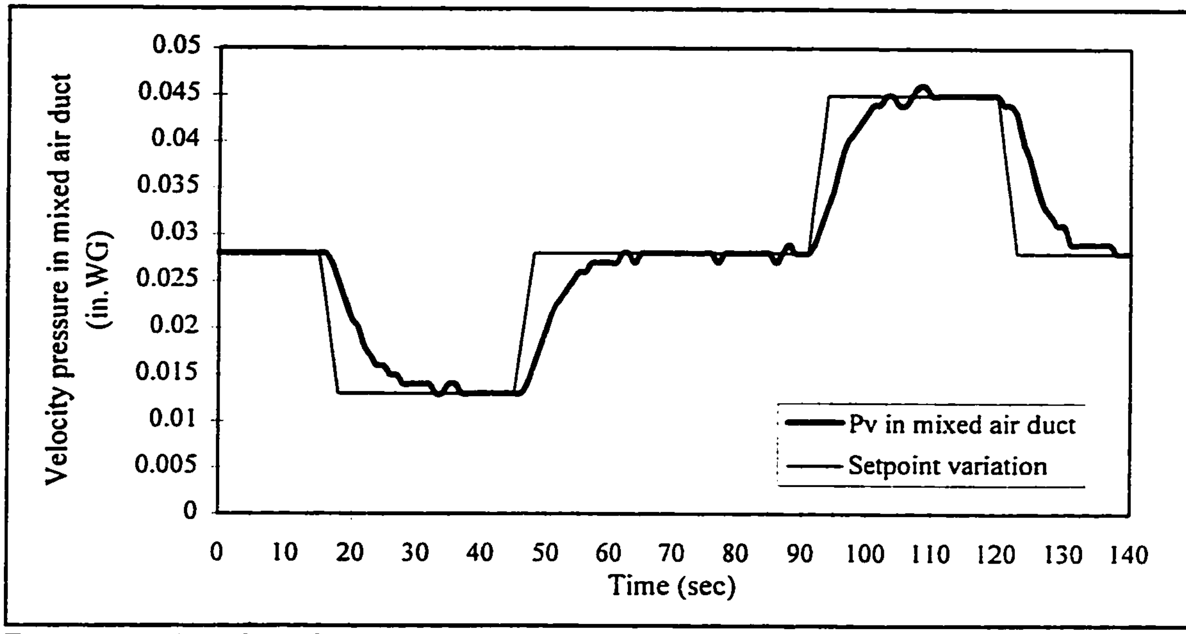

Figure 5.15 Fan Speed Loop Response to Setpoint Changes

5.2 Tuning the Valve Controller

The valve controller is a UDC-2000 from Honeywell and provides the flexibility of selecting which tuning parameters to be input: the proportional band or the proportional gain, and the integral time in minutes or the reset rate (1/minutes). During the configuration of the controller, the proportional gain and the integral time were chosen as the parameters to be input. Independently of the tuning method under study, the tests on the valve control loop were always started once the liquid temperature in the tank had settled to 6.5±0.2°C and with the valve fully closed. To analyze the effect of PI-control on the cooling valve control loop, the system was required to cool down the air in the cold duct to a temperature of 14.5°C, then two step changes in setpoint were performed: one to increase the temperature to 16.5°C and one to reduce the temperature back to 14.5°C. During the tests, all dampers were fully closed except for the cooling duct damper which was fully open, and the fan speed was set to its maximum value with the potentiometer. The effect of load disturbances on the valve control loop was not analyzed because it would have been too difficult to introduce the disturbances in the system.

5.2.1 Ziegler and Nichols' Tuning Rules

Ziegler and Nichols' tuning rules require to find the ultimate period of oscillation and the ultimate proportional band through several closed-loop tests in order to determine the tuning parameters.

a) Setting the Controller Gains

The product manual from Honeywell (32) also includes a section on manual tuning following Ziegler and Nichols' method. The step by step procedure is however somewhat different from the one described in the Eurotherm manual. First, the output must be adjusted so the process variable, the air temperature in the cold duct, is close to a desired value, 16.5°C. Then the derivative term must be set to zero and the integral time set to its maximum value, 50.00, to minimize their effects; this way the controller acts in proportional action only. The gain can then be increased significantly and the setpoint adjusted to be equal to the process variable. Finally, the setpoint value can be lowered to 14.5°C and the system response monitored. These steps must be repeated until the response oscillates with a constant period and amplitude which means the ultimate sensitivity is reached. The value of the gain, along with the value of the period can then be recorded and used to determine the tuning parameters.

From figures 5.16 and 5.17,

P = 80 and T = 0.54 min.

where, P = ultimate sensitivity
T = ultimate period of oscillation

Therefore, following the PI-control settings, the proportional band and the integral time to be input in the controller are:

$$K_p = 0.45P = 36$$
$$T_i = 0.8T = 0.45 \text{ min.}$$

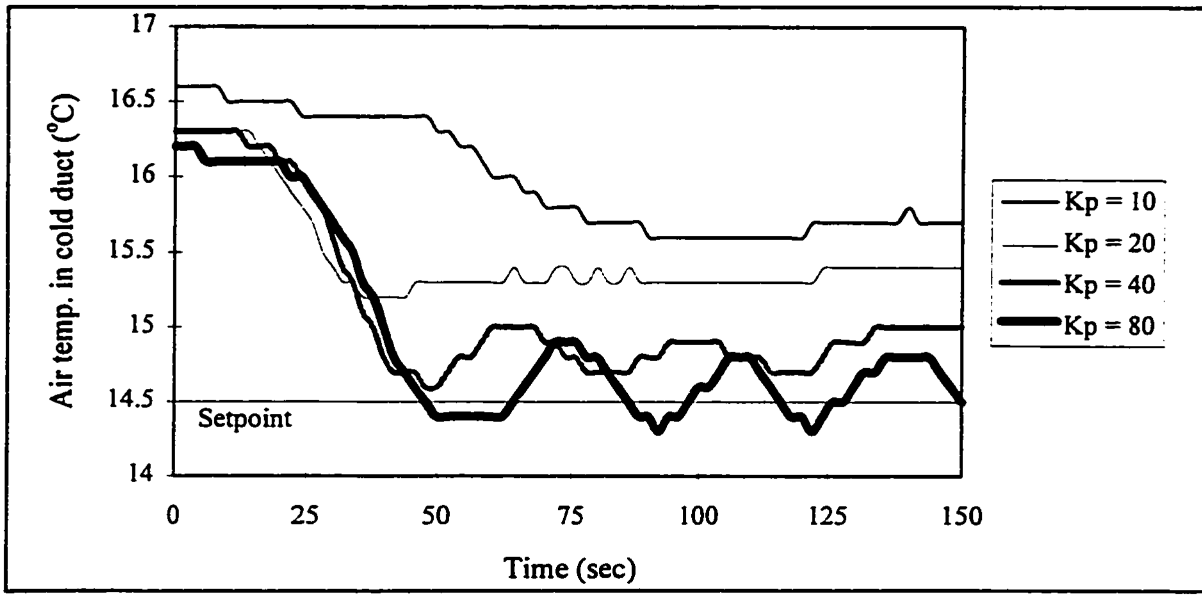

Figure 5.16 Closed-Loop Tests on Cooling Valve

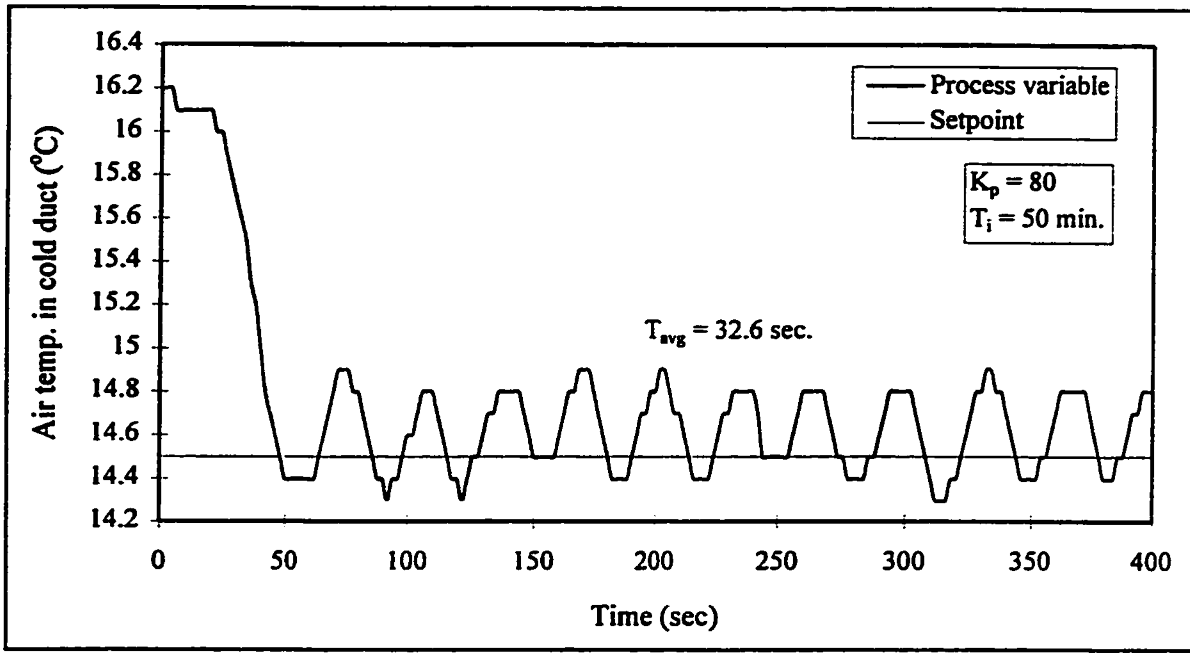

Figure 5.17 Closed-Loop Test on Cooling Valve (Ultimate Period of Oscillation)

b) Results

Considering that the time scale of thermal loops is much slower than the time scale of the air flow control loop, Ziegler and Nichols' settings provide a fast response to a cold air temperature setpoint change (figure 5.18). However, they induce a small oscillation of ±0.2°C. This is not deemed to be very important because it would not create discomfort and should not be of much influence on the energy consumption of the refrigeration system. In more details, an 8°C setpoint change from 22.5 to 14.5°C creates a 0.6°C undershoot and the setpoint is reached within 10 minutes. Such a large setpoint change should not occur often during system operation. Smaller setpoint changes of 2°C create a maximum undershoot/overshoot of 0.4°C and setpoint is reached within three minutes.

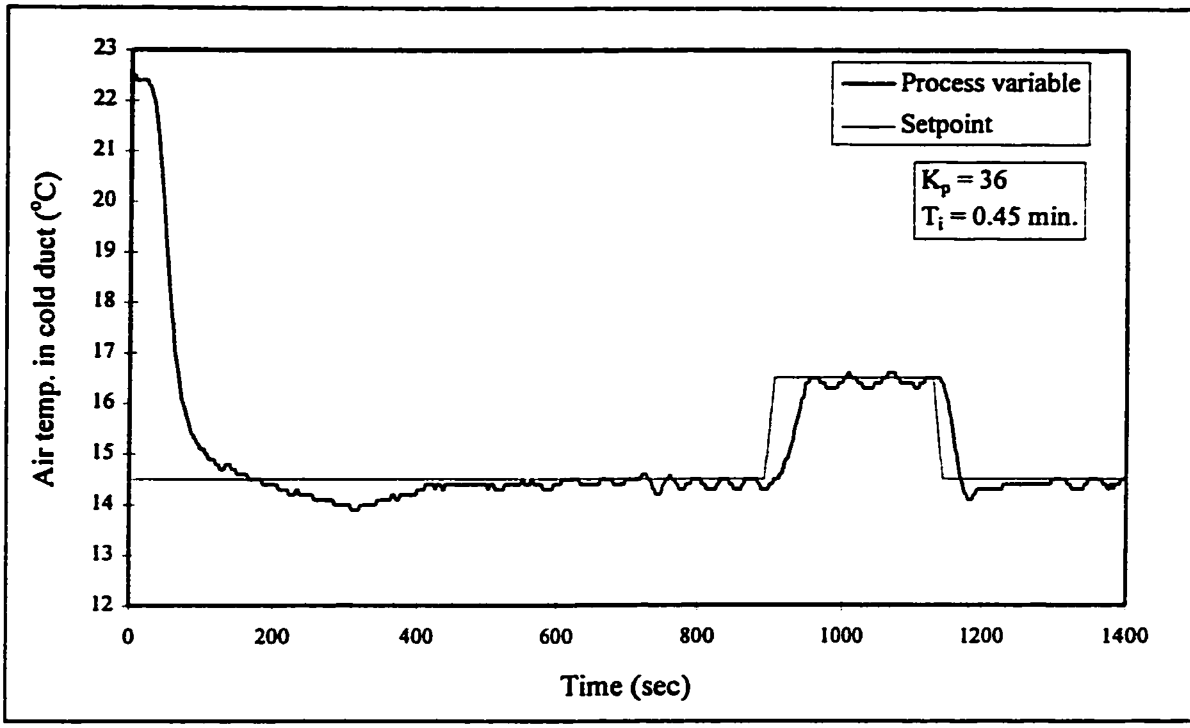

Figure 5.18 PI-Control of Cooling Valve (Ziegler and Nichols' Tuning Rules)

5.2.2 Simplified IMC-PID Tuning Rules

The dead time, time constant and reaction rate of the process must first be determined from an open-loop test before the controller can be tuned with the simplified IMC-PID rules.

a) Setting the Controller Gains

The open-loop test was performed by displacing the cooling valve after 30 seconds from a fully closed to a half open position with the liquid temperature in the tank at 6.5±0.2°C. Figure 5.19 shows how the air temperature in the cold duct varied with time due to the step change in valve position. A line through the maximum slope was drawn to define the dead time (L), the time constant (τ) and the reaction rate (R). The ratio of τ/L determines the set of simplified IMC-PID rules to be used. In this case, with:

$$\frac{\tau}{L}=\frac{19.0}{6.8}=2.8<3$$

The proportional gain and the time constant are given by:

$$K_p=\frac{1}{2RL}=\frac{1}{2\times2.17\times(6.8\div60)}=2.0$$

where, $$R=\frac{(\frac{18.2-22.4}{54.1-36.8})\times60}{(15.7-22.4)}=2.17\text{min.}^{-1}$$

and $$T_i=\tau=\frac{19.0}{60}=0.32\text{min.}$$

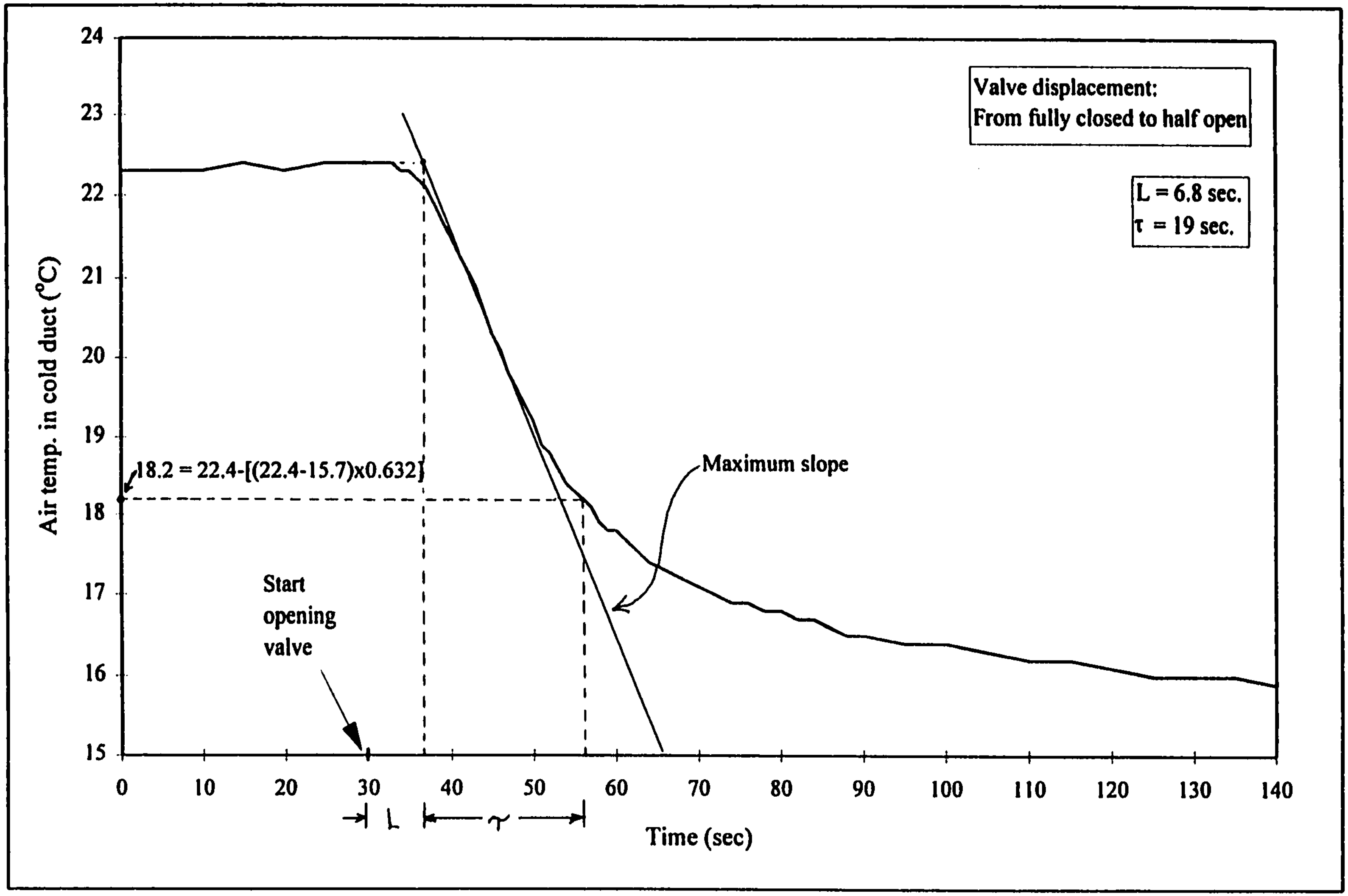

Figure 5.19 Variation of Temperature in Cold Duct Due to a Change in Valve Position

b) Results

Proportional and integral control of the cooling valve using the simplified IMC-PID tuning parameters provides a slower loop response than Ziegler and Nichols' parameters. For the first setpoint change where the process variable must decrease from room air temperature to 14.5°C, the undershoot is small at 0.4°C but it takes 17:24 minutes to reach steady-state. For smaller setpoint changes of 2°C the response is also slow: it takes almost nine minutes to reach 16.5°C and more than 13 minutes to return to 14.5°C (figure 5.20).

