DYNAMIC PERFORMANCE OF CONTROL LOOPS AND THEIR INTERACTIONS IN A VAV HVAC SYSTEM

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ABSTRACT

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Good HVAC control schemes in buildings help reduce energy use and maintain occupant comfort. During the last 10 years, VAV-HVAC systems are widely used in commercial buildings because they can reduce energy use significantly and provide good temperature control of the conditioned spaces compared to CV-HVAC systems. However, the controls for VAV systems are somewhat difficult and further research in this area still needs to be carried out.

This research will focus on local control loop interactions and operating strategies for VAV control systems. To reach this objective, experiments were conducted in a two-zone VAV test facility. The facility consists of six control loops (cooling valve, fan speed, two dampers, two electric heaters) activated by DDC PI controllers. Experiments were conducted in this facility and the results were processed, and analyzed in this thesis.

The steady state and dynamic characteristics of a VAV-HVAC control system has been studied. The interaction between control loops of a VAV system operating under (i) open- loop (ii) closed-loop control modes were evaluated. Also six local loops were tuned and optimal range of controller gains for single-loop and multi-loop VAV system operation were determined. Experiments were conducted to compare the operating performance of Pressure Independent Control (PIC) and Pressure Dependent Control (PDC) strategies. The results show that PIC strategy is more stable than PDC. However, a finely tuned PDC strategy gives good room temperature control compared to PIC.

The results of this research will help building engineers identify operating problems and interactions between local loops in VAV systems. In addition, the optimal range of controller gains obtained in this research can offer guidelines in PID controller tuning. Moreover, the tests conducted provide experimental data to building engineers and researchers for model development and control design.

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TABLE OF CONTENTS

LIST OF FIGURES .... **ix**
LIST OF TABLES .... **xii**
NOMENCLATURE .... **xiii**

CHAPTER 1 INTRODUCTION AND LITERATURE REVIEW **1**

1.1 Introduction 1

1.2 Dynamic Performance of VAV Systems and Components 4

1.3 Tuning Methods 6

1.4 Experimental Research in VAV System Controls 9

1.5 Potential Research Opportunities 18

1.6 Objective of the Research 19

CHAPTER 2 THE VAV HVAC TEST FACILITY **22**

2.1 Introduction 22

2.2 Mechanical System 23

2.2.1 Refrigeration System 25

2.2.2 Cooling and Dehumidifying Coil 25

2.2.3 Chilled Water Flow Control 25

2.2.4 Supply Fan 27

2.2.5 VAV Boxes 27

2.2.6 Zones 28

2.3 Control System 29

2.3.1 Local Loop Descriptions 30

2.3.2 System Architecture and Control System Components 35

2.3.3 I/O Point Configuration and Control Programming 39

2.3.4 Test Data Processing 39

CHAPTER 3 STEADY STATE AND DYNAMIC RESPONSES OF THE OPEN-LOOP VAV SYSTEM OPERATION 43

3.1 Introduction 43

3.2 Airflow System Responses 43

3.2.1 Steady-State Characteristics 44

3.2.2 Interactions between the Zone Airflow Rates 46

3.2.3 Dynamic Responses of the Airflow Subsystem 50

3.3 Dynamic Responses of the Discharge Air System 55

3.4 Dynamic Responses of the Environmental Zones 60

3.5 Summary and Conclusions 61

CHAPTER 4 TUNING OF VAV CONTROL LOOPS AND CLOSED-LOOP RESPONSES OF THE SYSTEM 63

4.1 Introduction 63

4.2 Tuning of Controllers 64

4.3 Closed-loop Responses of the DAT Control Loop 65

4.4 Closed-loop Responses of the Variable Speed Fan Control Loop 67

4.5 Closed-loop Responses of the Zone Damper Control Loops 68

4.6 Closed-loop Responses of the Overall VAV System 69

4.6.1 The Effect of Setpoint Changes on the VAV System Responses 72

4.7 Control Loop Interactions 76

4.8 Summary and Conclusions 80

CHAPTER 5 COMPARISON OF CONTROL STRATEGIES AND CONTROL LOOP OPTIMIZATION 81

5.1 Introduction 81

5.2 Simulation of PIC and PDC Strategies 81

5.3 Optimal Operation of Local Control Loops 86

5.4 Optimal Operation of Multiple Control Loops 88

5.5 Pressure Setpoints for PIC Operation 90

5.6 Summary and Conclusions 92

CHAPTER 6 CONCLUSIONS AND FUTURE WORK 93

6.1 Summary and Conclusions of the Present Study 93

6.1.1 Steady State and Dynamic Response of the Open-loop VAV System 93

6.1.2 Dynamic Response of the Closed-loop VAV System 94

6.1.3 Control Loop Interactions 96

6.1.4 Comparison of Control Strategies and Control Loop Optimization 97

6.2 Future Work 98

REFERENCES 99

APPENDICES

Appendix-1 I/O Point Description 102

Appendix-2 Listing of PPCL Program 105

Appendix-3 Listing of Typical Measured Data 108

Appendix-4 Listing of C++ Data Processing Program 112

Appendix-5 Listing of Typical Processed Data 116

LIST OF FIGURES

2.1 Schematic diagram of VAV HVAC and control system 24

2.2 Cooling coil, chilled water control valve and supply fan 26

2.3 VAV box and control wiring 28

2.4 Room1 temperature control loop 30

2.5 Room2 temperature control loop 31

2.6 Room1 temperature control by the electric heater1 32

2.7 Room2 temperature control by the electric heater2 32

2.8 Duct2 pressure control by the fan speed controller 34

2.9 Discharge air temperature control loop 35

2.10 Control system architecture 35

2.11 Lab building management and control system based on MBC 38

3.1.a Airflow rates to room1 at several damper open positions and fan speeds........... 45

3.1.b Airflow rates to room2 at several damper open positions and fan speeds............ 45

3.2.a The effect of VAV box interactions on airflow rates at high fan speed 47

3.2.b The effect of VAV box interactions on airflow rates at high fan speed 47

3.2.c The effect of VAV box interactions on airflow rates at high fan speed 48

3.3.a The effect of VAV box interactions on airflow rates at low fan speed 48

3.3.b The effect of VAV box interactions on airflow rates at low fan speed 49

3.3.c The effect of VAV box interactions on airflow rates at low fan speed 49

3.4.a Time responses of airflow rates to room 1 and 2 at different damper openings... 51

3.4.b Time responses of airflow rates to room 1 and 2 at different damper openings... 51

3.4.c Time responses of airflow rates to room 1 and 2 at different damper openings.. 52

3.4.d Time responses of airflow rates to room 1 and 2 at different damper openings.. 52

3.5.a Fan system dynamic 53

3.5.b Fan system dynamic 54

3.5.c Fan system dynamic 54

3.6 Steady state time for the airflow system to reach steady state subject to step changes in damper positions 55

3.7 Dynamic responses of the DATS 56

3.8.a DAT responses at several different chilled water flow control valve positions ... 58

3.8.b DAT responses at several different chilled water flow control valve positions .. 58

3.8.c DAT responses at several different chilled water flow control valve positions .. 59

3.8.d DAT responses at several different chilled water flow control valve positions .. 59

3.9.a Temperature responses of air in rooms 1 and 2 (full-load conditions) 60

3.9.b Relative humidity responses of air in rooms 1 and 2 (full-load conditions)........ 61

4.1.a Discharge air temperature response under full-load conditions 66

4.1.b Chilled water flow rate response under full-load conditions 66

4.2 Closed-loop response of the velocity pressure in duct2 68

4.3 Supply airflow rate response to rooms 1 and 2 in damper control mode 69

4.4.a Discharge air temperature response 70

4.4.b Supply airflow rate responses to room 1 and 2 71

4.4.c Room 1 and 2 air temperature responses 71

4.5.a Airflow rate responses subject to a 20% decrease in airflow rates to rooms 1 and 2 73

4.5.b DAT response subject to a 20% decrease in airflow rates to rooms 1 and 2 73

4.5.c Room temperature responses subject to a 20% decrease in airflow rates to room 1 and 2 74

4.6.a DAT response subject to 1°C step change in the DAT setpoint 75

4.6.b Airflow rate responses subject to 1°C step change in the DAT setpoint 75

4.6.c Room temperature responses subject to 1°C step change in the DAT setpoint ... 76

4.7.a Airflow rate responses subject to a 30% step change in the supply airflow rate. 77

4.7.b DAT response subject to a 30% step change in the supply airflow rate............. 77

4.8.a Duct airflow rate responses subject to a step change in the setpoint of damper-1 (case 1) 78

4.8.b Duct airflow rate responses subject to a step change in the setpoint of damper-1 (case 2) 78

4.9.a Duct airflow rate responses subject to a step change in the setpoint of damper-1 (case 3) 79

4.9.b Duct airflow rate responses subject to a step change in the setpoint of damper-1 (case 4) 79

5.1.a Room 1 and 2 temperature responses 84

5.1.b Responses of airflow rates to room 1 and 2 85

5.1.c Heat extraction (cooling rates) of room 1 and 2 85

5.1.d Discharge air temperature response 86

5.2.a Percent overshoot to controller gain for DATC loop.. 87

5.2.b Steady state time to controller gain for DATC loop 87

5.3 Steady state time to controller gain for VAV box damper control loops 88

5.4 Relationship between proportional gain and pressure setpoint........................... 89

5.5.a Relationship between duct1 pressure setpoint and room1 temperature setpoint. 91

5.5.b Relationship between duct2 pressure setpoint and room2 temperature setpoint. 91

LIST OF TABLES

2.1 Description of Sensor Legends 41

2.2 Description of Actuator and Transducer (Interface) Legends 42

NOMENCLATURE

T_{r1}	Room1 Temperature Measured by Sensor
T_{s1}	Room1 Temperature Setpoint
T_{r2}	Room2 Temperature Measured by Sensor
T_{s2}	Room2 Temperature Setpoint
P_{d2}	Duct2 Pressure Measured by Sensor
P_{s2}	Duct2 Pressure Setpoint
FVSD	Fan Variable Speed Drive
T_{d7}	Discharge Air Temperature Measured by Sensor
T_{s7}	Discharge Air Temperature Setpoint
SSR	Solid-State Silicon Rectifier
VAV	Variable Air Volume
MBC	Modular Building Controls
PID	Proportional-Integral-Derivative
PPCL	Powerful Process Control Language
DAT	Discharge Air Temperature
K_p	Proportional Gain
K_i	Integral Gain
K_d	Derivative Gain
$u(t)$	Control Variable
u_0	Initial Value of Control Variable

$e(t)$	Error in the Control Variable from its Setpoint
K_{pv}	Proportional Gain for Valve Control Loop
K_{iv}	Integral Gain for Valve Control Loop
K_{pd}	Proportional Gain for Damper Control Loop
K_{id}	Integral Gain for Damper Control Loop
PIC	Pressure-Independent-Control
PDC	Pressure-Dependent-Control
K_{pd1}	Proportional Gain for Damper1 Control Loop
K_{id1}	Integral Gain for Damper1 Control Loop
K_{pd2}	Proportional Gain for Damper2 Control Loop
K_{id2}	Integral Gain for Damper2 Control Loop

CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1.1 Introduction

HVAC systems provide comfort environments to the occupants of buildings. Usually, HVAC systems are designed for full capacity (design condition). However, most of time, they are not operated under design conditions, therefore, control systems must be employed and good control schemes should be implemented to maintain comfort conditions under changing loads. Good control strategies will also reduce energy use by keeping the process variables (temperature, pressure, relative humidity etc.) at their setpoints efficiently. The efficiency of a HVAC system depends on system type and control strategies such as the selection of appropriate setpoints, control sequence, and proper controller tuning.

In recent years, more and more variable air volume (VAV) HVAC systems have been designed and constructed because they reduce energy use significantly compared to other HVAC systems (CV systems). However, the controls for VAV systems are somewhat difficult. With the advent of inexpensive microprocessors, the use of computer energy management and control systems (EMCS) became popular for large buildings. The EMCS and DDC techniques allow the use of much more complex supervisory control strategies. HVAC control system operating strategy studies include energy savings, the dynamic performance of building and HVAC systems, tuning methods, interactions between local control loops, and system optimization.

During last decade, many researchers have studied design, dynamic performance, and control strategies for VAV HVAC systems. Simulation and experimental research indicate that in addition to lower utility cost and energy savings, VAV systems could also provide excellent temperature and humidity control and meet all criteria required for acceptable indoor air quality [1]. Although reduced energy consumption is possible. VAV systems require robust control schemes if they are to be as efficient as expected [2].

VAV systems became popular during the energy crisis of the 1970s because of their energy-saving potential. Unfortunately, like many new technologies, initial versions of these systems were improperly designed and controlled and caused many operating problems, which gave building engineers a negative lasting impression of the VAV systems. However, with the experience gained and improvement accomplished over the last decade, today's VAV systems can be used effectively throughout the industry under proper design and controls [3].

To understand the benefits of a VAV system, the theory, control system design criteria, and application of individual components must be understood. A VAV system uses the principle of varying the quantity of constant-temperature supply air to meet zone cooling or heating loads. It is typically applied by having a central system that varies the supply air volume in concert with zone volume control boxes. The benefits of a VAV system vary depending upon the type of VAV box selected by the designer and its control.

Zone control is accomplished in the VAV box through the use of a damper. The damper modulates to vary the quantity of air supplied to the zone. A call for cooling by the thermostat modulates the damper position to supply additional airflow through the VAV box to the zone.

When VAV systems are at their best, they provide excellent temperature and humidity control of the conditioned spaces. When VAV systems fail to operate as designed and/or intended, proper temperature and humidity control of the conditioned spaces is not achieved and the cost of system operation may increase to an unacceptable level [1]. The breakdown time, control hardware failures also add significantly to the operating costs. Therefore, research in VAV control systems continues to remain as one of the most important control problems in HVAC Industry.

To this end, this literature review attempts to summarize the developments in the area of VAV HVAC control system operating strategies. The subject is divided into three topic areas: (1) dynamic performance of VAV systems and components, (2) operating strategies, (3) tuning methods.

1.2 Dynamic Performance of VAV Systems and Components

For most VAV systems, the supply airflow rates to the rooms are controlled by the dampers or fan speed control. From a control point of view, the temperature of the zone settles down very slowly to the desired value because of thermal lag effects of enclosure elements.

T. Matsuba et al [4], 1998, developed a dynamic room model without infiltration or exfiltration, which is directly connected to a simple air-handling unit without an economizer. They used this room model to simulate a thermal system that is air-conditioned by a VAV control system. A numerical system model is formulated as bilinear system with time-delayed feedback, and a parametric analysis of the stability limit is presented. They quantified the stability limit and presented a criterion for the determination of control parameters so that hunting is avoided. This study does not consider actual control hardware and therefore not suitable for practical implementation in its current form.

Maxwell et al (1989) [5] performed an analytical and experimental study of the dynamics and control of a chilled water coil. They estimated a coil model from the transient behavior of the chilled water coil in a HVAC airflow test loop. PID controller gains were determined and implemented on a laboratory system. This study looks at the chilled water coil as a separate isolated system, which is not the case in real HVAC systems, where several control loops interact with each other.

Mehta (1987) [6] studied the effects of a building's system control dynamics on energy consumption and on the occupant's thermal acceptability in an occupied space with proportional-integral control. He developed a rational model which described the relationships between the throttling range, proportional band, reset time, coil capacity, and part-load operation. The effects of these relationships on the energy consumption in a fan coil system for a classroom and on thermal acceptability by the occupants were studied. These studies were used to derive control strategies for reducing energy consumption. However, the results are applicable to zone based fan coil units and not for the central VAV systems.

Virk et al (1991) [7] presented a performance comparison of an advanced technique-predictive control-with conventional on/off and PID methods for the control of temperature in a single zone. Based on a laboratory test cell, simulations are first carried out in which conventional on/off control and predictive on/off control are compared with experimental results. The study does not consider VAV used in multizone HVAC systems.

1.3 Tuning Methods.

Although the primary objective of this thesis is not focused on controller tuning methods, a few papers which discuss control interactions are reviewed here. A literature review on tuning methods is given in [8].

The PID controllers must be carefully tuned to avoid hunting. Tuning is a time consuming task for operators since PID parameters are normally determined by trial and error.

Huang and Lam (1997) [9] presented an adaptive learning algorithm based on genetic algorithm (GA) for automatic tuning of PID controllers in HVAC systems to achieve optimal performance. The simulation results show that this algorithm is useful for automatic tuning of PID controllers for HVAC systems, yielding minimum overshoot and settling time. An optimal PI controller-tuning program based on genetic algorithm was developed and implemented in a non-linear HVAC system. The results obtained show that the GA method yields a better performance than that of the traditional Ziegler-Nicoles method for controller tuning. Experimental work is needed to verify the accuracy of such methods.

Brandt (1986) [10] examined some of adaptive controller issues and demonstrated their effect on the performance of actual systems. He also discussed some of the practical considerations that must be resolved when an adaptive controller is used in a typical HVAC system.

Chen and Lee (1990) [11] developed an adaptive robust controller for a single-zone HVAC system that possesses modeling uncertainty and non-linearity. The adaptive robust control was designed based on the nominal portion of the uncertain bound. Simulation results depict a satisfactory transient performance under a significant deviation of the initial state from the comfort region. Compared to adaptive robust controls, classical PID controls are not robust against the uncertainty, the self-tuning PID controls are only applicable to constant uncertainty. The adaptive robust control was proved by theory to guarantee stability. However, this paper only provided theoretical study and the basic framework for further experimental verification, experimental research and study on industrial application is necessary to validate the claim.

Seem (1997) [12] used optimization methods to develop a pattern recognition adaptive controller (PRAC) that automatically adjusts the gain and integral time of PI controllers while under closed-loop control and successfully applied PRAC in tuning control systems for HVAC in various buildings.

Wallenborg (1991) [13] described a self-tuning controller, in which a discrete time process transfer function is calculated from the wave form of a periodic oscillation obtained with a relay feedback tuning experiment. A controller is designed using pole placement based on input-output models.

The author notes that in some cases it may be difficult to obtain the necessary steady-state conditions for a tuning experiment. In many HVAC systems there is also a close interaction between different control loops. Thus, when using an autotuner, it is important to select the order in which the loops are tuned so that the interaction is minimized. These drawbacks, however, also apply to other autotuners where the control design is based on the result from a tuning experiment, as well as to manual tuning of PID controllers.

Kasahara et al (2000) [40] presented a stability limit analysis and a new tuning method for proportional-integral-derivative (PID) controllers for bilinear systems with time-delayed feedback.

The author developed a normalized bilinear model of VAV systems and carried out the generalized parameter analysis of the stability limit. The results indicated that instabilities are not produced by the nonlinearity of VAV systems. However, the PID parameters for a linear system must be modified slightly by finding the gain reduction factors to suit real VAV systems. In addition, the author also derived the optimum tuning of controllers for VAV systems from optimization techniques.

1.4 Experimental Research in VAV System Controls

A number of proposed control systems that work well in theory fall short in practice. Therefore, studying operating strategies on practical VAV systems is very important. The following is a review of some studies in this area of research.

Outdoor Air Controls for VAV Systems

Accurately controlling the amount of outdoor air brought into a building is a major factor in ensuring good indoor air quality (IAQ). With CV systems, controlling the minimum ventilation volume is relative simple. Because the supply fan always provides the same amount of air, the minimum position of the outdoor air damper can be set so that it always supplies the specified minimum amount of outdoor air. However, a problem arises when this same control logic is applied to VAV systems, how to control this minimum ventilation volume is difficult in VAV systems.

Atkinson (1986) [14] discussed a dual-duct VAV system volume control techniques. Controlling the actual outside-air minimum volume independent of the variable supply fan delivery is necessary to meet minimum ventilation and positive building pressurization requirement. Energy can be saved by admitting only the required outside-air volume and by controlling the return-air volume for variable exhaust requirements, resulting in lower return-fan brake horsepower.

Interactions between control loops are generally a problem for system stability when control strategies are complex and integrated. When the supply volume oscillates during startup mode, if the return-fan tracks the supply-fan without "pushing" or pulling it, when the supply duct static pressure stabilizes at setpoint, the return-fan volume also stabilized. Thus the author recommends that return fan should track the supply fan in order to stabilize the airflow rates.

Kettler (1998) [15] reviewed several options for controlling the minimum ventilation volume in a VAV system and concluded that fan-tracking systems and fixed-area, flow-measuring stations sized to measure total outdoor airflow do not work in practice as they do in theory. He recommends plenum-pressure control as an alternate solution.

Seem et al (1998) [16] described a new control system for VAV Air Handling Units (AHU) that use volume matching to control the return fan. The purpose of the new control system is to prevent outdoor air from entering an AHU through the exhaust air outlet. The new control system links the position of the exhaust air damper and recirculation air damper, while the outdoor air damper is in the fully open position during occupied time. Traditional AHU control systems link the position of the exhaust air damper, recirculation air damper, and the outdoor air damper.

A common strategy for controlling the return fan is to maintain a constant difference between the supply and return airflow rates. This is called volume matching. The dampers are sequenced such that as the exhaust and outdoor air dampers begin to open, the recirculation air damper begins to close.

Simulation, laboratory, and field results presented in [16] demonstrate that the new control scheme prevents air from entering AHUs through the exhaust air outlet. However, the new control scheme will not prevent reverse airflow for poorly designed AHUs operating under extreme conditions.

Taylor (1996) [17] studied series fan-powered mixing box (with a parallel fan-powered and a cooling-only "shut-off" box) which includes a supply fan in series with VAV damper. The fan supplies a constant volume of air to the space consisting of primary air from the cooling system mixed with ceiling plenum return air.

The author observes advantages such that improved Indoor Air Quality, comfort, and air distribution performance and disadvantages such as energy inefficiency and cost of this system.

Janu et al (1995) [18] did research on outdoor airflow control for VAV systems. They focused on VAV central system controls and showed how online outdoor airflow measurement and control can be used to assure ventilation as well as meet space pressurization requirements. A practical method for online outdoor airflow control, which integrates the zone-level ventilation with outdoor air control, is proposed. In this method, the central system outdoor airflow controller's setpoint is reset such that the ventilation requirements of the critical zone are met.

Cohen (1994) [19] presented a procedure that maintains a constant minimum rate of outside air intake for all airflow rates in a VAV system by monitoring and adjusting the supply air and return airflow rates continuously.

Krarti et al (2000) [39] conducted an experimental test in a controllable environment using a laboratory equipped with a full-size HVAC system. The results of an experimental evaluation of four airflow measurement techniques and six control techniques used for maintaining minimum outside air intake rates in VAV systems were presented in his paper. The experimental study indicated that control strategies using the direct measurement of the outside airflow from an averaging pitottube array or an electronic thermal anemometer provided the best ventilation control.

Fan Operation and Supply Static Pressure Controls for VAV Systems

Static pressure control is needed to prevent overpressuring the duct system and is the key to energy savings [20]. The fan power required to distribute supply air is a major energy use in commercial buildings. The control strategy for fan control affects the total fan energy consumption [21]. How to reduce the fan energy consumption while assuring all zones receive adequate ventilation is an important issue in VAV system controls.

Williams (1988) [22] discussed some operating characteristics of VAV systems. He pointed out: "It is crucial that all system components, fans, terminal boxes, and controls are compatible to ensure achievement of the required total system operating characteristics. He recommended that the fan speed be controlled using single or multiple pressure sensors located at the end of the primary duct system. To protect the ductwork and the fan from over pressurization he suggests the use of a high-limit pressure sensor installed at the fan outlet.

Warren and Norford (1993) [21] presented a Direct Digital Control (DDC) static pressure reset strategy for supply fan operation in VAV supply air distribution systems. This strategy integrates the requirements of the VAV terminal units serving the building zones to minimize fan energy use. When the supply fan controller finds that several supply units are in the low airflow, the static pressure is reset upward in increment of 5% every 60s thus there is a continuous feedback between the fan and the VAV boxes.

Englander and Norford (1992) [23] proposed and evaluated two methods of controlling the supply fan in VAV systems to minimize duct static pressure and, therefore fan energy use without sacrificing occupant comfort or adequate ventilation. Both methods make use of HVACSIM+ Program to simulate feedback from local zone flow control loops.

According to the simulation results, both control methods: a modified PI control algorithm and a heuristic algorithm successfully minimize fan speed and (indirectly) static pressure, enable operation of the terminal box at its maximum damper opening, while maintaining the ability to meet flow requirements that change in response to loads. However, experiments were not conducted to verify the practical feasibility of the method.

Krakow and Lin (1995) [24] presented an analytical method for determining the PI coefficients of fan control loop in VAV system. The analysis is intended to provide a method of tuning that is based on pressure and flow rate characteristics of the fan and air distribution system as opposed to open-loop responses used in other methods. The tuning method was found to give good performance in a laboratory HVAC system.

Li et al (1996) [25] carried out research in minimizing the energy cost associated with the control of VAV systems. They developed a dynamic model of the system and analyzed its energy consumption characteristics. Experimental work was performed using a single zone VAV system. Conventional PI control, adaptive control, and optimal control algorithms were applied to the duct pressure control loop within the VAV system. A comparison of the performance of these control algorithms indicates that an optimal control strategy can save up to 30% of the energy consumed in a system that employs a conventional PI control scheme. However, such optimal control scheme cannot be applied on real systems which use PID controllers.

Maintaining proper differential pressure in lab spaces is one of the most challenging tasks facing the environmental control engineer according to Hitchings (1994) [26]. He reviewed lab space pressurization control systems, their operating strategies, and other practical issues that should be considered. From accuracy point of view, the differential volume control system offers slight advantage over differential pressure system according to the author.

Chilled Water Loop Controls for VAV Systems

Avery (1996) [27] discussed the advantages and disadvantages of using direct expansion cooling coil in VAV AHU. A method of control that minimizes the disadvantages, using a Face Damper Controlled DX System is described. By using coils with face dampers that close when the coil is deactivated, this system enables the engineer to utilize the advantages of direct expansion cooling on almost any VAV system.

The author shows that it is extremely difficult to maintain a relatively constant supply air temperature under partial load conditions without rapidly cycling the condensing equipment or artificially loading the system with hot gas.

Chen et al (2000) [41] presented a new method called "vote method" which can help decide the supply air temperature setpoint of a VAV system. This method employs control logic that chooses to either minimize the system deviation of room temperature control or minimize supply air volume according to its control objective. It may also link supply air temperature control with other control strategies, such as minimizing chilled/hot water volume control logic in the neutral zone of cooling/heating action.

The author conducted experiments in a real system and proved the controllability, stability, and practicability of the vote method.

Room Temperature and Air Volume Controls for VAV Systems

Hittle (1997) [2] reviewed some typical control strategies for a VAV system that provide multizone control with a single duct. This is a review article outlining existing VAV control schemes, which also discusses local loop tuning issues and approaches used for return/exhaust air volume control.

Brandemuehl and Kreider (1990) [28] built a facility for testing complete, full size HVAC systems and control systems for commercial buildings. They focus on the experimental measurement of the dynamic behavior of building energy systems, including building dynamics, HVAC system performance, and control system operation.

Cumali (1988) [29] made a detailed study of energy use in a high-rise building. He discussed the difficulty of calibrating a system and maintaining energy balances in interacting flow streams within acceptable bounds. He used a control system with enough sampling points to identify the interactions among the building systems and applying this information to the simulation of operation in real time to select operating strategies that are near optimal with respect to energy consumption.

1.5 Potential Research Opportunities

A review of the literature showed that a significant amount of research is conducted in component and system modeling, tuning methods, and operating strategies. However, if we narrow the focus to experimental research in HVAC systems, it is apparent that most efforts are directed towards (i) outdoor air control, (ii) energy saving strategies, and (iii) VAV flow control strategies.

The majority of experimental studies dealt with one aspect of HVAC system operation such as discharge air system control, static pressure control or outdoor air control. Even though these results are of interest in their own right, they often ignore or do not consider the impact of the control loop interaction in multiple loop HVAC systems.

A study of inter-loop interactions in VAV systems cannot be conducted satisfactorily in a real building environment. It requires a controlled laboratory test facility in which the effects of external variables can be minimized. Therefore, a major objective of this research is to identify and assess the control loop interactions in a VAV system by conducting experimental studies in a two-zone VAV test facility.

1.6 Objective of the Research

This research will focus on local loop interactions and operating strategies for VAV control systems, which are of interest in efficient operation of HVAC systems.

The main objectives are:

1. To study the steady state and dynamic characteristics of a VAV HVAC control system.
2. To identify and evaluate the interactions between control loops of a VAV system operating under (i) open-loop and (ii) closed-loop control modes.
3. To determine optimal range of controller gains for single-loop and multiple-loop VAV system operation.
4. To compare the performance of pressure-independent and pressure-dependent control strategies in terms of output regulation and stable operation.

To reach these objectives, experiments were conducted in a two-zone VAV test facility [30]. The facility consists of six control loops activated by DDC PI controllers, which maintain temperatures and flow rates at their setpoint levels.

The following work will be done in this research:

1. Tuning of the local loops and the system to find optimal range of PI control gains under several operating conditions.
2. Given a step input to one loop, monitor and analyze the interactions between the reference loop and other loops under open-loop and closed-loop conditions.

3. To compare and analyze experimental results of pressure dependent control and temperature dependent control schemes.

This thesis will document the experimental results, analysis, and conclusions mentioned above. The results will:

1. Help building engineers identify operating problems and interactions between local loops in VAV systems.
2. Offer guidelines in finding optimal controller gains.
3. And provide experimental data to building engineers and researchers for model development and control design.

CHAPTER 2

THE VAV-HVAC TEST FACILITY

2.1 Introduction

In order to analyze interactions occurring between different control loops and to study different operating strategies of HVAC systems, tests were conducted in a two-zone VAV Test Facility located in the Thermal Environment Control Lab of Concordia University [30]. The test facility includes both the mechanical system and the control system. The mechanical system represents the main section of a multi-zone AHU, that is, the supply fan, main ducts, sub-ducts and VAV boxes and two environmental zones. The control system consists of six local control loops: two VAV damper controls, two electric reheat controls, a fan speed control, and a chilled water flow rate control. All control variables are controlled by a high performance DDC (Direct Digital Control) controller manufactured by Landis & Staefa Inc.

The design, installation and commissioning details of the VAV-HVAC Test Facility are described in [30]. Here a brief description of the system architecture, system functions, system components, and control schemes useful in understanding the results presented in this thesis are presented.

2.2 Mechanical System

A schematic diagram of the two-zone VAV test facility is shown in Figure 2.1. It consists of two 10X10X10 ft thermal simulation chambers. Each chamber has a diffuser and a return air grille. There are two VAV boxes for both of the chambers, thereby allowing the supply air volume to the chamber to be varied. Each VAV box has an electric heater inside, which is used for re-heating and humidity control. An automatically operated fan motor speed control is used to adjust total airflow rate in the system. Two airflow dampers are used to modulate the airflow into the rooms. Chilled water is supplied to the cooling coil from a 2-ton water-cooled chiller and a storage tank unit.