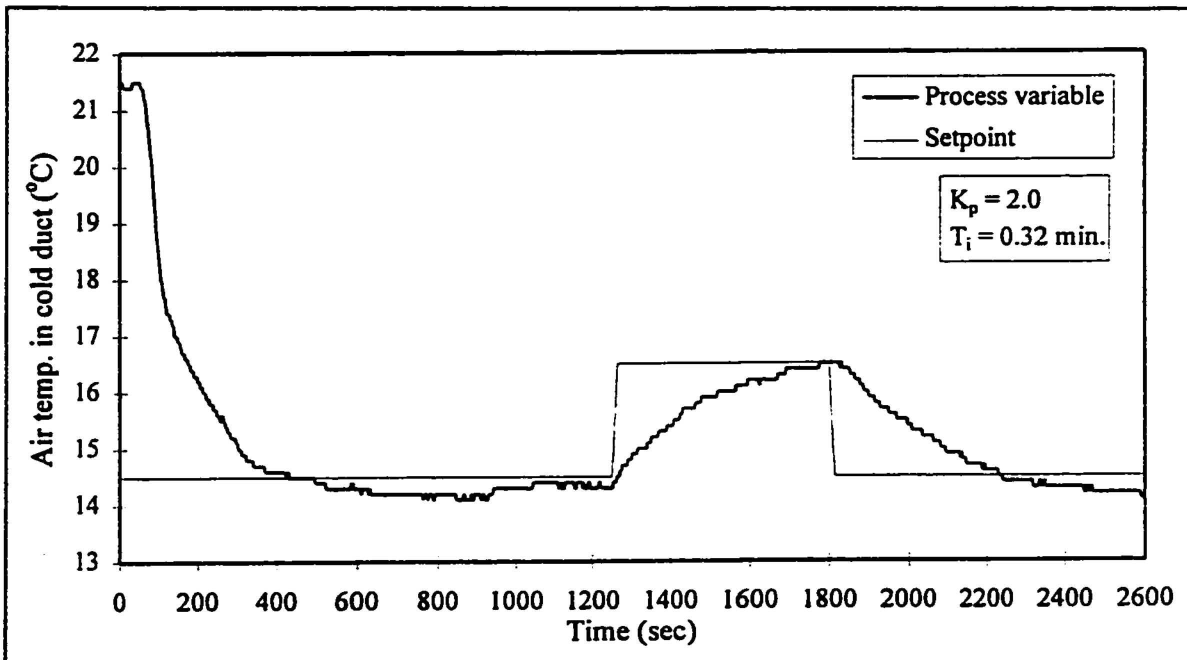

Figure 5.20 PI-Control of Cooling Valve (Simplified IMC-PID Tuning Rules)

5.2.3 Bekker et al. Tuning Rules

The last tuning method to be tested on the cooling valve controller was from Bekker et al.. The required process dead time and time constant have already been obtained in the open-loop test performed for the simplified IMC-PID tuning rules (figure 5.19).

a) Setting the Controller Gains

The set of tuning rules from Bekker et al. used for the cooling valve controller differ from the one used for the fan speed controller because the valve actuator is slow-acting, that is, the dead time is less than half the time constant. The tuning parameters were set to:

$$K_p = \frac{\tau}{LK_sK_{GR}}e^{-1} = \frac{19.0\times e^{-1}}{6.8\times 2.34\times 2.19} = 0.20$$

where, $$K_s = \frac{C_{ss}}{u} = \frac{15.7}{22.4-15.7} = 2.34$$

$$K_{GR} = a_6 X^{[a_1 lnX + a_3 lnY + a_4]} Y^{[a_2 lnY + a_5]} = 2.19$$

with
C_{ss} = process steady-state value
u = step input
a_1 = +0.1092
a_2 = +0.0121
a_3 = -0.0760
a_4 = -0.2685
a_5 = +0.1026
a_6 = +1.1814
X = L/τ = 6.8/19 = 0.358
Y = T_v/τ = 60/19 = 3.16
T_v = actuator operation time = 60 sec.
K_{GR}= gain reduction factor

and $$K_i = \frac{1}{LK_sK_{GR}}e^{-1} = \frac{e^{-1}}{(6.8\div 60)\times 2.34\times 2.19} = 0.633\text{min.}^{-1}$$

so $$T_i = \frac{K_p}{K_i} = \frac{0.20}{0.633} = 0.32\text{min.}$$

b) Results

The loop response obtained when the cooling valve controller is tuned with Bekker et al. rules is very slow (figure 5.21). After 36.5 minutes the process variable settled at 14.2°C which is 0.3°C below the setpoint. There are two possibilities to explain this offset: either the system was not given enough time to reach setpoint, or the discrepancy between the process variable and the setpoint is due to the data acquisition system which does not always precisely follow the controller reading. The difference between the two readings can go up to 0.2°C. Since the response to a large setpoint change is much slower than with the other tuning rules, it was not considered relevant to run a test with smaller setpoint changes of 2°C.

It is possible that the loop response is slow because of the gains reduction performed. There were actually two constraints described in Bekker et al.: one allowing to reduce the gain and another constraint which stated that the system did not require gain reduction if the actuator operation time was less than 20τ, which is the case with the cooling valve. If no reduction is applied to the gains, the proportional gain doubles and becomes 0.4 while the integral time remains unchanged. Compared with the simplified IMC-PID tuning parameters, the integral times are equal but the new proportional gain is still conservative. Therefore, the new loop response without gains reduction will remain slower than the response obtained using the simplified IMC-PID tuning parameters.

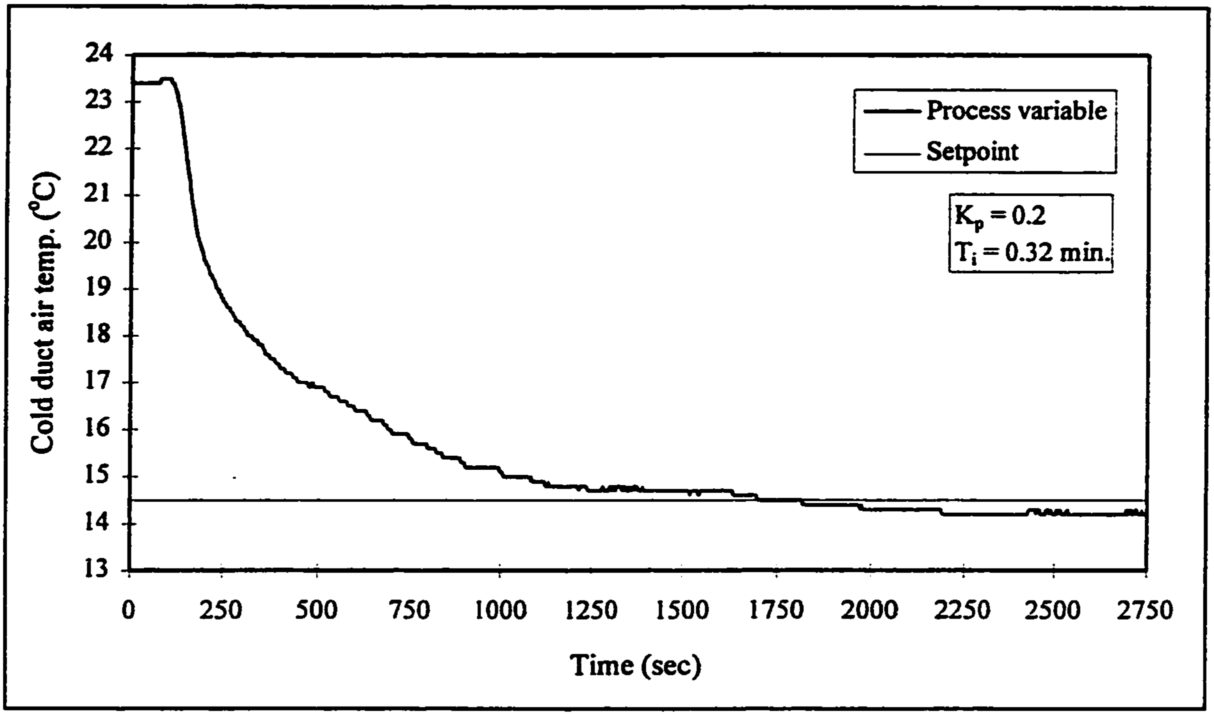

Figure 5.21 PI-Control of Cooling Valve (Bekker et al. Tuning Rules)

5.2.4 Observations

Comparing the tests performed on the cooling valve, Ziegler and Nichols' tuning parameters surely provide the fastest response (figure 5.22). The undershoot they create for a large setpoint change might be more important than the other sets of parameters, but the process variable stays below 14.3°C for only 1:42 minutes compared to 5:06 minutes for the simplified IMC-PID parameters and 8:24 minutes for Bekker et al. parameters. Note that for both Ziegler and Nichols and the simplified IMC-PID tuning methods, the steady-state response time is longer for a 2°C setpoint decrease than for a 2°C setpoint increase. Table 5.4 presents a summary of the tuning parameters used during the different tests. It can be observed that the simplified IMC-PID and Bekker et al. proportional gain would need to be increased considerably to obtain the optimum response.

Table 5.4 Summary of Tuning Parameters for the Valve Controller

	Ziegler and Nichols	Simplified IMC-PID	Bekker et al.
Proportional term (K_p)	36.0	2.0	0.2
Integral term (T_i)	0.45 minute	0.32 minute	0.32 minute

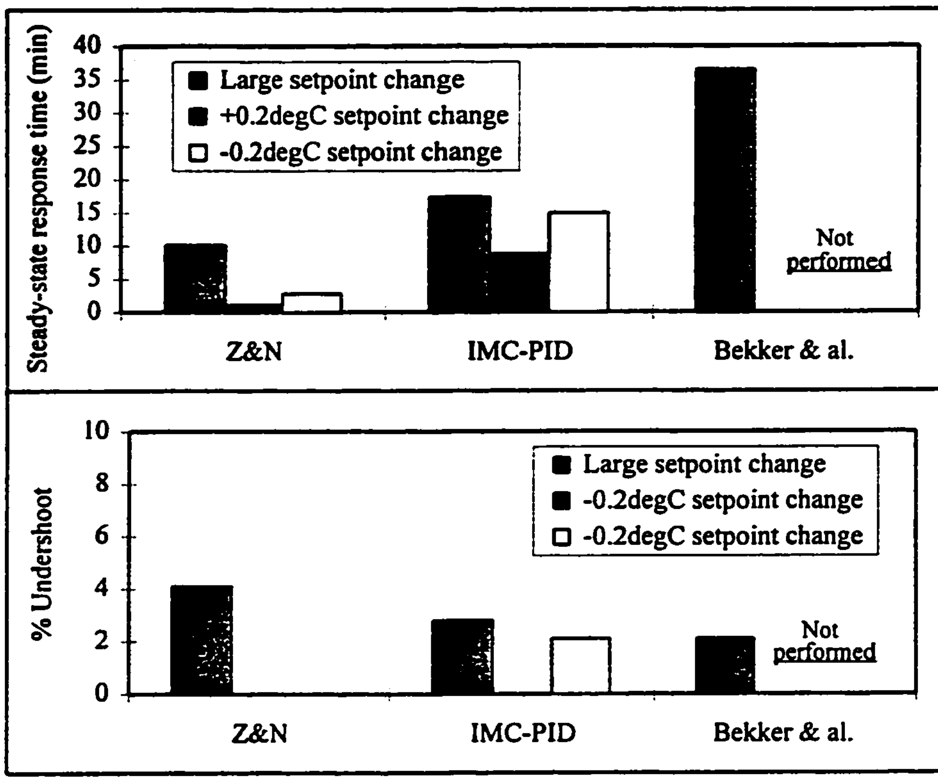

Figure 5.22 Summary of Results

Therefore, of the three tuning rules, Ziegler and Nichols' parameters provide the best response: it is fast, the undershoot is not too large, and the oscillation present in the response is very small.

5.3 Tuning the Hot Duct Damper Controller

The hot duct damper controls the air temperature in the mixed air duct. Because it is not possible to operate the heater in open-loop, it was first tried to tune the damper controller with cooling only but it was very difficult to produce a satisfying mixed air temperature change by mixing room air with cold air at 14.5°C. Therefore, in order to perform conclusive tests, it was preferable to have simultaneous heating and cooling. Cooling was created by keeping the valve position half opened and the liquid temperature in the tank at 6.5±0.2°C, and the fan speed was set to its maximum value with the potentiometer. Like for the valve control loop, the effect of load disturbances is not analyzed because it would be too difficult to introduce the disturbances in the system.

5.3.1 Ziegler and Nichols' Tuning Rules

Since the damper controller is also a UDC 2000 from Honeywell, the same steps as with the cooling valve were taken to determine Ziegler and Nichols' tuning parameters. However, in this case there are two closed loops involved: the damper and the electrical heater. Because the procedure requires to go back and forth from one setpoint to another until the ultimate sensitivity is reached, the effect of the heater controller trying to keep its own setpoint has a large influence on the control of mixed air temperature. It was therefore decided that Ziegler and Nichols' tuning rules were unapplicable in the case of the damper controller.

5.3.2 Simplified IMC-PID Tuning Rules

The fact that the heater can only be operated in closed-loop will also affect the determination of tuning parameters according to the simplified IMC-PID rules (11). However, since the test is performed in one step the effect of the heater controller should be less than with the tests required by Ziegler and Nichols' rules.

a) Setting the Controller Gains

The open-loop test was performed by displacing the hot duct damper after 20 seconds from a fully closed position to a ¾ open position. Prior to the test the liquid temperature in the tank is provided time to settle to 6.5±0.2°C and the heater is switched on. While the hot duct damper is fully closed, its exhaust damper is open to avoid overheating of the coils. At that time, the heater is always on because the controller does not receive a smaller error signal; the hot duct damper being closed, the heat cannot reach the sensors. When the test is started, the exhaust damper is closed and the heater is always on for the important section of the test (the first 60 seconds) because the air temperature in the hot duct has not reached the setpoint yet. Figure 5.23 shows how the air temperature in the mixed air duct varied with time due to the step change in damper position. A line through the maximum slope was drawn to define the dead time (L), the time constant (τ) and the reaction rate (R). The ratio of τ/L determines the set of simplified IMC-PID rules to be used. In this case, with:

$$\frac{\tau}{L}=\frac{41}{1}=41>3$$

The proportional gain and the time constant are given by:

$$K_p = \frac{1}{2RL} = \frac{1}{2\times3.44\times(1\div60)} = 8.7$$

where, $$R = \frac{\left(\frac{20.54-15.8}{42-31}\right)\times60}{(23.3-15.8)} = 3.44\text{min.}^{-1}$$

and $$T_i = 5L = \frac{5\times1}{60} = 0.08\text{min.}$$

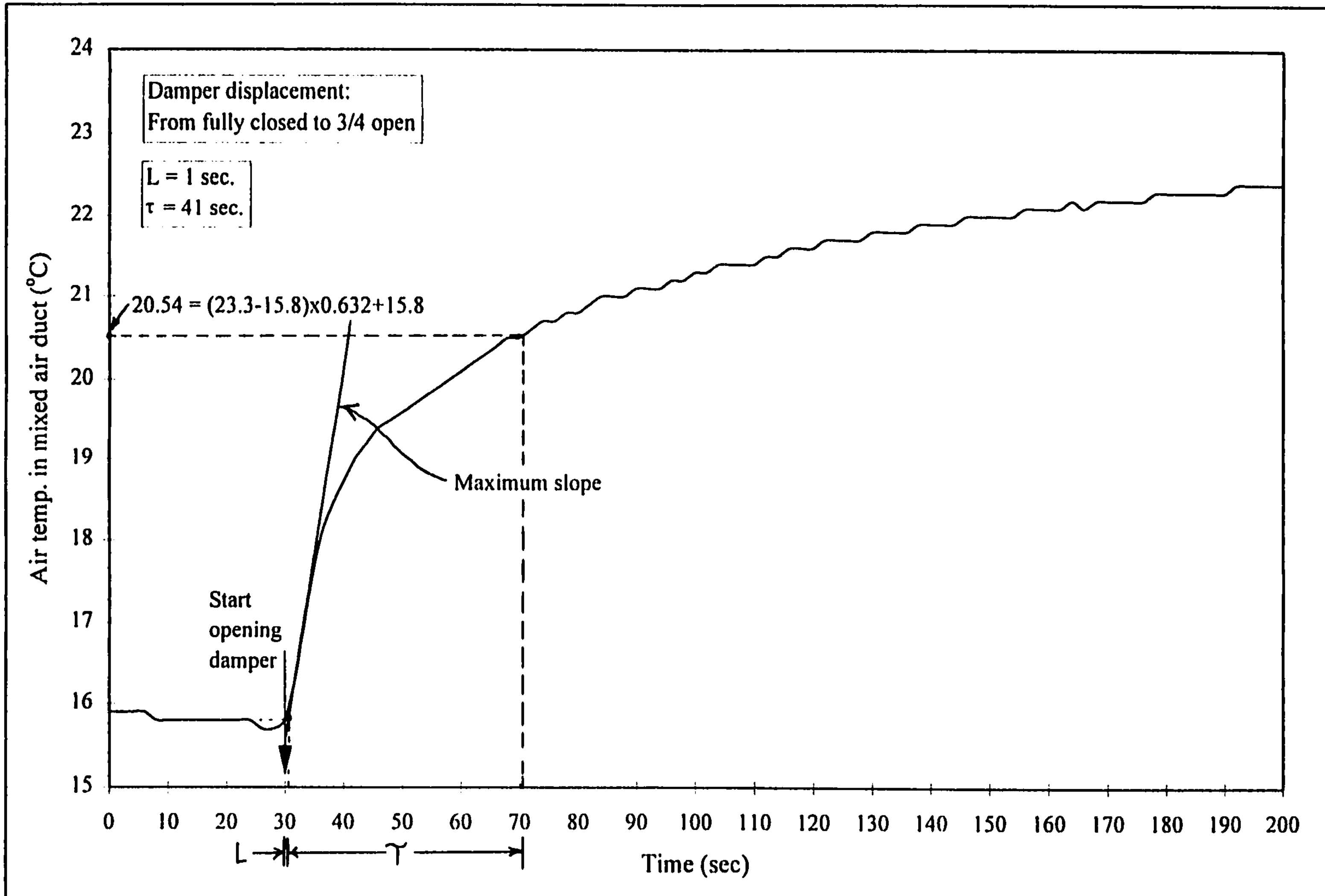

Figure 5.23 Variation of Temperature in Mixed Air Duct Due to a Change in Hot Duct Damper Position

b) Results

The loop response obtained for damper control with the simplified IMC-PID tuning parameters is presented in figure 5.24. A 5.5°C setpoint change causes the process variable to first reach the setpoint of 21°C in 1:20 minutes, then overshoots for 2:46 minutes before it finally settles to the setpoint within 10 minutes. The maximum value of the overshoot is 0.4°C above the setpoint.