When the system is operating, supply air is cooled to a specific temperature in the cooling coil. The cooled air is then driven by the fan to VAV boxes where the air flow rate and humidity or even temperature can also be changed by modulating dampers and/or by reheating by electric heaters if necessary to meet the cooling load requirements of zone.

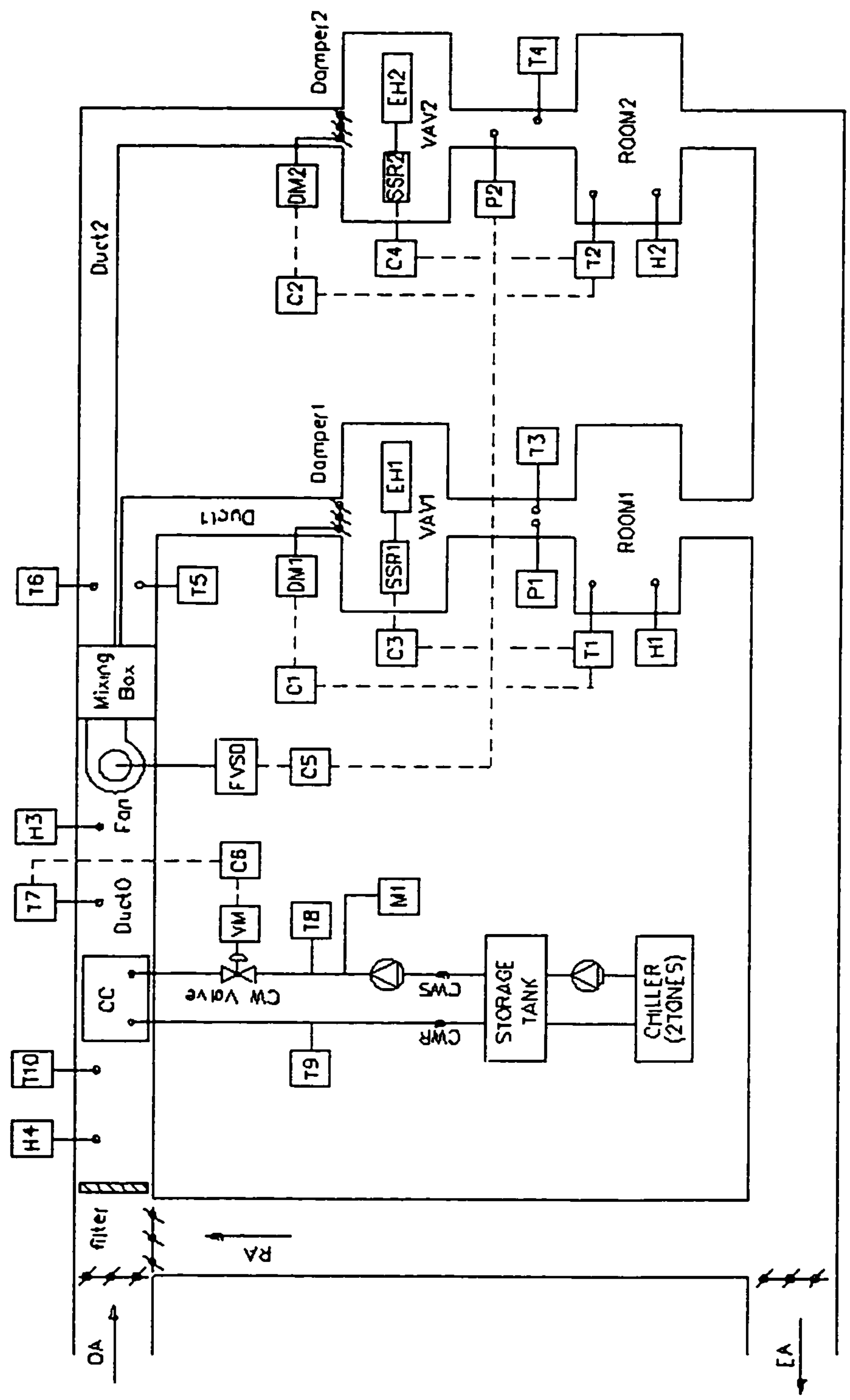

Figure 2.1: Schematic diagram of the VAV HVAC and controls system

2.2.1 Refrigeration System

The refrigeration system consists of a chiller and a storage tank. It can provide 2 tons of refrigeration and uses Freon-22 as refrigerant. Pressure in the suction line was set to 87 psig so that a water temperature of 6.5C can be obtained under 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 a manual ON/OFF switch. Antifreeze solution was added to the water (1/3:1) to prevent corrosion and deterioration of the evaporator and the cooling coil.

2.2.2 Cooling and Dehumidifying Coil

The cooling coil shown in Figure 2.2 has a capacity of 2 tons (24,000BH). It has 4 rows (12 fins per inch) and measures 17"X21"X8" (length X height X thickness). Design airflow rate through the coil is 400cfm. Chilled water runs through the coil in counter-cross flow mode. This type of design provides higher coil efficiency than parallel-flow design because of a higher overall temperature difference between air and water throughout the coil.

2.2.3 Chilled Water Flow Control

The flow control valve is shown in Figure 2.2. It is a Johnson Controls three-way valve with ½ inch diameter pipe connections. This valve is used to adjust supply air temperature in the duct by changing its opening thereby varying the mass flow rate of chilled water in the coil. Its actuator uses a reversible synchronous motor and magnetic clutch to accurately position the valve. The actuator provides a proportional stroke in relation to the control input signal from the electronic controller.

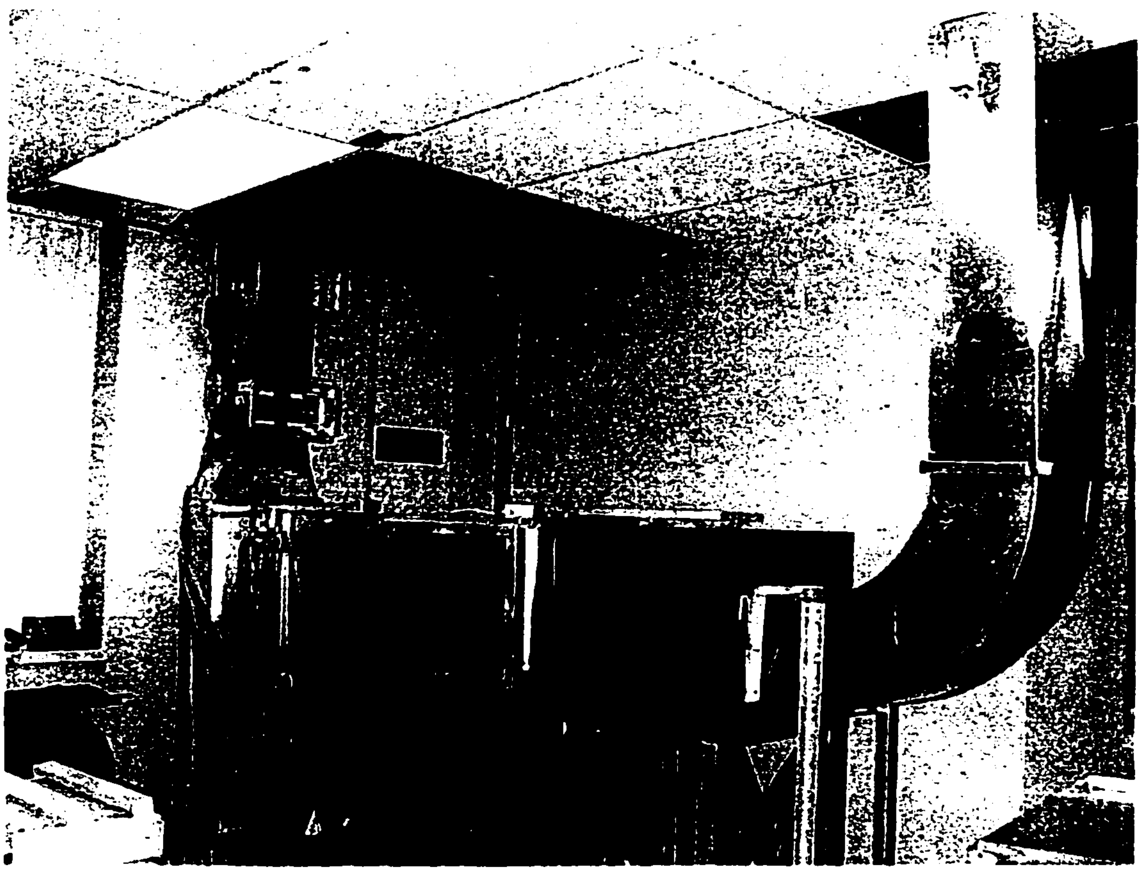

Figure 2.2: Cooling coil, chilled water control valve and supply fan

2.2.4 Supply Fan

A draw-through centrifugal fan (shown in Figure 2.2) equipped with a variable speed D/C motor supplies air to the laboratory HVAC system. The fan speed could be varied between 0 RPM and 1750 RPM producing a maximum airflow of 600 CFM, at an input power of 1 horsepower.

2.2.5 VAV Boxes

There are two identical VAV boxes in the HVAC system. Figure 2.3 just shows one of them. Each VAV box has a damper and an electric heater inside. The damper is used to regulate airflow entering into the zone while the electric heater is used for reheating and dehumidifying for temperature and humidity control.

Each damper (opposed blades) measures 12" X 12". The reheat coil is an electric heater (500W). The output of the heater is controlled by the MBC through a SSR device. Static pressure sensors and inter-lock circuits in the ducts and VAV boxes respectively prevent overheating when the fan is off or the airflow rate in the ducts is less than a minimum threshold value.

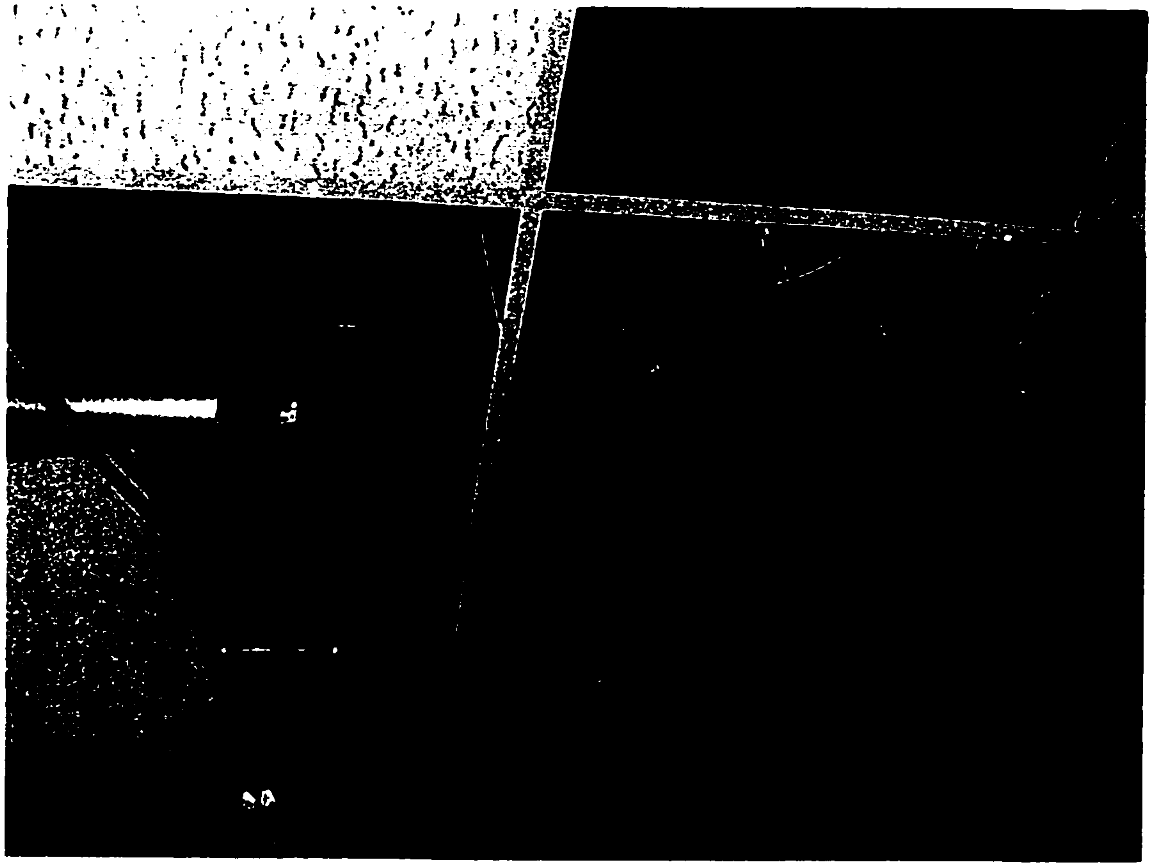

Figure 2.3: VAV box and control wiring

2.2.6 Zones

Two 9'X9'X9' identical zones (chambers) are used in the laboratory so that different HVAC system operating strategies can be tested simultaneously and compared with each other conveniently. Each zone has one diffuser (9"X9"), one grille. A 2KW electric heater and humidifier are installed in each zone to simulate sensible and latent loads.

2.3 Control System

The monitoring and control equipment includes one PC, one Modular Building Controller (MBC), several sensing devices, interfaces, and actuators such as damper motors, fan variable speed drive, valve motor, etc. It has the following functions [31]:

- Data collection and processing
- Local sequencing and control logic
- Operating points and control actions monitoring
- Advanced PID tuning and control

Figure 2.1 shows locations of sensors, actuators, equipment and relations between them in the HVAC system. In Figure 2.1, temperature sensors T3, T4, T5, T6, T8, T9, T10 (see Table 2.1 for nomenclatures), humidity sensors H3, H4, pressure sensors P1, and chilled water flow rate meter M1 are only used for monitoring and calculation. Temperature sensors T1, T2, T7, humidity sensors H1, H2, and volume-measuring station P2 are not only used for monitoring and calculation but also for real-time control. All controllers are in MBC. All sensors, actuators, and devices are connected to the MBC by wires and cables.

During operation MBC receive signals directly from the sensors. The signals are then converted into special data format, which is readable by the computer. These data can either be stored for future analysis or used directly in the control algorithms. Once the value of the desired control parameter is calculated, an output signal is sent from the computer (or MBC) to the actuating device, and a proper control action is carried out to the system.

2.3.1 Local Loop Descriptions

There are six local closed control loops in the VAV system [30].

Loop1: room1 temperature control by modulating the position of damper1 (Figure 2.4).

The purpose of the damper1 controller is to vary the damper1 position in the VAV box for room1 temperature is modulated through a damper actuator, in order to maintain the air in the room1 at a constant temperature. The room temperature sensor T1 measures the value of room1 temperature Tr1 and sends it to controller1 (C1). C1 compares this value to the setpoint Ts1. If there is a difference, then C1 calculates the required control parameter and sends an output signal to damper1 motor (DM1). The DM1 adjusts damper1 position in accordance with its control signal to maintain Tr1 as close as possible to the setpoint.

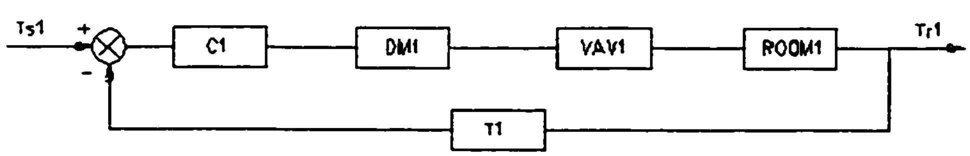

Figure 2.4: Room1 temperature control loop

Loop2: room2 temperature control by modulating the position of damper2 (Figure 2.5).

The damper2 position in the VAV box for room2 is modulated through a damper actuator, in order to maintain the air in the room2 at a constant temperature. The room temperature sensor T2 measures the value of room2 temperature Tr2 and sends it to controller2 (C2). C2 compares this value to the setpoint Ts2. If there is a difference, then C2 calculates the required control value and sends an output signal to damper2 motor (DM2). The DM2 modulates damper2 position to maintain Tr2 close to the setpoint.

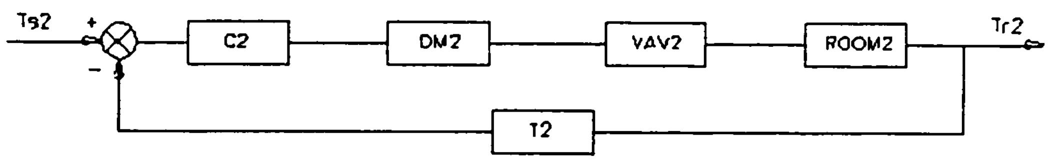

Figure 2.5: Room2 temperature control loop

Loop3: room1 temperature control by reheat via the electric heater1 (Figure 2.6).

In summer, when dehumidification of air is needed, sometimes this control action may cause a problem: the room temperature is lower than we expected. In this case, a reheat coil is needed to warm air. The purpose of the heater1 controller is to vary the output of the electric heater1, through a control device, in order to maintain the air temperature in the room1 at the desired temperature. The room temperature sensor T1 measures the value of room1 temperature Tr1 and sends it to controller3 (C3). C3 compares this value to the setpoint Ts1. If there is a difference, then C3 calculates the value of the required control parameter and send an output signal to SSR1. The SSR1 adjust output of electric heater1 to satisfy the requirement.

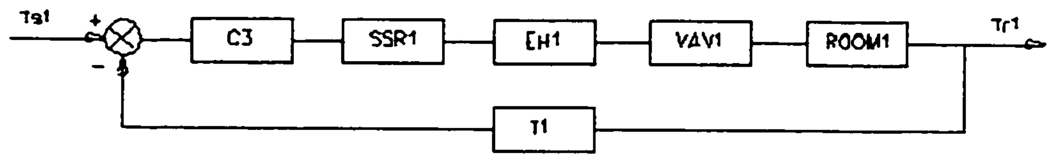

Figure 2.6: Room1 temperature control loop by the electric heater1

Loop4: Room2 temperature control by the electric heater2 (Figure 2.7).

In this case, Heater2 controller is used to vary the output of the electric heater2, through a control device, in order to maintain the air temperature in the room2 at the desired temperature. This is achieved through controller C4 shown in the Figure 2.7.

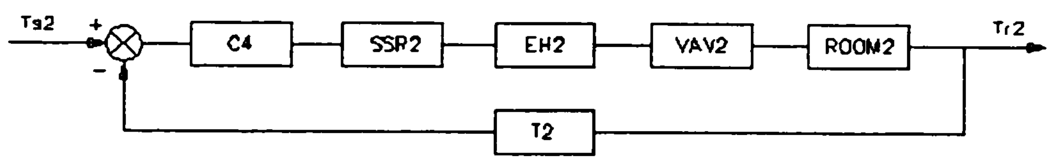

Figure 2.7: Room2 temperature control loop by the electric heater2

Loop5: duct2 pressure control by modulating the fan speed (Figure 2.8).

Since the supply air fan use is one of major energy consumption in buildings, controlling fan is a very important issue in HVAC system operations, especially for VAV HVAC systems. In order to reduce the fan energy consumption to the lowest level possible without affecting comfort, a lot of research were carried out on this subject in recent years. Those studies indicated that three important factors were identified as affecting the energy consumption of a fan:

1. The static pressure sensor location (design issue, will not be discussed here);
2. The controller type to be used (P, PI, PID);
3. Fan type (how the flow rate is to be controlled) to be used.

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. The Test Facility [30] is equipped with a centrifugal fan and a variable speed control device; it is controlled by a PI-controller.

The purpose of the fan speed controller is to vary the fan speed, through a control device (or actuator), in order to maintain static pressure in the duct2 at a desired value. In figure 2.8, the pressure sensor P2 measures the value of duct2 pressure Pd2 and sends it to controller5 (C5). C5 compares this value to the setpoint Ps2 (since Ps2<Ps1). The error is used to generate an output signal to fan variable speed drive (FVSD). The FVSD

modulates fan speed in accordance with its frequency to maintain Pd2 within a specific range.

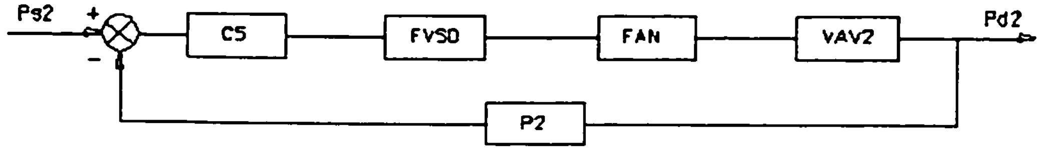

Figure 2.8: Duct2 pressure control by the fan speed controller

Loop6: Discharge air temperature (T7) control by modulating the chilled-water flow rate through the valve (Figure 2.9).

The purpose of the chilled water valve controller is to vary chilled water flow rate, through a valve actuator, in order to maintain the air in the duct0 at the constant temperature. The duct temperature sensor T7 measures the value of duct0 temperature Td7 and sends it to controller6 (C6). C6 compares this value to the setpoint Ts7 and generates and sends an output signal to the chilled-water valve motor (VM). The VM modulates chilled-water flow rate through adjusting the valve plug in accordance with the opening of the chilled-water valve to maintain Td7 within a specific range. For example, when Td7 is higher than its setpoint Ts7, the opening of the chilled-water valve will increase; when Td7 is lower than Ts7, the opening of the chilled-water valve will decrease

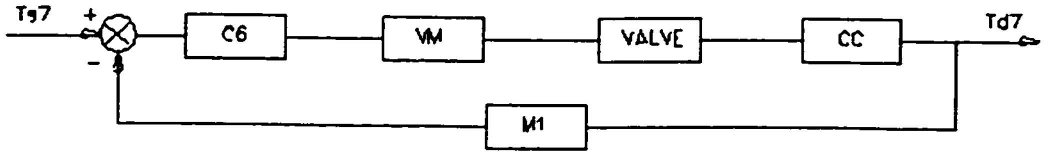

Figure 2.9: Discharge air temperature control loop

2.3.2 System Architecture and Control System Components

A schematic diagram showing that the control system architecture, sensors, actuators and controllers together with the range of control input signals is shown in Figure 2.10 [30].

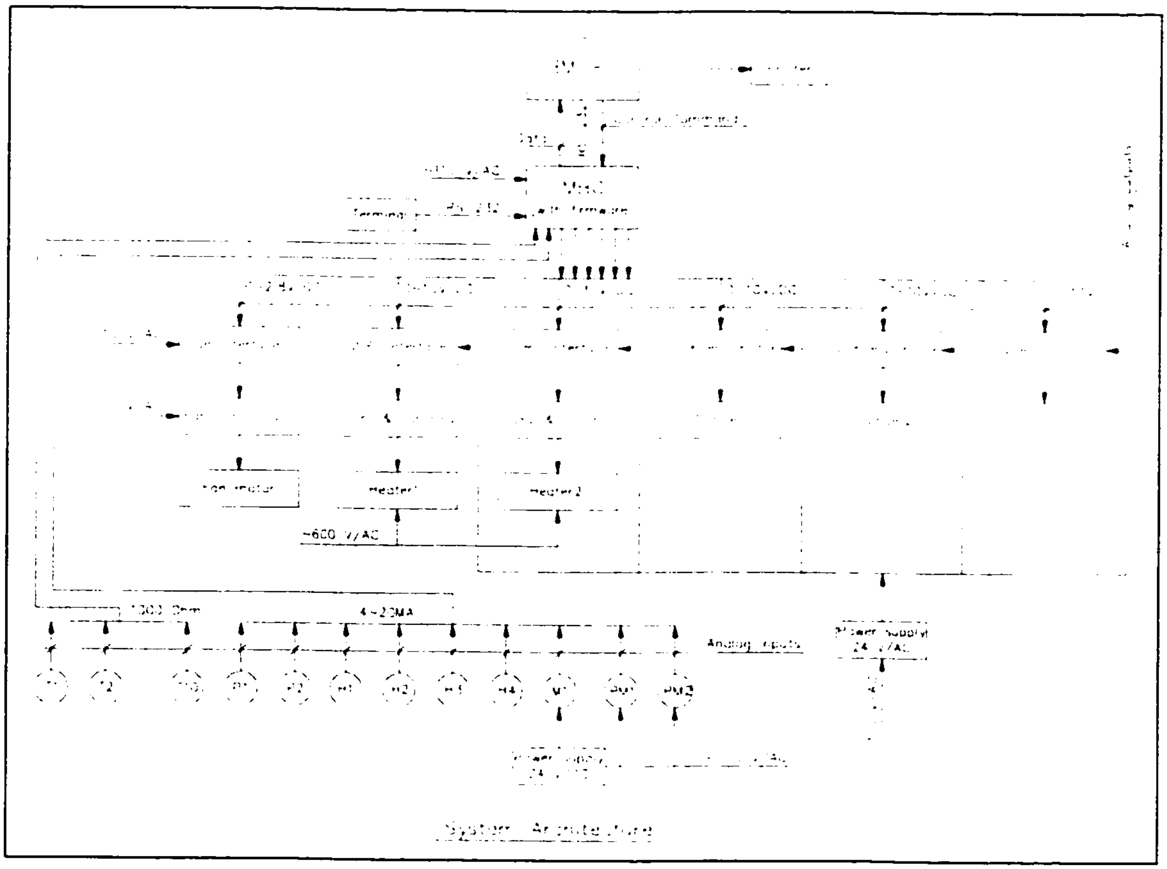

Figure 2.10: Control system architecture [30]

a) Sensors and Transmitters

The type of sensors, their range, and output signals transmitted by the sensors are given in Ref [30]. Details on the calibration and accuracy of the sensors are also described in Ref [30]. Description of Sensor and Legends are presented in Table 2.1.

b) Actuators and Transducers

When we try to integrate a control system that consists of components and devices from different vendors, interoperability is a very important issue. In this project, we used several vendors' control products. To make them work together well, some interface devices were employed. Among the six control loops, the two damper actuators and valve actuator can accept 0~10VDC signal directly from MBC and don't need extra interfaces or transducers. However, the other three loops, two electric heaters and fan controller (FVSD) cannot be controlled by MBC directly, therefore, three corresponding interfaces were selected, which are shown in Figure 2.10.

Normally, actuators and transducers are fed by the power from DDC controllers, but sometimes, extra power supply to actuators and transducers is needed. In this control system, the pressure transducers and water flow meter require a 24V DC power supply to transmit signals to the controller. Likewise a 24 V AC power supply provides power to the actuating devices such as damper motors, valve motor and the solid-state silicon rectifiers (SSR).

Table 2.2 summarizes and lists all actuators and transducers (interfaces) used in this project.

c) Modular Building Controller

The Modular Building Controller (MBC) is an integral part of the Landis & Staefa Building Management and Control System [31]. It is a high performance, modular Direct Digital Control (DDC) supervisory field panel. The field panel operates stand-alone or networked to perform complex control, monitoring and energy management functions without relying on a higher-level processor. Figure 2.11 shows a lab Building Management and Control System and its wirings based on MBC. PC and MBC communicate through RS-232 interface [30].

Combined with Powers Process Control Language (PPCL) [32] and APOGEE Insight software, MBC can implement many powerful functions such as:

- Custom program sequences to match equipment control applications
- Advanced Proportional Integral Derivative (PID) loop tuning algorithms for HVAC Built-in energy management applications and DDC programs for complete facility management

- Optimizing the use of the equipment.
- Energy Control and Management.
- Comprehensive alarm management, historical data trend collection, operator control and monitoring function

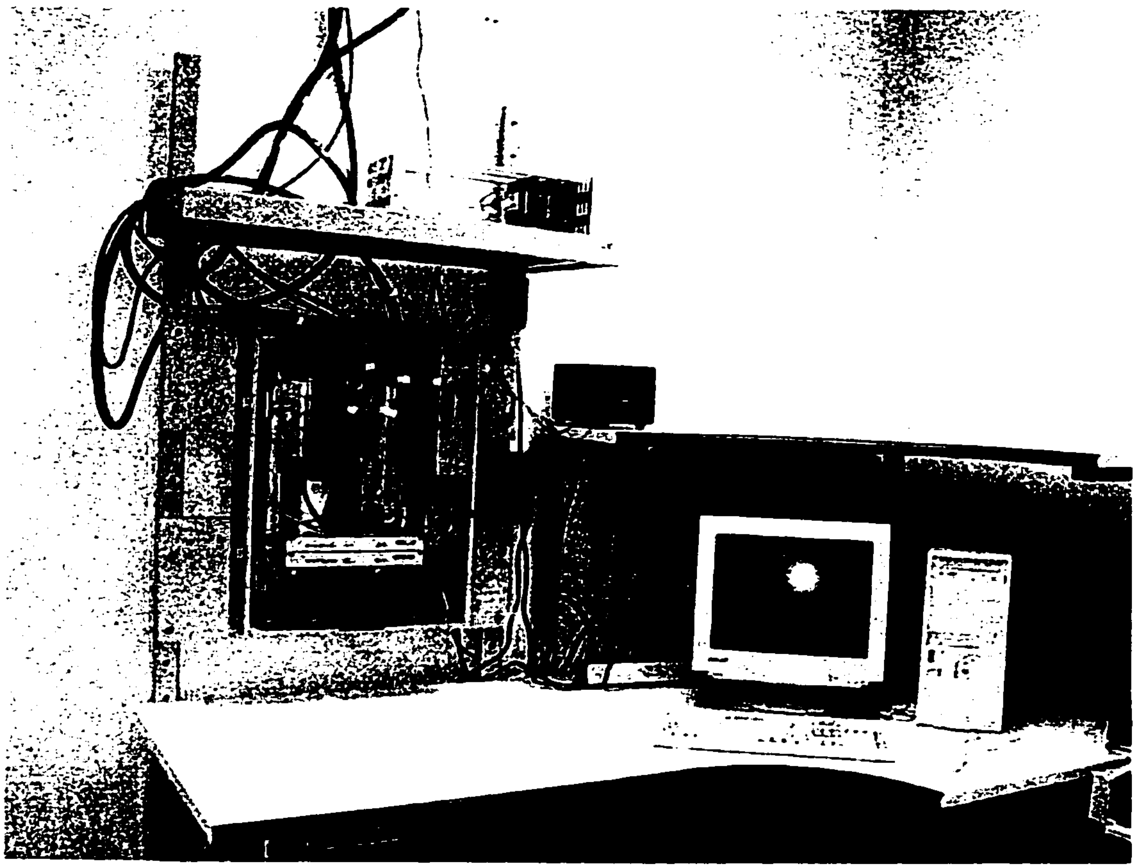

Figure 2.11: Lab building management and control system based on MBC

The MBC contains four major components: an enclosure assembly, a power module, point termination modules, and an open processor [31].