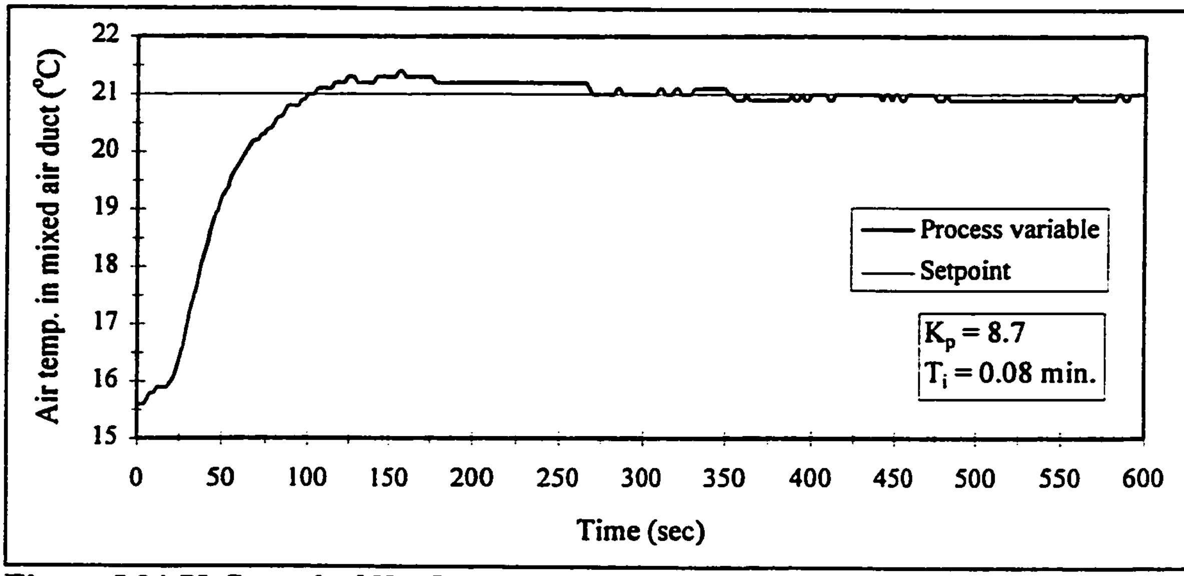

Figure 5.24 PI-Control of Hot Duct Damper (Simplified IMC-PID Tuning Rules)

5.3.3 Bekker et al. Tuning Rules

Like in the case of the other controllers previously tuned, the process parameters found in the open-loop test required by the simplified IMC-PID rules, figure 5.23, were used to determine the tuning parameters of Bekker et al. rules.

a) Setting the Controller Gains

Since the damper travels from fully closed to fully open faster than the valve, and considering the valve actuator provided a conservative response, Bekker et al. gains will not be reduced and the tuning parameters are set to:

$$K_p = \frac{\tau}{LK_s} e^{-1} = \frac{41 \times e^{-1}}{1 \times 3.11} = 4.85$$

where $$K_s = \frac{c_{ss}}{u} = \frac{23.3}{23.3 - 15.8} = 3.11$$

with C_{ss} = process steady-state value
u = step input

and $$K_i = \frac{1}{LK_s} e^{-1} = \frac{e^{-1}}{\frac{1}{60} \times 3.11} = 7.1 \text{min.}^{-1}$$

so $$T_i = \frac{K_p}{K_i} = \frac{4.85}{7.1} = 0.68 \text{min.}$$

b) Results

Figure 5.25 shows the system response to a 5.5°C change in the damper controller setpoint when the latter is tuned with Bekker et al. parameters. The process variable increases and first reaches the setpoint of 21°C in 10 minutes. Thereafter, it slowly fluctuates between 20.8°C and 21.4°C.

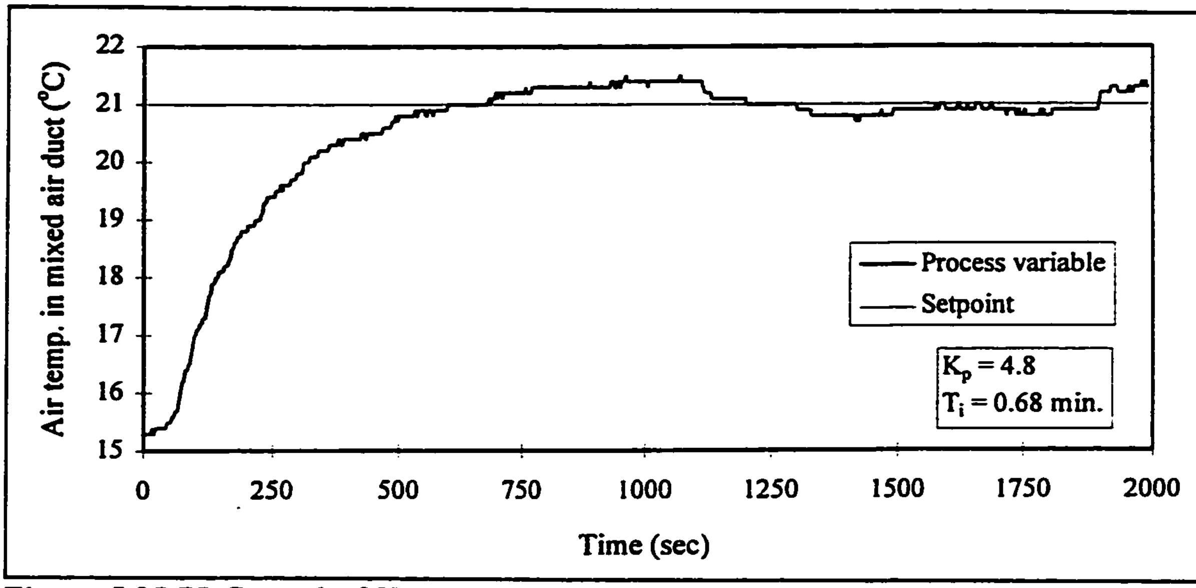

Figure 5.25 PI-Control of Hot Duct Damper (Bekker et al. Tuning Rules)

5.3.4 Observations

Table 5.5 summarizes the different tuning parameters used to control the hot duct damper loop. With a higher proportional gain and a lower integral time, the simplified IMC-PID tuning rules provide a more sensitive response than Bekker et al. rules. Even if it created an overshoot within the first 10 minutes, the process variable settled very close to the setpoint, fluctuating between 20.9 and 21°C only. The response with Bekker et al. parameters only reached setpoint after 20 minutes and induced mixed air temperature fluctuations varying between 20.8 and 21.3°C. Figure 5.26 summarizes the results obtained by both methods. Clearly, the simplified IMC-PID tuning method provides the best response for the damper control.

To verify if the simplified IMC-PID tuning parameters will provide good loop response under operating conditions, two setpoint changes were performed in addition to the first setpoint increase which brought the mixed air temperature to 21°C (figure 5.27). The figure shows that the process variable undershoots or overshoots always stay within 3% of the setpoints, whether the loop went through a large or small setpoint change. Also, the process variable requires about 3:30 minutes to reach steady-state for both the 2°C setpoint changes. Of course, the steady-state response time is longer, almost 5 minutes, for the large setpoint change.

Table 5.5 Summary of Tuning Parameters for the Damper Controller

	Simplified IMC-PID	Bekker et al.
Proportional term (K_p)	8.7	4.8
Integral term (T_i)	0.08 minute	0.68 minute

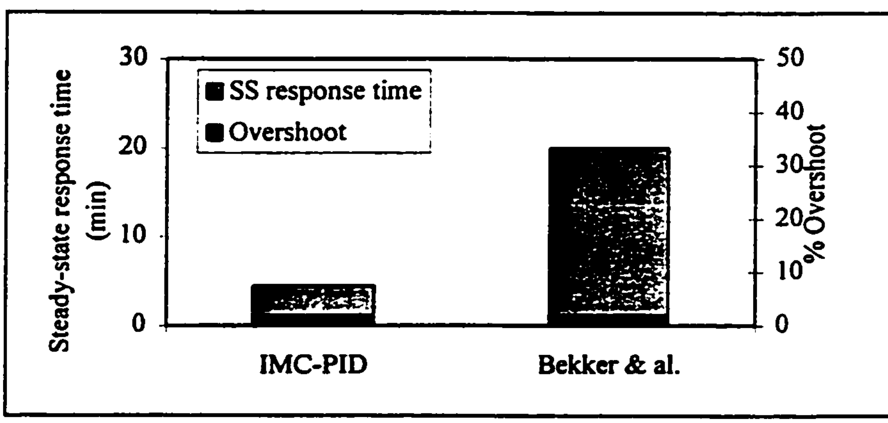

Figure 5.26 Summary of Results

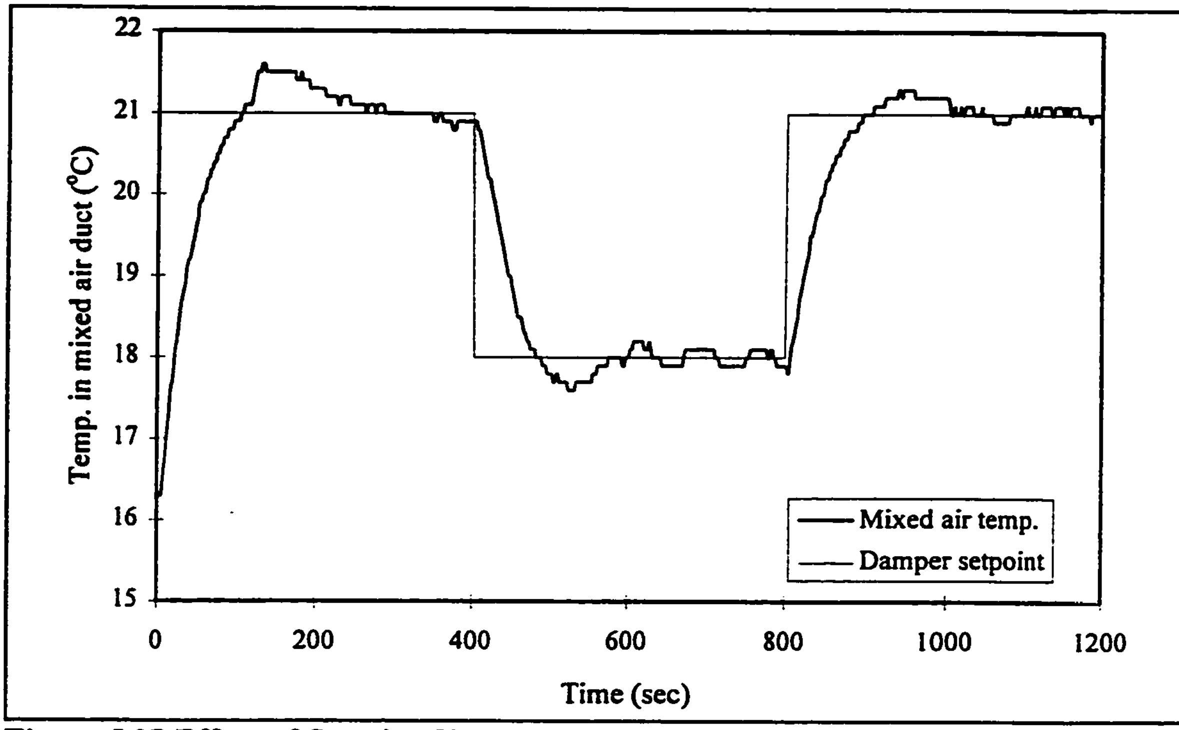

Figure 5.27 Effect of Setpoint Changes on Damper Control Loop

5.4 Tuning the Heater Controller

With the present installation, it is not possible to perform open-loop tests with the heater because it will heat until it reaches its upper limit and shut itself off to prevent overheating. A thermostat would need to be installed in order to test the heater in open loop.

5.4.1 Ziegler and Nichols Tuning Rules

The heater controller is also from Honeywell, therefore the method explained in the cooling valve section is again followed to perform the closed-loop tests with a setpoint change from 26.5°C to 28°C. During the tests, the hot duct damper was half open and all the other dampers were fully closed; the fan speed was set at its maximum value with the potentiometer and there was no cooling. The results of those tests are shown in figure 5.28. Even with a proportional gain as high as 640, it is not possible to produce an oscillatory response. The heater controller will consequently be tuned with proportional action only. A proportional gain of 320 is chosen because it provides a good response with very little fluctuations around the setpoint (figure 5.29). Steady-state for the small 1.5°C setpoint change is reached very quickly (less than 15 sec.) without any offset.

5.4.2 Yuwana and Seborg on-line Controller Tuning Method

Yuwana and Seborg on-line controller tuning method (9) would allow to determine the process parameters required by Bekker et al. tuning rules to determine the proportional gain and the integral time. However, this method requires to obtain, with proportional action only, an underdamped response that oscillates around the setpoint before stabilizing on it. Since this was not possible with the heater, the method could not be applied.

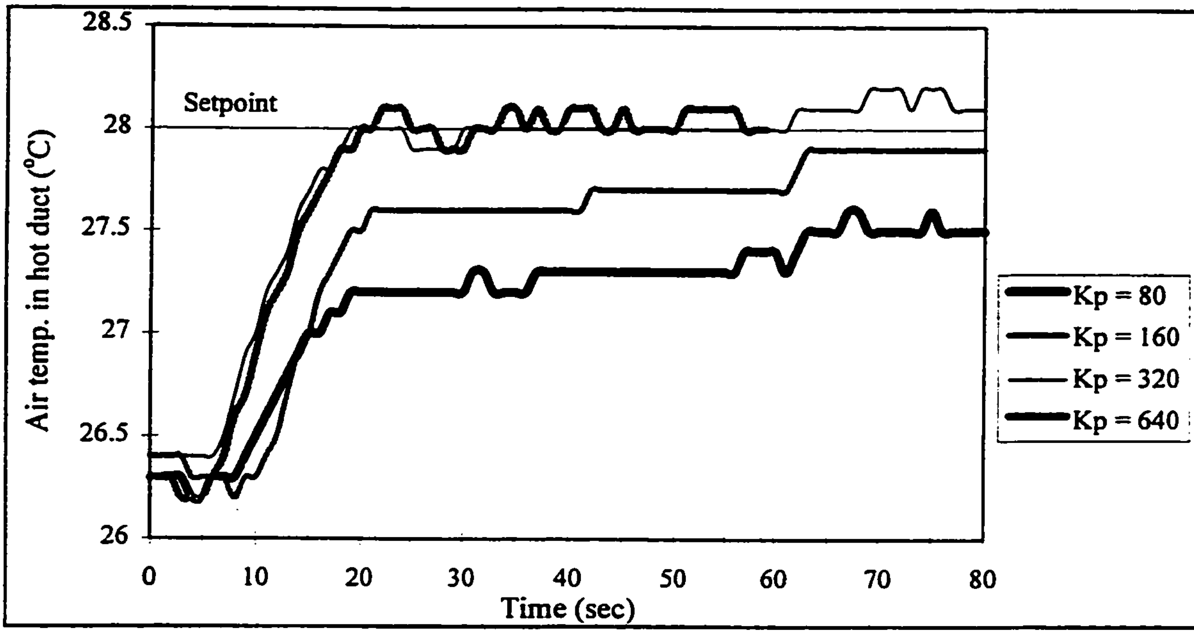

Figure 5.28 Closed-Loop Tests on Heater

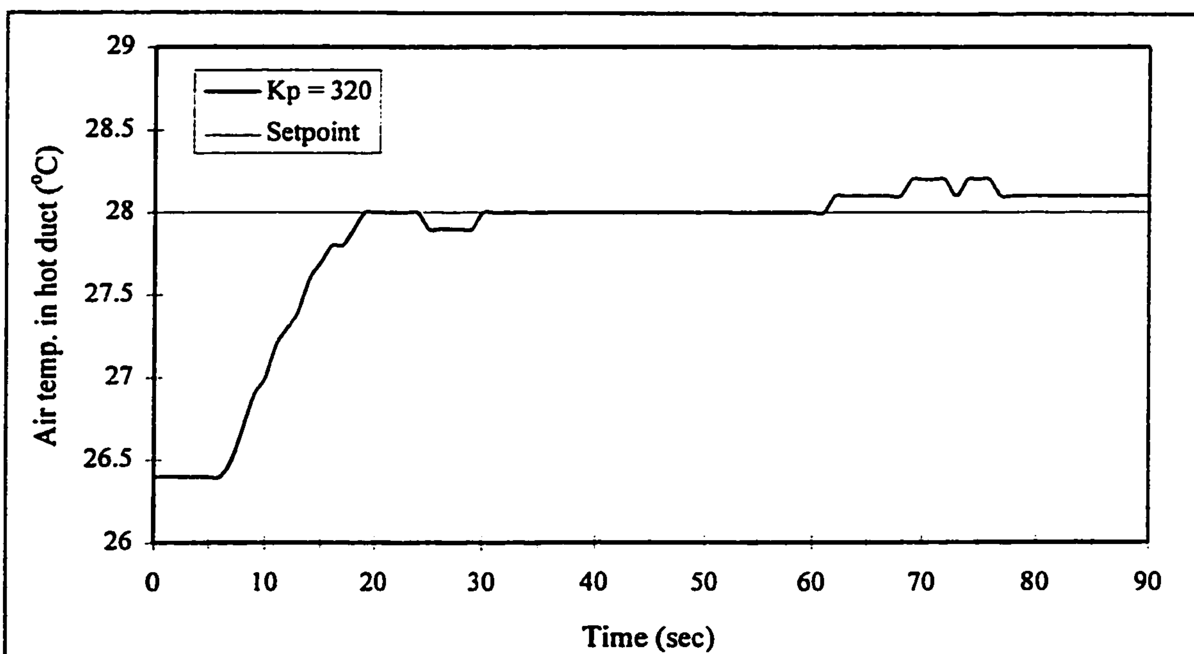

Figure 5.29 P-Control of Electrical Heater

5.5 Comparison Between Closed and Open Loop Responses

Open-loop tests performed in Chapter 4 showed that a new setpoint can be reached very efficiently. However, it would be extremely tedious to determine the appropriate positions of the fan speed potentiometer, valve or damper to obtain a specific setpoint. Controllers will perform this task automatically but may alter the quality of the response. Of course, if the controllers are poorly tuned, poor responses will result. Therefore, each controller will be configured with its best set of tuning parameters. In order to obtain corresponding open-loop and closed-loop curves, the open-loop curves were first obtained simulating an average load; then the closed-loop curves were obtained by performing the appropriate setpoint change on the controllers. Figure 5.30 shows that the fan speed controller considerably slows down the air distribution system response and the valve and damper controllers cause undershoot/overshoot in the thermal system response. The laboratory system requires triple the time to reach the velocity pressure steady-state when the fan speed controller is activated (figure 5.30a). Part of the delay is due to the controller waiting for the setpoint change to be input, through the arrow keys, before acting. The effect of the other controllers, is not as important; the valve controller causes a minor delay of 20 sec. and an undershoot of 6% (figure 5.30b) while the damper controller only causes an overshoot of 1% (figure 5.30c).

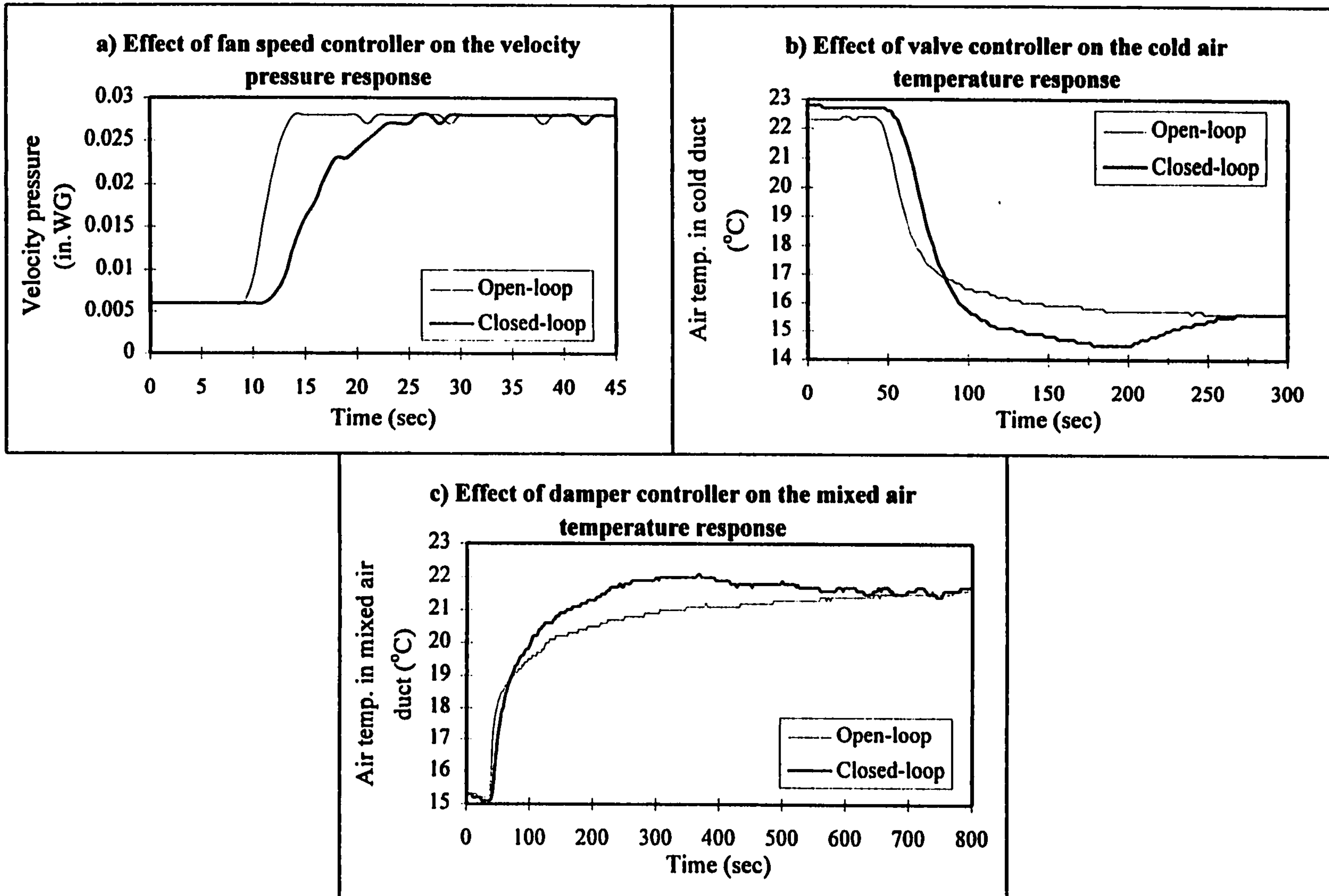

Figure 5.30 Comparison Between Open-Loop Reponse and Closed-Loop Response

5.6 Summary

The different tuning experiments conducted on the laboratory HVAC system, demonstrated that Ziegler and Nichols, and the simplified IMC-PID tuning rules generally provide acceptable loop responses. On the other hand, Bekker et al. tuning rules are too conservative for such a small system and provide slow loop responses. Table 5.6 identifies the tuning rules selected for each control loop. The parameters are not optimum and the responses obtained could be improved through fine-tuning. All three tuning methods were easy to apply, however Ziegler and Nichols' ultimate sensitivity method required several tests before the tuning parameters can be determined. One may easily assume the loop considered has reached ultimate sensitivity when in fact it has not. This error may considerably affect the quality of the response. In the simplified IMC-PID and Bekker et al. tuning methods, only one open-loop response was required. The determination of the maximum slope was not found to be critical in these cases. The maximum slope was determined by observation, without any calculations, and still, good responses were obtained for the simplified IMC-PID method. However, if the open-loop tests had been unstable then the on-line method from Yuwana and Seborg should have been used. The stability of the system obviously facilitated tuning the different control loops. With this system, the effort expended while tuning a control loop depends more on the controller brand than on the loop itself. A very flexible controller will be more complex to configure because it will involve the selection of more parameters. The fan loop required less preparation than the other loops because no cooling or heating was involved. It also required less time to run a test because of its fast time scale. Time delays were not a factor because of the small size of the system. Note that it was not always possible to test all the rules on two control loops due to the system limitations.

Table 5.6 Tuning Parameters Selected for Each Control Loop

Control loop	Tuning rule	Type of control	Tuning parameters
Fan speed	Simplified IMC-PID	PI	PB = 274% , T_i = 1 sec.
Valve position	Ziegler & Nichols	PI	K_p = 36 , T_i = 0.45 min.
Damper position	Simplified IMC-PID	PI	K_p = 8.7 , T_i = 0.08 min.
Heater	Ziegler & Nichols	P	K_p = 320 , T_i = 50 min.

The different tests also permit to comment on the time scale of the different control loops. Assuming that the valve and damper operate in the same time scale, table 5.7 indicates that based on small setpoint changes, a 1°C difference in setpoint change represents an extra 30 seconds for the process variable to reach steady-state. Applying this reasoning to large setpoint changes means that a 2.5°C difference in setpoint change should require an extra 75 seconds to reach steady-state. This is less than 30 seconds away from the actual value (100 seconds); therefore, it can be stated that the valve and the damper operate in very similar time scales. The fan speed control loop was found to operate in a time scale approximately 10 times faster than the two thermal control loops time scales. Finally, it is difficult to determine the heater time scale because only one test with a small setpoint change was performed; however, the heater seems to operate in a faster time scale than the valve and the damper. Therefore, it seems Ziegler and Nichols' tuning method is more appropriate for control loops operating in slow time scales, while the simplified IMC-PID tuning method would be more appropriate for control loops operating in fast time scales.

Table 5.7 Valve and Damper Steady-State Response Time

	Large setpoint change		Small setpoint change	
	Setpoint change of:	Steady-state response time	Setpoint change of:	Steady-state response time
Valve control	8.0°C	400 sec.	2°C	180 sec.
Damper control	5.5°C	300 sec.	3°C	210 sec.
Difference:	2.5°C	100 sec.	1°C	30 sec.

With the experience gained from tuning four control loops, selecting the proper tuning rule was not found to be the most difficult task in tuning a system with constant tuning parameters. Choosing the proper design conditions is more complex. Setpoints and load conditions must be selected carefully to recreate the operating conditions which will occur most often. Indeed, part-load tests performed on the fan speed controller showed that tuning design conditions different from operating conditions affected the quality of the system response.

6.0 OPERATING THE HVAC SYSTEM UNDER REAL CONDITIONS

After each of the four control loops were tuned according to one of the three methods studied, the system was analyzed as a whole. This part of the study determined whether the HVAC system was able to maintain the four process variables at their setpoints. First, the system was operated under the design conditions already established. Then, interaction effects was analyzed by introducing oscillation in each of the following control loops, one loop at a time.

- Fan speed controller
- Valve controller
- Damper controller

To induce oscillation in the loop under consideration, the controllers' parameters were severely modified. In the case of the heater controller, it was shown during the tuning stage that oscillation could not be produced under proportional-only control. To obtain an oscillatory response for that loop, PI-control would be required. Therefore, this controller was disregarded when analyzing the effect of poorly tuned controllers.

The purpose of all these tests was to determine between which loops interaction occurs and to what extent.

6.1 System Response Under Full Operation

The system was operated with the four control loops running simultaneously. Assuming that operating conditions were identical to the design conditions, the controllers' setpoint was set to the same values as those chosen during the tuning phase. The purpose of such a test is to determine if several loops working together affect the efficiency of the system to reach the desired setpoints. To perform this test, the water in the tank was first allowed to reach 6.5°C. During that time, the cooling valve was kept shut by setting the valve controller setpoint to 27.5°C. Once the water reached the required temperature, the heater and the fan were switched on and the valve controller setpoint was reduced to 16.5°C. After 420 seconds, another valve setpoint change was performed in order to reduce the cold air temperature to 14.5°C. The first setpoint change could represent morning start up of the system while the second setpoint change could represent new requirements, which for example could be due to higher occupancy.

The system response obtained during the test can be analyzed through four curves representing valve, fan, damper and heater control loops (figure 6.1). The valve and damper control loops have similar time scales, while the fan and the heater control loop have a much faster time scale. Comparing the curve of figure 6.1 with the different results obtained while tuning each controller one at a time, it can be said that:

(a) The overshoot/undershoot of the mixed air temperature response have increased and it seems the damper control is slower;

(b) The large undershoot of 2.1°C for the air temperature in the cold duct must be considered;

(c) There is a small oscillation in the velocity pressure response which was not present when all the other loops were open;

(d) The air temperature in the hot duct is not very much affected by the other loops.

Identifying the causes of the effects listed above will allow an understanding of the interaction between the different loops. First, considering cold air temperature and mixed air temperature (figure 6.1a and c), it can be observed that they both vary in a similar pattern. Drawing the two curves on the same set of axes (figure 6.2) shows that the mixed air temperature is directly affected by the valve setpoint changes. When the valve setpoint is reduced to 16.5°C, there is a short delay before the valve opens which is caused by the time required to reduce the setpoint from 27.5°C to approximately 23°C. Then, the temperature in the cold duct starts decreasing and so does the temperature in the mixed air duct. Both process variables drop below their respective setpoint to produce an undershoot, however the mixed air temperature reaches its minimum sooner because its setpoint (21°C) is encountered first. That is, while the cold air temperature is still decreasing to reach its setpoint, the damper has already been activated to bring back the temperature in the mixed air duct to 21°C. The peak in velocity pressure (figure 6.1b) shows that the damper was first activated at 45 seconds, at which time the cold air temperature was still above its setpoint. The mixed air temperature dropped 1.5°C below its setpoint while the cold air temperature dropped 2.1°C below its setpoint, but this large cold air temperature undershoot did not prevent the latter from settling to its setpoint one minute before the mixed air temperature. This is due to the direct effect the air temperature in the cold duct has on mixed air temperature; for the mixed air temperature to settle to its setpoint, the cold air temperature must be steady or varying slightly. For example, the mixed air temperature overshoot between 180 and 300 seconds is partly due to the cold air temperature increasing which consequently warms up the temperature in the mixed air duct. At time 420 seconds the valve setpoint is reduced to 14.5°C. This causes an undershoot of 0.5°C in the cold air temperature which reaches its setpoint in 152 seconds. The setpoint change also causes a drop of 0.4°C in mixed air temperature before it returns its to setpoint in 124 seconds. For this smaller setpoint change, the mixed air temperature returns to steady state faster than the cold air temperature. While the valve is opening to cool the air in the cold duct down to 14.5°C, the damper is already in movement to counteract this effect and keep the temperature in the mixed air duct at 21°C. From these observations it can be concluded that valve displacements interact with the control of mixed air temperature.

Previously, it was noted that the velocity pressure can indicate damper movements. At 45 seconds, the velocity pressure peaks, indicating a disturbance in the system. This disturbance can only be due to the damper opening in order to let more hot air mix with cold air. At 64 seconds, the fan controller kicks in to bring back the velocity pressure to the setpoint. Thereafter, the damper movements are small and produce only 0.001in.WG disturbances before the velocity pressure returns to its setpoint due to the controller action on fan speed. After 110 seconds it becomes extremely difficult to link velocity pressure fluctuations with mixed air temperature because noise also disturbs velocity pressure readings by 0.001in.WG and therefore cannot be disassociated from velocity pressure fluctuations due to damper movements (figure 6.3). Surely the damper control loop interacts with the control of velocity pressure; however, its effect is so small that it is difficult to quantify in these conditions.

The last loop to be considered is the heater control loop. It can be observed from figure 6.1d that valve setpoint changes have no major effect on the air temperature in the hot duct. Furthermore, once the damper is open to allow air flow in the hot duct, the temperature in that duct is not affected by the change in damper position. The ±0.1°C fluctuations can be neglected and velocity pressure changes are too small to conclude if interaction occurs or not between the fan and the heater loops.

From all these observations, it can be concluded that fan speed fluctuations, temperature in the hot duct and mixed air temperature drop cannot be the cause of the large cold air temperature undershoot encountered at first setpoint change. Therefore, the undershoot is not a result of interaction, but is due to the valve controller parameters which were set to reach a setpoint of 14.5°C, not 16.5°C. It can still reach the setpoint but the undershoot is more important.

Even if interaction were found to occur between certain loops, all control loops reached their setpoint within ±1% and they all have an acceptable response time and tolerable overshoots/undershoots. Running the system under full operation actually eliminated the oscillation present in the valve control loop response during individual tuning (figure 5.18 vs figure 6.1a). Therefore, this test tends to show that for a small system like the one under study, with the system operating conditions identical to the design conditions, the interaction between the loops is minimal. To have a better understanding of loop interaction, it will be necessary to introduce a large disturbance or parameter change in each loop one at a time. Furthermore, the mixed air temperature and the cold air temperature responses could benefit from fine-tuning of the valve and damper controllers.