2.3.3 I/O Point Configuration and Control Programming

Control software is the head of a HVAC control system. Before commissioning the system, I/O point database needs to be created and control program has to be made.

Appendix-1 lists all I/O point configuration information, which includes point type (analog or digital, input or output), point description, point priority, point alarm setting, point high/low limit, point engineering units, point communication addresses, and point calibration parameters. This I/O database was generated through MBC Operator Interface and stored in non-volatile memory of MBC [32].

Based on I/O point Configuration and PPCL, control program was easy to be developed. Appendix-2 lists all control program used in this thesis.

2.3.4 Test Data Processing

The results presented in this thesis are based on capturing the data from the MBC in terminal mode of operation and post processing the data by developing appropriate computer program and the use of Excel software packages.

Since the real test data cannot be trended from MBC without APOGEE software, we used Hyperterminal function provided by Windows95 operating system to monitor test results.

The controller updates control signals every second and output data is printed on terminal every 4 seconds [33]. The data scrolling on the screen can be saved into text files. Appendix-3 lists a set of typical measured data.

In order to analyze the test results, the format of these text files shown in the Appendix-3 needs to be converted into standard format that can be identified by Excel. Therefore a C++ data-converting program (see Appendix-4) was developed and used to process the raw test data. The typical processed data is shown in Appendix-5.

Table 2.1: Description of Sensor Legends

Number	Sensor symbol	Variable measured	Units
1	T_1	Room1 air temperature	^oC
2	T_2	Room2 air temperature	^oC
3	T_3	Supply air temperature to room1	^oC
4	T_4	Supply air temperature to room2	^oC
5	T_5	Temperature of air entering the VAV box-1	^oC
6	T_6	Temperature of air entering the VAV box-2	^oC
7	T_7	Discharge air temperature	^oC
8	T_8	Supply chilled water temperature	^oC
9	T_9	Return chilled water temperature	^oC
10	T_{10}	Air temperature entering the cooling coil	^oC
11	H_1	Room1 relative humidity	%
12	H_2	Room2 relative humidity	%
13	H_3	Discharge air relative humidity	%
14	H_4	Relative humidity of air entering the coil	%
15	P_1	Pressure (flow station) of air entering room1	in WG
16	P_2	Pressure (flow station) of air entering room2	in WG
17	M_1	Mass flow rate of chilled water	gpm

Table 2.2: Description of Actuator and Transducer (Interface) Legends

Number	Symbol	Controlling Variable	Interface	Signal Range
1	DM1	Room1 air temperature	Not needed	0~10VDC
2	DM2	Room2 air temperature	Not needed	0~10VDC
3	SSR1	Supply air temperature to room1	Needed	0~10VDC
4	SSR2	Supply air temperature to room2	Needed	0~10VDC
5	FVSD	Duct2 Pressure	Needed	0~2.8VDC
6	VM	Discharge air temperature	Not needed	0~10VDC

CHAPTER 3

STEADY STATE AND DYNAMIC RESPONSES OF THE OPEN-LOOP VAV SYSTEM OPERATION

3.1 Introduction

Before studying the closed-loop responses of the VAV system, it is instructive to study the open-loop responses of the system subjected to different cooling loads with constant control inputs. From the open-loop test results we can establish the operating range of the system, quantify the system time constants and study interactions among the local control loops. In this chapter the following sub-system responses will be evaluated.

1. Steady state and dynamic responses of the airflow system.
2. Dynamic responses of the discharge air system.
3. Dynamic responses of the environmental zones.

3.2 Airflow System Responses

The experimental results presented in this section describe the steady state and dynamic responses of the airflow sub-system decoupled from the thermal sub-system. In other words, the cooling loads on the zones, the chiller and the coil loop were turned off. The three important variables influencing the airflow rates in the system and to the rooms are: fan speed and damper 1 and 2 positions in respective VAV boxes.

3.2.1 **Steady-State Characteristics**

Figure 3.1.a-b show the supply airflow rates to rooms 1 and 2 as a function of damper opening at different fan speeds. In these experiments, with the chosen fan speed and damper 1 and 2 positions, the airflow system was turned on and allowed to reach steady state. The resulting steady state responses are depicted in Figure 3.1.a-b. The results show that airflow rates to the rooms are equal with room1 receiving slightly higher airflow compared to room2. At maximum speed room1 airflow rate is about 3% higher than room2 airflow rate. Also to note from the figures is fact that with damper openings of less than 25%, airflow rates to the rooms are practically zero. From control point of view, damper positions between 40~100% appear to be effective. Establishing such operating range of the system is very helpful in designing good control strategies for VAV systems.

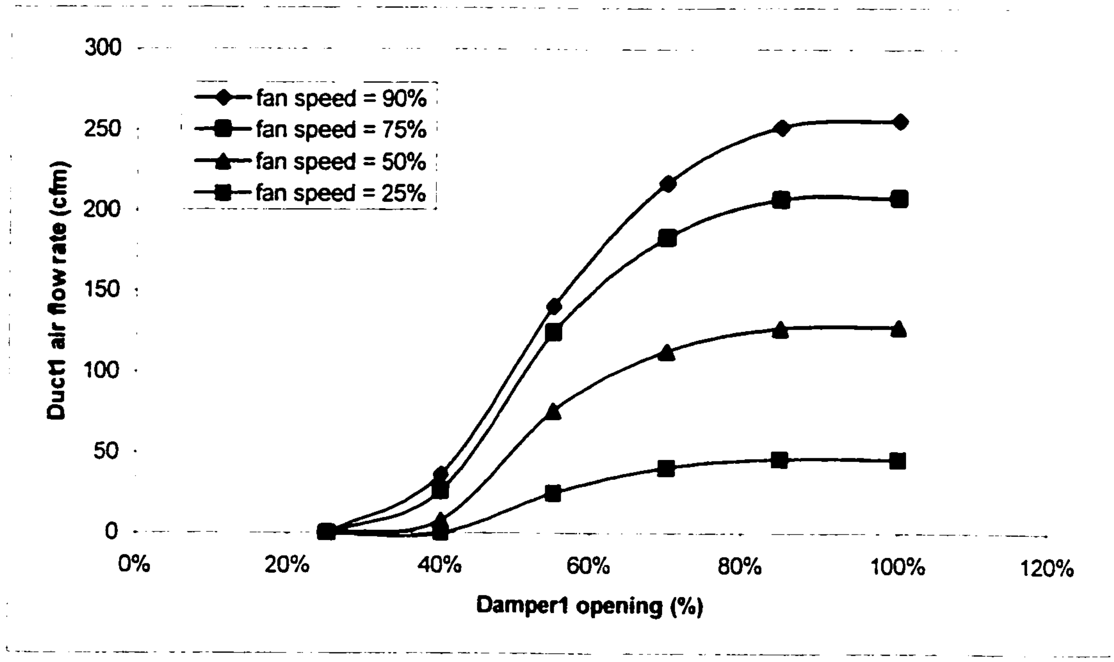

Figure 3.1.a: Airflow rates to room1 at several damper open positions and fan speeds

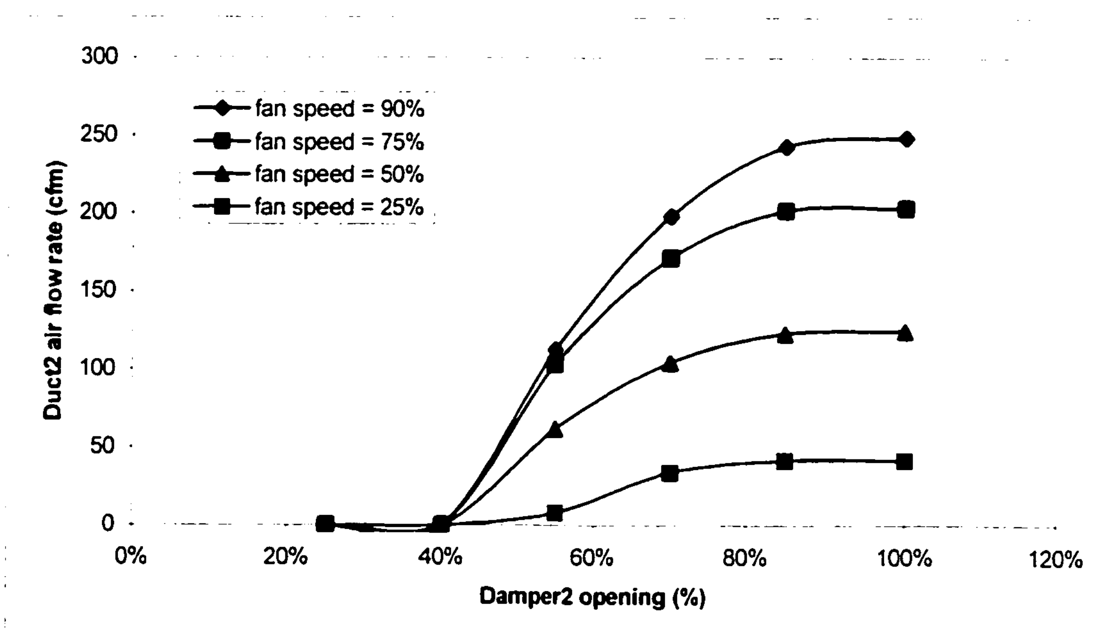

Figure 3.1.b: Airflow rates to room2 at several damper open positions and fan speeds

3.2.2 Interactions between the Zone Airflow Rates

One of the important question that arises in the operation of VAV systems is the effect one VAV box has on the airflow rates delivered by the neighboring VAV box. In other words, how significant are the interactions between two adjacent VAV boxes measured in terms of change in airflow rates delivered to their respective zones. To study and quantify these effects several experiments were conducted. The results are plotted in Figure 3.2.a-c and 3.3.a-c.

These figures show the effect of changing the damper-1 position while the damper-2 is held open at a fixed position and fan speed is also kept constant. Figure 3.2.a-c depict the results at full speed (about 90% of the rated maximum) and corresponding results at low speed (30% of the maximum) are shown in Figure 3.3.a-c. At higher fan speeds a change in damper-1 position from minimum to maximum open condition (25%~100%) can cause a 16% reduction in airflow rate to room2 through a fully open damper-2. On the other hand, at lower fan speeds (Figure3.3.a-c) this reduction in airflow rate to room2 significantly increases to about 22% (Figure 3.3.a). Furthermore, the results also show that at lower fan speed (30%) the wide open damper-1 could cause zero airflow rates through the half-open damper-2 (Figure 3.3.c). From these results we can state that tuning of one damper due to variations in load automatically necessitates tuning of the neighboring VAV dampers. And, at low loads (and therefore low fan speeds) the interaction effects are more significant than at full load. This means control of VAV system at part-loads is more difficult than at higher loads.

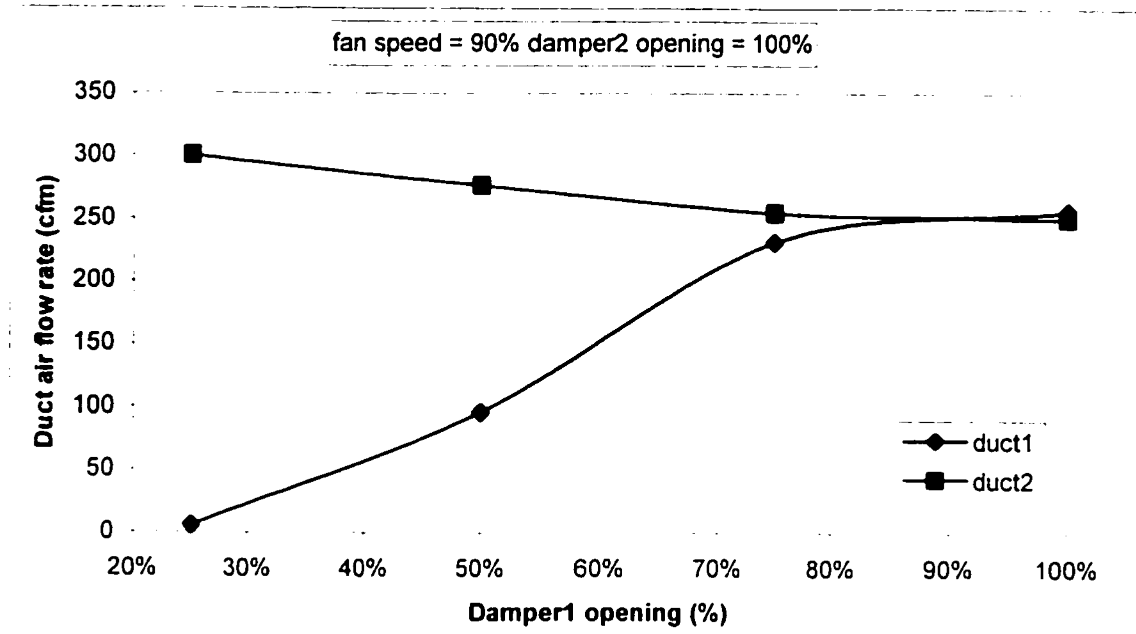

Figure 3.2.a: The effect of VAV box interactions on airflow rates at high fan speeds

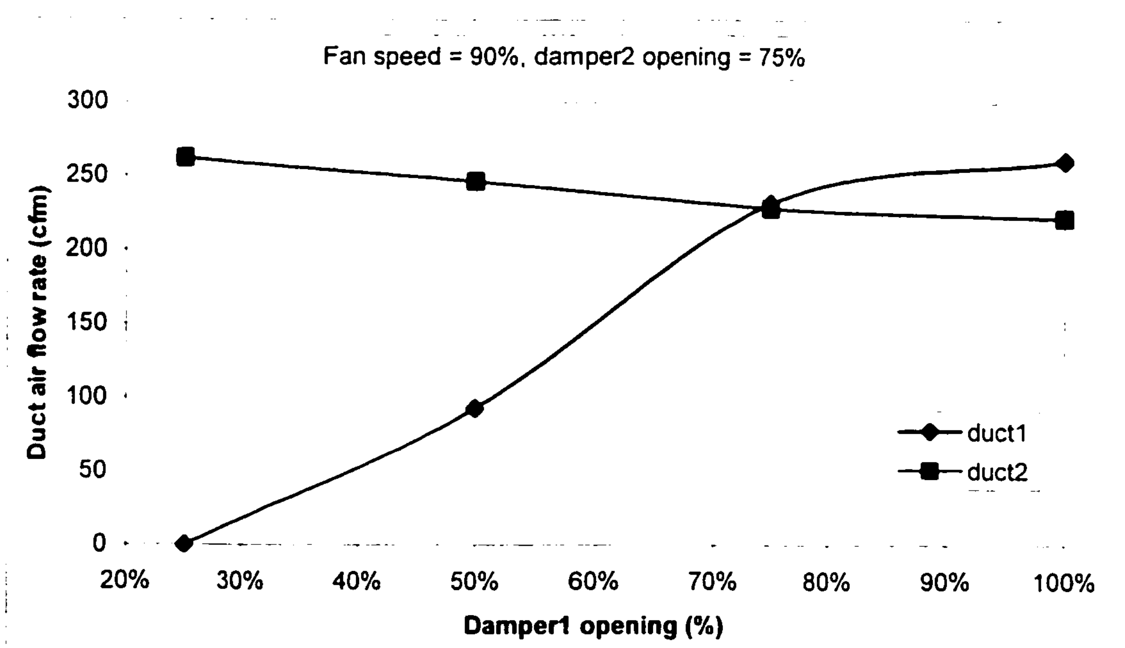

Figure 3.2.b: The effect of VAV box interactions on airflow rates at high fan speeds

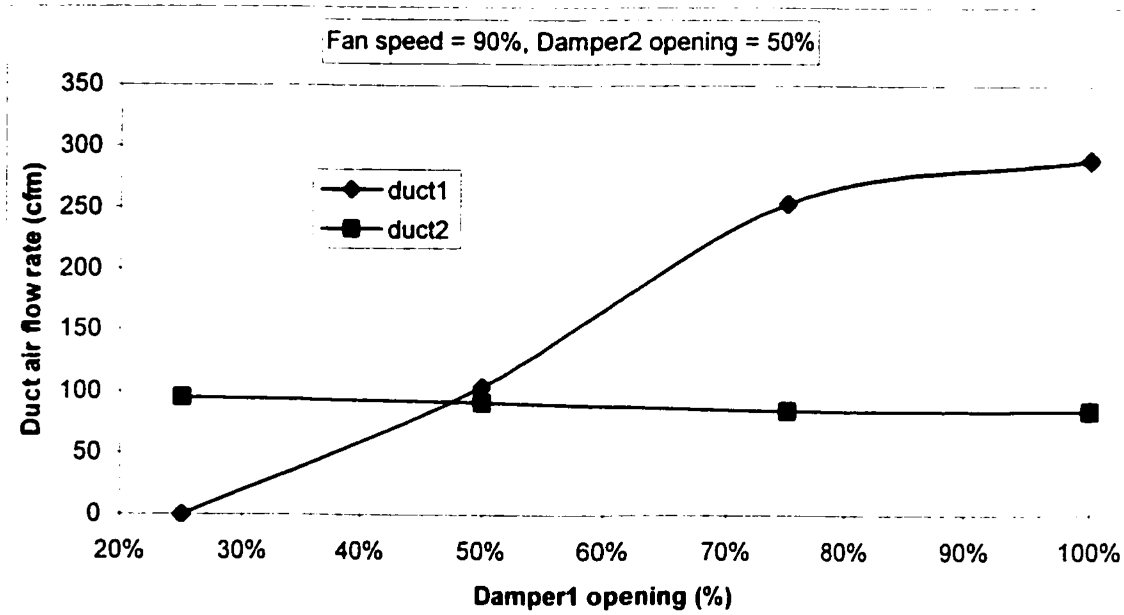

Figure 3.2.c: The effect of VAV box interactions on airflow rates at high fan speeds

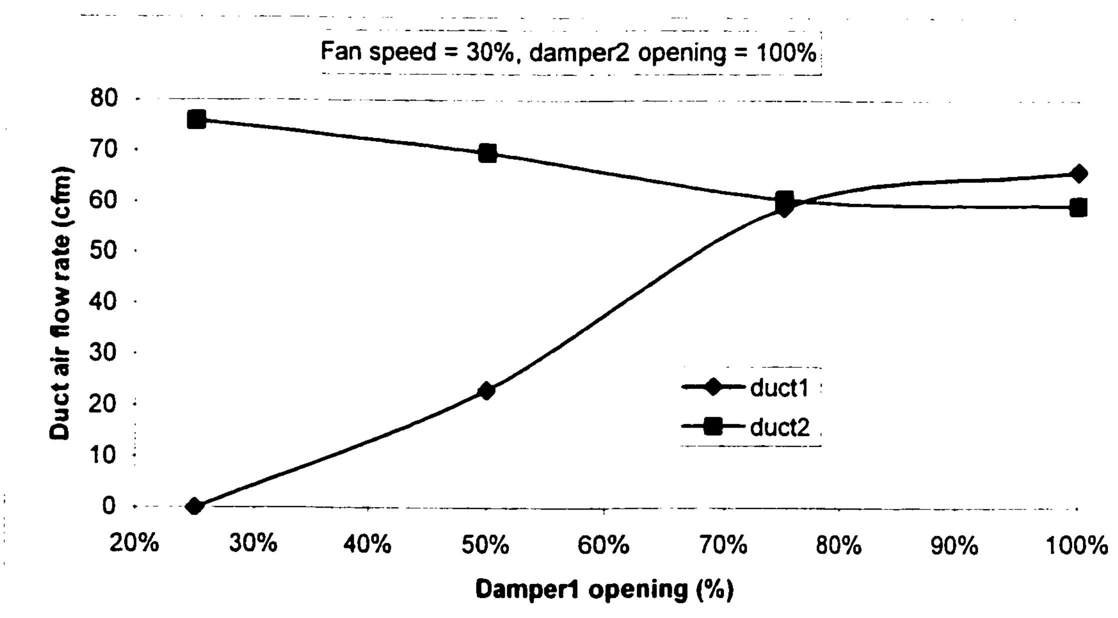

Figure 3.3.a: The effect of VAV box interactions on airflow rates at low fan speeds

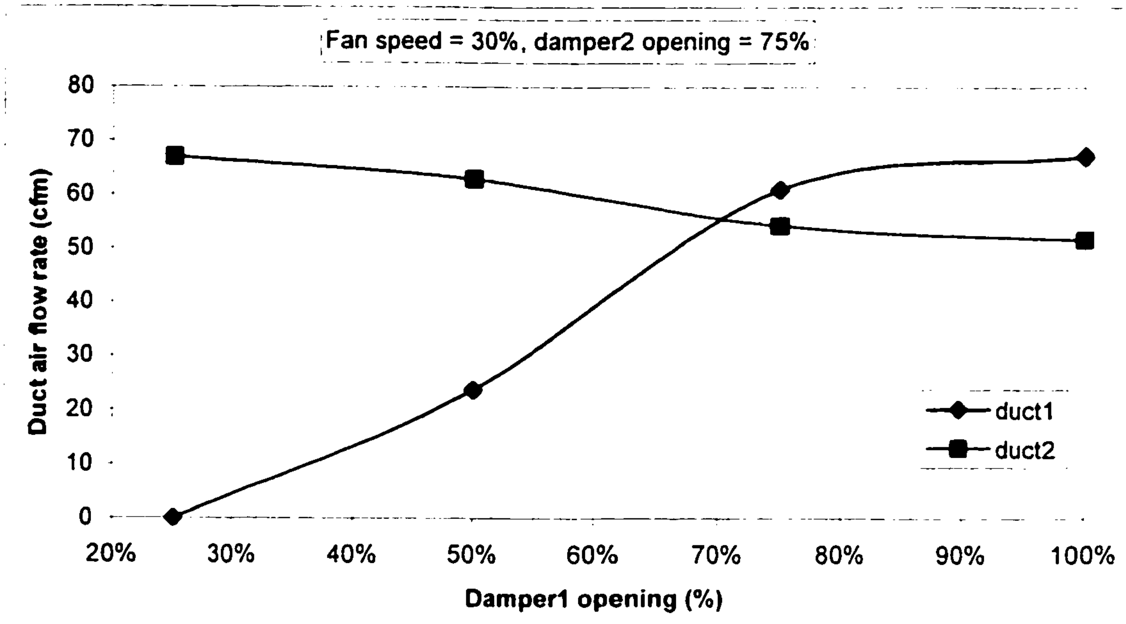

Figure 3.3.b: The effect of VAV box interactions on airflow rates at low fan speeds

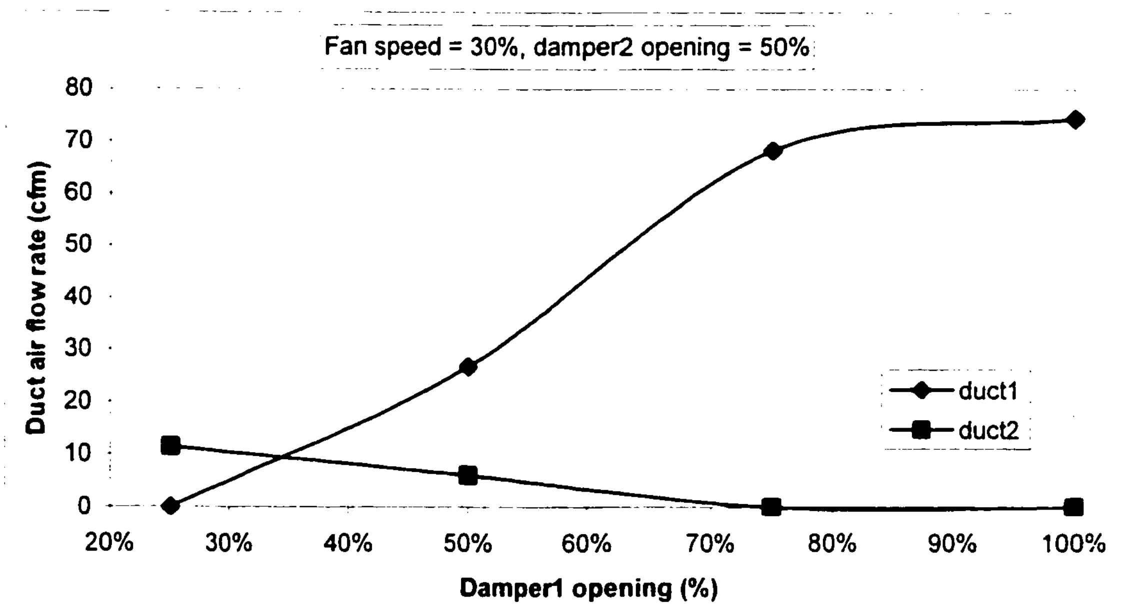

Figure 3.3.c: The effect of VAV box interactions on airflow rates at low fan speeds

3.2.3 Dynamic Responses of the Airflow Subsystem

In Figure 3.4.a-d, airflow rates to rooms 1 and 2 as a function of time at several damper opening positions are plotted. From these figures, we note that the time needed for the system to reach steady state increases as the fully closed damper opens towards fully open position. In other words, as shown in Figure 3.4.a between 80~100 seconds are needed for the system to reach steady state when the damper-1 is opened from fully closed position to full-open position. Likewise, when the damper-1 is opened from 0 to 50%, the airflow rates are stabilized in about 40 seconds (Figure 3.4.c). Figure 3.4.d depicted the airflow rate responses as a function of time when damper-1 opens from 0~100% open position while the damper-2 simultaneously close from 100~0%. From this figure we can also note that a minimum of 20 seconds are lapsed from the time the control signal to the damper is given to the time we notice a measurable airflow rate at airflow measuring station. Thus the 20 seconds is the 'dead-time' of the airflow sub-system.

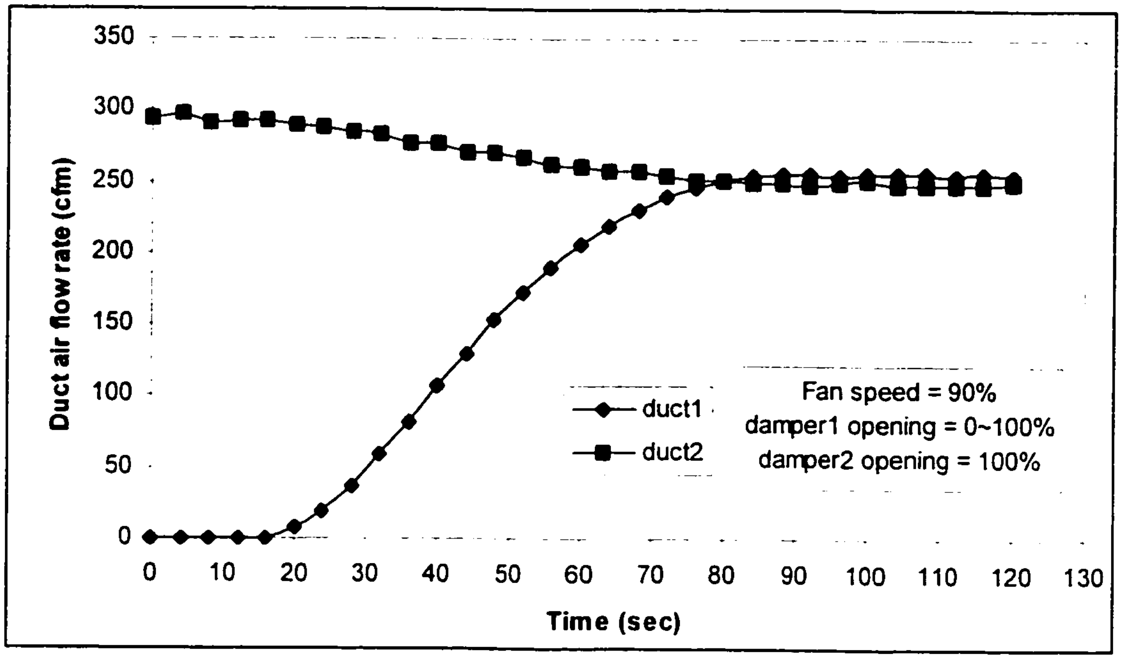

Figure 3.4.a: Time responses of airflow rates to rooms 1 and 2 at different damper openings

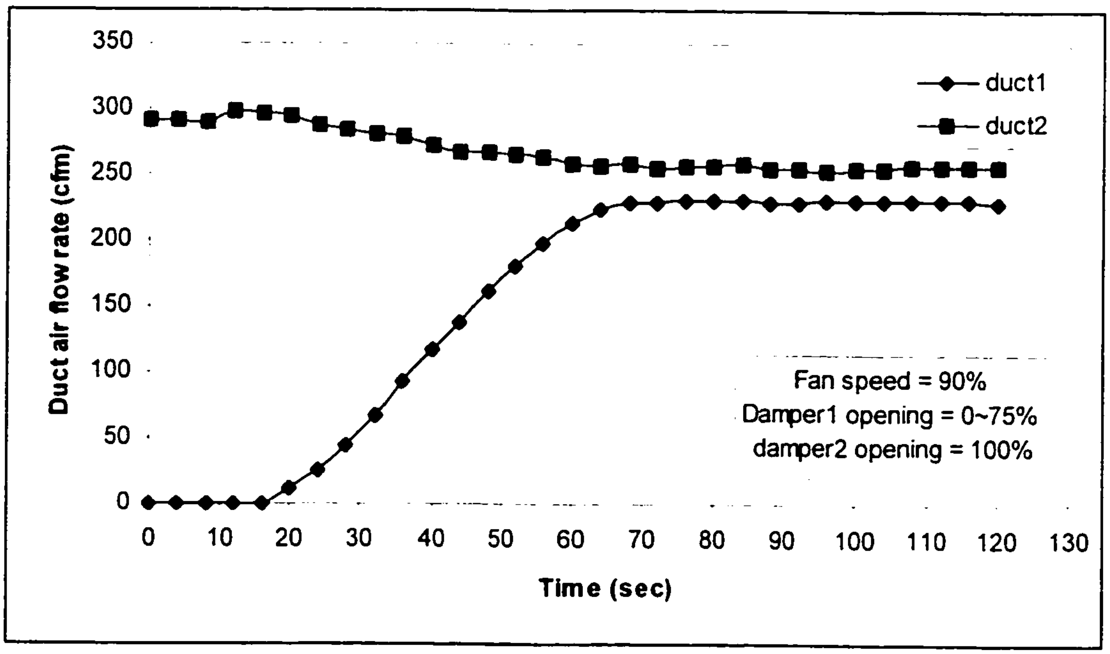

Figure 3.4 .b: Time responses of airflow rates to rooms 1 and 2 at different damper openings

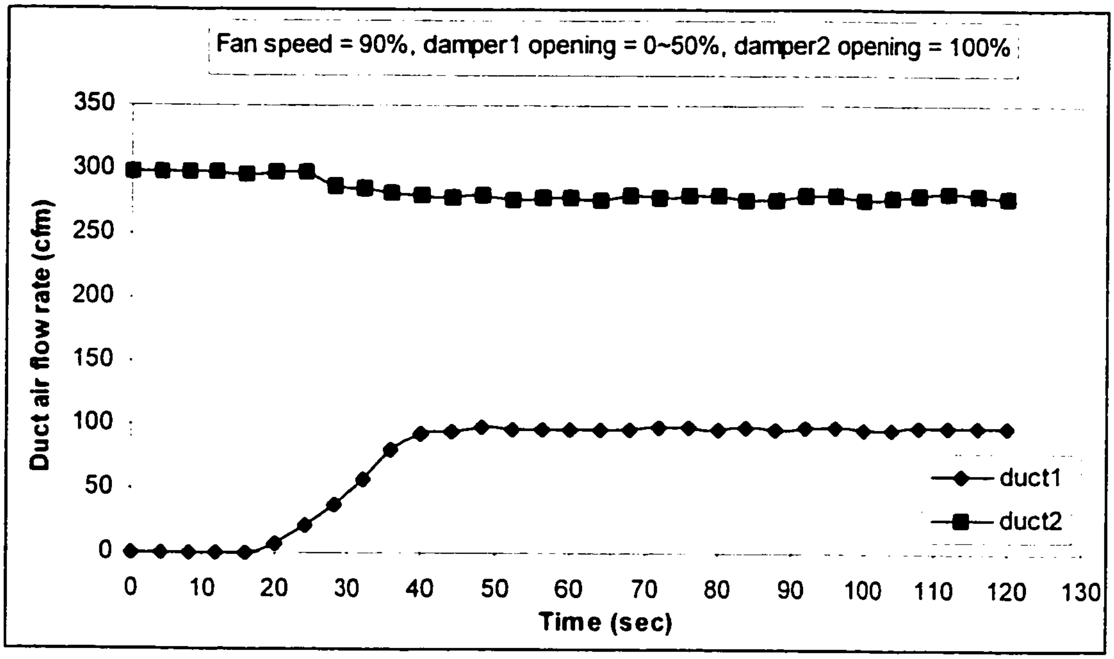

Figure 3.4.c: Time responses of airflow rates to rooms 1 and 2 at different damper openings

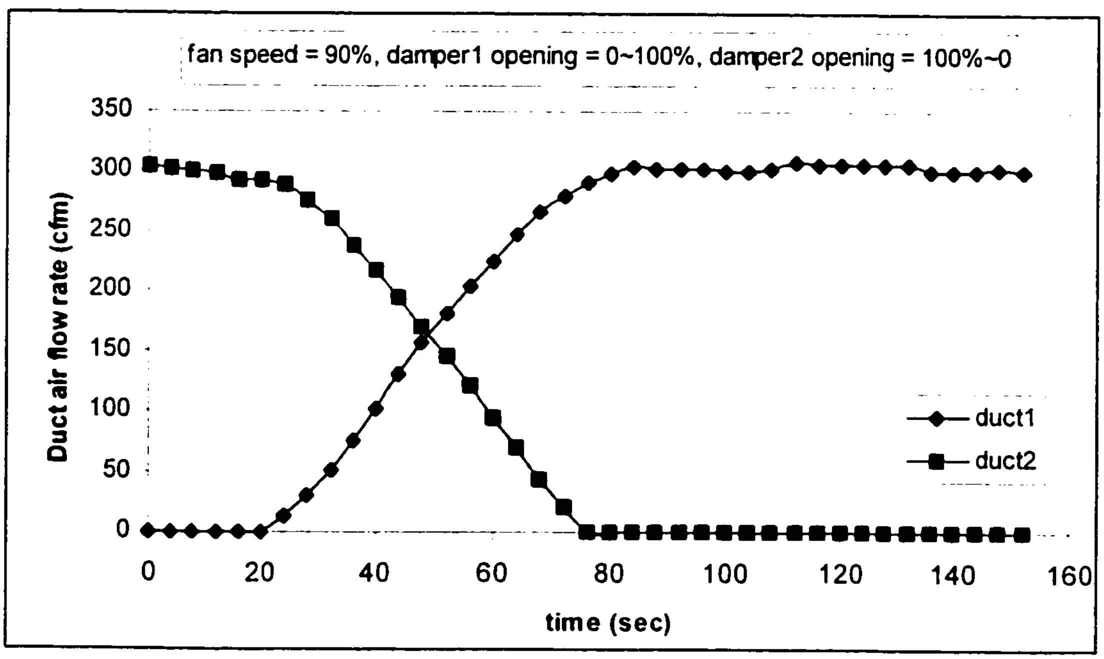

Figure 3.4.d: Time responses of airflow rates to rooms 1 and 2 at different damper openings

We have also measured the time response of the airflow system to step changes in fan speed. That is, while keeping the dampers 1 and 2 full open, the fan speed was increased from 0 to 90%, 0 to 60% and o to 30%, as shown in Figure 3.5.a-c. It is apparent from these figures that the fan dynamics are much faster. The system reaches steady state in about 10 seconds, which is much faster than 80 seconds needed to reach steady state by modulating the dampers. This means that damper dynamics tend to govern the airflow rate responses in VAV system. A summary of steady state times of the airflow system as a function of damper opening and fan speed is depicted in Figure 3.6. The results show that in a normal operating mode (damper opening ranging between 40~100%) the steady state time is influenced more by the damper opening position than the fan speed. For the VAV system the time needed for the airflow rate to stabilize range between 40~90 seconds.

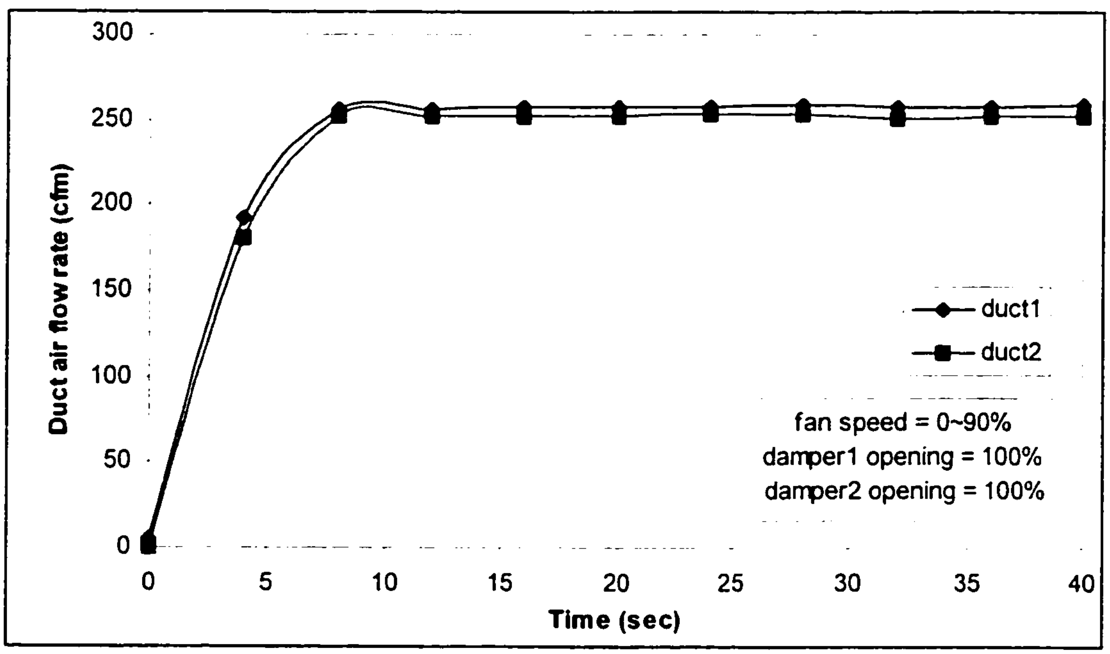

Figure 3.5.a: Fan system dynamics

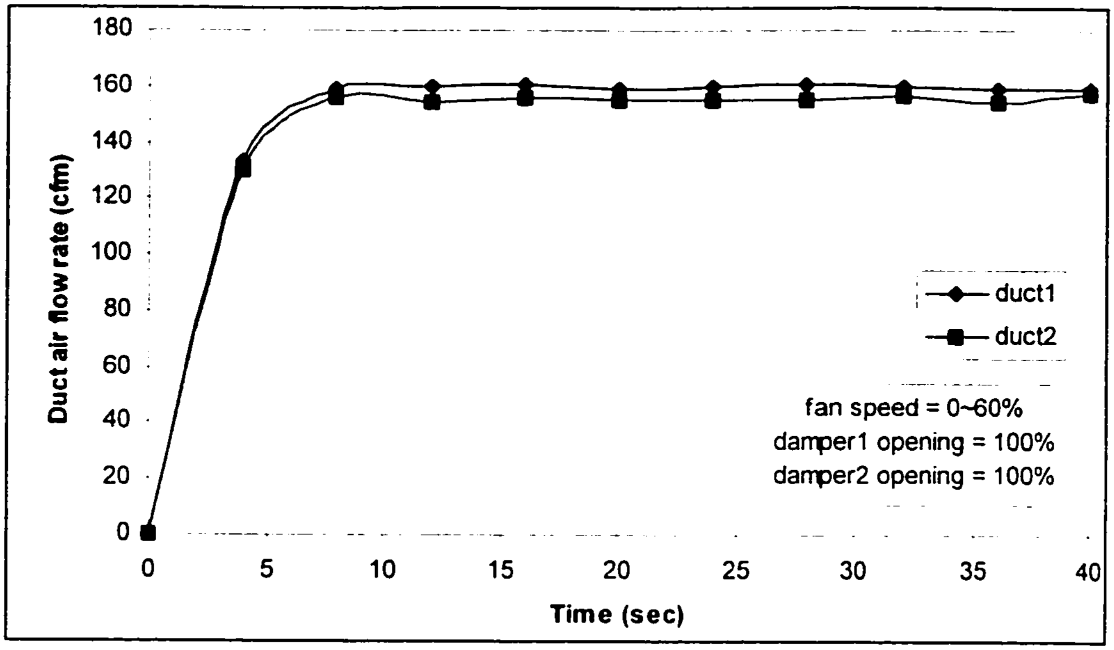

Figure 3.5.b: Fan system dynamics

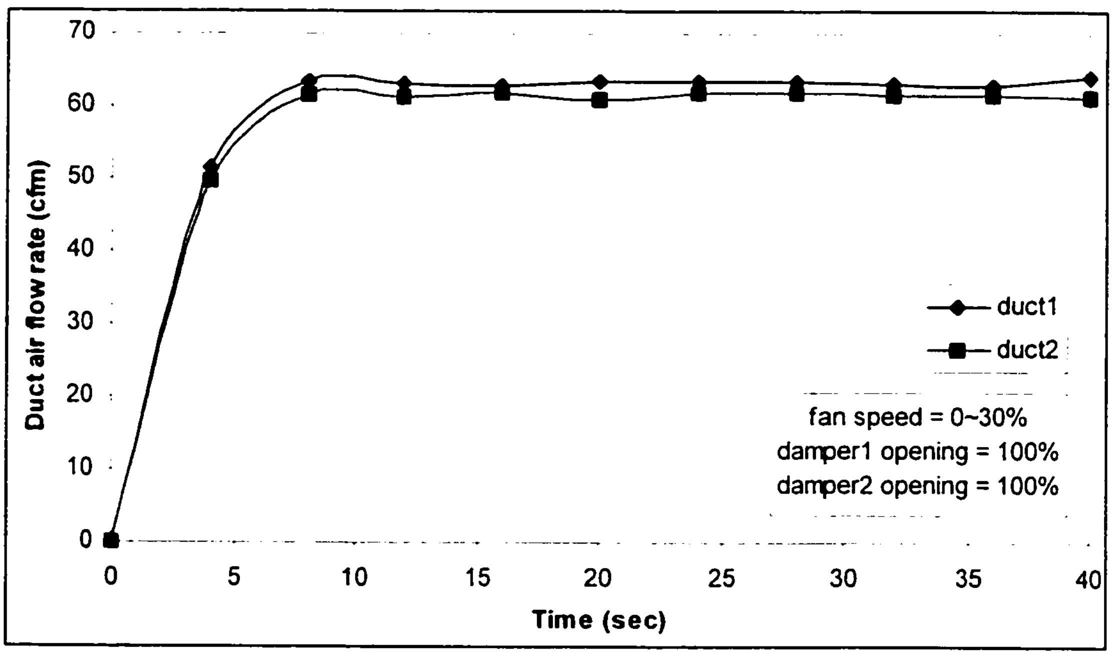

Figure 3.5.c: Fan system dynamics

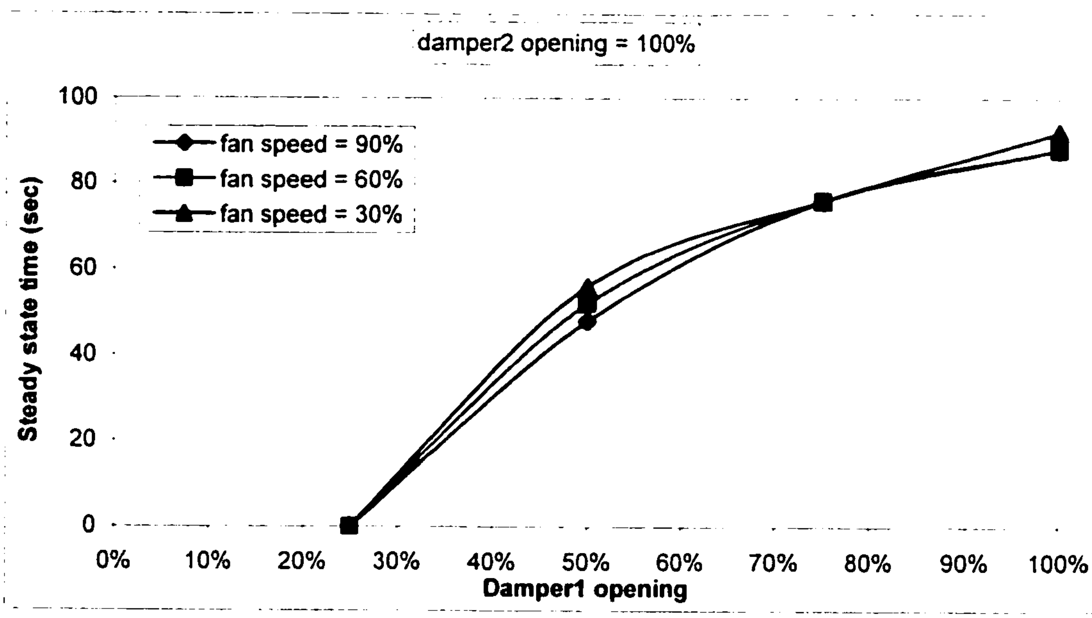

Figure 3.6: Steady state time for the airflow system to reach steady state subject to step changes in damper positions

3.3 Dynamic Responses of the Discharge Air System

That part of the system in which cooling and dehumidification of air occurs (cooling coil) is often known as discharge air temperature system (DATS). The inputs to the system are the condition of air entering the coil, airflow rate, chilled water temperature and its flow rate. The outputs of the system are the temperature and humidity ratio of the air leaving the coil. In order to study the open-loop dynamics of the DATS we have conducted several experiments.

The dynamic responses of the DATS under full-load conditions (cooling load = 100%, fan speed = 90%) are depicted in Figure 3.7. With the system under full-load conditions, the chilled water valve was opened from 0 to 90% open position. The resulting responses are plotted in Figure 3.7. The discharge air temperature reaches steady state value of 13°C in about 400 seconds. Chilled water supply temperature remains constant between 8.5~9°C. The steady state return water temperature from the coil was about 12°C. Also shown in the figure is the chilled water flow rate as a function of time. We note that at 90% valve open position the water flow rate was about 6gpm. Also the 'dead-time' of the water flow system is about 20 seconds and water flow rate reaches steady state in 90 seconds. The results reveal that the steady state time of the airflow sub-system is about 4 times faster than the discharge air sub-system.

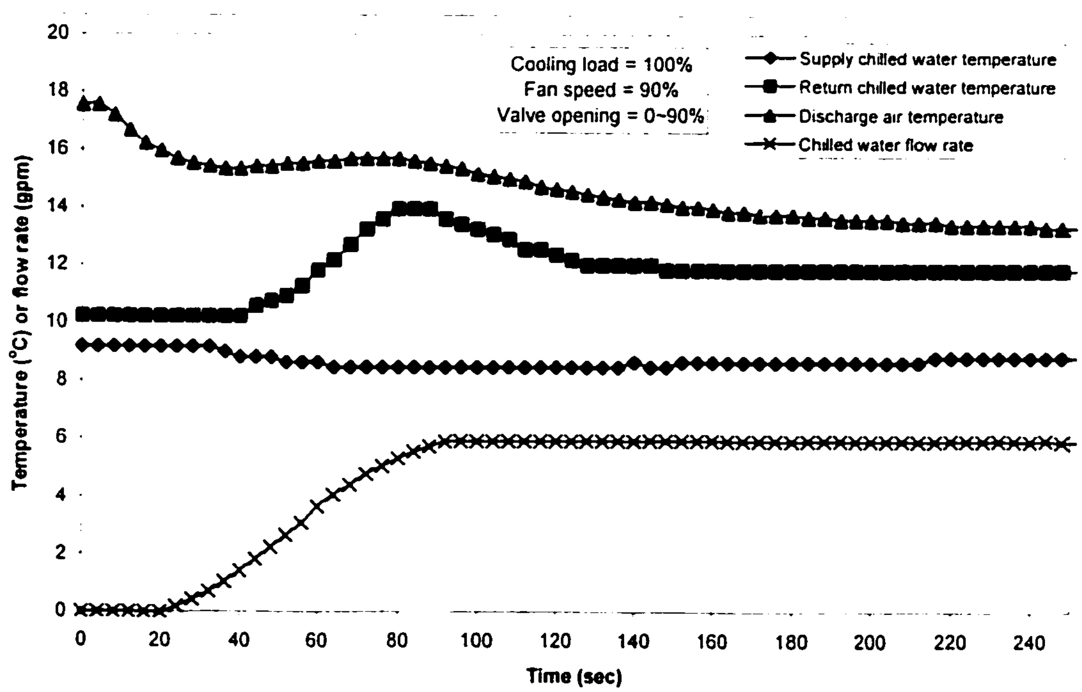

Figure 3.7: Dynamic responses of the DATS

The effect of chilled water flow rates (at several different cooling loads and fan speeds) on the discharge air temperature responses are depicted in Figure 3.8a-d. The results show that the steady state time remains constant between 350~400 seconds irrespective of the load (The load dependence is not very significant). Furthermore, the operating range of the chilled water valve between 50% to 90% is able to achieve discharge air temperature ranging between 10 to 14°C when the chilled water supply temperature is held between 6.5 to 8°C. It should be noted that essentially the dynamics of the discharge air system are coupled with the dynamics of the zone and the airflow system. Therefore, the steady state time of 400 seconds for the DATS does include and therefore is influenced by the zone dynamics. Even with this coupling effect it is a useful indicator of the time constant of the DATS.

Another important consideration in our experiments was the need to create desired initial conditions in the rooms. This is somewhat time consuming exercise. Initial conditions do have an effect on the time needed by the system to reach steady state. In the results presented in Figured 3.8.a-d most of the time we were able to keep the initial conditions within +1°C except for one case in Figure 3.8.b in which the maximum variation in initial conditions was noted to be about 2°C.

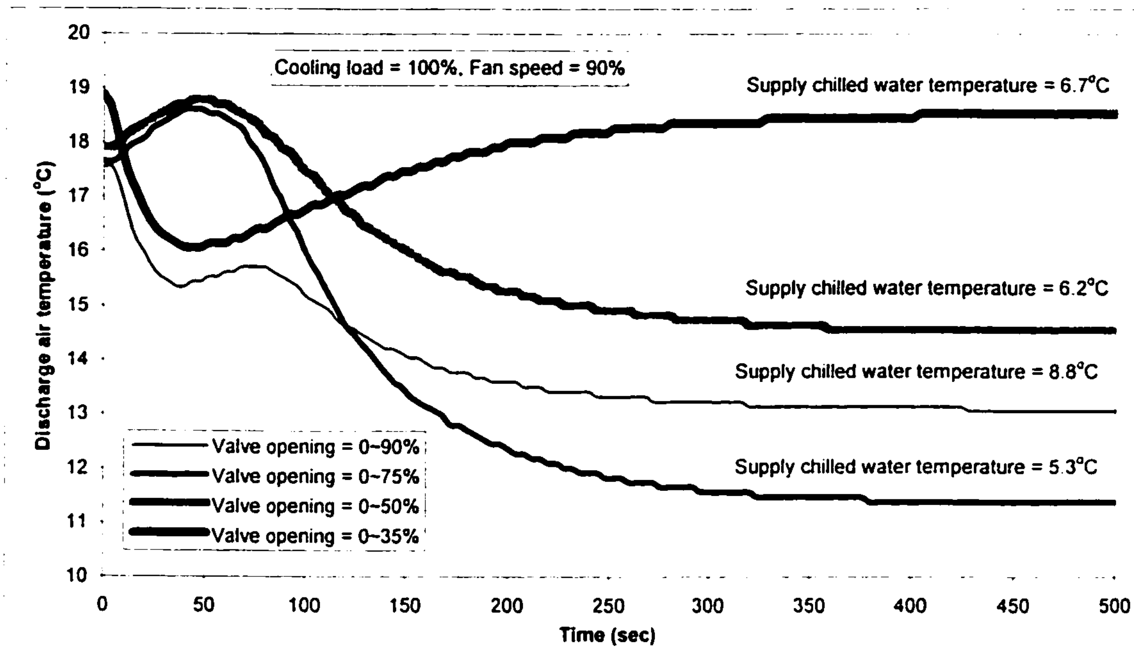

Figure 3.8.a: DAT responses at several different chilled water flow control valve positions

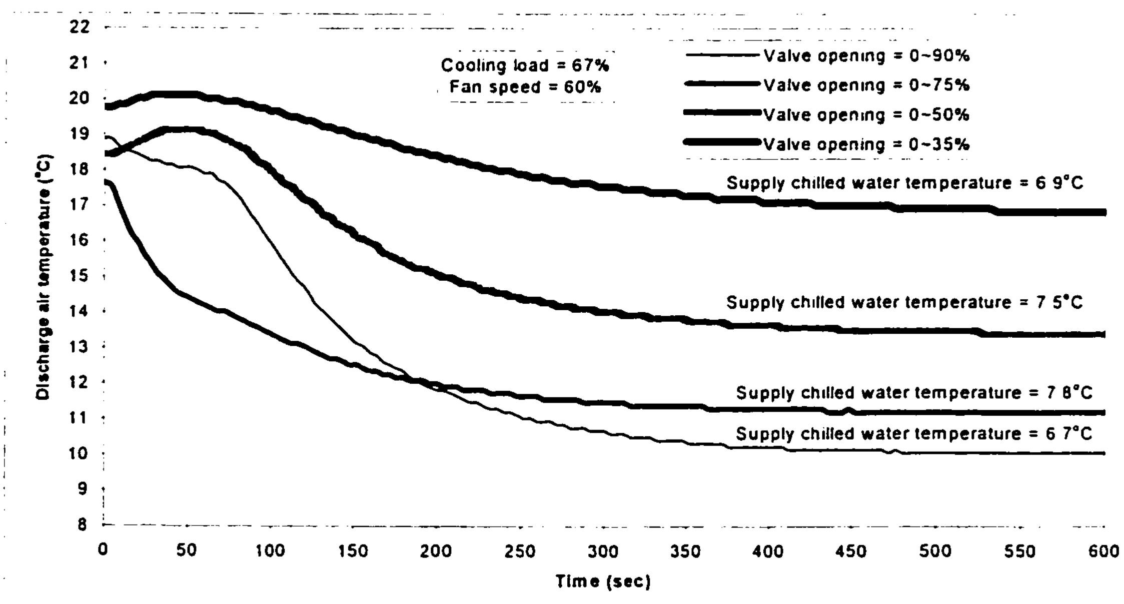

Figure 3.8.b: DAT responses at several different chilled water flow control valve positions

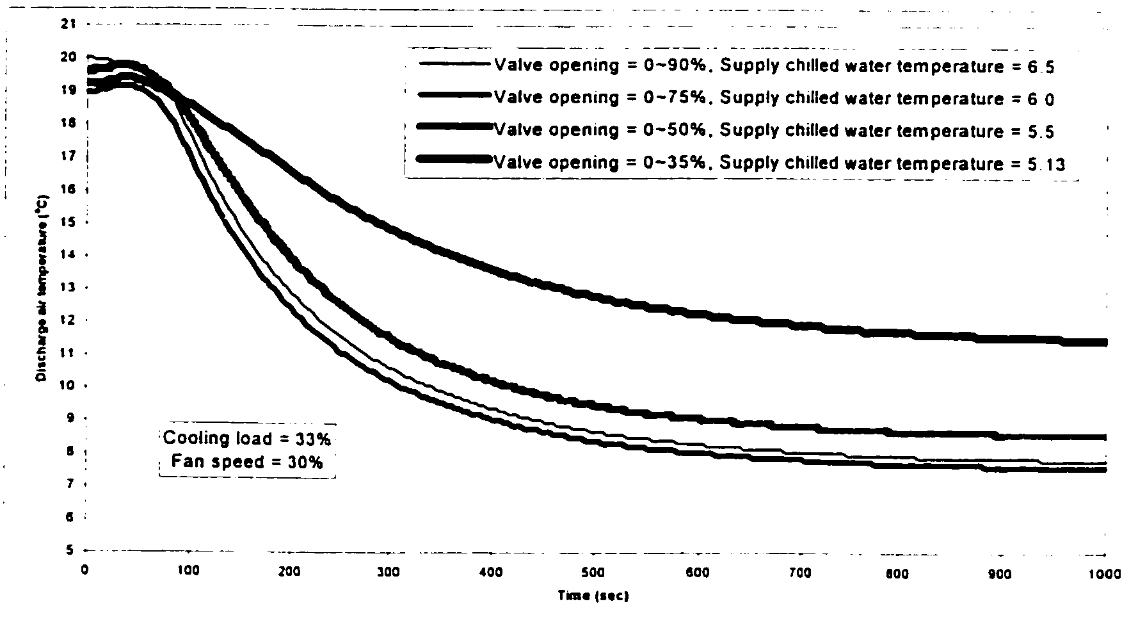

Figure 3.8.c: DAT responses at several different chilled water flow control valve positions

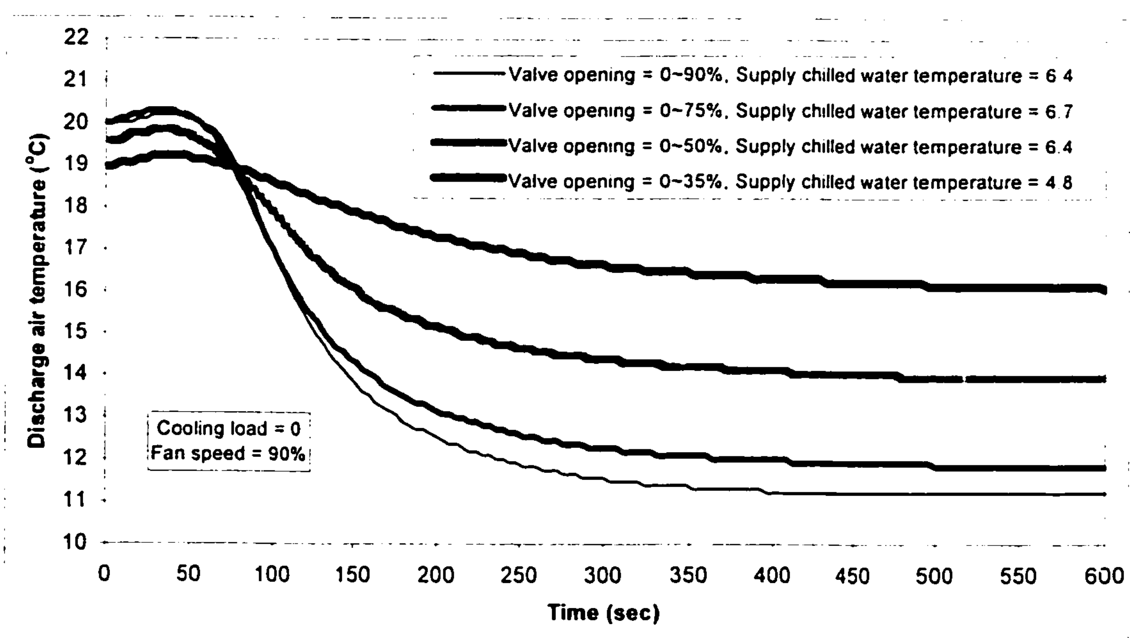

Figure 3.8.d: DAT responses at several different chilled water flow control valve positions

3.4 Dynamic Responses of the Environmental Zones

The room air temperature and relative humidity responses of both rooms 1 and 2 as a function of time are depicted in Figure 3.9.a-b. These open-loop tests were conducted under full load conditions (about 2KW on each room) at maximum fan speed and full-open chilled water valve. It can be seen from the responses that the system reaches steady state in about 1200 seconds. The room1 temperature decreases from 29°C to about 22.7°C. However, the bulk of the temperature decrease occurs in the first 300~450 seconds which corresponds to the steady state time of the DATS. Therefore, it could be reasoned that the slow dynamics between 400~1200 seconds are influenced by the dynamics of the enclosure elements.