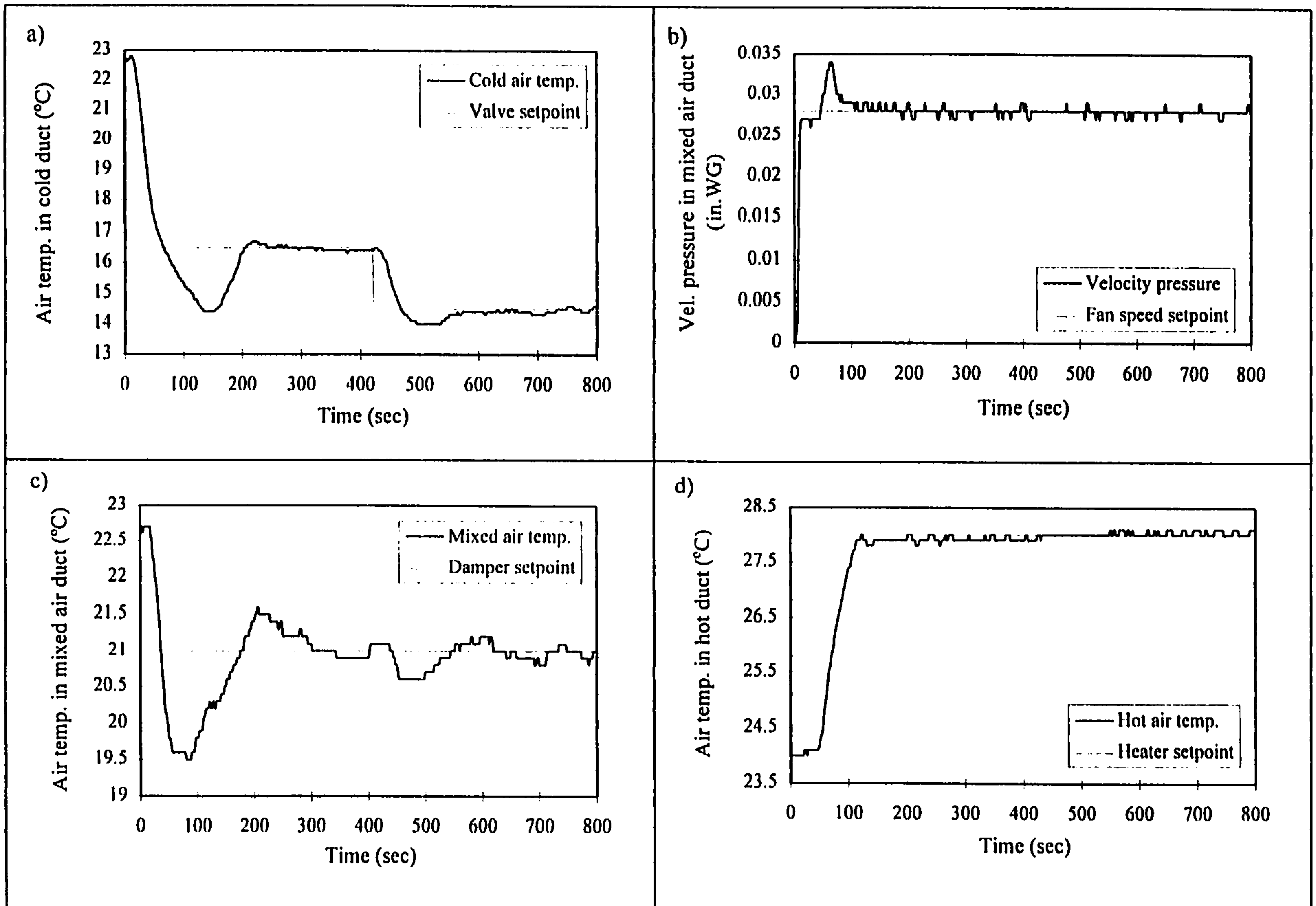

Figure 6.1 Base Case Results

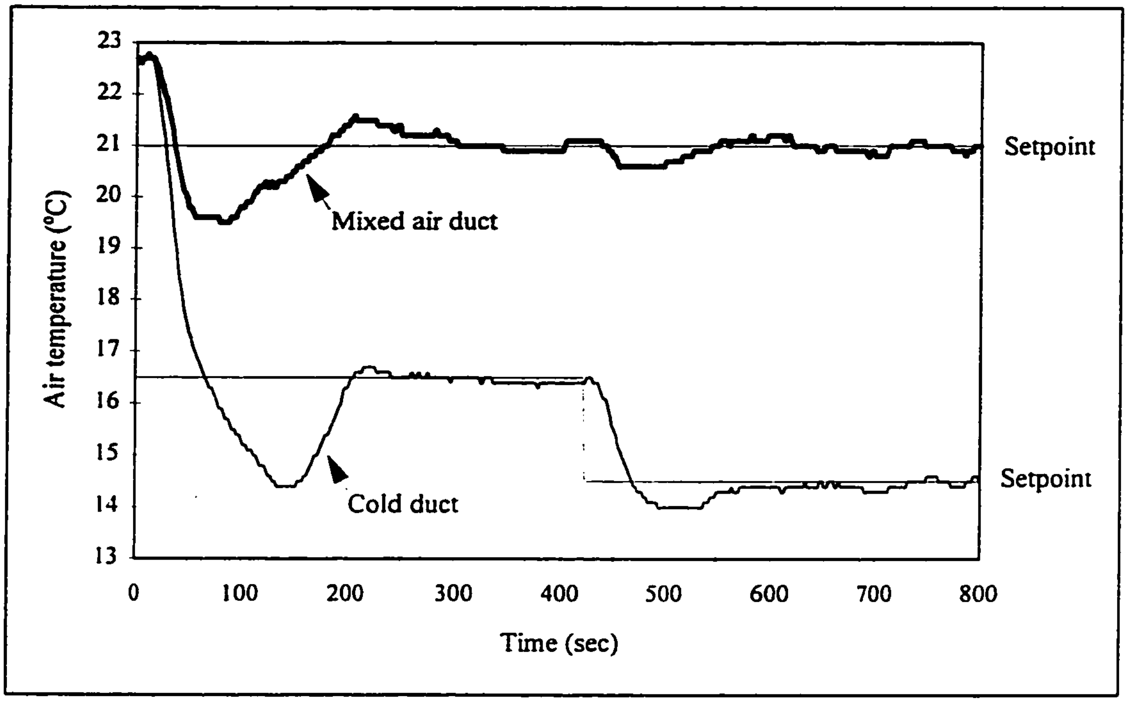

Figure 6.2 Cold Air Temperature and Mixed Air Temperature Responses

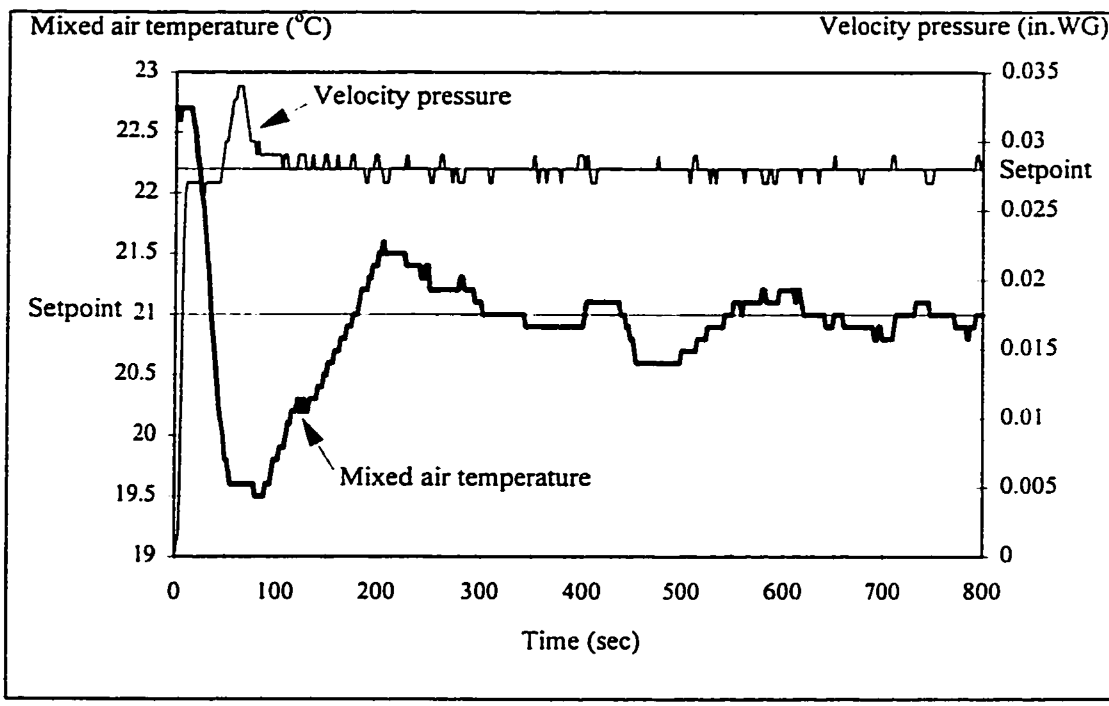

Figure 6.3 Velocity Pressure and Mixed Air Temperature Responses

6.2 Effect of Poorly Tuned Controllers on System Response

During the life of a building, changes such as building use or occupancy type can occur thereby affecting the system loads considerably. In such cases, the HVAC system is not properly tuned anymore and the system response will obviously be affected. Three tests were run to identify how a poorly tuned controller can affect the other control loops. These tests will therefore provide a better understanding of the interaction occurring between the loops. They consist of modifying the parameters of the fan, valve and damper controllers, one controller at a time, in such a way that the controller will provoke an oscillatory response in its loop. A poorly tuned controller can produce a sluggish or an oscillatory response; however, only the latter is considered because it is expected to produce the worst effect. The rest of the procedure is the same as the one described in the previous section. Note that the parameters of the heater controller were not modified because it was impossible to create an oscillatory response in the hot duct with P-control only.

6.2.1 Poorly Tuned Fan Speed Controller

Oscillation was introduced in the velocity pressure control loop by significantly reducing the proportional band of the fan speed controller (see table 6.1). The value was actually chosen as a possible error made at the tuning stage during the selection of the ultimate sensitivity (Ziegler and Nichols' tuning rules). The corresponding integral time, compared to the original integral time, requires only two more seconds per reset. This difference is insignificant and will not affect the response. Figure 6.4 shows the system response the fan speed controller is poorly tuned. The horizontal axis of the velocity pressure response, figure 6.4a, stops at 200 seconds instead of 800 in order to show its rapid oscillation period. In fact, the oscillation is so rapid that the other loops do not have the time to react to the increase or decrease in velocity pressure, and therefore are not affected by the poorly tuned controller. The air temperature in the mixed air duct and the hot duct do fluctuate more than in the base case results, but the fluctuations are always within 1% of their respective setpoint. Therefore, it is very difficult to link those fluctuations to interaction between the control loops.

Even if this type of fan control does not affect the other loops, it is not recommended because it causes premature wear and tear of the fan motor.

Table 6.1 Fan Speed Controller's Parameters

Tuning parameters	Values for proper tuning	Values for poor tuning
PB	274%	29%
T_i	1 sec.	3 sec.

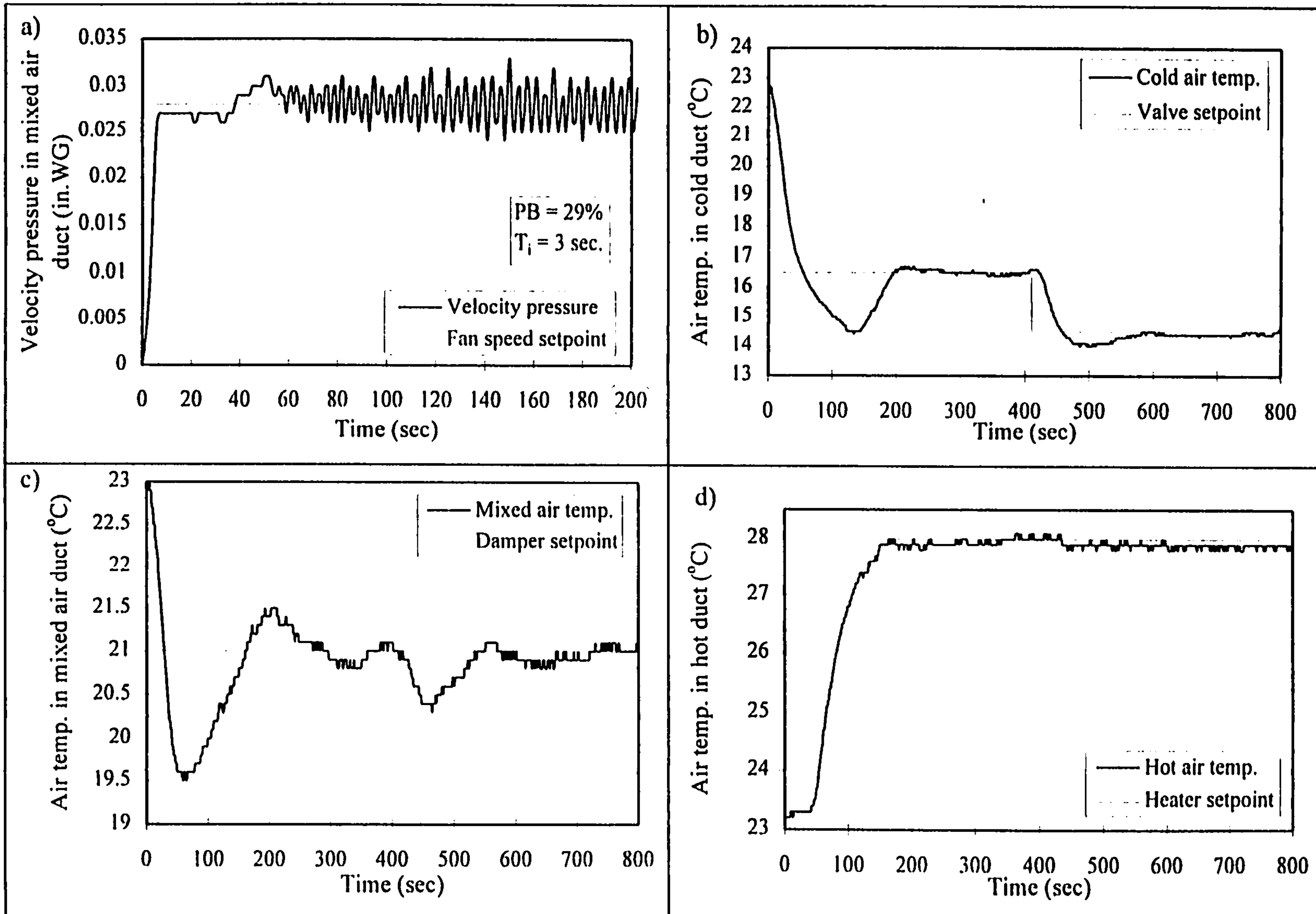

Figure 6.4 Effect of Poorly Tuned Fan Controller on System Response

6.2.2 Poorly Tuned Valve Controller

Oscillation was introduced in the cold air temperature control loop by considerably increasing the proportional gain of the cooling valve controller and keeping the integral time at a constant value (see table 6.2). Figure 6.5 shows the system response when the valve controller is poorly tuned. The oversensitive controller activates the valve continuously, causing the temperature in the cold duct to oscillate (figure 6.5a). Its effect on mixed air temperature is instantaneous; after the first undershoot, the temperature in the mixed air duct starts oscillating without any delay, following the same period (approximately 100 sec.) as the cold air temperature (figure 6.6). The amplitude is smaller though, about half the amplitude of the cold air temperature curve, showing that the damper tries to reduce the mixed air temperature oscillation. The displacements of the damper are slow and interrupted, therefore not significantly affecting velocity pressure (figure 6.7). The poorly tuned valve does not considerably disturb the temperature in the hot duct either (figure 6.5d). Therefore, introducing oscillation in the temperature of the cold duct has major repercussion on the mixed air temperature control, but no significant effect on velocity pressure or air temperature in the hot duct.

Table 6.2 Cooling Valve Controller's Parameters

Tuning parameters	Values for proper tuning	Values for poor tuning
K_p	36	248
T_i	0.45 min.	0.45 min.

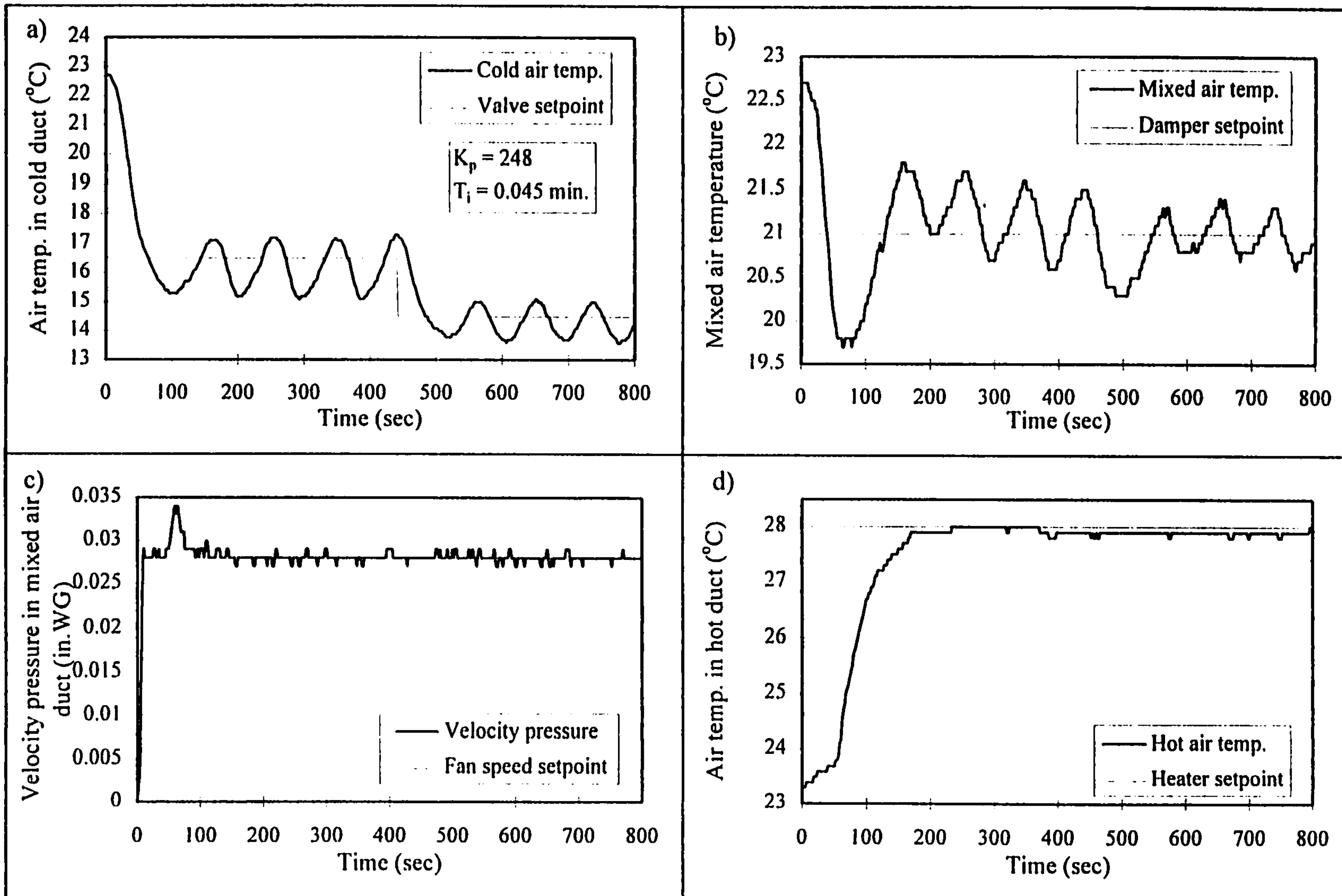

Figure 6.5 Effect of Poorly Tuned Valve Controller on System Response

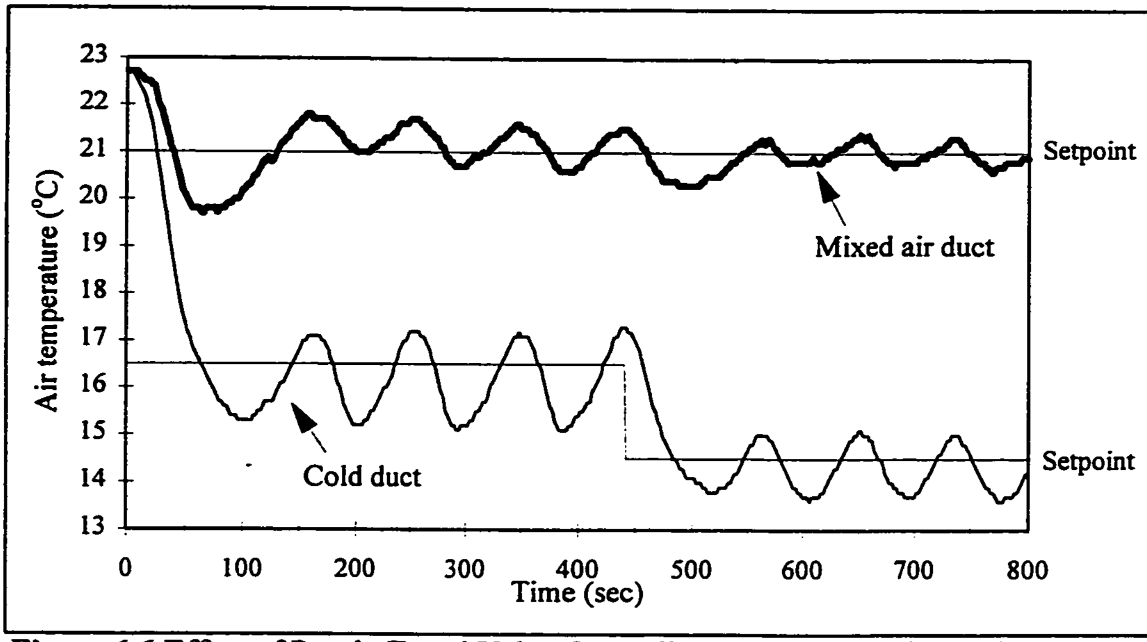

Figure 6.6 Effect of Poorly Tuned Valve Controller on Cold and Mixed Air Temperature

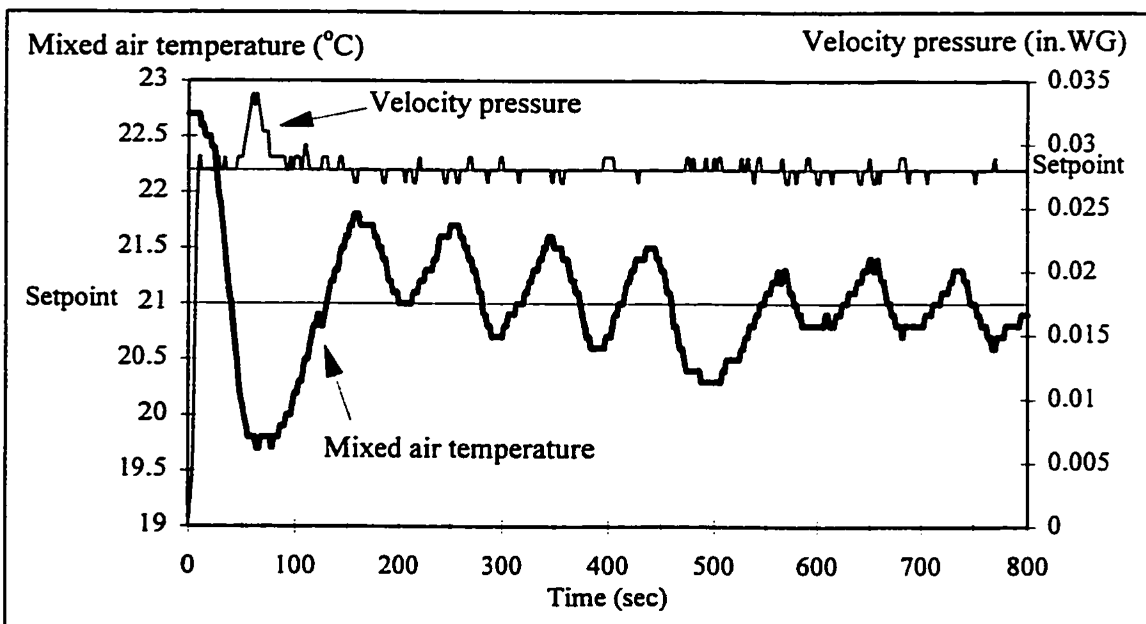

Figure 6.7 Effect of Mixed Air Temperature Oscillation on Velocity Pressure

6.2.3 Poorly Tuned Damper Controller

Increasing the damper controller proportional band to 150 (table 6.3) introduced a sustained oscillation in the damper control loop. Figure 6.8 shows how the system responded to the poorly tuned damper controller. The mixed air temperature oscillations are a consequence of the oversensitive damper moving too rapidly. The oscillation has a fast period (approximately 10 sec.) and a small amplitude so the temperature is kept within ±0.2°C of the setpoint. In terms of comfort, this might be very good, but it will result in premature wear and tear of the mechanical parts of the damper and the fan motor. Indeed, the fast movements of the damper has a direct impact on the velocity pressure control (figure 6.9). The air distribution system reacts very rapidly to changes in air flow caused by the damper in order to keep the velocity pressure at its setpoint. During the first valve setpoint change, the damper responds to the rapid drop in mixed air temperature and opens widely causing a higher velocity pressure peak than usual (figure 6.10). Actually, the damper opens before the mixed air temperature has reached setpoint, whereas in the base case, the damper waits for the temperature to be almost 1°C below setpoint before opening. The hot air temperature (figure 6.8d) tends to show, like in the response for poorly tuned fan speed controller, that the temperature is affected by oscillation of velocity pressure. However, the temperature oscillation is still very small and not worth considering. Finally, the cold air temperature is not affected by the poorly tuned damper controller (figure 6.8b).