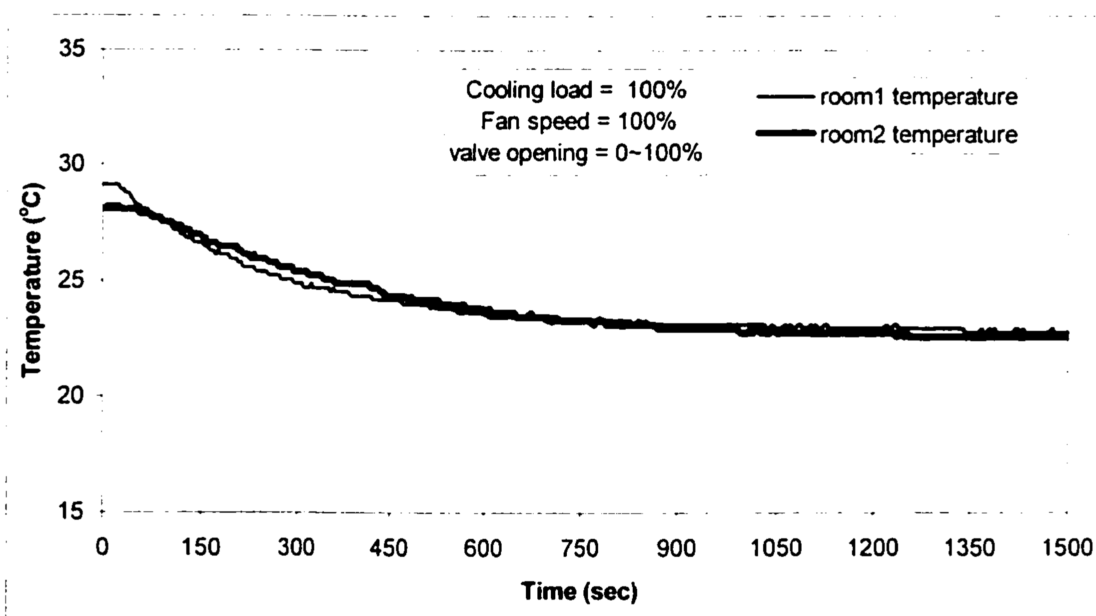

Figure 3.9.a: Temperature responses of air in rooms 1 and 2 (full-load conditions)

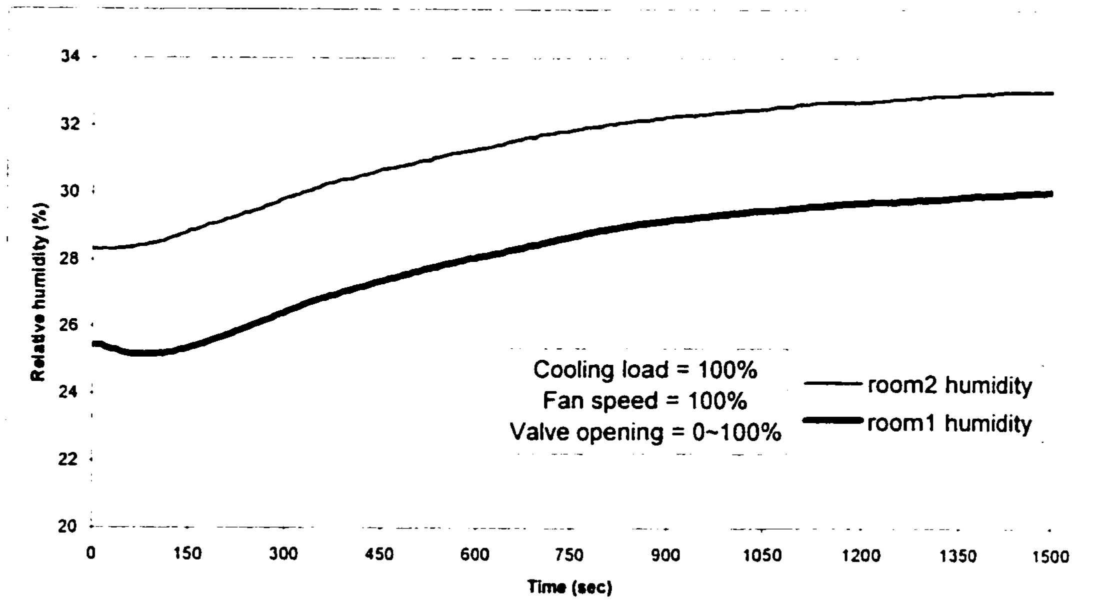

Figure 3.9.b: Relative humidity responses of air in room 1 and 2 (full-load conditions)

Also shown in Figure 3.9.b are the relative humidity responses in room 1 and 2. We note that during this test the humidifiers in the rooms were kept off. As such the change in relative humidity shown in Figure 3.9.b is not significant but nevertheless does change due to dehumidifying coil from 25.5 to 33%.

3.5 Summary and Conclusions

A series of systematic open loop experiments were conducted in order to study the steady state and dynamic responses of three sub-systems: (i) airflow system, (ii) discharge air system and (iii) environmental zones. Through these tests it was possible to identify and quantify the operating range of the VAV system under several different cooling load conditions. Also, the effects of interactions in the airflow rates during VAV system operation were analyzed. From the results presented in this chapter we draw the following important conclusions.

(i) The operating range of the dampers in the VAV boxes 1 and 2 for good flow control was found to be between 40~100%.

(ii) The modulation of dampers in one VAV box influences the airflow rate from its neighboring VAV box. It was found that as much as 16~22% reduction in airflow rate is likely at higher fan speeds. This effect is much more pronounced under part-load (low fan speed) conditions.

(iii) The time needed for the airflow system to stabilize is influenced more by the damper movement than changes in the fan speed.

(iv) The operating range of the chilled water valve for water flow rate control was found to be between 35~90%

(v) The steady state time of the discharge air system was measured and is of the order of 400~450 seconds.

(vi) Both room air temperature and relative humidity responses in rooms 1 and 2 show that about 1200 seconds are needed for the environmental zones to reach steady state.

(vii) A comparison of the steady state times shows that the airflow sub-system is the fastest followed by the DATS and environmental zones. The time scales in order are: 100:400:1200 seconds.

CHAPTER 4

TUNING OF VAV CONTROL LOOPS AND CLOSED LOOP RESPONSES OF THE SYSTEM

4.1 Introduction

There are several methods available for tuning of PID (Proportional-Integral-Derivative) controllers. Among these the more widely used techniques are: (i) Ziegler – Nichols tuning rules [34], (ii) Cohen and Coon [35] and IMC-PID rules [36]. These methods were developed for single-input-single-output systems. The control industry has adopted these same basic rules of tuning but with certain modifications unique to their hardware structure.

In the developed test facility, we have used a modular building controller (MBC) (manufactured by Landis and Steafa) for the operation and control of the VAV system. The MBC has several PID controllers. In the VAV test facility [30] there are six local control loops. They are:

(i) Discharge air temperature control

(ii) Variable speed fan control

(iii) Airflow rate control to room-1 (VAV box-1 damper control)

(iv) Airflow rate control to room-2 (VAV box-2 damper control)

(v) Reheat control in VAV box for room-1

(vi) Reheat control in VAV box for room-2

Each of these loops can be regulated by their respective PID controllers. The results presented in this thesis include all control loops except for the two reheat controllers. Although we tuned the reheat controllers they were not used in the simulated tests because of the fact that humidity control of VAV system is not examined in this thesis.

In this chapter the following issues will be studied: (i) dynamic responses of the individually tuned local feedback control loops, (ii) dynamic responses of the overall VAV system with multiple loop feedback control and (iii) the nature and effect of interactions between control loops in the VAV system.

4.2 Tuning of Controllers

The PID controllers were tuned using the method prescribed in the Landis and Staefa's MBC operation manual [33]. The controller output equation is

$$u(t) = k_p e(t) + k_i \int e(t)dt + k_d \frac{de}{dt} + u_0 \tag{4.1}$$

Where k_p, k_i and k_d are the proportional, integral, and derivative gains of the controller; e(t) is the error in the control variable from its setpoint. It has been shown in many studies [8] that derivative control action, the third term in the right hand side of Equation (4-1), does not improve the control performance in slow systems such as the HVAC systems. To this end, we have implemented PI control actions in the tuning process. Therefore, Equation (4-1) for PI control reduces to

$$u(t) = k_p e(t) + k_i \int e(t)dt + u_0 \qquad (4.2)$$

The controller gains k_p and k_i were computed for each control loop one at a time. This is a two-step process: (a) the calculation proportional band and (b) the determination of integral gain from tuning rules [33]. We have found that the recommended tuning rules [33] did not produce good results in multiple loop system operation. This issue will be addressed in the next chapter.

4.3 Closed-loop Response of the DAT Control Loop

Figure 4.1.a-b show the discharge air temperature response under full load conditions. During this test, the fan speed was held constant at 90% of its maximum value and the room 1 and 2 VAV dampers were full open. At an arbitrary initial condition of DAT equal to 16.7°C the chilled water valve was put into feedback control mode. The resulting closed loop response of the DAT is depicted in Figure 4.1.a. The DAT reaches setpoint (13°C) in about 600S. In the process the maximum deviation from the setpoint was found to be –0.9°C to +0.3°C. How the mass flow rate of chilled water to the coil was regulated during this test is depicted in Figure 4.1.b. Note that the chilled water valve remains fully open initially due to large error from the setpoint and reaches steady state. In steady state the mass flow rate of chilled water under full load conditions was 4.7 GPM.

It is worth noting the delay between high and low values of chilled water mass flow rate and corresponding DAT responses. That is there is a delay of 150S in the DAT response from the time the valve initiates a change in the water flow rate as shown in Figure 4.1.a-b.

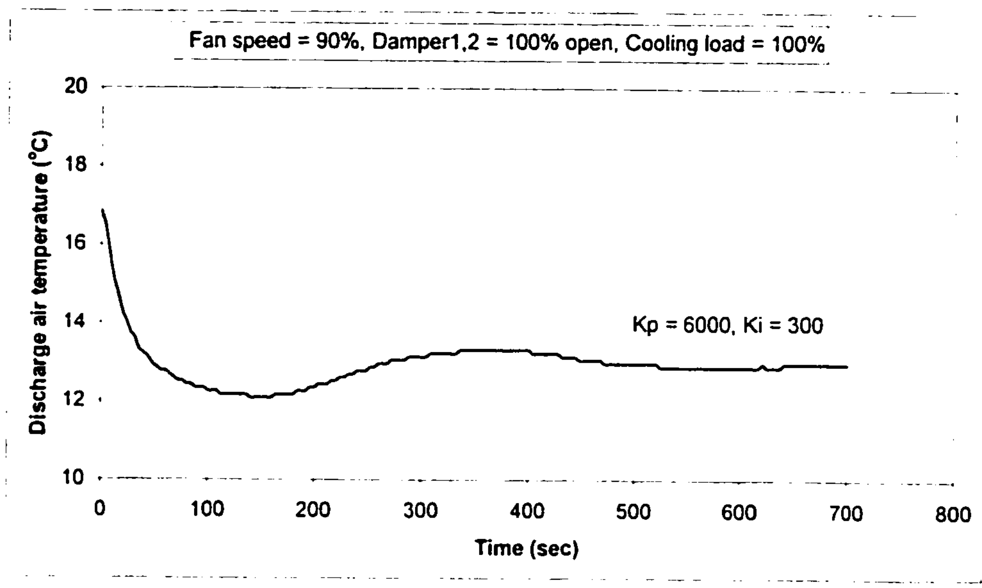

Figure 4.1.a: Discharge air temperature response under full load conditions

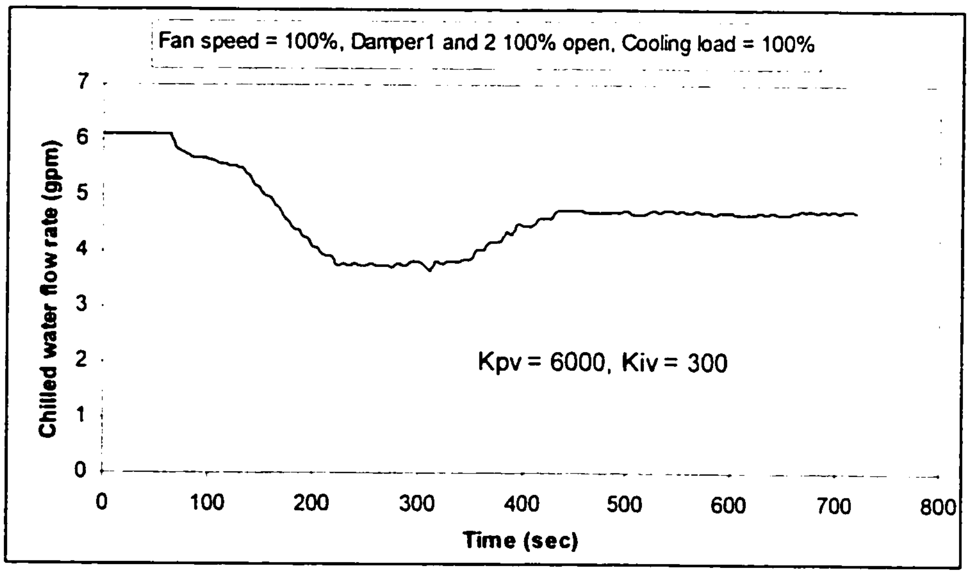

Figure 4.1.b: Chilled water flow rate response under full load conditions

4.4 Closed-Loop Response of the Variable Speed Fan Control Loop

Airflow rates in VAV systems are modulated by either VAV box damper control or by fan speed control or by simultaneous control of the VAV box dampers and the fan speed. Here, we present fan speed control loop response under full load conditions with VAV box dampers 1 and 2 in full open position. The fan speed controller was configured to receive feedback signals from the pressure sensor located in the duct carrying airflow to room2. This pressure sensor is the one farthest from the fan and therefore is used to ensure sufficient airflow rates to rooms away from the fan.

With an arbitrarily chosen setpoint of 0.0375 in WG (velocity pressure) the fan loop was turned on. The resulting closed loop response of the velocity pressure in duct-2 is depicted in Figure 4.2. The maximum overshoot is 66% above the setpoint and the duct-2 pressure reaches steady state in about 40S. The fan loop is the fastest of all control loops in the VAV system. From the response shown in Figure 4.2 we note that large step changes in the pressure setpoint could cause severe overshoot problems and even oscillations if the controller gains are not updated.

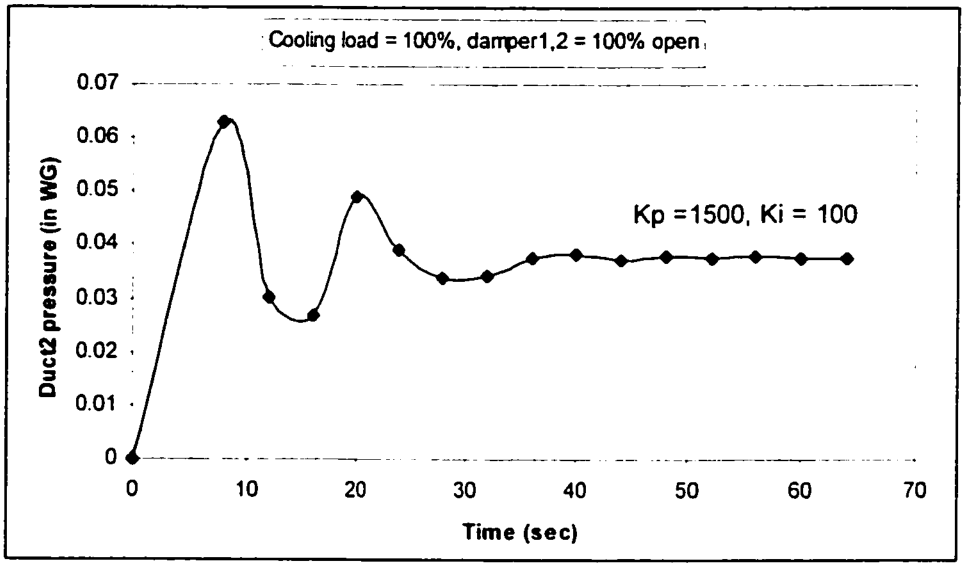

Figure 4.2: Closed loop response of the velocity pressure in duct-2.

4.5 Closed-Loop Response of the Zone Damper Control Loops

Airflow control at the zone level by the use of VAV boxes (damper control) is a widely practiced strategy in buildings. Here, we have examined the closed loop control of room 1 and 2 dampers while the fan speed was held constant at 90%. The damper control loops were configured to simulate the pressure-independent control strategy meaning the feedback signals to the damper controllers were transmitted by their respective pressure sensors. Thus the room 1 and 2 dampers are modulated to track their respective pressure setpoint.

Figure 4.3 shows the supply airflow rate responses to room 1 and 2 in damper control mode. Note that airflow rates stabilize in about 120S. The maximum variation in room1 airflow rate was found to be 10% of the setpoint (250cfm). The zone closer to the fan (room1 in the test facility) experience somewhat higher fluctuations in airflow rates than those farther from the fan. This is consistent with the fact that the zones closer to the fan receive air at higher pressure than those farther from the fan. The airflow rate responses also reveal that the steady state time of the damper control loop is greater than the fan speed control loop in stabilizing the airflow rates in the system following a step change in the pressure setpoints.

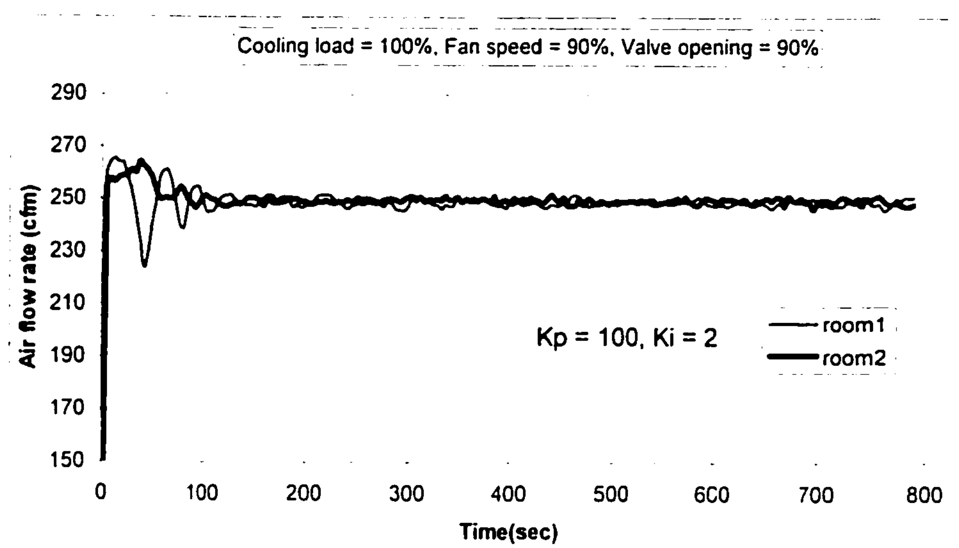

Figure 4.3: Supply airflow rate responses to room 1 and 2 in damper control mode.

4.6 Closed Loop Response of the Overall VAV System

Having examined the closed loop responses of the local control loops one at a time, we now focus on the overall VAV system with several control loops working simultaneously. To this end, the operation of VAV system with the following control loops was studied: (i) the DAT control, (ii) airflow rate control to room1 (VAV damper-1 control), and (iii) airflow rate control to room2 (VAV damper-2 control). The simulation tests were conducted under full load conditions. In other word, room 1 and 2 were subjected 2KW each of sensible cooling load and fan speed was held constant at 90% of its maximum value. Note that the room 1 and 2 dampers were set to pressure-independent control mode. The resulting closed loop responses of the VAV systems are shown in Figure 4.4.a-c.

Figure 4.4.a shows the DAT responses. The DAT rises to 15.2°C and dips to 12.1°C and eventually reaches the setpoint value equal to 13°C. The time taken by the DAT to reach steady state is about 1200S. A comparison between Figure 4.4.a and 4.1.a shows that the DAT control response isolated from the other control loops (Figure 4.1.a) is faster and gives less overshoot compared to the case when DAT control is operating in a multiple loop control environment (Figure 4.4.a).

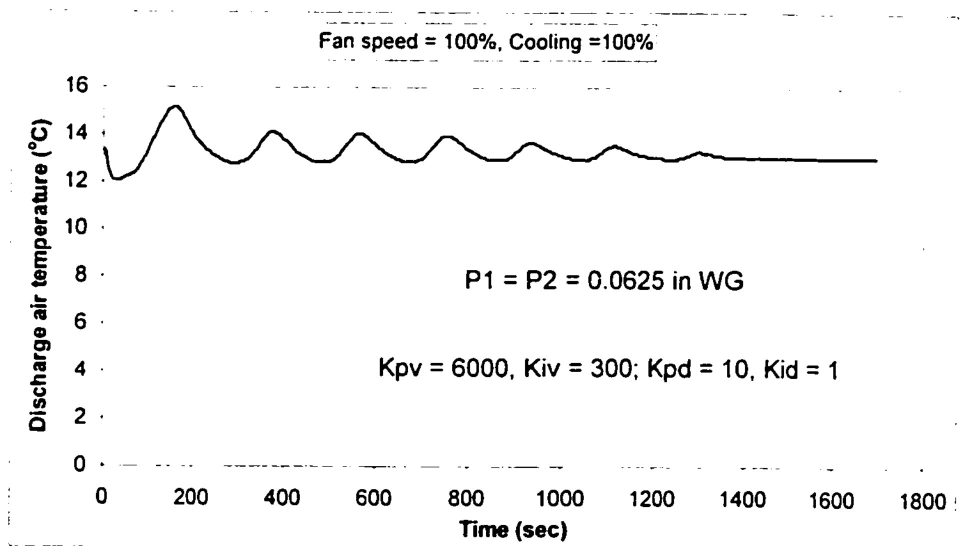

Figure 4.4.a: Discharge air temperature response

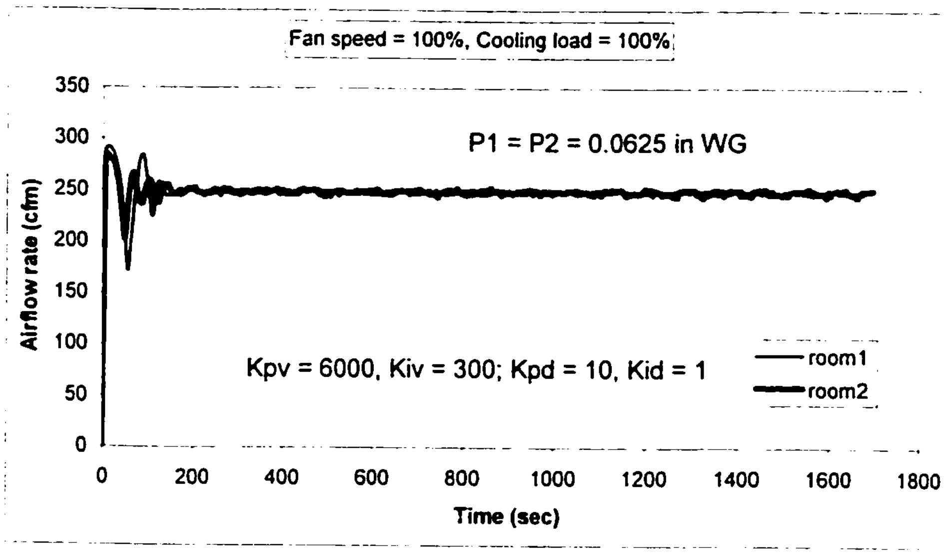

Figure 4.4.b: Supply airflow rate responses to room 1 and 2.

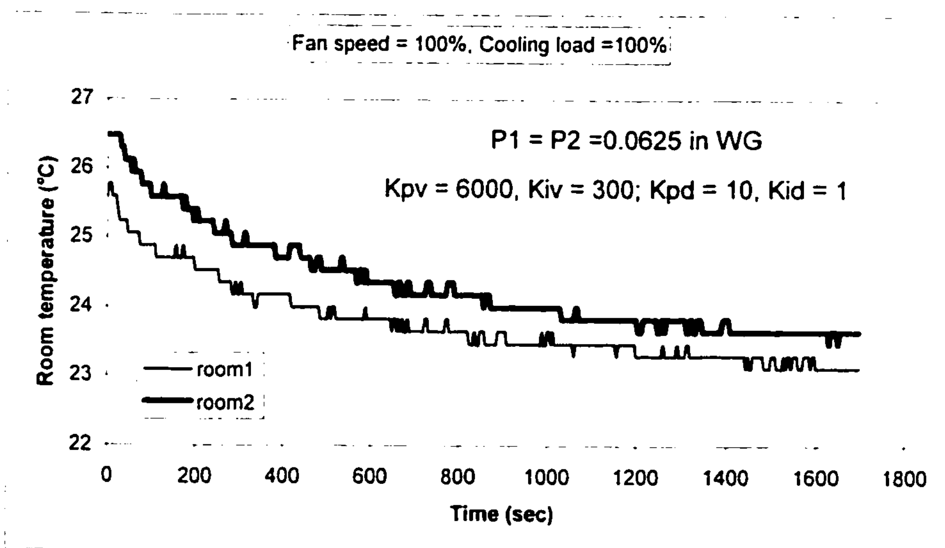

Figure 4.4.c: Room 1 and 2 air temperature responses.

The supply airflow rate responses to room 1 and 2 are depicted in Figure 4.4.b. The damper control loops stabilize fast and deliver between 245 to 250 cfms to room 1 and 2. Under these conditions, the resulting room air temperature responses are depicted in Figure 4.4.c. Both room 1 and 2 air temperature reach steady state in about 1500S. Note that room1 air temperature remains about 0.5°C below room2 air temperature.

4.6.1 The Effect of Setpoint Changes on the VAV System Responses

During the operation of the VAV system, it is realistic to assume that operating setpoint of the VAV damper loops and the DAT loop could be varied to match the changing cooling loads. In order to simulate these cases, two cases were examined: (i) a setpoint change in airflow rates, and (ii) a step change in the DAT setpoint.

Figure 4.5.a-c show the VAV system responses subject to a 20% decrease in airflow rates to room 1 and 2. To simulate this test the system was first allowed to reach steady state and then a step change was introduced. It can be seen from Figure 4.5.a that following the step change in the airflow rate at 1280S, the dampers react rapidly; the airflow rates dip to about 186cfm before reaching steady state value of 210cfm. The effect of this step change on the DAT can be seen in Figure 4.5.b. The DAT rises to 14°C (1°C overshoot) even though the DAT setpoint was not changed. The resulting room air temperature responses are depicted in Figure 4.5.c, which are not significantly affected by a 20% decrease in the airflow rates. The only thing that is noticeable is a slight rising trend in room 1 and 2 temperatures from 1280S onwards which is quite small (+0.2°C).

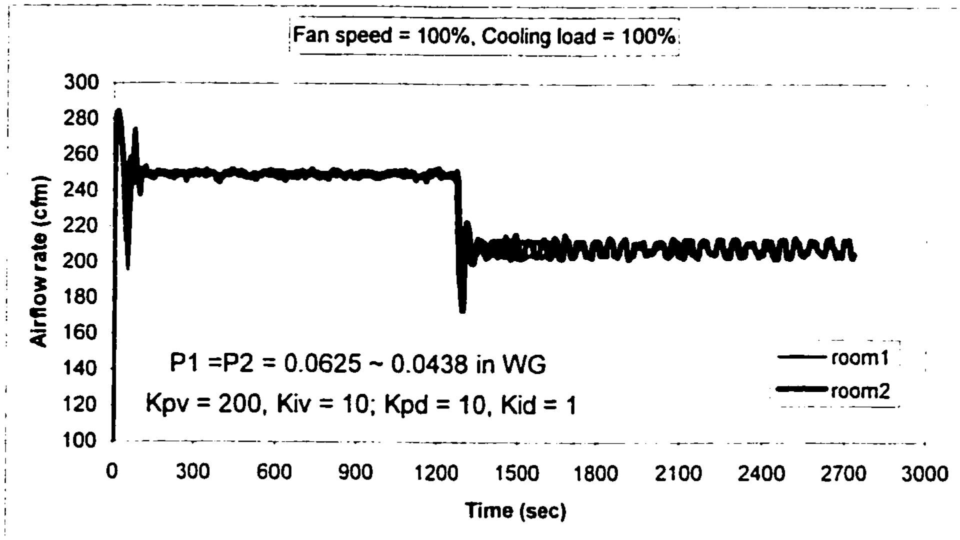

Figure 4.5.a: Airflow rate responses subject to a 20% decrease in airflow rates to room 1 and 2.

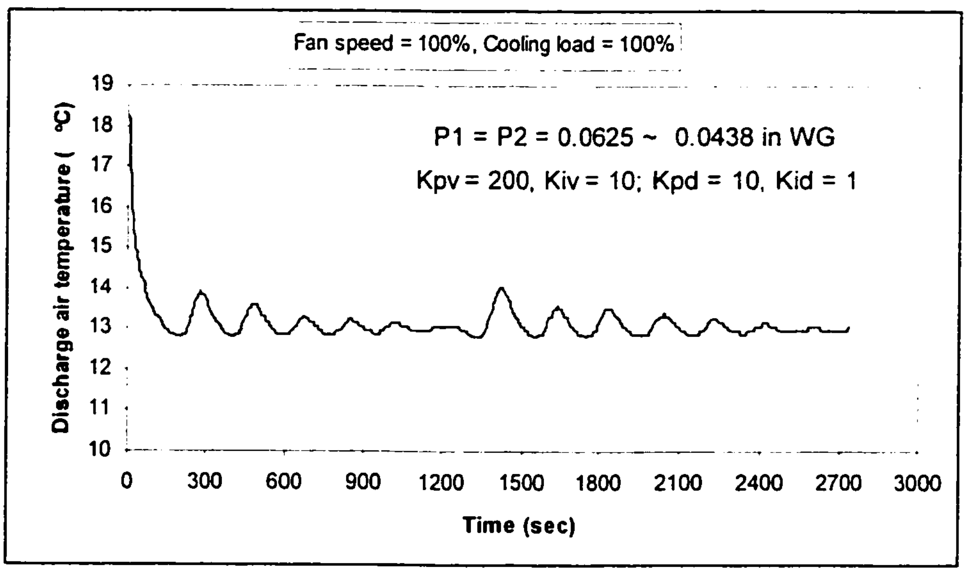

Figure 4.5.b: DAT responses subject to a 20% decrease in airflow rates to room 1 and 2.

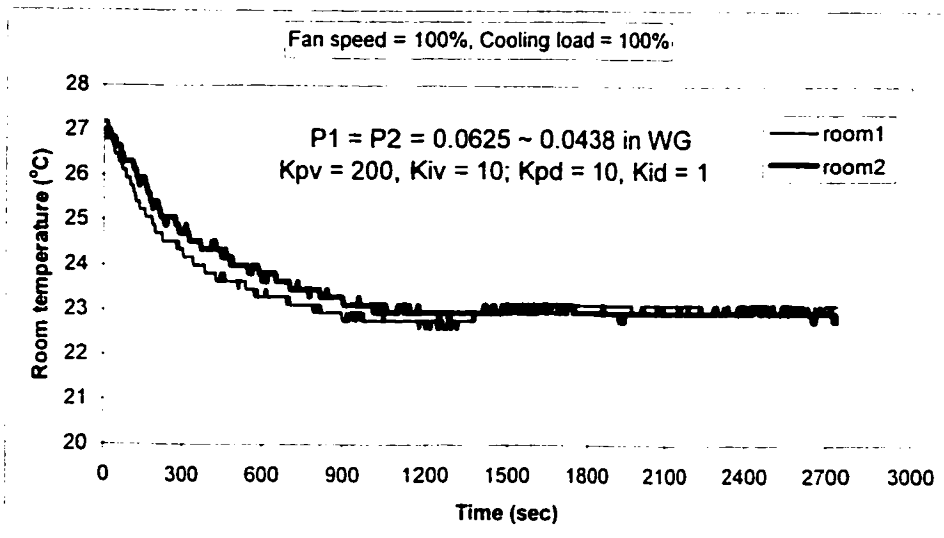

Figure 4.5c: Room temperature responses subject to a 20% decrease in airflow rates to room 1 and 2

The effect of a 1°C step change in the DAT setpoint (from 13°C to 14°C) and the corresponding closed loop responses are depicted in Figure 4.6.a-c. Note that following the setpoint change at 1460S (Figure 4.6a) the DAT overshoot and in the process reaches as high as 16.6°C before settling towards the new setpoint of 14°C. The corresponding airflow rates responses to room 1 and 2 are depicted in Figure 4.6.b and the room air temperature in Figure 4.6.c. It can be noted that both airflow rates remained constant. However, room air temperatures do show a slight increase after 1460S since somewhat warm air is now being supplied to rooms 1 and 2.

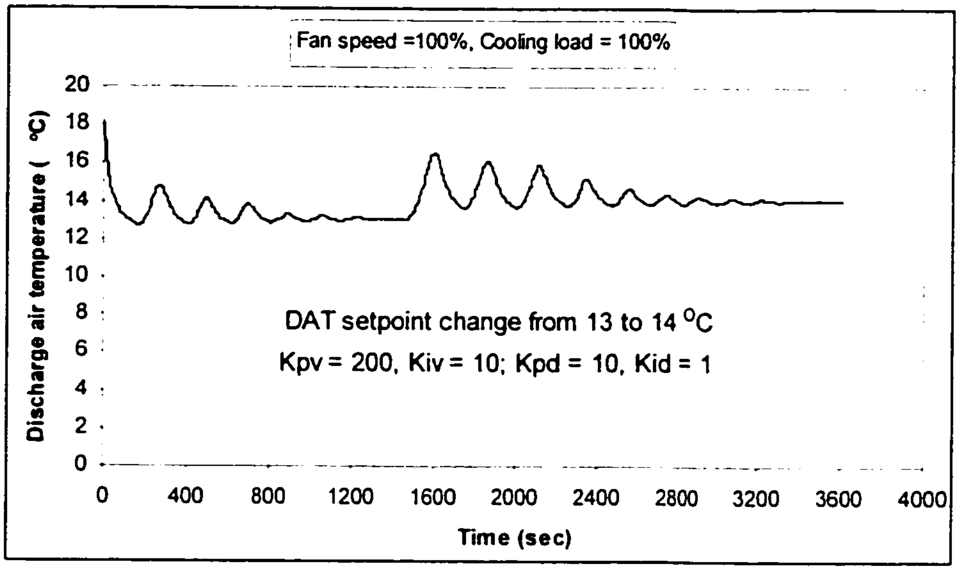

Figure 4.6.a: DAT response subject to 1°C step change in the DAT setpoint.

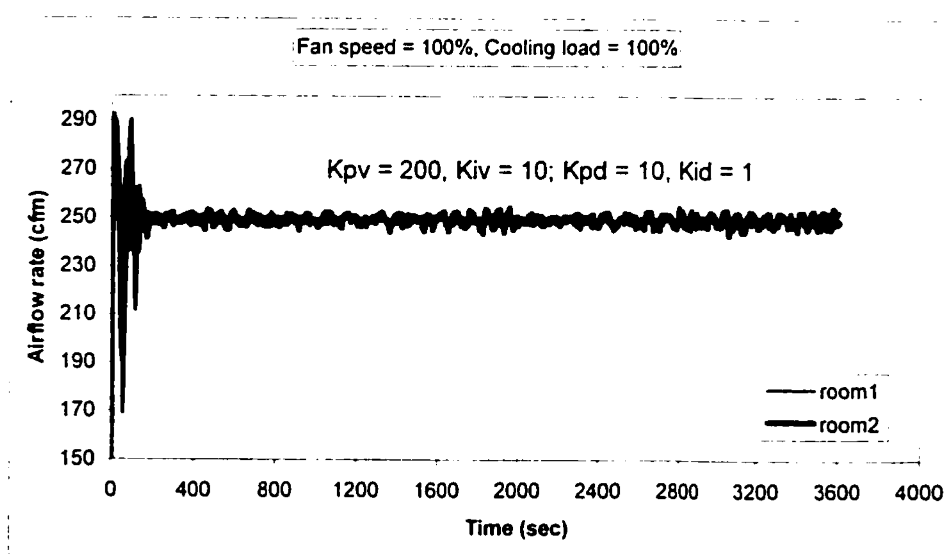

Figure 4.6.b: Airflow rate responses subject to 1°C step change in the DAT setpoint.

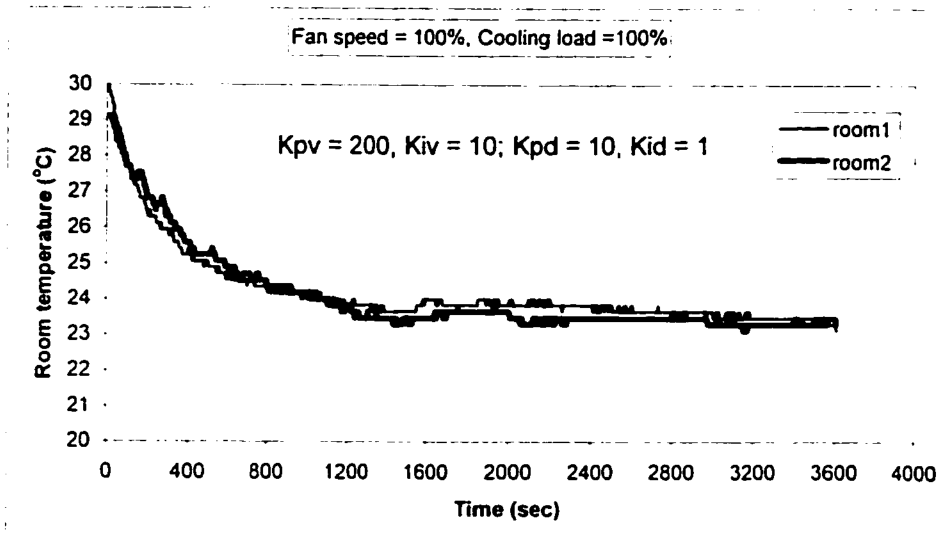

Figure 4.6c: Room temperature responses subject to 1°C step change in the DAT setpoint.

4.7 Control Loop Interactions

From the results presented in Figures 4.5.a-b it can be stated that the damper loops 1 and 2 affect DAT control loop. On the other hand, the DAT control loop does not affect the damper loops. This is expected since a change in airflow rates in the system (and therefore through the cooling coil) has an effect on the DAT and not the other way around.

To examine the severity of this interaction, we have simulated a test case in which a 30% step change in the supply airflow rate as shown in Figure 4.7.a was introduced at 1320S. Following this step change, airflow rates decrease to about 106cfm and quickly restored to the new setpoint of 175cfm. However, what is of interest is the effect of such a step change on the DAT response. As shown in Figure 4.7.b the DAT, which was stable before the application of the step change in the airflow rate setpoint, goes into sustained oscillatory response, which should be avoided to protect the actuators. From this result we conclude that VAV damper loops could significantly interact with the DAT loop.

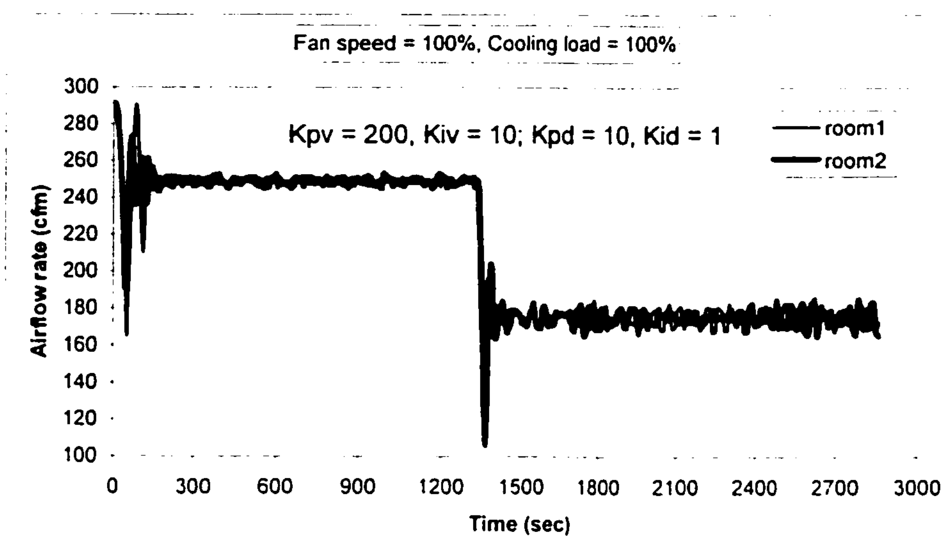

Figure 4.7.a: Airflow rate responses subject to a 30% step change in the supply airflow rate

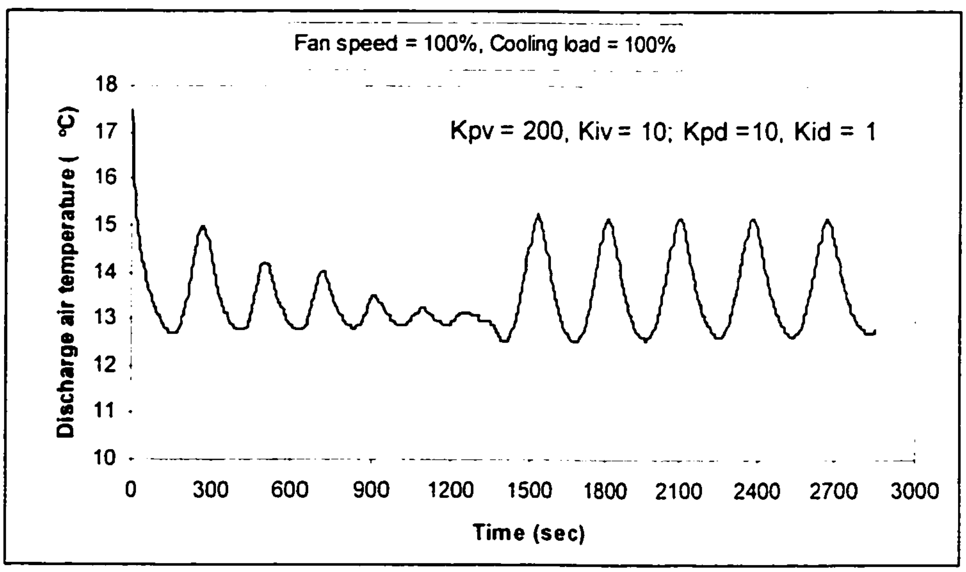

Figure 4.7.b: DAT responses subject to a 30% step change in the supply airflow rate

From the results presented in Figure 3.1.a-b it was noted that under open loop conditions, the effect of VAV box1 on the other VAV box could be as much as 16%. To study whether or not this holds true in closed loop control operation, we have conducted several tests. In the tests conducted, the VAV system was first allowed to reach steady state. At this time, a step change in the setpoint of one VAV damper was given and the corresponding airflow rates were recorded. The results are plotted in Figures 4.8.a-b (fan speed = 90%) and Figures 4.9a-b (fan speed = 60%).

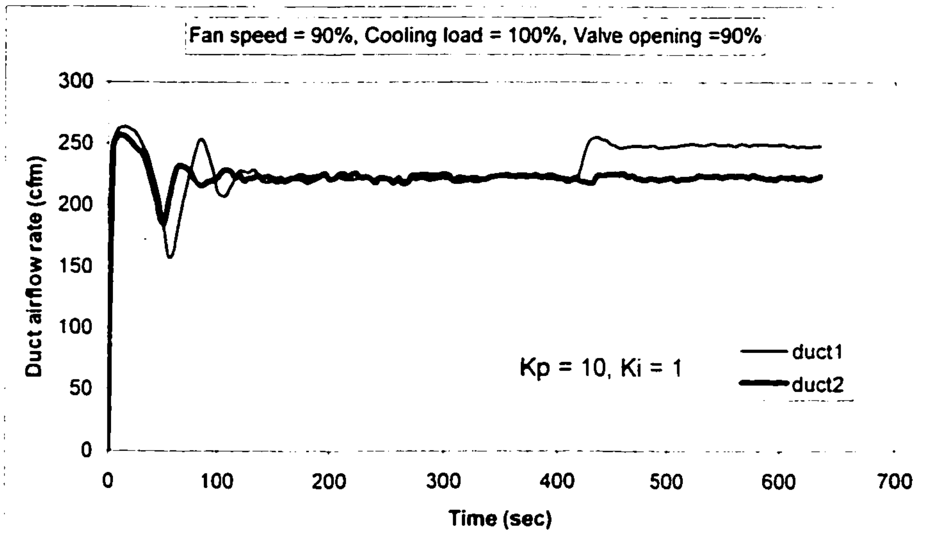

Figure 4.8.a: Duct airflow rate responses subject to a step change in the setpoint of damper-1 (Case 1).

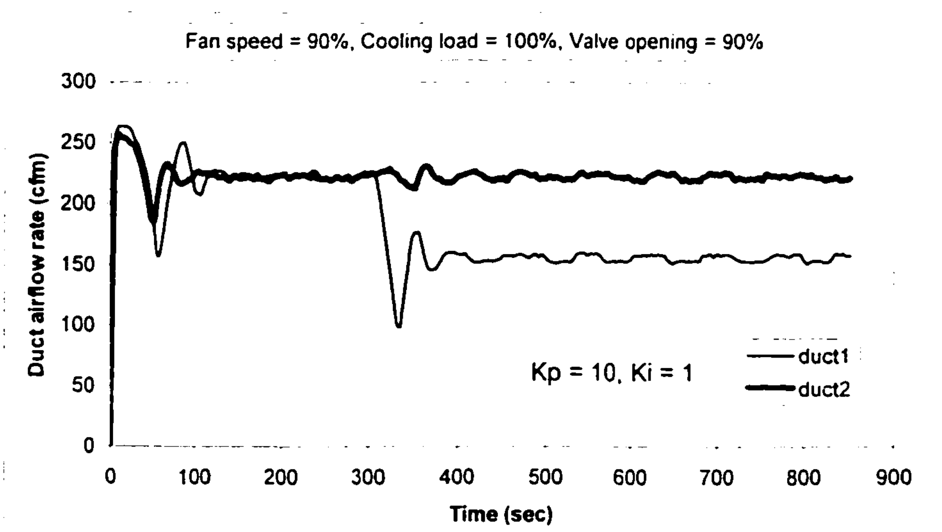

Figure 4.8.b: Duct airflow rate responses subject to a step change in the setpoint of damper-1 (Case 2).

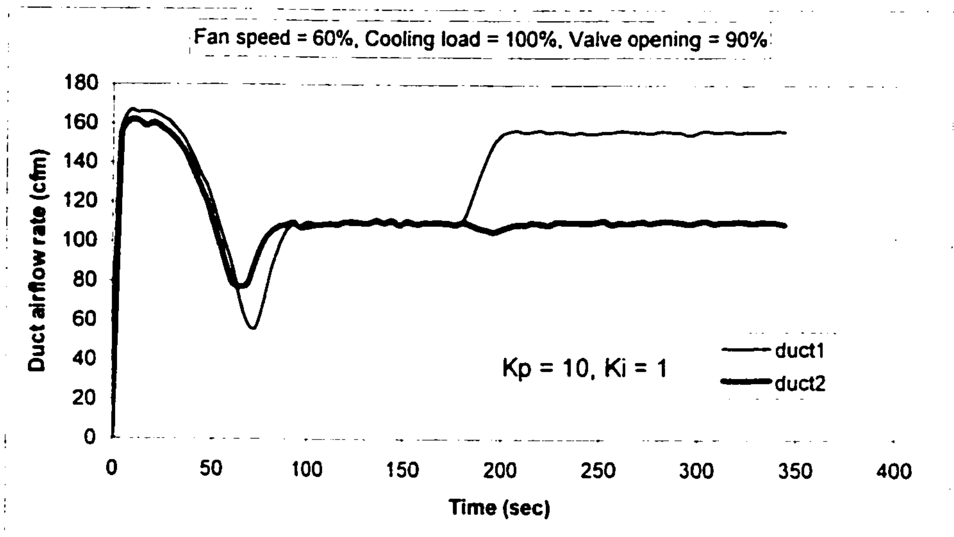

Figure 4.9.a: Duct airflow rate responses subject to a step change in the setpoint of damper-1 (Case 3).

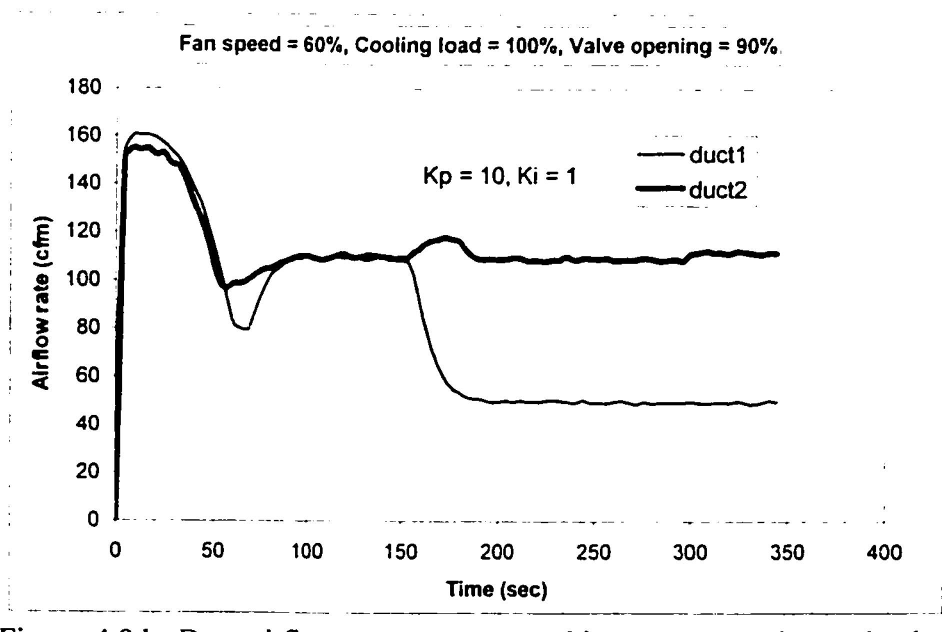

Figure 4.9.b: Duct airflow rate responses subject to a step change in the setpoint of damper-1 (Case 4).

As shown in Figure 4.8a, a 10% increase in room1 airflow rate setpoint caused a 2% increase in room2 airflow rate. A 30% decrease in room1 airflow rate setpoint, caused a 5% change in room2 airflow rates. Results from the low speed case (fan speed = 60%) depicted in Figures 4.9a-b also show some fluctuations in the airflow rates to one room when the neighboring damper tracks its setpoint. Since both VAV boxes are now operating in feedback mode, the effect of one loop on the other in terms of airflow rates is not as significant as was the case under open loop operating conditions.

4.8 Summary and Conclusions

The local loops were tuned following the method described in [33]. The closed loop responses of fan loop, DAT loop, damper loops due to step change were measured. The steady state times of the control loops respectively are 40:120:600:1500S. The research show that under closed loop control the interactions between room 1 and 2 VAV dampers range between 2 to 5%. It was found that a step change in VAV damper loop could cause significant oscillations in DAT loop. Therefore along with the tuning of VAV damper loops the DAT loop must also be retuned otherwise the DAT could become unstable.

CHAPTER 5

COMPARISON OF CONTROL STRATEGIES AND CONTROL LOOP OPTIMIZATION

5.1 Introduction

The VAV Test Facility [30] enables comparison of two different control strategies, one in each room of the test facility under similar load conditions. To this end, the results obtained with (i) Pressure-Independent-Control (PIC) and (ii) Pressure-Dependent-Control (PDC) strategies will be examined in this chapter. Furthermore, several experiments were conducted to determine near-optimal range of controller gains in both single and multi-loop operating modes.

5.2 Simulation of PIC and PDC Strategies

To recapitulate, in a PIC strategy static pressure signal is used to control the VAV box. On the other hand, in a PDC strategy, room temperature signal is used for controlling the VAV box dampers. There are no studies which show clearly the advantages of one strategy over the other [37]. It is, however known that most operating engineers prefer the use of PIC over PDC for the reason that it gives somewhat stable operation of VAV box controls.

To investigate the operation of VAV system under PIC and PDC modes, the PIC strategy was implemented to control the VAV box-2 supplying air to room-2, and PDC to control the VAV box-1. During this test the following operating conditions were maintained: cooling load = 2KW sensible in each room; Fan speed = maximum operating speed (1650 rpm); Chilled water supply temperature = 6~7°C; Room-1 temperature setpoint = 23°C and pressure setpoint to VAV box-2 = 0.063 in WG (velocity pressure); The discharge air temperature setpoint = 13°C.

Note that it is easier to choose temperature setpoint as required in PDC strategy than the pressure setpoint needed in implementing the PIC strategy. The VAV system responses under the above noted operating conditions and setpoints are depicted in Figures 5.1.a-d. Figures 5.1.a-b show the room temperature responses and corresponding airflow rates to rooms 1 and 2. As shown in Figure 5.1.a, PDC strategy maintains room-1 temperature very close to the setpoint (23°C). From about 1100S onwards more or less a perfect setpoint control is achieved. On the other hand, in the PIC strategy used in room-2 the room temperature is allowed to float as such it drifts as a function of load. This means the choice of pressure setpoint is critical in PIC. One method to mitigate this issue is to define pressure as a function of temperature as practiced in some VAV systems, since such a function even if defined may not be optimal leading to concerns about loss of energy.

The airflow rate responses under these conditions are plotted in Figure 5.1.b. Notice that after some initial transients over the first 200S, airflow rate to room-2 attains steady state and is held constant thereafter. This is to be expected since PIC strategy tracks the pressure setpoint. The PDC strategy on the other hand keeps room-1 temperature constant while modulating the airflow rate to room-1. As shown in Figure 5.1.b airflow rate to room-1 varies between 300 to 200 cfms and is continuously modulated. It is interesting to note that even though the airflow has fast response characteristics (steady state time less than 100S), when coupled with room thermal dynamics it too slows down with a cycle time of 1200S. This latter characteristic is critical in the sense that following a change in load the PDC strategy is susceptible and could become unstable. For this reason, PIC is a safer alternative to PDC. However, if PDC gains can be updated using advanced control design techniques then it may be possible to achieve better performance using PDC. Such design methods are part of other research projects presently, under investigation [38].

In order to evaluate PIC and PDC strategies, not only temperature regulation but energy used should also be considered. To this end, the heat extraction rates (cooling rates) of room 1 and 2 were computed using the measured data and these are plotted in Figure 5.1.c. Disregarding the initial transients, it appears that both control strategies might use nearly the same rate of energy. However, careful experiments with precise load control are needed to accurately quantify the results. Additional sensors are being installed to carry out such measurement in the future.

From the results presented in Figure 5.1.a-c it may be concluded that when precise room temperature control is required it is preferable to use PDC strategy with good online adaptive strategies to optimize the loop performance. On the other hand, if room temperature variations are acceptable then PIC strategy appears to be easier to implement provided that some reasonable means of updating the pressure setpoint is available a priori from past performance data.

Also shown in Figure 5.1.d is the discharge air temperature (DAT) response. It should be noted that several trial runs were needed to achieve stable response from the DAT loop. Note that the proportional gain for the valve control one that produced the response shown in Figure 5.1.d is between 1/15 to 1/30 of the value used when both VAV boxes were operated using PIC strategy.

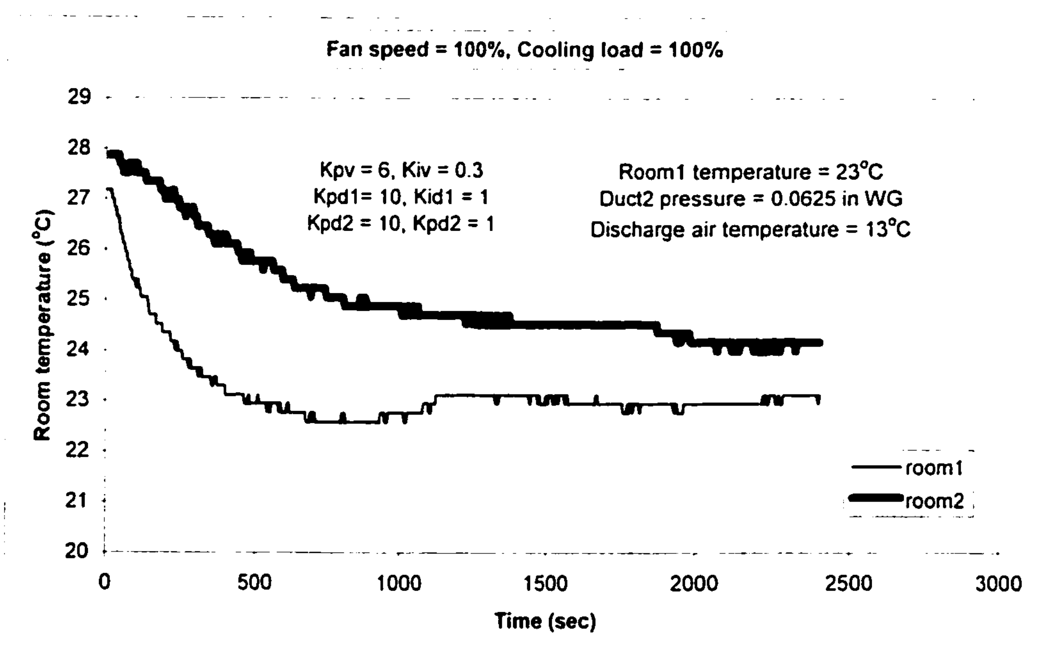

Figure 5.1.a: Room 1 and 2 temperature responses

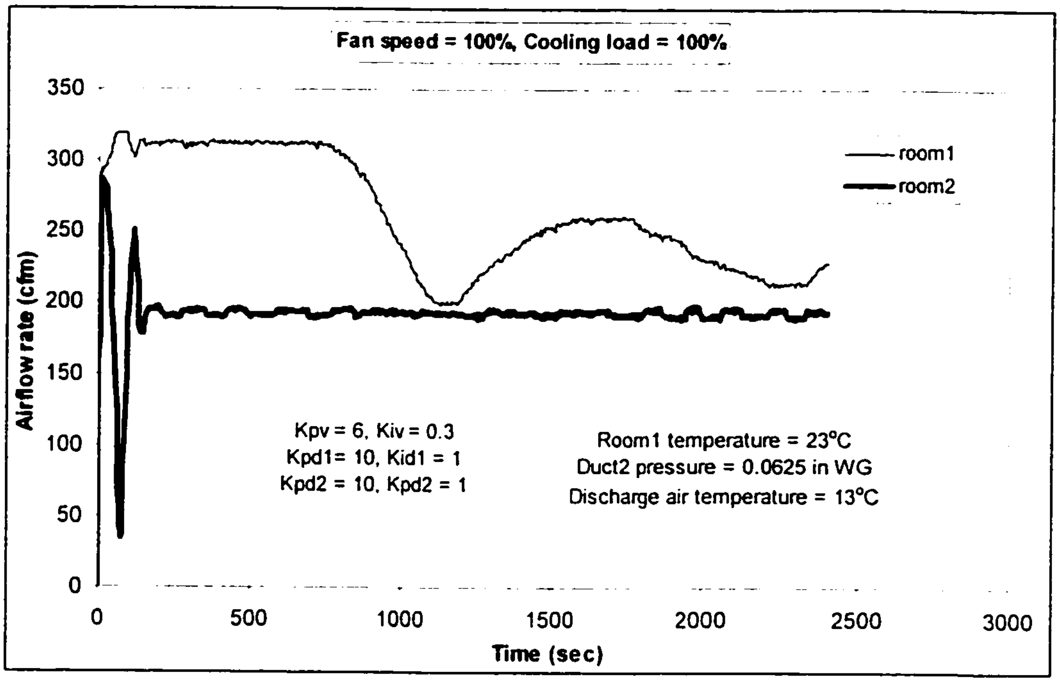

Figure 5.1.b: Responses of airflow rates to rooms 1 and 2

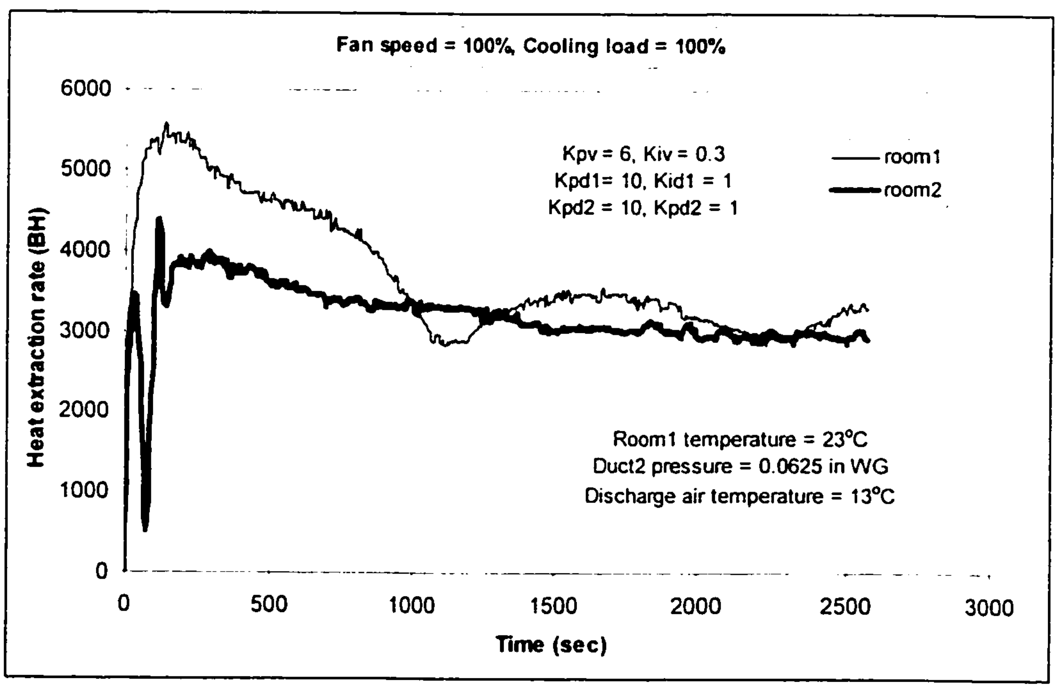

Figure 5.1.c: Heat extraction rates (cooling rates) of room 1 and 2

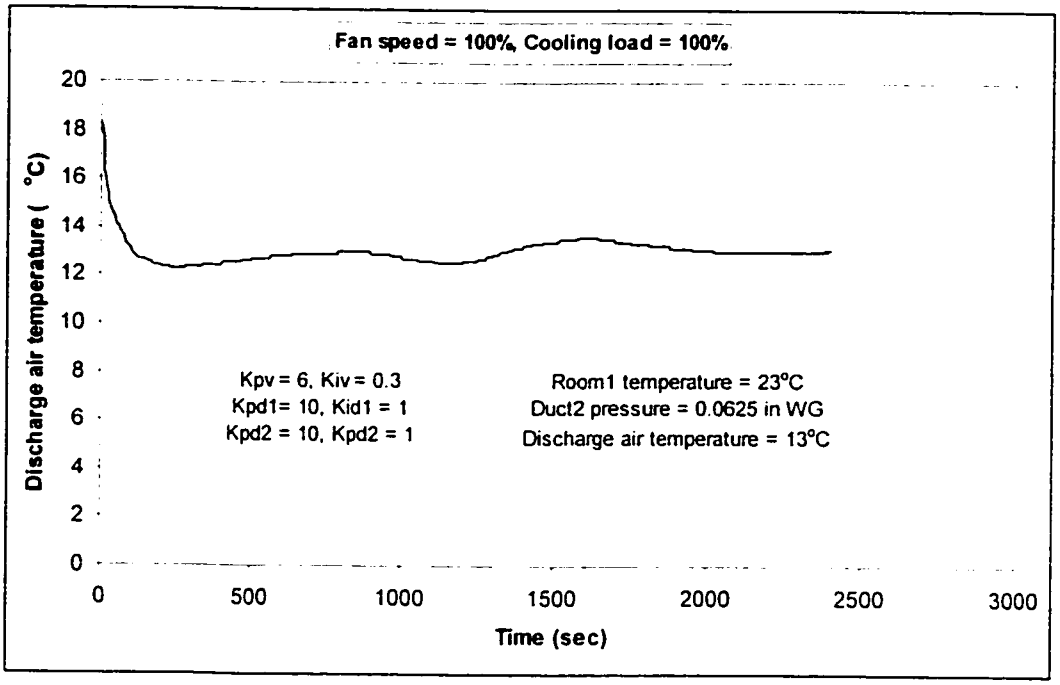

Figure 5.1.d: Discharge air temperature response

5.3 Optimal Operation of Local Control Loops

By conducting a series of experiments in which the controller gains were varied from small to large values, output responses of the local control loops were recorded. The steady state time and/or percent overshoot were used as output parameters. The results obtained from such experiments are summarized in the following.