Table 6.3 Damper Controller's Parameters

Tuning parameters	Values for proper tuning	Values for poor tuning
K_p	8.7	150
T_i	0.08 min.	0.08 min.

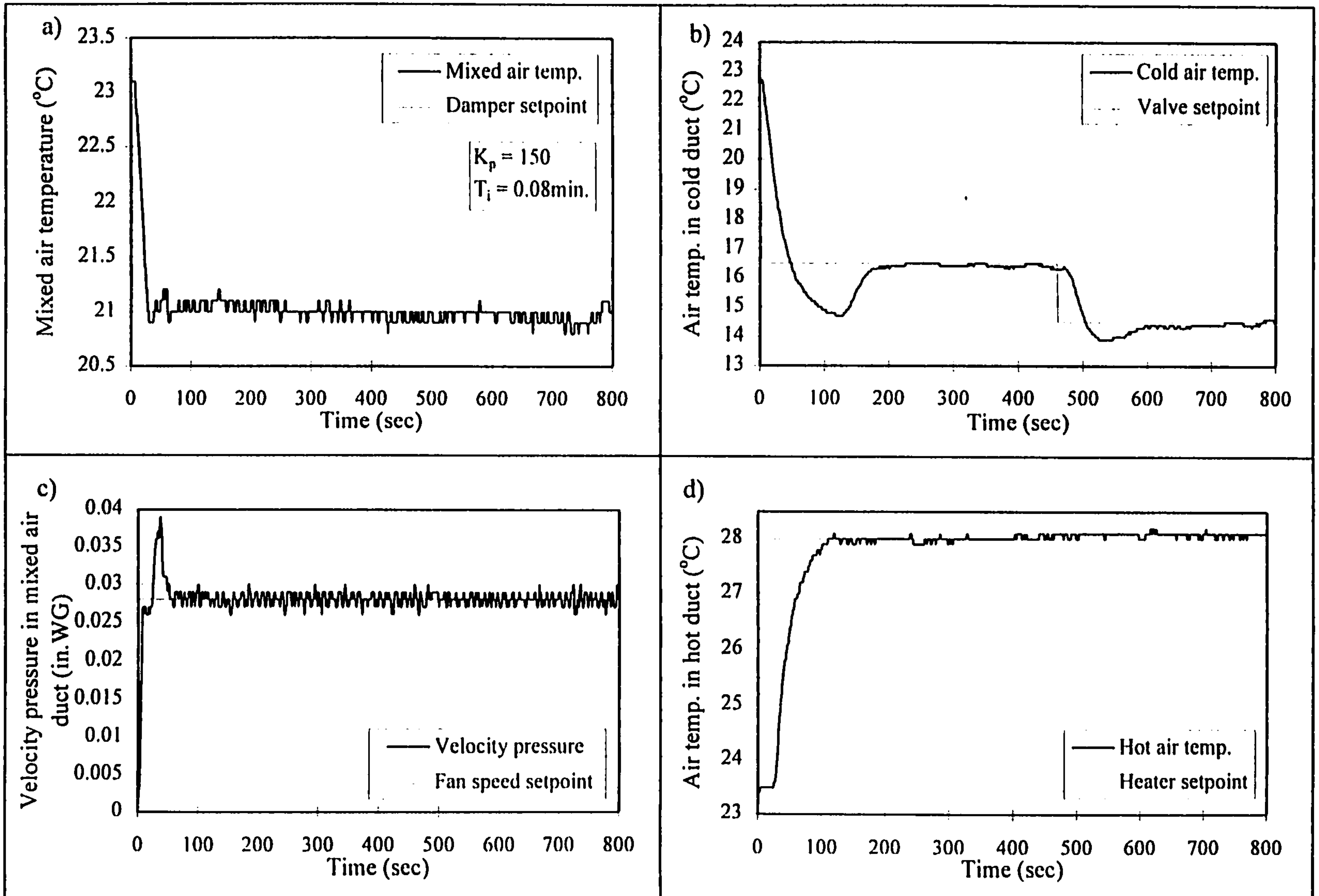

Figure 6.8 Effect of Poorly Tuned Damper Controller on System Response

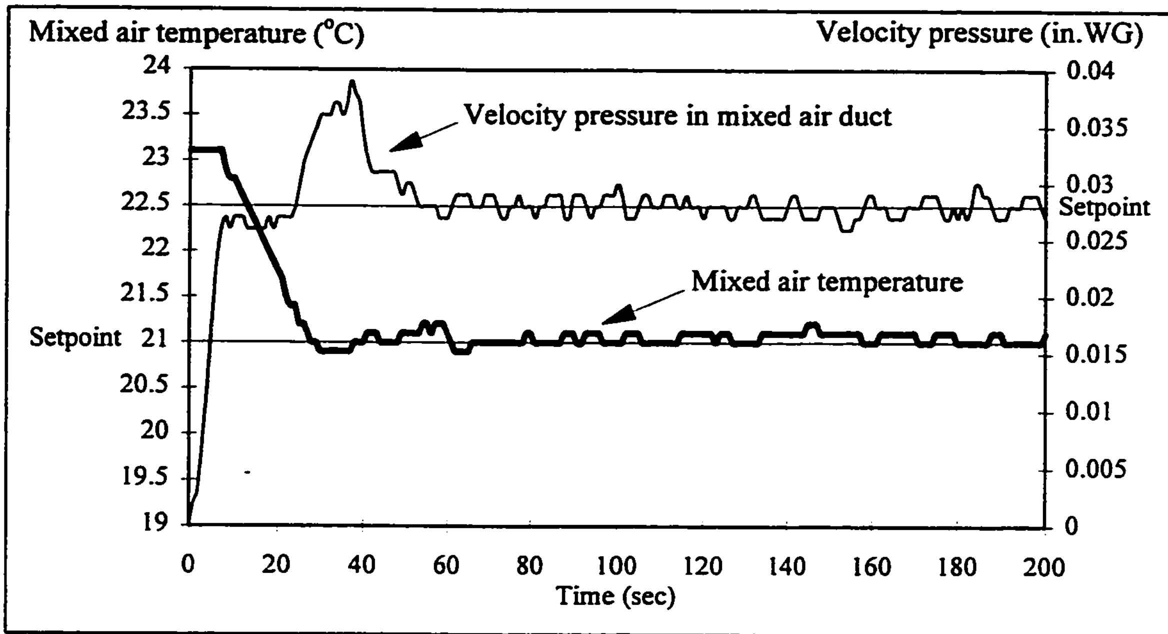

Figure 6.9 Effect of Damper Movement on Velocity Pressure

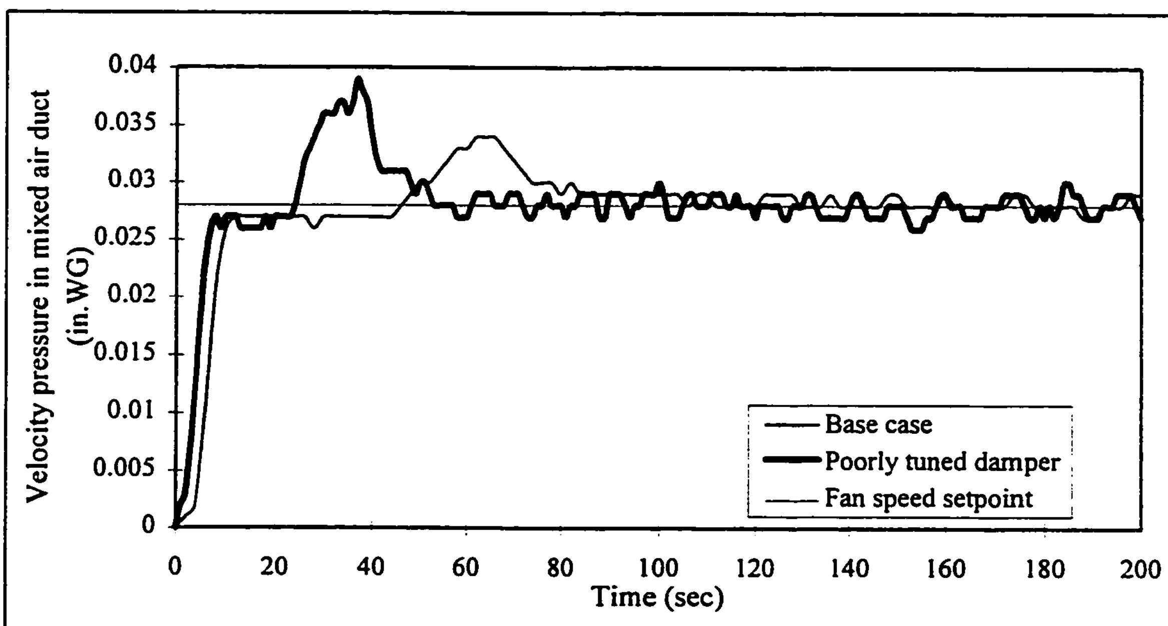

Figure 6.10 Effect of Poorly Tuned Damper Controller on Velocity Pressure

6.3 Fine-Tuning Method for Damper and Valve Controllers

Observing the results obtained with the HVAC system running under design conditions (figure 6.1), the damper and valve control loops are the two loops that would benefit the most from fine-tuning. For the damper and valve controllers, an optimizing set of curves was developed taking into consideration the interaction the damper has with fan speed and the interaction the valve has with the mixed air temperature (figures 6.11 and 6.12). The curves were obtained by varying only the proportional band of the controller in question. Proportional band was selected instead of integral time because it is the most error prone parameter during the tuning phase. Since it was already shown that the hot air temperature is not affected by the other loops, it was not considered in this analysis.

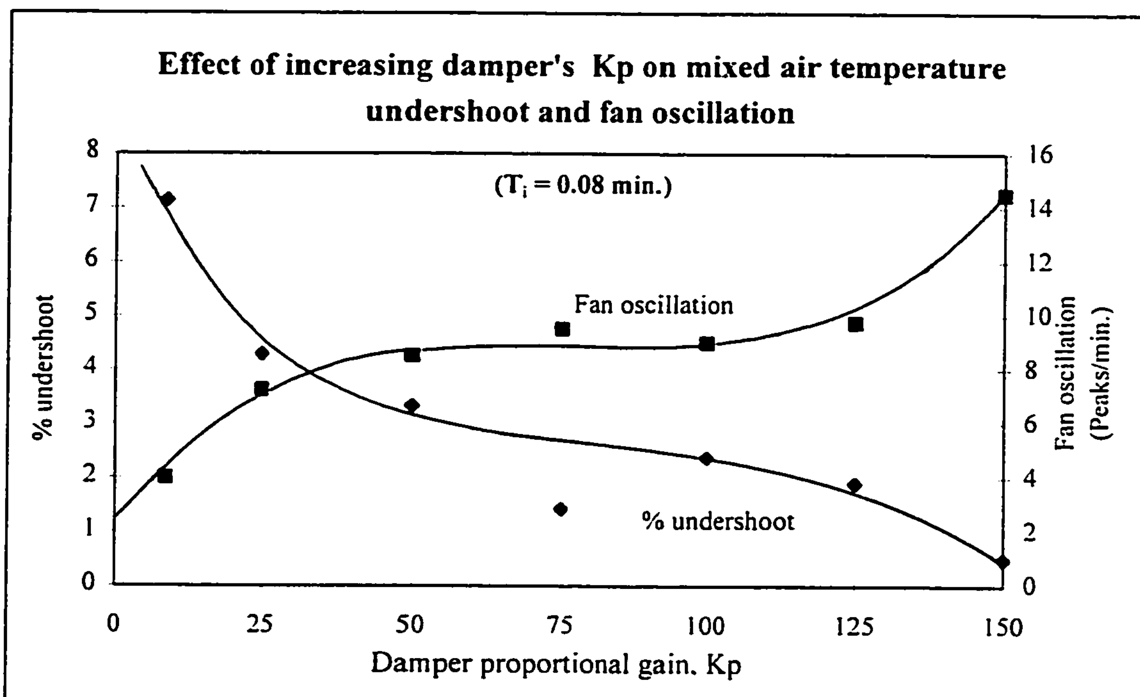

Figure 6.11 Fine-Tuning Curves for Damper Controller

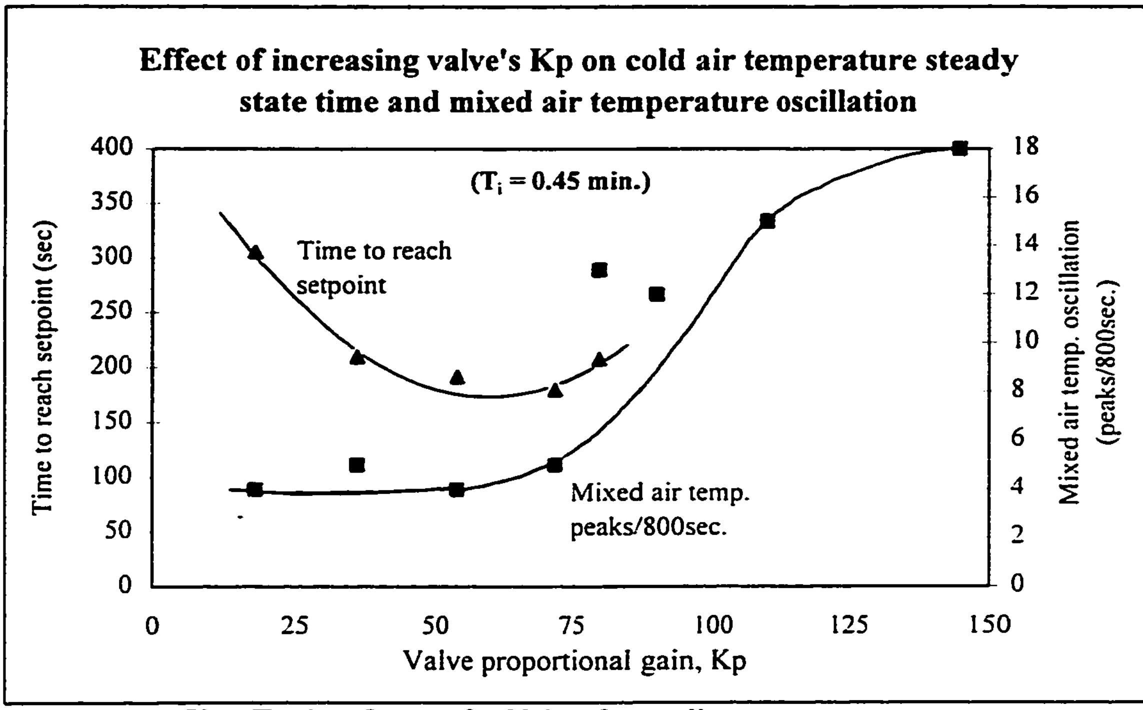

Figure 6.12 Fine-Tuning Curves for Valve Controller

6.3.1 Optimizing System Response Through Damper Control Loop

Before optimizing the system response by modifying the damper controller parameters, the interaction between damper control and fan speed control must be quantified. Several tests with different values of proportional gain for the damper controller have been conducted, noting for each test the mixed air temperature percent undershoot and the fan oscillation. Figure 6.11 shows the optimization curves for the damper controller resulting from these tests. Increasing K_p reduces the percent undershoot but increases the fan oscillation. Therefore, the optimum damper controller setting occurs when the mixed air temperature percent undershoot is minimum without considerably increasing velocity pressure oscillation. Figure 6.13 shows the results when K_p=20; this represents an undershoot of 5% and a reasonable velocity pressure oscillation. Comparing the mixed air temperature curve with the base case response (figure 6.14) shows that its first steady-state response time is reduced by 20% while its second steady-state response time is reduced by 55%. The proportional band adjustment also reduces the first undershoot by 2.5%. The cold air is not influenced as much. During the first valve setpoint change there is a small overshoot just before reaching steady state, but steady-state response time remains the same. This is not the situation for the steady state-response time of the second valve setpoint change which is reduced by 25% (figure 6.15). For velocity pressure, the difference between the two cases occurs at startup (figure 6.16). Both curves reach steady-state at the same time but velocity pressure in the optimized case varies between 0.028 and 0.031 in.WG before settling down to its setpoint, while in the base case, velocity pressure settles at 0.027in.WG for 32 seconds then peaks to 0.034 in.WG before returning to the setpoint. From these observations it can be concluded that increasing the damper controller proportional gain from 8.7 to 20 improves mixed air temperature response and still provides good loop response for cold air temperature and velocity pressure.