Shown in Figures 5.2.a-b are the curves of percent overshoot and steady state time as a function of controller gain for the Discharge Air Temperature Control (DATC) loop. The results depict the loop performance without closed-loop interactions from other VAV control loops. In other words, the other control loops were set in the open-loop mode with only the DATC in the closed-loop control mode of operation. This is sometimes referred to as optimization of one-loop-at-a-time in the literature. Although optimal gains obtained from such tests may not be optimal when all control loops are working simultaneously, nevertheless the technique is the easiest and gives good range of controller gains. As shown in Figures 5.2.a-b controller gains in the range between 4000~6000 produced lower overshoot and faster response. The integral gain was set at 5% of the proportional gain for the results shown in the Figure 5.2.a-b.

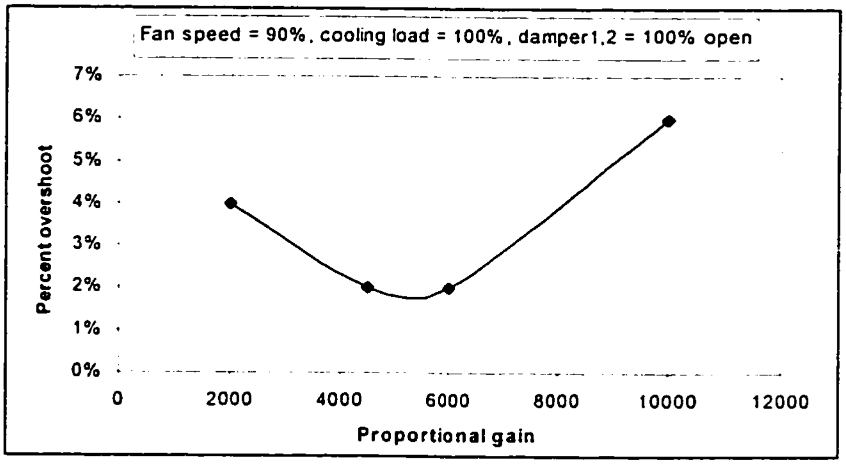

Figure 5.2.a: Percent overshoot to controller gain for DATC loop

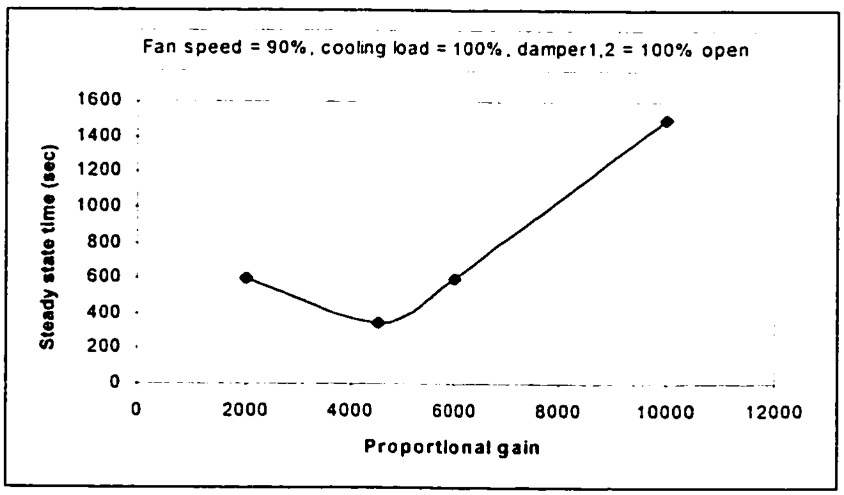

Figure 5.2.b: Steady state time to controller gain for DATC loop

Similar tests were conducted with VAV box damper control loops one-at-a-time. Both damper loops gave nearly same results as such only one set is shown in Figure 5.3. The results show that a wide-range of damper control gains (between 10~120) gives fast response. In other words, the damper control loop was found to exhibit stable response

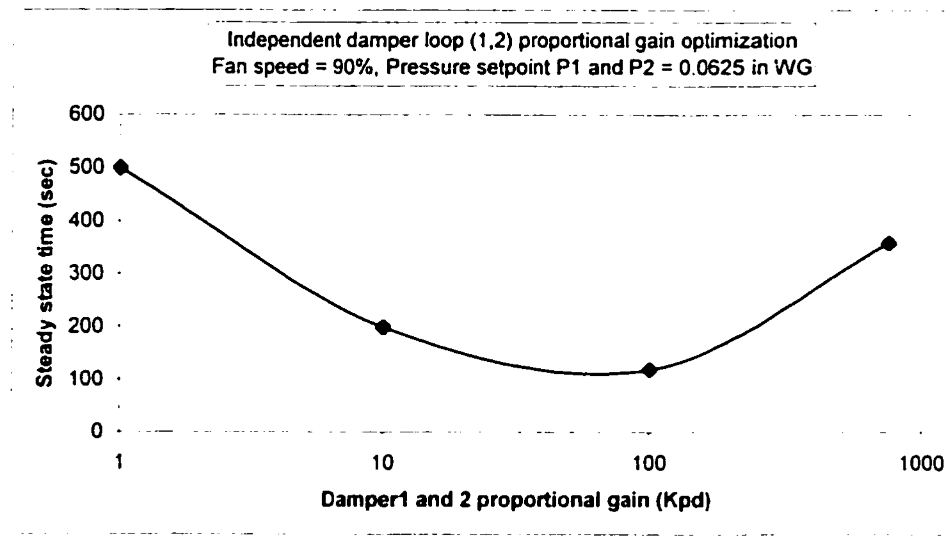

Figure 5.3: Steady state time to controller gain for VAV Box damper control loops

over a wide range of gains when operated in PIC mode. This clearly shows the advantage of PIC strategy in so far as maintaining stable operation over a wide operating range is concerned. Tests conducted on the damper control loops using PDC strategy gave a very narrow range (k_{pv} = 4~6). Even then the control loops were very sensitive to small variations. The results are omitted since they were found to be not reliable.

5.4 Optimal Operation of Multiple Control Loops

The results presented in Chapter 4 showed that there is strong interaction between VAV damper control loop and the DATC loop. In other words, tuning or updating one control loop affects the performance of the other control loop. Therefore, both of these control loop gains should be tuned or updated simultaneously. This is somewhat more difficult to achieve compared to the one-loop-at-a-time approach used earlier. To alleviate this difficulty an alternate method was employed to seek optimal gains. In the method used, the damper control gains were kept constant, a near optimal value of $k_{pd} = 10$ from Figure 5.3 was selected. Using this optimal value, several tests were conducted each time varying the valve proportional gain for one pressure setpoint. From this set, that value of k_{pv}, which gave good DAT response, was selected. By repeating the tests at several different pressure setpoints the set of good k_{pv} values were obtained. These are plotted in Figure 5.4 as a function of pressure setpoint. It is apparent that as the airflow rates are increased somewhat higher k_{pv} values could be used to achieve good control. Note that the optimal values of k_{pv} in a multiple-loop control mode are significantly lower than those in a one-loop-at-a-time operating mode.

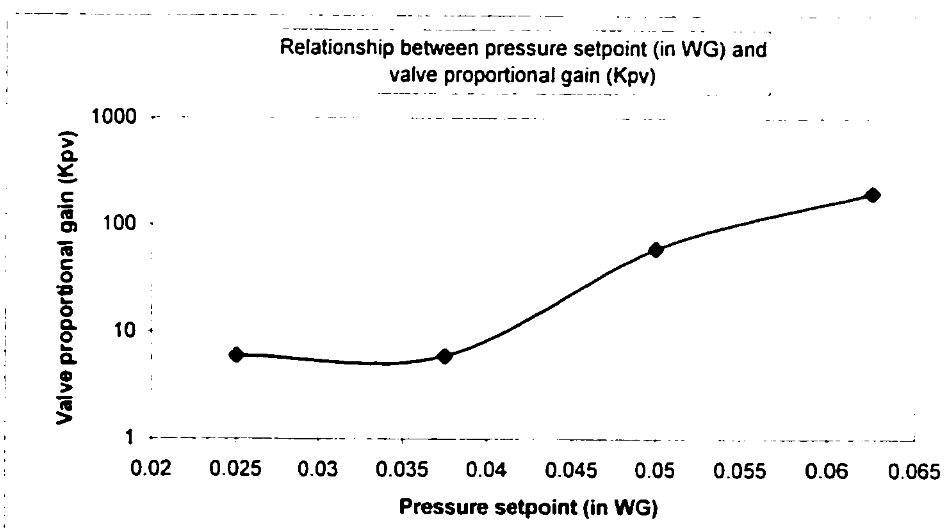

Figure 5.4: Relationship between proportional gain and pressure setpoint.

5.5 Pressure Setpoints for PIC Operation

Although PIC strategy gives good stable operation, problems still remain in selecting appropriate pressure setpoints. Since the airflow rates to the zones are function of cooling load, discharge air temperature and zone temperature, seeking a optimal balance between these variables is necessary. Here, we describe the method used to achieve such a balance in the experiments conducted and present the results.

During the experiments the DAT setpoint was kept constant at 13°C. The optimal gains for valve and damper control were chosen from Figure 5.4. Tests were conducted at several pressure setpoints (range from 0.0125 to 0.0625 in WG) and each time the room temperature was allowed to reach steady state and the final value was recorded. The tests were conducted at two different loads: full load = 100% and at part-load = 67%. The results obtained are summarized in Figure 5.5.a-b. The results show that as the pressure setpoint, hence the airflow rate, is increased the room temperature decreases. The higher the load, higher will be the room temperature. The stationary relationships depicted in Figure 5.5.a-b reveal that at full load, room-1 temperature can be varied between 25.6~23.3°C by modulating the airflow rate by choosing an appropriate pressure setpoint. The results presented in Figure 5.5.a-b serve as useful guidelines in this regard. It should be noted that real time updating of pressure setpoints is complicated by the fact that prediction of zone loads is required. Therefore, simple and accurate methods have to be developed which can be implemented using the available hardware. This problem is presently being explored as a part of separate study.

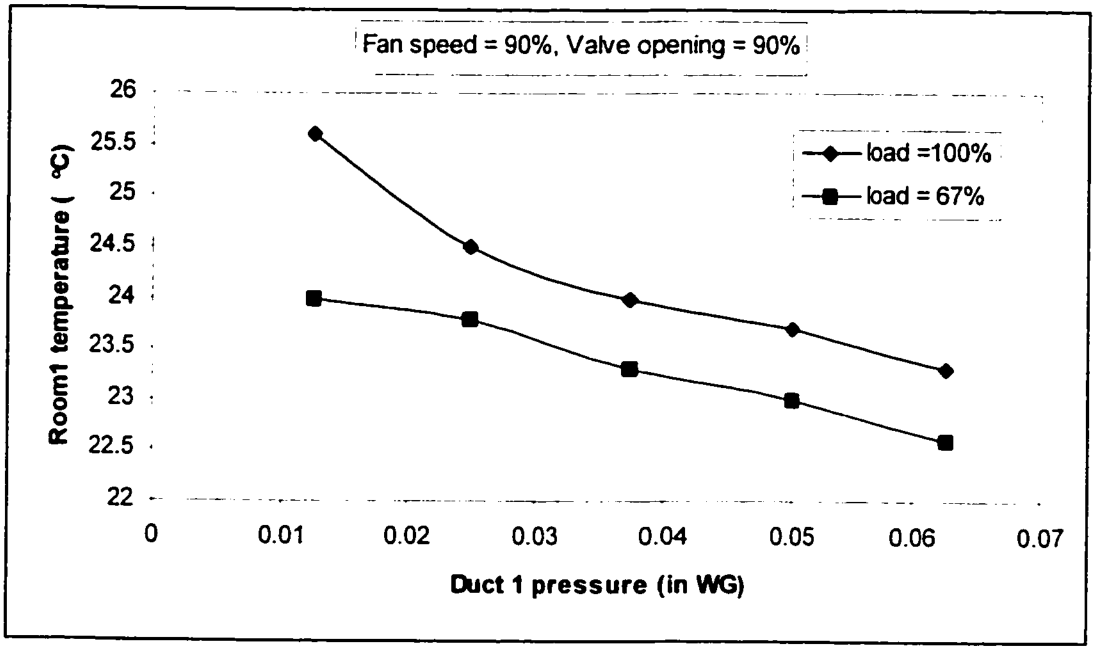

Figure 5.5.a: Relationship between duct1 pressure setpoint and room1 temperature setpoint

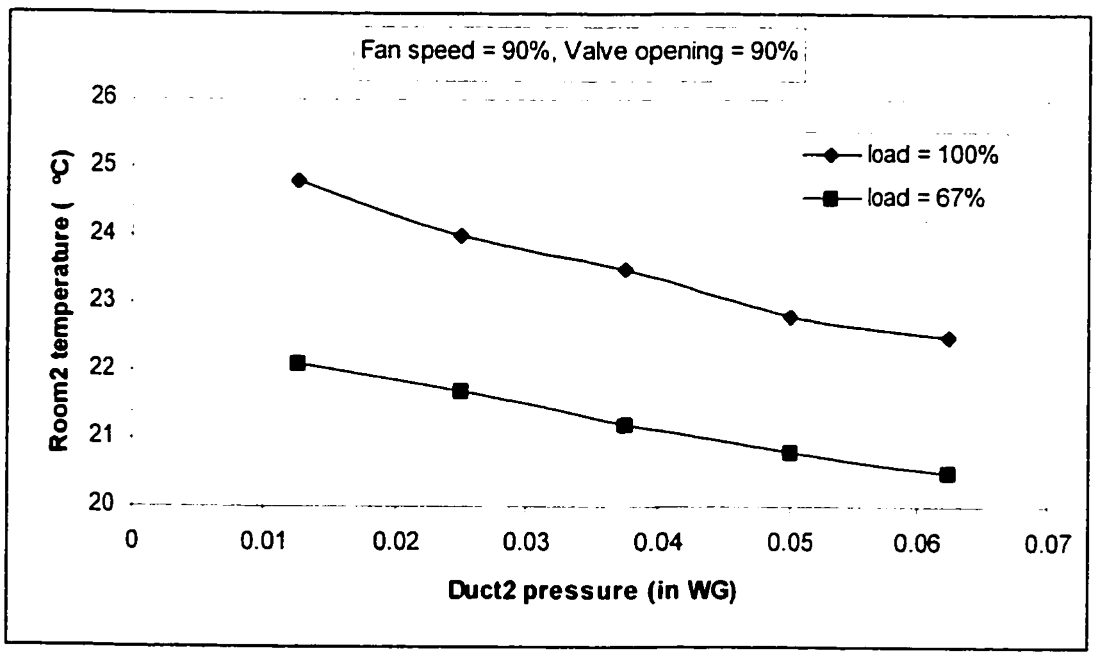

Figure 5.5.b: Relationship between duct2 pressure setpoint and room2 temperature setpoint

5.6 Summary and Conclusions

Experiments were conducted to compare the operating performance of PIC and PDC strategies. Also, controller gains for good operation of VAV control loops were determined. The results show that PIC strategy is more stable than PDC. However, a finely tuned PDC strategy gives better room temperature control compared to PIC. Experimental results show that for good performance the optimal range of controller gains in multiple-loop control mode are significantly lower than single-loop optimal control gains.

CHAPTER 6

CONCLUSIONS AND FUTURE WORK

An experimental study of loop tuning, inter-loop interactions, and operating strategies in a two-zone VAV system was conducted in a laboratory test facility [30] which consists of six local control loops. Chapter 2 described the experimental facility and the experimental results and analysis are documented in Chapter 3, 4, 5. In this Chapter, we will summarize conclusions of the present study and future works on this topic.

6.1 Summary and Conclusions of the Present Study

6.1.1 Steady State and Dynamic Response of the Open-loop VAV System

A series of systematic open loop experiments were conducted on three sub-systems: (i) airflow system, (ii) discharge air system and (iii) environmental zones. Through these tests it was possible to identify and quantify the operating range of the VAV system under several different cooling load conditions. From the results presented in this thesis we draw the following important conclusions.

(i) The operating range of the dampers in the VAV boxes 1 and 2 for good flow control was found to be between 40~100%.

(ii) The time needed for the airflow system to stabilize is influenced more by the damper movement than changes in the fan speed.

(iii) The operating range of the chilled water valve for water flow rate control was found to be between 35~90%.

(iv) The steady state time of the discharge air system was measured and is of the order of 400~450 seconds.

(v) Both room air temperature and relative humidity responses in rooms 1 and 2 show that about 1200 seconds are needed for the environmental zones to reach steady state.

(vi) A comparison of the steady state times shows that the airflow sub-system is the fastest followed by the DATS and environmental zones. The time scales in order are: 100:400:1200 seconds.

6.1.2 Dynamic Response of the Closed-loop VAV System

A series of systematic closed loop experiments were individually conducted on three sub-systems: (i) discharge air system, (ii) air flow system, and (iii) environmental zones. In addition, we focus on the overall VAV system with several control loops working simultaneously. Through these tests it was possible to identify and quantify the steady state time, overshoot and stability of the VAV system under full load conditions. From the results presented in this thesis we draw the following important conclusions.

(i) Under proper tuning, the discharge air system could be stably controlled. The DAT reaches setpoint (13°C) in about 600S. In the process the maximum deviation from the setpoint was found to be –0.9°C to +0.3°C. These values are normal and acceptable in real commercial building controls.

(ii) The fan loop is the fastest of all control loops in the VAV system. Closed loop response of the velocity pressure in duct-2 indicates that the maximum overshoot is 66% above the setpoint and the duct-2 pressure reaches steady state in about 40S. The test results also show that large step changes in the pressure setpoint could cause severe overshoot problems and even oscillations if the controller gains are not updated.

(iii) Airflow rates to the rooms stabilize in about 120S. The maximum variation in room1 airflow rate was found to be 10% of the setpoint (250cfm). The zone closer to the fan (room1 in the test facility) experienced higher fluctuations in airflow rates than the room 2 farther from the fan. The airflow rate responses also reveal that the steady state time of the damper control loop is greater than that of the fan speed control loop and discharge air temperature control loop.

(iv) With good-tuned PID parameters, the four control loops in VAV system were stabilized and operated very well. The overall system closed loop response shows that the DAT control response isolated from the other control loops is faster and gives less overshoot compared to the case when DAT control is operating in a multiple loop control environment. Obviously, this is because of interactions among the local loops.

6.1.3 Control Loop Interactions

(i) The modulation of dampers in one VAV box influences the airflow rate from its neighboring VAV box. It was found that as much as 16~22% reduction in airflow rate is likely at higher fan speeds in open loop control mode. This effect is much more pronounced under part-load (low fan speed) conditions.

(ii) In closed loop control operation, a 10% increase in room1 airflow rate setpoint caused a 2% increase in room2 airflow rate. With a 30% decrease in room1 airflow rate setpoint, caused a 5% fluctuations in room2 airflow rates. Results from the low speed case (fan speed = 60%) also shown some fluctuations in the airflow rates to one room when the neighboring damper tracks its setpoint. The effect of one loop on the other in terms of airflow rates is not as significant as was the case under open loop operating conditions.

(iii) The airflow rates setpoint step change (20%) has a considerable effect on the DAT (1°C overshoot) even though the DAT setpoint was not changed. However, the room temperature is not significantly affected by this step change. On the contrary, when a 1°C step change was initiated to the DAT setpoint, both airflow rate and room air temperatures are not greatly affected. They almost remained constant.

(iv) When 30% step change in the supply airflow was introduced, the DAT response, which was stable before the application of the step change in the airflow rate setpoint, goes into sustained oscillatory response. This phenomenon should be avoided to protect the actuators. From this result we conclude that VAV damper loops could significantly interact with the DAT loop.

6.1.4 Comparison of Control Strategies and Control Loop Optimization

Experiments were conducted to compare the operating performance of PIC and PDC strategies. Also, controller gains for good operation of VAV control loops were determined. The results show:

(i) PIC strategy is more stable than PDC. However, a finely tuned PDC strategy gives good room temperature control compared to PIC.

(ii) For good performance the optimal range of controller gains in multiple-loop control mode are significantly lower than single-loop optimal control gains.

The results presented above will:

(i) help building engineers identify operating problems and interactions between local loops in VAV systems,

(ii) offer guidelines in finding optimal controller gains,

(iii) provide experimental data to building engineers and researchers for model development and control design, and,

(iv) be helpful for commissioning engineers and building engineers to determine VAV system operating strategies.

6.2 Future Work

Room humidity controls should be introduced in the VAV system. A study on interactions between humidity control loop and other loops should be carried out. Energy conservation issue should be considered in VAV control system in analyzing and selecting operating strategies. High performance adaptive tuning controller needs to be developed to eliminate control loop interactions.

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APPENDIX-1: I/O POINT DESCRIPTION

The following is the I/O point list, which was used by the MBC in LAB tests. Name, type and other configuration information of I/O points are presented in the following table.

name	type	descriptor	value	condition	priority	alarmable	high limit	low limit	eng. units	address	digital val.	slope	intercept	COV limit	sensor type
ARDTS	LAI	AIR R T	22.06	N	NONE	NO			DEG. C	1000033	14360	0.02205	-294.57	0.0441	I
ASDTS	LAI	AIR S T	21.89	N	NONE	NO			DEG. C	1000032	14352	0.02205	-294.57	0.0441	I
ATS001	LAO	SETPOINT	25	N	OPER	NO			DEG. C	01000S00	25	1	0	1	N/A
ATS002	LAO	SETPOINT	25	N	OPER	NO			DEG. C	01000S11	25	1	0	1	N/A
CCIHS	LAI	COIL IN H	30.44	N	NONE	NO			PER.	1000016	9352	0.003255	0	0.00651	I
CCITS	LAI	COIL IN T	21.53	N	NONE	NO			DEG. C	1000040	14336	0.02205	-294.57	0.0441	I
CCOHS	LAI	COIL OUT H	29.76	N	NONE	NO			PER.	1000017	9144	0.003255	0	0.00651	I
CCOTS	LAI	COIL OUT T	21.53	N	NONE	NO			DEG. C	1000041	14336	0.02205	-294.57	0.0441	I
CWFRM	LAI	C WATER RATE	4.037	N	NONE	YES	22.23	3.5	MA	1000021	3644	0.000625	1.76	0.00125	I
CWFRM1	LAI	C W METER	4.037	N	NONE	YES	20.5	3.5	MA	1000021	3644	0.000625	1.76	0.00125	I
CWRTS	LAI	WATER R T	21.53	N	NONE	NO			DEG. C	1000037	14336	0.02205	-294.57	0.0441	I
CWSTS	LAI	WATER S T	21.18	N	NONE	NO			DEG. C	1000036	14320	0.02205	-294.57	0.0441	I
DAMP01	LAO	CV	100	N	NONE	NO			PCT	01000D01	100	1	0	1	N/A
DAMP02	LAO	CV	100	N	NONE	NO			PCT	01000D06	100	1	0	1	N/A
DAMP1	LAO	DAMPER1 CON	10	N	NONE	YES	10.5	1.499	VDC	1000045	30769	0.000325	0	0.00065	N/A
DAMP2	LAO	DAMPER2 CON	10	N	NONE	YES	10.5	1.499	VDC	1000049	30769	0.000325	0	0.00065	N/A
DPS001	LAI	DUCT1 P	4.02	N	NONE	NO			MA	1000004	3616	0.000625	1.76	0.00125	I
DPS002	LAI	DUCT2 P	3.96	N	NONE	NO			MA	1000012	3520	0.000625	1.76	0.00125	I
DTS001	LAI	DUCT1 T	21.71	N	NONE	NO			DEG. C	1000025	14344	0.02205	-294.57	0.0441	I
DTS002	LAI	DUCT2 T	21.89	N	NONE	NO			DEG. C	1000029	14352	0.02205	-294.57	0.0441	I
D1KD	LAO	DER	0	N	OPER	NO			BITS	01000D05	0	1	0	1	N/A
D1KI	LAO	INT	0.1	N	OPER	NO			BITS	01000D04	0	1	0	1	N/A
D1KP	LAO	PROP	1	N	OPER	NO			BITS	01000D03	0	1	0	1	N/A
D1SP	LAO	SET POINT	14	N	OPER	NO			MA	01000D02	14	1	0	1	N/A
D2KD	LAO	DER	0	N	OPER	NO			BITS	01000D10	0	1	0	1	N/A
D2KI	LAO	INT	0.1	N	OPER	NO			BITS	01000D09	0	1	0	1	N/A
D2KP	LAO	PROP	1	N	OPER	NO			BITS	01000D08	0	1	0	1	N/A
D2SP	LAO	SET POINT	14	N	OPER	NO			MA	01000D07	14	1	0	1	N/A
FAN	LAO	FAN CONTROL	0	N	OPER	YES	10	0	V	1000052	0	0.000325	0	0.00065	N/A

FANKD	LAO	DER	0	N	OPER	NO			BITS	01000F05	0	1	0	1	N/A
FANKI	LAO	INT	50	N	OPER	NO			BITS	01000F04	50	1	0	1	N/A
FANKP	LAO	PROP	1000	N	OPER	NO			BITS	01000F03	1000	1	0	1	N/A
FANSP	LAO	SET POINT	10	N	OPER	NO			MA	01000F02	10	1	0	1	N/A
FANSPD	LAO	CV	28	N	NONE	NO			PCT	01000F01	28	1	0	1	N/A
HEAT1	LAO	HEATER1 CON	0	N	OPER	YES	10.5	-0.2	VDC	1000044	0	0.000325	0	0.00065	N/A
HEAT2	LAO	HEATER2 CON	0	N	OPER	YES	10.5	-0.2	VDC	1000048	0	0.000325	0	0.00065	N/A
RHS001	LAI	ROOM1 H	26.27	N	NONE	NO			PER.	1000005	8072	0.003255	0	0.00065	I
RHS002	LAI	ROOM2 H	35.18	N	NONE	NO			PER.	1000013	10808	0.003255	0	0.00065	I
RPM001	LAI	ROOM1 P M	4	N	NONE	NO			MA	1000008	3584	0.000625	1.76	0.00125	I
RPM002	LAI	ROOM2 P M	3.994	N	NONE	NO			MA	1000009	3576	0.000625	1.76	0.00125	I
RTS001	LAI	ROOM1 T	20.83	N	NONE	NO			DEG. C	1000024	14304	0.02205	-294.57	0.0441	I
RTS002	LAI	ROOM2 T	17.65	N	NONE	NO			DEG. C	1000028	14160	0.02205	-294.57	0.0441	I
VALKD	LAO	DER	0	N	OPER	NO			BITS	01000V05	0	1	0	1	N/A
VALKI	LAO	INT	0.03	N	OPER	NO			BITS	01000V04	0	1	0	1	N/A
VALKP	LAO	PROP	0.6	N	OPER	NO			BITS	01000V03	0.6	1	0	1	N/A
VALSP	LAO	SET POINT	13	N	OPER	NO			DEG. C	01000V02	13	1	0	1	N/A
VALOP	LAO	CV	100	N	NONE	NO			PCT	01000V01	100	1	0	1	N/A
VALVE	LAO	VALVE CON	10	N	NONE	YES	10.2	-0.1	V	1000053	30769	0.000325	0	0.00065	N/A
WTS001	LAO	DER	1	N	NONE	NO			BITS	01000S06	1	1	0	1	N/A
WTS002	LAO	DERI	1	N	NONE	NO			BITS	01000S14	1	1	0	1	N/A
XTS001	LAO	CV	58.01	N	NONE	NO			PCT	01000S04	58	1	0	1	N/A
XTS002	LAO	CV	60.55	N	NONE	NO			PCT	01000S10	61	1	0	1	N/A
YTS001	LAO	PROP	3500	N	OPER	NO			BITS	01000S01	3500	1	0	1	N/A
YTS002	LAO	PROP	3500	N	NONE	NO			BITS	01000S12	3500	1	0	1	N/A
ZTS001	LAO	INT	50	N	OPER	NO			BITS	01000S02	50	1	0	1	N/A
ZTS002	LAO	INT	50	N	NONE	NO			BITS	01000S13	50	1	0	1	N/A