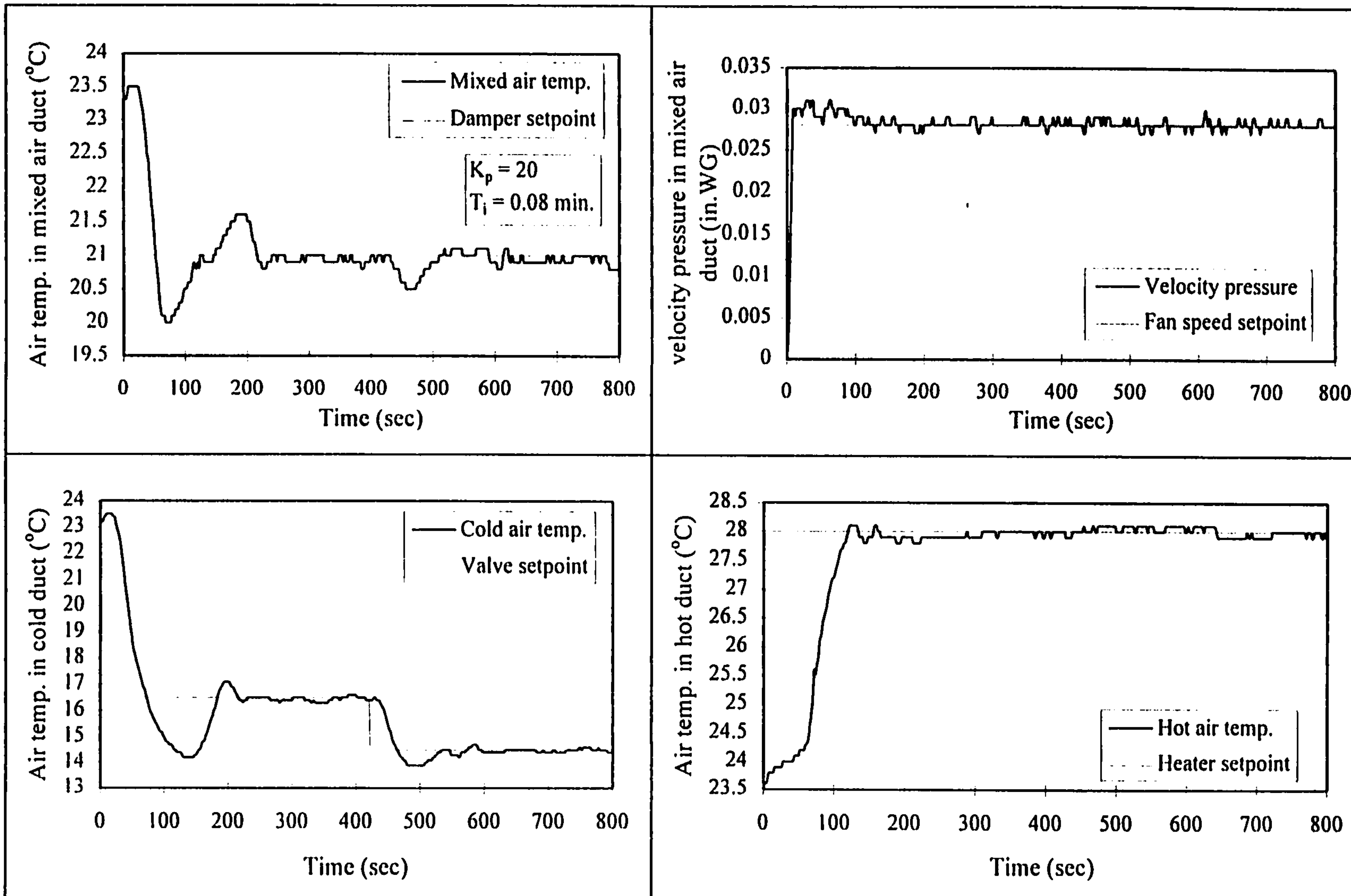

Figure 6.13 System Response to Damper Control Loop Optimization

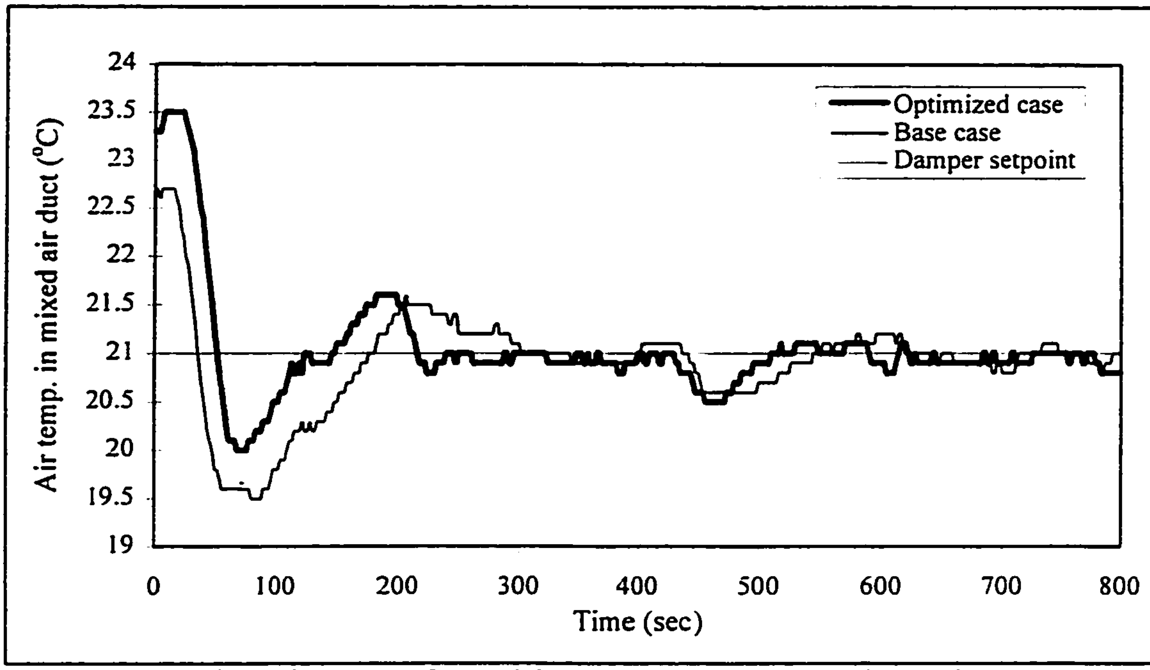

Figure 6.14 Effect of Damper Control Loop Optimization on Mixed Air Temperature

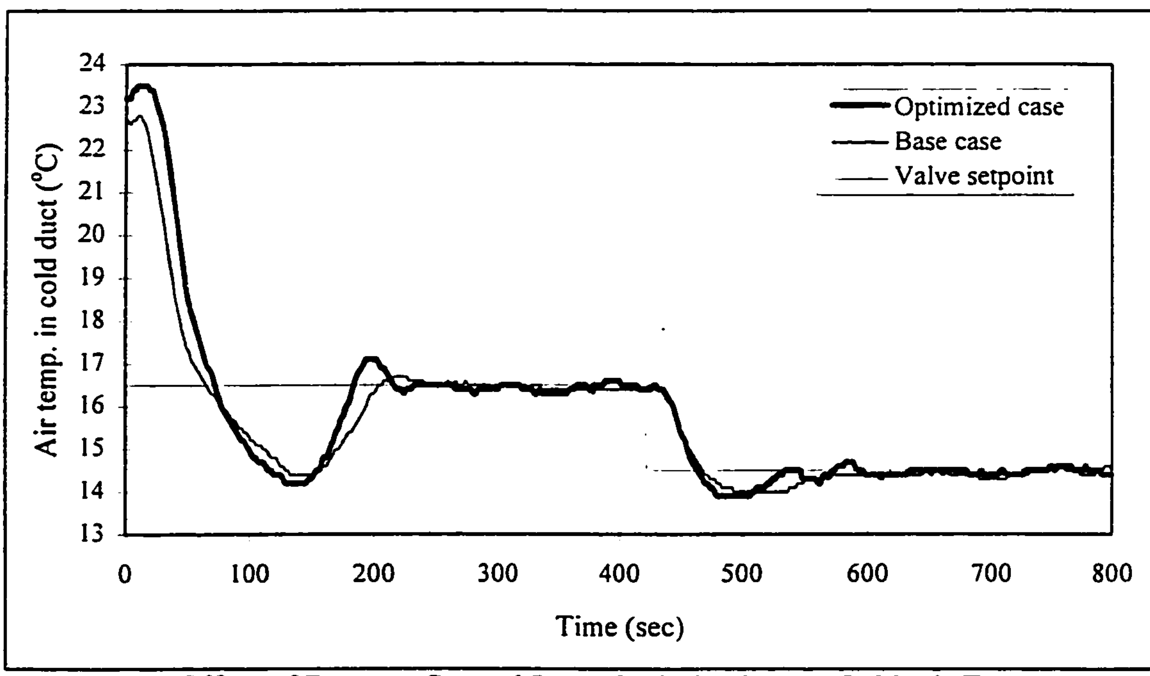

Figure 6.15 Effect of Damper Control Loop Optimization on Cold Air Temperature

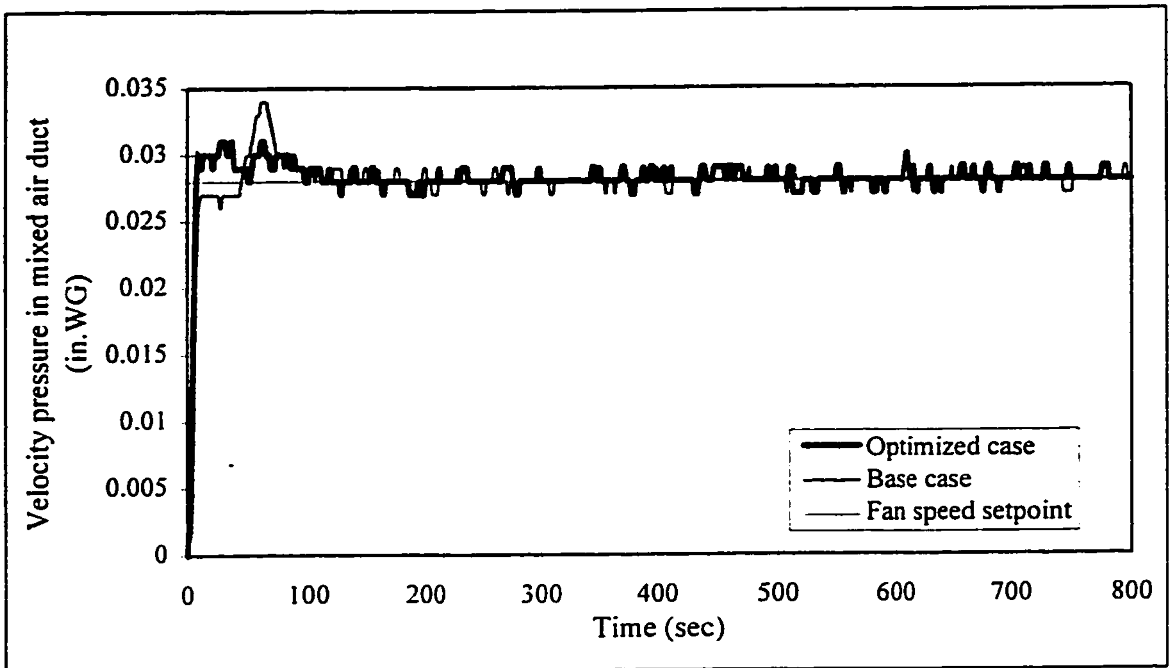

Figure 6.16 Effect of Damper Control Loop Optimization on Velocity Pressure

6.3.2 Optimizing System Response Through Valve Control Loop

Before optimizing the system response by modifying the valve controller parameters, the interaction between the valve control loop and the damper control loop must be quantified. Several tests with different values of proportional gain for the valve controller also have been conducted, noting for each test the cold air temperature steady state response time (for the second valve setpoint change) and the mixed air temperature oscillation. Figure 6.12 shows the optimization curves for the valve controller resulting from these tests. Increasing K_p reduces the cold air temperature steady state response time until a sustained oscillation is introduced in the control loop; it also increases the mixed air temperature oscillations. Therefore, the optimum valve controller setting occurs when the cold air temperature steady state response time is minimum without significantly increasing the mixed air temperature oscillation. A value of K_p=50 was selected to improve the system response. Figure 6.17 shows the results obtained. The proportional band adjustment introduces a small oscillation in the cold air temperature response which remains within 2% of the setpoint, and will therefore have little effect on comfort or valve mechanical parts. Comparing the cold air temperature curve with the base case response (figure 6.18) shows that the two steady state response times remain unchanged and that the undershoots vary by less than 2%. The mixed air temperature response however has improved compared to the base case. The steady-state times are reduced by 28% and 23% for each valve setpoint change respectively while the undershoots vary by less than 2%. Finally, the system still provides a good velocity pressure response because the fluctuations have not increased substantially. Therefore, from these observations it can be concluded that increasing the proportional band of the valve controller from 36 to 50, has little effect on the cold air temperature and velocity pressure control loop but improves the mixed air temperature control loop.

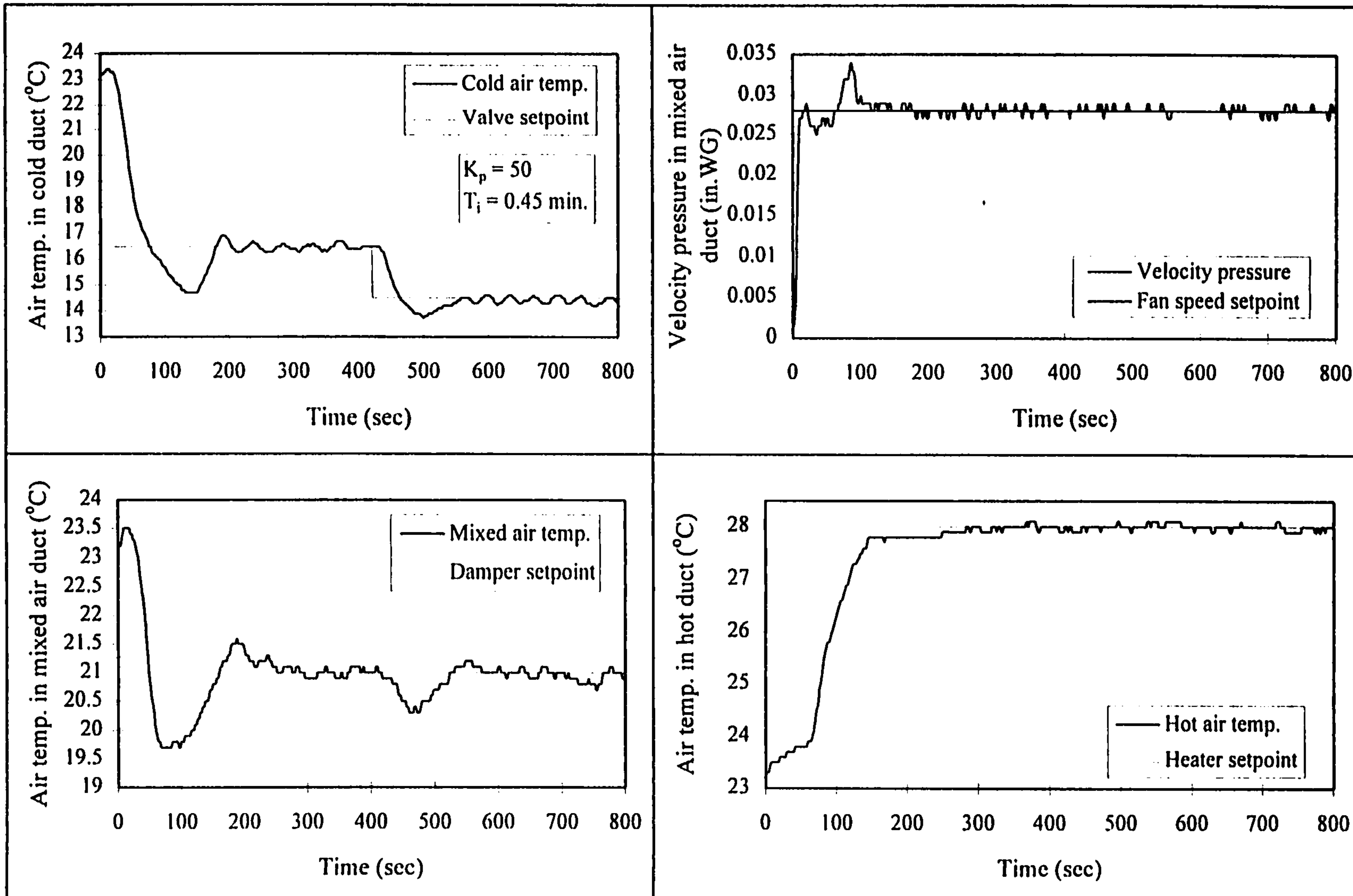

Figure 6.17 System Response to Valve Control Loop Optimization

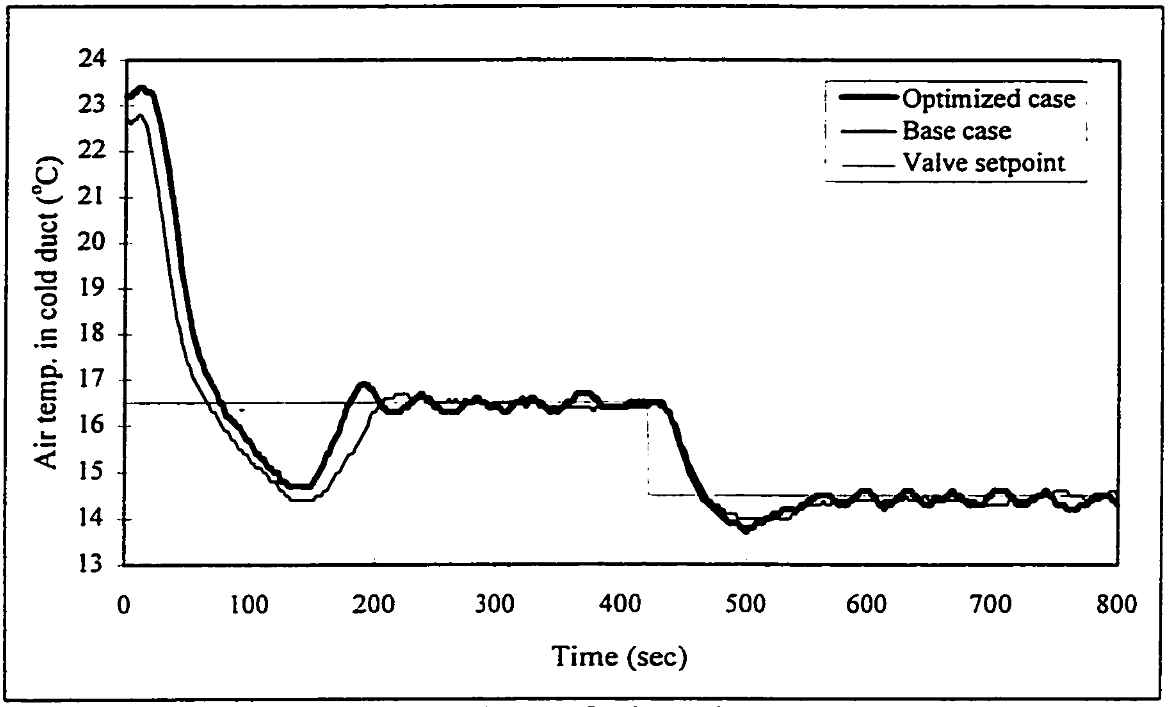

Figure 6.18 Effect of Valve Control Loop Optimization on Cold Air Temperature

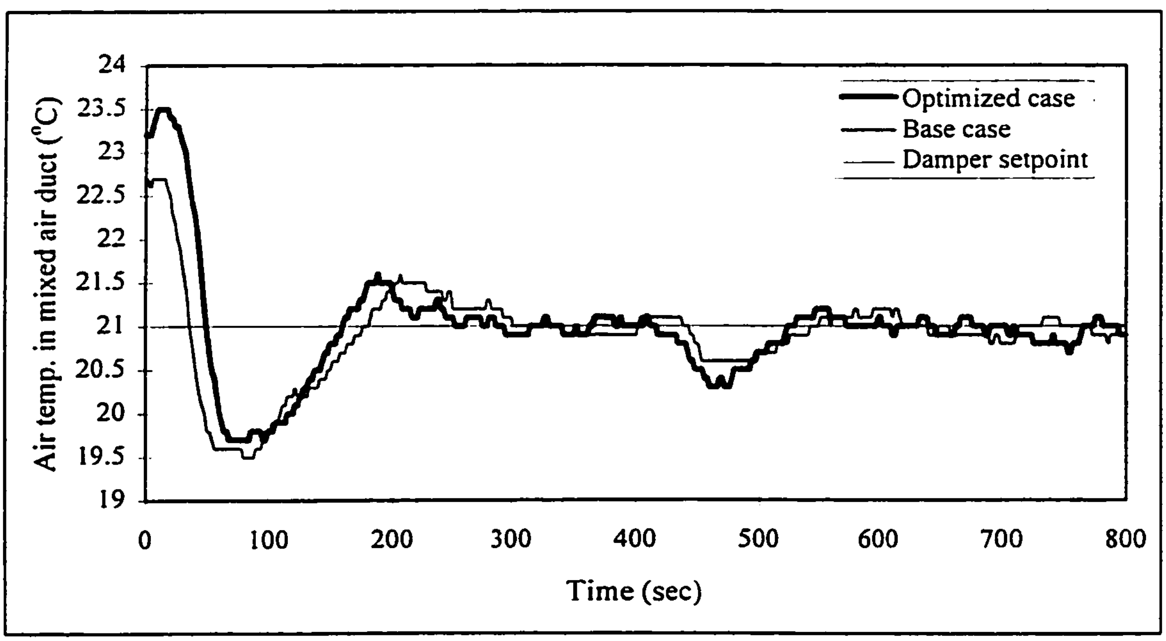

Figure 6.19 Effect of Valve Control Loop Optimization on Mixed Air Temperature

6.4 Summary

Operating the laboratory HVAC system under design conditions showed that interaction is minimized for a well designed system. To better understand interaction between control loops, a sustained oscillation was introduced in each control loop, one loop at a time. The heater controller was disregarded because it was impossible to induce oscillation with its proportional-only control. For the dual duct system under study, severe interaction occurred between the valve and damper control loop. An oscillating cold air temperature caused the mixed air temperature to oscillate as well following the same period. It was also discovered that damper movements interact with fan speed control. The oversensitive damper significantly increased the velocity pressure fluctuations and therefore, fan speed variations. It is interesting to note that the oscillation induced in the mixed air temperature response by a poorly tuned valve controller significantly differs from the oscillation induced by the poorly tuned damper controller. A poorly tuned valve controller introduces an oscillation which has a period 10 times slower and an amplitude at worst twice the one created by a poorly tuned damper controller. Finally, an oscillating fan speed has little effect on the thermal loops because it operates in a much faster time scale. The oscillations are so rapid that the thermal loops do not have the time to react to it.

Two sets of fine-tuning curves were defined to optimize the system response taking into account interaction effects. One set of curves was used to optimize the system response by modifying the damper controller proportional band, while the other set of curves was used to optimize the system response by modifying the valve controller proportional band. In both cases, the damper control loop benefited the most from the optimization. However, the best system response is obtained when the damper control loop is fine-tuned because it also improves cold air temperature response. In the case of fine-tuned valve controller, the cold air temperature response is not improved as expected. The steady-state response times remain unchanged and a small oscillation within 2% of the setpoint occured.

7.0 CONCLUSION AND RECOMMENDATIONS

PI-control of a laboratory size dual duct system was analyzed. The air handling unit included four decentralized control loops which allowed the comparison between three different tuning methods and the evaluation of interaction among the loops. Two sets of optimizing curves were then developed considering interaction. However, the first step of this work was to operate the system in open-loop in order to determine the system dynamics.

7.1 System Performance in Open-Loop

Studying the system performance under open-loop conditions before tuning the controllers (Chapter 4), not only helps understand the system dynamics but also provides the system operating range, and may indicate if a component of the system has been improperly sized. While it was rather easy to perform open-loop tests for a small scale system because all the equipment is located in the same room, the tests would be more time consuming for a real HVAC system due to the distance separating the different components to be manually controlled such as valves and dampers. For the small scale dual duct system, transient and steady-state analyses performed on the air and water distribution systems showed that they both are stable under open-loop conditions. However, the system has a much faster response to fan speed step change than to step change in valve position. The tests indicated that the air flow can vary from 220 to 900 cfm and cold air temperature can vary from 13°C to 17.5°C, depending on the load. If one of the system components had been poorly selected, it would have been noticed by finding operating ranges too narrow or too wide for proper control.

7.2 Comparison Between Tuning Methods

Applying different tuning rules to four control loops instead of just one allows to verify for which type of loops each rule is more suitable. In Chapter 5, three tuning rules were compared. Unfortunately, due to system limitations, one or two tuning rules could not be applied on two of the controllers. Still, it seems Ziegler and Nichols' tuning method would be more appropriate for control loops operating in slow time scales, while the simplified IMC-PID tuning method would be more appropriate for control loops operating in fast time scales. Bekker et al. tuning rules are too conservative for the small scale system and provide a slow response for all control loops. All three tuning methods were easy to apply, however Ziegler and Nichols' ultimate sensitivity method is time consuming because of the several tests it requires to determine the tuning parameters. It is also more prone to errors; one may easily assume ultimate sensitivity was reached when in fact it has not, which may considerably affect the response. The other two tuning methods required only one open-loop test and the determination of the maximum slope was not found to be critical for the simplified IMC-PID method. A more accurate method to calculate maximum slope should be used for Bekker et al. tuning rules to determine if an approximate slope causes the sluggishness of the responses.

Closed-loop tests also showed that the control loops operate in different time scales. As it was found in open-loop tests, the fan speed control loop has a faster time scale than the valve or damper control loops. However, the difference between the time scales was reduced by a factor of two. Furthermore, the valve and the damper operate in very similar time scales while the heater operates in a faster time scale than the other two thermal loops.

Tuning controllers properly does not only involve finding the proper gains, it also involves the careful selection of the design conditions which should represent the operating conditions occurring most often. It was shown, through tests on the fan speed control loop, that operating at conditions different than the design conditions will affect loop response. In this experiment, it was found that tuning the controller at medium-load conditions would provide the best response at any operating conditions. Tuning the controller at low-load conditions induced oscillation in the response for other part low-load conditions. This contradicts one of Nesler and Stoecker (15) suggestions which stated that:

> "The first step in controller tuning is to identify the operating region with highest process gain. In most systems, this occurs at low-load conditions. All controller tuning should be made within the region of high gain so that system stability is maintained throughout the operating range of the process."

7.3 Effect of Interaction on System Control

Interaction among control loops occurs in systems for which the controllers have been tuned manually or with autotuners. In Chapter 6, it was shown that interaction is minimized when the small scale HVAC system operates under design conditions. To obtain a more complete picture of interaction between control loops, each loop response had to be disturbed. This was performed by introducing oscillation in each loop response, one loop at a time. There was one exception, the heater control loop, because it was impossible to introduce an oscillation with its proportional-only control. For the dual duct system under study, the conclusions which can be stated are the following:

1. Severe interaction occurs between the valve and damper control loop. An oscillating cold air temperature causes the mixed air temperature to oscillate as well. However, the opposite is not true.
2. Damper movements interact with fan speed control. The numerous damper movements significantly increases the number of fan speed changes.
3. An oscillating fan speed has little effect on the thermal loops because it operates in a much faster time scale.

In order to optimize system response, two sets of fine-tuning curves were defined taking into account interaction between the damper and the fan and between the valve and the damper. One set of curves optimizes the system response by modifying the damper controller proportional band, while the other set of curves optimizes the system response by modifying the valve controller proportional band. The benefits of optimizing the control loop responses were small for the small scale system because the system was already operating properly and with minimized interaction. Still, sets of curves such as the ones developed in Chapter 6 have the advantage of improving some loop responses without rendering other loops unstable. Therefore the curves narrow the range of K_p values for which the system can be operated and still provide good response.

7.4 Recommendations

For the small scale dual duct system with mixed air temperature controlled by a damper and equiped with an electrical heater:

1. If oscillation in mixed air temperature response is found and it has a fast period, the damper controller should be retuned.
2. If oscillation in mixed air temperature response is found and it has slow period, the valve controller should be retuned.
3. If velocity pressure oscillates, it may be caused by a poorly tuned fan speed controller or a poorly tuned damper. In the case of the former, oscillation will subsist when heating and cooling are switched off.

Generalized case of manual tuning or tuning with autotuners:

1. Loop response optimization should not be performed considering only the loop in question. It is important to know with which other loop it may interact, and the controller must be fine-tuned following curves such as the ones defined in Chapter 6.
2. Maintenance is a very important factor for good control in HVAC systems. Readings of sensors and controllers must be checked regularly and may require periodical recalibration. Also, current and voltage fluctuations may have a negative effect on controllers' readings. Although these problems were not assessed during this research,

tests had to be repeated after noticing misreadings from the damper controller when compared to a thermometer placed at the sensing station of the mixed air duct.

7.5 Future Work

The following ideas could be pursued:

1. Provide manual control of the heater so all three tuning methods could be tested on the four control loops.
2. Tune all the controllers with Ziegler and Nichols' tuning method, then with the simplified IMC-PID tuning method and compare system response with the one obtained in section 6.1. It would then be possible to discuss if it is worthwhile to use several tuning methods on one HVAC system.
3. Obtain the maximum slope from open-loop response more accurately, and observe if Bekker et al. tuning method provides better response. If it is the case, then this method would be very sensitive to the determination of the maximum slope.
4. Add humidity control and observe if the system response is still adequate.
5. Study more complex control schemes such as master/slave controllers and sub-control loops.

APPENDIX A: SYSTEM COMPONENTS

A.1 Fan Speed Control Loop Equipment

CENTRIFUGAL	FAN
Company:	Produits 3B & S Inc.
Model:	122 Vent-Pak
Blade type:	Backwardly inclined
Operating pressure:	1" SP
Average flow:	860 cfm
rpm range:	360 to 1600 rpm
Fan power:	0.21 BHP
Drive:	Variable speed DC drive

VARIABLE	SPEED DC CONTROL
Company:	DART Controls Inc.
Model:	IM-250G-0592
Signal Input:	Current signal from the fan speed controller
Signal Output:	Voltage signal sent to the fan motor

FAN SPEED	CONTROLLER
Company:	Eurotherm Corporation
Model:	Model 847
Input signal:	Voltage signal coming from transducer Range: -10 to 50mV
Output signal:	Current signal going to speed control device Range: 4 to 20mA

A.2 Valve Control Loop Equipment

COOLING COIL	
Company:	Enersol
Dimension:	12" x 12"
Capacity:	12,000 BH (3517 W)
Flow direction:	Counter flow

VALVE	
Company:	Honeywell
Model:	V5011F1048
Service:	Two-way
Pipe size:	½ in.

VALVE ACTUATOR	
Company:	Honeywell
Model:	ML784
Type of actuator:	Cervo-system type
Type of stem lift:	Linear travel

VALVE CONTROLLER	
Company:	Honeywell
Model:	UDC 2000 Mini-Pro
Input signal:	Thermocouple T type extension grade Voltage range: -2.225 to 12.572mV
Output signal:	Current signal going to the valve actuator Range: 4 to 20mA

A.3 Heater Control Loop Equipment

HEATING COIL	
Company:	WATTCO ELECTRIC
Dimension:	12" x 12"
Capacity:	4,800 Watt (4x 1200W)
Voltage input:	120 Volt

HEATER CONTROLLER	
Company:	Honeywell
Model:	UDC 2000 Mini-Pro
Input signal:	Thermocouple T type extension grade Voltage range:-2.225 to 12.572mV
Output signal:	Current signal going to heater control block Range: 4 to 20mA

HEATER CONTROL DEVICE	
Company:	HALMAR ROBICON
Model:	Series 100Z-C
Current rating:	15 Amperes resistive
Input signal:	4 to 20 mA
Output signal:	Volts
Status indicator:	Red flashing LED when unit is on

A.4 Damper Control Loop Equipment

DAMPERS	
Position:	Opposed operation
Dimension:	12" x 12"
Leakage:	Not defined

DAMPER ACTUATOR	
Company:	Honeywell
Model:	M984 versadrive
Actuator type:	Cervo-system type

DAMPER CONTROLLER	
Company:	Honeywell
Model:	UDC 2000 Mini-Pro
Input signal:	Thermocouple T type extension grade Voltage range: -2.225 to 12.572mV
Output signal:	Current signal going to the damper motor Range: 4 to 20mA

APPENDIX B: CONTROLLER CONFIGURATION RECORD SHEETS

EUROTHERM, MODEL 842 (34)
(Air flow indicator)

Mnemonic	Parameters		

ALARM CHANNEL 1

H AL 1	High alarm setpoint	50.0	
L AL 1	Low alarm setpoint	None	
SP 1	Deviation setpoint	None	
d AL 1	Deviation band	None	
HyS 1	Hysteresis setting	None	
Hi r1	High alarm relay action	LAtch	
Lo r1	Low alarm relay action	None	
d r1	Deviation alarm relay action	None	

ALARM CHANNEL 2

H AL 2	High alarm setpoint	None	
L AL 2	Low alarm setpoint	-1.0	
SP 2	Deviation setpoint	None	
d AL 2	Deviation band	None	
HyS 2	Hysteresis setting	None	
Hi r2	High alarm relay action	None	
Lo r2	Low alarm relay action	LAtch	
d r2	Deviation alarm relay action	None	

INPUT PARAMETERS

InPUt	Input type	1 dc	
dP	Decimal point position	0.0	
unitS	Display units	None	
OFSEt	Calibration offset	N/A	
C-F	Degree C or F selection	N/A	
CJC	Cold junction compensation	N/A	
Filtr	Input filter constant	0.0	

VOLTAGE INPUTS

Proc	Type of operation selection	nlin	
Vin 0-9	Voltage inputs	See table B1	
Eout 0-9	Engineering units equivalents to voltage inputs	See table B1	
dSP H	Maximum display value	2832	
dSP L	Minimum display value	-2.0	

Table B1 shows the values entered during configuration so that air flow in cfm could be read directly from the display. Conversion of the voltage signal into cfm is performed through a 5th order polynomial curve fit linearization using ten voltage input points and the corresponding output cfm values (34). Each voltage input represents a specific pressure which is related to air flow using the following equations:

$$V = 4005 \times \sqrt{p_v} \tag{B1}$$

$$Q = V \times A \tag{B2}$$

where V = air velocity, ft/min
p_v = velocity pressure, in. WG
Q = air flow, cfm
A = duct cross-sectional area, ft²

Table B1 Nonlinear Configuration of the Eurotherm Model 842 Indicator

Applied pressure in. WG	Voltage input Volts	Display reading cfm
0.00	$V_{in0} = 0.00$	$E_{out0} = 0.0$
0.01	$V_{in1} = 0.01$	$E_{out1} = 400.5$
0.02	$V_{in2} = 0.02$	$E_{out2} = 566.4$
0.03	$V_{in3} = 0.03$	$E_{out3} = 693.7$
0.04	$V_{in4} = 0.04$	$E_{out4} = 801.0$
0.05	$V_{in5} = 0.05$	$E_{out5} = 895.5$
0.06	$V_{in6} = 0.06$	$E_{out6} = 981.0$
0.07	$V_{in7} = 0.07$	$E_{out7} = 1059.6$
0.08	$V_{in8} = 0.08$	$E_{out8} = 1132.8$
0.09	$V_{in9} = 0.09$	$E_{out9} = 1201.5$

EUROTHERM, MODEL 847 (33)

Fan speed controller

Mnemonic	Parameters			
ALARM PARAMETERS				
Hi Al	High alarm setpoint	200.0		
Lo Al	Low alarm setpoint	1.0		
d Al	Deviation alarm setpoint	50.0		
H AO	High alarm output	nLat		
L AO	Low alarm output	nLat		
d AO	Deviation alarm output	off		
CONTROL PARAMETERS				
ProP	Proportional band	*		
Int.t	Integral time constant	*		
dEr.t	Derivative time constant	*		
rEL.C	Relative cool gain	N/A		
H c.t	Heat cycle time	N/A		
C c.t	Cool cycle time	N/A		
H cb	High cutback point	N/A		
L cb	Low cutback point	N/A		
SETPOINT LIMITS				
SP L	Setpoint low limit	0.0		
SP H	Setpoint high limit	200.0		
OUTPUT POWER LIMITS				
H PL	Maximum power limit	100.0		
SnbP	Sensor break power level	0.0		
MEASURED VALUE ATTRIBUTES				
OFSt	Calibration offset	0.0		
C F	degree C or F selection	N/A		
INPUT SENSOR SELECTION				
Sn	Sensor selection	lin		
COMMUNICATIONS CONFIGURATION				
Addr	Instrument address	0.0		
bAud	Baud rate	9600		
GENERAL CONFIGURATION				
idno	Identification number	1		
Ctrl	Control type	PID		
SPrr	Setpoint ramping speed	N/A		
OP 1	Output 1 configuration	4-20 mA		
OP 2	Output 2 configuration	Off		
A H	Auto/manual enable	HAnd		
CJC	CJC reference selection	N/A		
Pb d	Proportional band display	Pct		
PH-L	Prop. band scale factor	100**		
t SU	Tune on startup	no		
Cb O	Cutback operation	Auto		
LINEAR PROCESS INPUTS				
Act	Control action	reverse		
Hi L	High sensor break point	200.0		
Lo L	Low sensor break point	-10.0		
Fil	Input filter constant	0.1		

* Will vary according to tuning rules

** P_v reading when the output is at 20mA => 0.1 in.WG

HONEYWELL, UDC 2000 MINI-PRO (32)

		Controller		
		Damper	Valve	Heater
Mnemonic	Parameters			
TUNING				
GAIN	Proportional gain	*	*	*
RATE T	Rate time	*	*	*
I MIN	Integral time	*	*	*
MANRST	Manual reset	N/A	N/A	N/A
CYC TI	Deviation alarm relay action	N/A	N/A	N/A
LOCK	Lock access to parameters	CONF	CONF	CONF
ALGORITHM				
CTRALG	Control algorithm selection	PIDA	PIDA	PIDA
OUTALG	Output type	CUR	CUR	CUR
SETPOINT RAMP				
SPRAMP	Setpoint ramp	DISABL	DISABL	DISABL
TI MIN	Setpoint ramp time	0	0	0
FINLSP	Final setpoint value	0	0	0
SP PROG	Setpoint ramp programming	DISABL	DISABL	DISABL
AUTOTUNE				
AT ENB	Autotune function	DISABL	DISABL	DISABL
OUTSTP	Output step size	N/A	N/A	N/A
AT ERR	Error prompts	None	None	None
INPUT 1				
DECMAL	Decimal point location	888.8	888.8	888.8
UNITS	Temperature units	C	C	C
IN1TYP	Input 1 actuation type	T L	T L	T L
IN1 HI	Input 1 high range value	260	260	260
IN1 LO	Input 1 low range value	-62.2	-62.2	-62.2
BIAS 1	BIAS on input 1	-0.1	0.3	0.0
FILTR1	Input filter constant	0.0	0.0	0.0
EMISS	Emissivity for radiamatic inputs	N/A	N/A	N/A
BRNOUT	Burnout protection	None	Up	Up
FREQ	Power line frequency	60	60	60
DISPLY	Display configuration	PR N	N/A	PR N
CONTROL PARAMETERS				
SP SRC	Setpoint source	1LOC	1LOC	1LOC
PWR UP	Setpoint at power up	ALSP	ALSP	ALSP
SP HI	Setpoint high limit	35	35	100
SP LO	Setpoint low limit	7	7	13
ACTION	Control output direction	Reverse	Direct	Reverse
OUT HI	High output limit	105%	105%	100%
OUT LO	Low output limit	-5%	-5%	0%
FAILSF	Failsafe output value	0	0	0
PB OR GN	Proportional band units	GN	GN	GN
MIN RPM	Reset units	Min	Min	Min

*Will vary according to the tuning rules

APPENDIX C: CALIBRATION INSTRUMENTS

ICE BATH	
Company:	Omega
Model:	TRCIII ICE POINT reference cell
Accuracy:	0°C ±0.1°C
Use:	INPUT calibration of the two Honeywell controllers (UDC 2000 Mini-Pro)

DIGITAL T/C CALIBRATOR/INDICATOR	
Company:	Doric Scientific, division of Emerson electric co.
Model:	Model 477
Accuracy:	This instrument was not used for accuracy
Use:	Feed mV to the Honeywell controllers (UDC 2000 Mini-Pro) during their input calibration

MICROMANOMETER	
Company:	Dwyer Instrument Inc.
Model:	Microtector
Accuracy:	±0.00025 in WG
Use:	Check if portable transducers (Air Neotronics) needed calibration Calibration of Dwyer transducer (although they may not be used)

MULTIMETER	
Company:	Fluke
Model:	8840A
Use:	Provide accurate voltage readings during input calibration of the Honeywell controllers Provide mA reading during output calibration

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