APPENDIX-2: LISTING OF PPCL PROGRAM

The following is a PPCL program list, which was used by the MBC to implement Lab tests shown in this thesis. Refer to Appendix-1 for I/O point description; Appendix-3 for experiment measured data format; Appendix-4 for experiment conditions and processed data format.

```
Program Report     Cabinet 10    Lines 1 to 5000    11:54 15-Mar-2000
------------------------------------------------------------------------

DT    20     LOOP (128,RTS001,XTS001,ATS001,YTS001,ZTS001,WTS001,1,50.0,
                   0.0,100.0,0)

DT    30     TABLE (XTS001,HEAT1,0.0,0.0,100.0,10.0)

DT    40     LOOP (128,RTS002,XTS002,ATS002,YTS002,ZTS002,WTS002,1,50.0,
                   0.0,100.0,0)

DT  50     TABLE (XTS002,HEAT2,0.0,0.0,100.0,10.0)

DT  100    GOTO 10

DT  220    IF (TIME .GE. 13:05) THEN DAMP1 = 5.0

DT  230    IF (TIME .GE. 13:25) THEN DAMP1 = 2.5

DT  300    IF (TIME .GE. 14:40) THEN DAMP1 = 10.0

DT  310    IF (TIME .GE. 14:45) THEN DAMP1 = 0.0

DT  330    IF (TIME .GE. 15:00 .AND. TIME .LE. 15:03) THEN DAMP1 = 5.0

DT  340    IF (TIME .GE. 15:07) THEN DAMP1 = 0.0

DT  400    IF (TIME .GE. 10:42 .AND. TIME .LE. 10:45) THEN FAN = 2.52

DT  500    IF (TIME .GT. 15:05) THEN FAN = 2.8

DT  600    IF (TIME .GT. 15:05) THEN VALVE = 9

DT  700    IF (TIME .GE. 12:55 .AND. TIME .LE. 13:39) THEN DAMP2 = 10.0

DT  710    IF (TIME .GE. 13:00 .AND. TIME .LE. 13:04) THEN DAMP1 = 10.0

DT  720    IF (TIME .GT. 13:04) THEN DAMP1 = 0.0
```

```
DT  730   IF (TIME .GE. 13:07 .AND. TIME .LE. 13:11) THEN DAMP1 = 7.5
DT  740   IF (TIME .GT. 13:11) THEN DAMP1 = 0.0
DT  750   IF (TIME .GE. 13:14 .AND. TIME .LE. 13:18) THEN DAMP1 = 5.0
DT  760   IF (TIME .GT. 13:18) THEN DAMP1 = 0.0
DT  770   IF (TIME .GE. 13:21 .AND. TIME .LE. 13:25) THEN DAMP1 = 2.5
DT  780   IF (TIME .GT. 13:25) THEN DAMP1 = 5.0
DT  790   IF (TIME .GE. 13:28 .AND. TIME .LE. 13:32) THEN DAMP1 = 10.0
DT  800   IF (TIME .GT. 13:32) THEN DAMP1 = 2.5
DT  810   IF (TIME .GE. 13:35 .AND. TIME .LE. 13:39) THEN DAMP1 = 7.5
DT  820   IF (TIME .GT. 13:39) THEN DAMP1 = 0.0
DT  830   IF (TIME .GT. 13:39) THEN DAMP2 = 5.0
DT  840   IF (TIME .GE. 13:42 .AND. TIME .LE. 13:46) THEN DAMP1 = 10.0
DT  850   IF (TIME .GT. 13:46) THEN DAMP2 = 0.0
DT  855   IF (TIME .GT. 14:55) THEN DAMP1 = 10.0
DT  860   IF (TIME .GE. 14:00 .AND. TIME .LE. 14:05) THEN DAMP1 = 0.0
DT  870   IF (TIME .GE. 14:00 .AND. TIME .LE. 14:05) THEN DAMP2 = 10.0
DT  890   IF (TIME .GT. 14:08) THEN DAMP2 = 7.5
DT  900   IF (TIME .GE. 14:11 .AND. TIME .LE. 14:16) THEN DAMP1 = 7.5
DT  910   IF (TIME .GE. 14:11 .AND. TIME .LE. 14:16) THEN DAMP2 = 0.0
DT  920   IF (TIME .GT. 14:19) THEN DAMP1 = 5.0
DT  925   IF (TIME .GE. 14:21 .AND. TIME .LE. 14:26) THEN DAMP1 = 0.0
DT  930   IF (TIME .GE. 14:21 .AND. TIME .LE. 14:26) THEN DAMP2 = 5.0
DT  940   IF (TIME .GT. 14:29) THEN DAMP1 = 10.0
DT  950   IF (TIME .GE. 14:32 .AND. TIME .LE. 14:37) THEN DAMP1 = 5.0
DT  960   IF (TIME .GE. 14:32 .AND. TIME .LE. 14:37) THEN DAMP2 = 10.0
DT  970   IF (TIME .GT. 14:40) THEN DAMP2 = 5.0
DT  980   IF (TIME .GE. 14:43 .AND. TIME .LE. 14:48) THEN DAMP1 = 10.0
DT  990   IF (TIME .GE. 14:43 .AND. TIME .LE. 14:48) THEN DAMP2 = 10.0
DT  2000  LOOP (0,RTS001,DAMP01,D1SP,D1KP,D1KI,D1KD,1,50.0,0.0,
```

```
              100.0,0)

ET  2050  LOOP (128,DPS001,DAMP01,D1SP,D1KP,D1KI,D1KD,1,50.0,0.0,
              100.0,0)

ET  2100  TABLE (DAMP01,DAMP1,0.0,0.0,100.0,10.0)

DT  2110  IF (DAMP1 .LT. 4.0) THEN DAMP1 = 4.0

DT  2200  LOOP (0,RTS002,DAMP02,D2SP,D2KP,D2KI,D2KD,1,50.0,0.0,
              100.0,0)

ET  2250  LOOP (128,DPS002,DAMP02,D2SP,D2KP,D2KI,D2KD,1,50.0,0.0,
              100.0,0)

ET  2300  TABLE (DAMP02,DAMP2,0.0,0.0,100.0,10.0)

D   2310  IF (DAMP2 .LT. 4.0) THEN DAMP2 = 4.0
DT  2360  IF (TIME .GE. 16:50) THEN FAN = 2.52

DT  2400  LOOP (128,DPS002,FANSPD,FANSP,FANKP,FANKI,FANKD,1,14.0,0.0,
              28.0,0)

DT  2500  TABLE (FANSPD,FAN,0.0,0.0,28.0,2.8)

DT  2600  LOOP (0,CCOTS,VALOP,VALSP,VALKP,VALKI,VALKD,1,50.0,0.0,
              100.0,0)

DT  2700  TABLE (VALOP,VALVE,0.0,0.0,100.0,10.0)

ET  3000  GOTO 1999

DT  3100  GOTO 2350
```

End of Report

APPENDIX-3: LISTING OF TYPICAL MEASURED DATA

The following is a sample data list and format, which was downloaded from the controller by HyperTerminal. Refer to Appendix-1 for I/O point description. Refer to Appendix-4 for experiment conditions and processed data format.

```
19:44:21 01-Mar-2000
CWSTS   (WATER S T    ) 8.132     DEG. C -N-       P:NONE
CWRTS   (WATER R T    ) 9.720     DEG. C -N-       P:NONE
CCOTS   (COIL OUT T   ) 12.01     DEG. C -N-       P:NONE
DPS001  (DUCT1 P      ) 4.05      MA     -N-       P:NONE
DPS002  (DUCT2 P      ) 3.99      MA     -N-       P:NONE
CCIHS   (COIL IN H    ) 30.15     PER.   -N-       P:NONE
CCOHS   (COIL OUT H   ) 42.23     PER.   -N-       P:NONE
RTS001  (ROOM1 T      ) 26.83     DEG. C -N-       P:NONE
RTS002  (ROOM2 T      ) 25.59     DEG. C -N-       P:NONE
DTS002  (DUCT2 T      ) 16.24     DEG. C -N-       P:NONE
ARDTS   (AIR R T      ) 26.65     DEG. C -N-       P:NONE
ASDTS   (AIR S T      ) 14.30     DEG. C -N-       P:NONE
CCITS   (COIL IN T    ) 22.24     DEG. C -N-       P:NONE
CWFRM   (C WATER RATE) 18.73      MA     -N-       P:NONE
DTS001  (DUCT1 T      ) 16.07     DEG. C -N-       P:NONE
RHS002  (ROOM2 H      ) 25.88     PER.   -N-       P:NONE
RHS001  (ROOM1 H      ) 19.21     PER.   -N-       P:NONE
Delaying...

19:44:25 01-Mar-2000
CWSTS   (WATER S T    ) 8.308     DEG. C -N-       P:NONE
CWRTS   (WATER R T    ) 9.543     DEG. C -N-       P:NONE
CCOTS   (COIL OUT T   ) 12.01     DEG. C -N-       P:NONE
DPS001  (DUCT1 P      ) 7.97      MA     -N-       P:NONE
DPS002  (DUCT2 P      ) 7.875     MA     -N-       P:NONE
CCIHS   (COIL IN H    ) 30.13     PER.   -N-       P:NONE
CCOHS   (COIL OUT H   ) 42.21     PER.   -N-       P:NONE
RTS001  (ROOM1 T      ) 27.00     DEG. C -N-       P:NONE
RTS002  (ROOM2 T      ) 25.59     DEG. C -N-       P:NONE
DTS002  (DUCT2 T      ) 16.24     DEG. C -N-       P:NONE
ARDTS   (AIR R T      ) 26.65     DEG. C -N-       P:NONE
ASDTS   (AIR S T      ) 14.30     DEG. C -N-       P:NONE
CCITS   (COIL IN T    ) 22.15     DEG. C -N-       P:NONE
CWFRM   (C WATER RATE) 18.73      MA     -N-       P:NONE
DTS001  (DUCT1 T      ) 16.24     DEG. C -N-       P:NONE
RHS002  (ROOM2 H      ) 25.85     PER.   -N-       P:NONE
RHS001  (ROOM1 H      ) 19.21     PER.   -N-       P:NONE
Delaying...
```

```
19:44:29 01-Mar-2000
CWSTS    (WATER S T    ) 8.308      DEG. C -N-        P:NONE
CWRTS    (WATER R T    ) 9.367      DEG. C -N-        P:NONE
CCOTS    (COIL OUT T   ) 11.83      DEG. C -N-        P:NONE
DPS001   (DUCT1 P      ) 8.54       MA     -N-        P:NONE
DPS002   (DUCT2 P      ) 8.244      MA     -N-        P:NONE
CCIHS    (COIL IN H    ) 30.05      PER.   -N-        P:NONE
CCOHS    (COIL OUT H   ) 42.34      PER.   -N-        P:NONE
RTS001   (ROOM1 T      ) 27.00      DEG. C -N-        P:NONE
RTS002   (ROOM2 T      ) 25.59      DEG. C -N-        P:NONE
DTS002   (DUCT2 T      ) 16.24      DEG. C -N-        P:NONE
ARDTS    (AIR R T      ) 26.65      DEG. C -N-        P:NONE
ASDTS    (AIR S T      ) 14.30      DEG. C -N-        P:NONE
CCITS    (COIL IN T    ) 22.24      DEG. C -N-        P:NONE
CWFRM    (C WATER RATE) 18.72       MA     -N-        P:NONE
DTS001   (DUCT1 T      ) 16.07      DEG. C -N-        P:NONE
RHS002   (ROOM2 H      ) 25.88      PER.   -N-        P:NONE
RHS001   (ROOM1 H      ) 19.21      PER.   -N-        P:NONE
Delaying...

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19:49:53 01-Mar-2000
CWSTS    (WATER S T    ) 8.308      DEG. C -N-        P:NONE
CWRTS    (WATER R T    ) 10.60      DEG. C -N-        P:NONE
CCOTS    (COIL OUT T   ) 10.77      DEG. C -N-        P:NONE
DPS001   (DUCT1 P      ) 8.0        MA     -N-        P:NONE
DPS002   (DUCT2 P      ) 6.02       MA     -N-        P:NONE
CCIHS    (COIL IN H    ) 29.16      PER.   -N-        P:NONE
CCOHS    (COIL OUT H   ) 45.28      PER.   -N-        P:NONE
RTS001   (ROOM1 T      ) 25.06      DEG. C -N-        P:NONE
RTS002   (ROOM2 T      ) 24.89      DEG. C -N-        P:NONE
DTS002   (DUCT2 T      ) 14.83      DEG. C -N-        P:NONE
ARDTS    (AIR R T      ) 26.65      DEG. C -N-        P:NONE
ASDTS    (AIR S T      ) 13.07      DEG. C -N-        P:NONE
CCITS    (COIL IN T    ) 23.65      DEG. C -N-        P:NONE
CWFRM    (C WATER RATE) 18.52       MA     -N-        P:NONE
DTS001   (DUCT1 T      ) 14.13      DEG. C -N-        P:NONE
RHS002   (ROOM2 H      ) 25.98      PER.   -N-        P:NONE
```

```
RHS001 (ROOM1 H      ) 19.55      PER.    -N-        P:NONE
Delaying...

19:49:57 01-Mar-2000
CWSTS  (WATER S T    ) 8.308      DEG. C -N-         P:NONE
CWRTS  (WATER R T    ) 10.60      DEG. C -N-         P:NONE
CCOTS  (COIL OUT T   ) 10.77      DEG. C -N-         P:NONE
DPS001 (DUCT1 P      ) 8.005      MA      -N-        P:NONE
DPS002 (DUCT2 P      ) 6.015      MA      -N-        P:NONE
CCIHS  (COIL IN H    ) 29.16      PER.    -N-        P:NONE
CCOHS  (COIL OUT H   ) 45.31      PER.    -N-        P:NONE
RTS001 (ROOM1 T      ) 25.06      DEG. C -N-         P:NONE
RTS002 (ROOM2 T      ) 24.89      DEG. C -N-         P:NONE
DTS002 (DUCT2 T      ) 14.83      DEG. C -N-         P:NONE
ARDTS  (AIR R T      ) 26.65      DEG. C -N-         P:NONE
ASDTS  (AIR S T      ) 13.07      DEG. C -N-         P:NONE
CCITS  (COIL IN T    ) 23.65      DEG. C -N-         P:NONE
CWFRM  (C WATER RATE) 18.54       MA      -N-        P:NONE
DTS001 (DUCT1 T      ) 14.13      DEG. C -N-         P:NONE
RHS002 (ROOM2 H      ) 25.98      PER.    -N-        P:NONE
RHS001 (ROOM1 H      ) 19.58      PER.    -N-        P:NONE
Delaying...

19:50:01 01-Mar-2000
CWSTS  (WATER S T    ) 8.308      DEG. C -N-         P:NONE
CWRTS  (WATER R T    ) 10.60      DEG. C -N-         P:NONE
CCOTS  (COIL OUT T   ) 10.77      DEG. C -N-         P:NONE
DPS001 (DUCT1 P      ) 8.045      MA      -N-        P:NONE
DPS002 (DUCT2 P      ) 6.01       MA      -N-        P:NONE
CCIHS  (COIL IN H    ) 29.14      PER.    -N-        P:NONE
CCOHS  (COIL OUT H   ) 45.31      PER.    -N-        P:NONE
RTS001 (ROOM1 T      ) 25.06      DEG. C -N-         P:NONE
RTS002 (ROOM2 T      ) 24.71      DEG. C -N-         P:NONE
DTS002 (DUCT2 T      ) 14.83      DEG. C -N-         P:NONE
ARDTS  (AIR R T      ) 26.65      DEG. C -N-         P:NONE
ASDTS  (AIR S T      ) 13.07      DEG. C -N-         P:NONE
CCITS  (COIL IN T    ) 23.65      DEG. C -N-         P:NONE
CWFRM  (C WATER RATE) 18.56       MA      -N-        P:NONE
DTS001 (DUCT1 T      ) 14.13      DEG. C -N-         P:NONE
RHS002 (ROOM2 H      ) 25.98      PER.    -N-        P:NONE
RHS001 (ROOM1 H      ) 19.58      PER.    -N-        P:NONE
Delaying...

19:50:05 01-Mar-2000
CWSTS  (WATER S T    ) 8.308      DEG. C -N-         P:NONE
CWRTS  (WATER R T    ) 10.60      DEG. C -N-         P:NONE
CCOTS  (COIL OUT T   ) 10.77      DEG. C -N-         P:NONE
DPS001 (DUCT1 P      ) 8.02       MA      -N-        P:NONE
DPS002 (DUCT2 P      ) 5.98       MA      -N-        P:NONE
CCIHS  (COIL IN H    ) 29.14      PER.    -N-        P:NONE
CCOHS  (COIL OUT H   ) 45.33      PER.    -N-        P:NONE
RTS001 (ROOM1 T      ) 25.06      DEG. C -N-         P:NONE
RTS002 (ROOM2 T      ) 24.71      DEG. C -N-         P:NONE
DTS002 (DUCT2 T      ) 14.83      DEG. C -N-         P:NONE
ARDTS  (AIR R T      ) 26.65      DEG. C -N-         P:NONE
ASDTS  (AIR S T      ) 13.07      DEG. C -N-         P:NONE
CCITS  (COIL IN T    ) 23.65      DEG. C -N-         P:NONE
```

```
CWFRM  (C WATER RATE) 18.54    MA      -N-      P:NONE
DTS001 (DUCT1 T     ) 14.13    DEG. C  -N-      P:NONE
RHS002 (ROOM2 H     ) 25.98    PER.    -N-      P:NONE
RHS001 (ROOM1 H     ) 19.58    PER.    -N-      P:NONE
Delaying...
```

APPENDIX-4: LISTING OF C++ DATA PROCESSING PROGRAM

The following is a C++ program list, which was used to process raw data (shown in Appendix-3) to processed data (shown in Appendix-5). Refer to Appendix-3 for typical measured data and its format. Refer to Appendix-5 for corresponding processed data and its format.

```
// ************************************************************
// *   Data conversion Version 1.0                            *
// *                                                          *
// *   by : Ziyu Fan                                          *
// *   Building Department of Concordia University            *
// *   Date: Nov. 25, 1999                                    *
// ************************************************************


#include <iostream>
#include <fstream>
#include <string>
#include <math.h>
#include <stdlib.h>
#include <iomanip>
using namespace std;


float ExtractData(char buffer[]);

int main(){
        int i=0;
        int number;
        int Timer[2000];

        float Data[20][2000];
        string date;
        string TimeStart;
        string TimeEnd;
```

```
char FileNameIn[80];
char FileNameOut[80];
char buffer[100]; //used to store one line of record of the data file
char value[6]; //used to extract values from buffer

cout<<"\nEnter name of file to process: ";
cin>>FileNameIn;

cout<<"\nEnter name of file to save results: ";
cin>>FileNameOut;
//to judge if "scores.dat" exists
ifstream fin(FileNameIn);
if(!fin){
        cerr<<"Cannot open the file!"<<endl;
        return 1;
}

//open a new file "averages.dat" to write
ofstream fout(FileNameOut);

cout<<"experiment date: ";
cin>>date;
fout<<"experiment date: "<<date<<endl; //write the date of experiment
cout<<"nexperiment starting time: ";
cin>>TimeStart;
fout<<"experiment starting time: "<<TimeStart<<endl; //write the time of
experiment
cout<<"nexperiment end time: ";
cin>>TimeEnd;
fout<<"experiment ending time: "<<TimeEnd<<endl; //write the time of
experiment

cout<<"\n\n\nProcessing data, please wait...\n\n";

for(int index=0; index<2000; index++)
        Timer[index]=0;

while(!fin.eof()){
        loop1: fin.getline(buffer, 80);

        for(int j=0;j<5; j++)
                value[j]=buffer[j+22];

        long flag=atof(value);

        if(!flag)
```

```
			goto loop1;

		Data[0][i]=atof(value);

		for(j=1; j<17; j++){
			fin.getline(buffer, 80);
			Data[j][i]=ExtractData(buffer);
		}

		Data[3][i]=0.1*(Data[3][i]-4.0)/16.0-0.000625;//convert 'ma' to "WG
		Data[4][i]=0.1*(Data[4][i]-4.0)/16.0-0.0004625; //convert 'ma to "WG
		Data[13][i]=6.0*(Data[13][i]-4.0)/16.0-0.015; //convert 'ma' to 'GPM'

		fin.ignore(80,'\n');
		fin.ignore(80,'\n');
		fin.ignore(80,'\n');
		cout<<". ";
		Timer[i+1]=Timer[i]+4;
		++i;
		number=i;
	}

	cout<<"\n\nComplete data processing";
	cout<<"\nNow the data is saving in <"<<FileNameOut<<">...\n";


/*
	cout<<"Time\t"<<"CWSTS\t"<<"CWRTS\t"<<"CCOTS\t"<<"DPS001\t"
		<<"DPS002\t"<<"CCIHS\t"<<"CCOHS\t"<<"RTS001\t"<<"RTS002\t"
		<<"DTS002\t"<<"ARDTS\t"<<"ASDTS\t"<<"CCITS\t"<<"CWFRM\t"
		<<"DTS001\t"<<"RHS002\t"<<"RHS001"<<endl<<endl;
*/

	fout<<"\nTime\t"<<"CWSTS\t"<<"CWRTS\t"<<"CCOTS\t"<<"DPS001\t"
		<<"DPS002\t"<<"CCIHS\t"<<"CCOHS\t"<<"RTS001\t"<<"RTS002\t"
		<<"DTS002\t"<<"ARDTS\t"<<"ASDTS\t"<<"CCITS\t"<<"CWFRM\t"
		<<"DTS001\t"<<"RHS002\t"<<"RHS001"<<endl<<endl;

	for(int k=0; k<number; k++){
/*		cout<<Timer[k];
		for(int j=0; j<17; j++){
			if(Data[j][k]<=0.0005)
				cout<<"\t"<<"0.000";
			else cout<<"\t"<<setprecision(4)<<Data[j][k]; //
		}
		cout<<endl;
```

```
*/
			fout<<Timer[k];
			for(int j=0; j<17; j++){
				if(Data[j][k]<=0.0005)
					fout<<"\t"<<"0.000";
				else fout<<"\t"<<setprecision(4)<<Data[j][k];
			}

			fout<<endl;
			cout<<". ";

		}

		cout<<"\nPress any key to exit";
		cin.get();
		cin.get();
		return 0;
}

//************************************************************************
float ExtractData(char buffer[100]){
		char temp[6];
   for(int j=0;j<5; j++){
						temp[j]=buffer[j+22];
					}
		return atof(temp);
}
```

APPENDIX-5: LISTING OF TYPICAL PROCESSED DATA

The following is a sample data list and format, which was processed by a data processing program from measured data showed in Appendix-3. Refer to Appendix-3 for measured data and its format.

== inter-damper-loop3.out ==

experiment date: 01/03/2000
experiment starting time: 19:44:21
experiment ending time: 19:50:05

== control variables ==

fan speed: 60%
valve opening: 90%
damper1 opening: variable
damper2 opening: variable
heater1 output: 0
heater2 output: 0

== tuning loop status ==

fan loop status: off
valve loop status: off
damper1 loop status: on
damper2 loop status: on
heater1 loop status: off
heater2 loop status: off

====================================== parameter description ======================================

CWSTS(C. deg.): supply chilled water temperature
CWRTS(C. deg.): return chilled water temperature
CCOTS(C. deg.): temperature of air coming out of coil

Q1(cfm): air flow rate in duct 1
Q2(cfm): air flow rate in duct 2
CCIHS(%): relative humidity of air going in coil
CCOHS(%): relative humidity of air coming out of coil
RTS001(C. deg.): air emperature in room 1
RTS002(C. deg.): air emperature in room 2
DTS002(C. deg.): air emperature in duct 2
ARDTS(C. deg.): return air emperature in duct(common)
ASDTS(C. deg.): supply air emperature in duct(common)
CCITS(C. deg.): temperature of air going in coil
CWFRM(GPM): chilled water flow rate
DTS001(C. deg.): air emperature in duct 1
RHS002(%): air relative humidity in room 2
RHS001(%): air relative humidity in room 1

== data ==

Time	CWSTS	CWRTS	CCOTS	Q1	Q2	CCIHS	CCOHS	RTS001	RTS002	DTS002	ARDTS	ASDTS	CCITS	CWFRM	DTS001	RHS002	RHS001
0	8.132	9.72	12.01	0	0	30.15	42.23	26.83	25.59	16.24	26.65	14.3	22.24	5.509	16.07	25.88	19.21
4	8.308	9.543	12.01	155.7	154.3	30.13	42.21	27	25.59	16.24	26.65	14.3	22.15	5.509	16.24	25.85	19.21
8	8.308	9.367	11.83	166.8	161.6	30.05	42.34	27	25.59	16.24	26.65	14.3	22.24	5.505	16.07	25.88	19.21
12	8.308	9.19	11.66	165.9	161.8	29.97	42.42	26.83	25.59	16.07	26.83	14.13	22.33	5.49	15.89	25.85	19.19
16	8.308	9.19	11.57	166.5	159.1	29.94	42.44	26.83	25.59	16.07	26.83	13.95	22.42	5.505	15.71	25.85	19.19
20	8.308	9.19	11.48	166	160.7	29.92	42.47	26.65	25.41	15.89	26.83	13.95	22.59	5.498	15.54	25.88	19.19
24	8.308	9.19	11.39	163.8	158.3	29.92	42.49	26.65	25.41	15.71	27	13.77	22.68	5.501	15.54	25.88	19.19
28	8.308	9.19	11.3	161.5	155.4	29.89	42.55	26.65	25.41	15.71	27	13.77	22.77	5.501	15.36	25.88	19.19
32	8.308	9.367	11.21	156.9	151.2	29.86	42.57	26.47	25.59	15.71	27	13.6	22.86	5.512	15.36	25.88	19.19
36	8.308	9.543	11.21	151.9	145.9	29.84	42.6	26.65	25.41	15.54	27	13.6	22.86	5.498	15.18	25.88	19.19
40	8.308	9.543	11.21	144	137.8	29.81	42.6	26.47	25.41	15.54	27	13.6	22.95	5.498	15.18	25.88	19.21

44	8.308	9.72	11.21	135	128.3	29.79	42.65	26.47	25.41	15.54	27	13.6	23.03	5.501	15.18	25.88	19.21
48	8.308	10.07	11.13	127.4	118.9	29.79	42.7	26.47	25.41	15.54	27	13.6	23.03	5.509	15.18	25.88	19.21
52	8.132	10.07	11.13	114.8	105.5	29.76	42.73	26.47	25.41	15.54	27	13.6	23.03	5.516	15.18	25.88	19.21
56	8.308	10.24	11.13	102	92.85	29.73	42.73	26.3	25.41	15.54	27	13.42	23.12	5.501	15.18	25.91	19.21
60	8.308	10.24	11.13	88.5	79.16	29.73	42.78	26.3	25.24	15.54	27	13.42	23.12	5.512	15.18	25.91	19.19
64	8.308	10.24	11.13	70.58	77.39	29.73	42.81	26.3	25.24	15.54	27	13.42	23.12	5.494	15.18	25.88	19.19
68	8.308	10.42	11.13	58.44	79.16	29.71	42.86	26.3	25.24	15.54	27	13.6	23.12	5.505	15.18	25.91	19.19
72	8.308	10.42	11.04	56.53	90.11	29.71	42.89	26.3	25.41	15.54	27	13.6	23.21	5.501	15.18	25.91	19.21
76	8.308	10.42	11.04	68.32	99.69	29.68	42.94	26.47	25.24	15.54	27	13.6	23.21	5.479	15.18	25.91	19.21
80	8.132	10.42	11.04	84.14	104.7	29.68	42.99	26.3	25.24	15.54	27	13.6	23.21	5.475	15.18	25.91	19.21
84	8.132	10.24	10.95	97.11	107.8	29.66	43.04	26.47	25.24	15.54	27	13.42	23.21	5.479	15.18	25.91	19.21
88	8.132	10.24	10.95	105.3	108.7	29.66	43.09	26.3	25.24	15.54	27.18	13.6	23.3	5.46	15.18	25.91	19.21
92	8.308	10.24	10.95	109.4	110	29.63	43.15	26.47	25.24	15.36	27	13.42	23.3	5.479	15.18	25.91	19.21
96	8.308	10.07	10.86	108.4	107.3	29.63	43.17	26.47	25.24	15.54	27.18	13.42	23.3	5.475	15.18	25.91	19.21
100	8.308	10.07	10.86	110.1	108	29.6	43.25	26.3	25.24	15.36	27.18	13.42	23.39	5.486	15.18	25.91	19.21
104	8.308	10.07	10.86	109.3	108.6	29.6	43.3	26.3	25.24	15.36	27.18	13.42	23.39	5.486	15.18	25.91	19.21
108	8.132	10.07	10.77	109.1	109	29.58	43.35	26.3	25.24	15.36	27.18	13.42	23.39	5.449	15.18	25.91	19.21
112	8.132	10.07	10.77	109.7	109.4	29.58	43.41	26.3	25.24	15.36	27.18	13.24	23.39	5.445	15.18	25.91	19.21
116	8.132	10.07	10.77	109.5	109	29.58	43.46	26.3	25.24	15.36	27.18	13.24	23.47	5.423	15.01	25.91	19.21
120	8.132	10.07	10.77	109.7	110.3	29.55	43.48	26.3	25.24	15.36	27.18	13.42	23.47	5.434	15.18	25.91	19.24
124	8.308	10.07	10.69	109.5	110.4	29.53	43.54	26.3	25.24	15.18	27.18	13.24	23.47	5.445	15.01	25.91	19.24
128	8.308	10.07	10.69	109.5	110	29.53	43.59	26.3	25.24	15.36	27.18	13.24	23.47	5.441	15.01	25.91	19.24
132	8.308	10.07	10.69	109.8	109.6	29.53	43.64	26.3	25.24	15.18	27.18	13.24	23.47	5.445	15.01	25.93	19.27
136	8.308	10.24	10.69	110.8	111.1	29.5	43.69	26.12	25.24	15.18	27.18	13.24	23.47	5.438	15.01	25.93	19.27
140	8.308	10.07	10.69	109.7	109.7	29.5	43.74	26.12	25.24	15.18	27.18	13.24	23.47	5.434	15.01	25.93	19.27
144	8.308	10.07	10.69	110.1	111	29.47	43.77	26.12	25.24	15.18	27.18	13.24	23.56	5.456	15.01	25.93	19.27
148	8.308	10.24	10.69	108.5	108.6	29.47	43.82	26.12	25.24	15.18	27.18	13.24	23.56	5.441	15.01	25.93	19.27
152	8.308	10.24	10.6	110	110.7	29.47	43.88	26.12	25.24	15.18	27.18	13.24	23.56	5.49	15.01	25.93	19.27
156	8.308	10.24	10.6	109.7	109.3	29.45	43.9	25.94	25.24	15.18	27.18	13.24	23.56	5.516	15.01	25.93	19.27
160	8.308	10.24	10.6	109.8	110	29.45	43.95	25.94	25.06	15.18	27.18	13.24	23.56	5.512	14.83	25.93	19.27
164	8.308	10.24	10.6	109.3	109.9	29.42	44.01	25.94	25.06	15.01	27.36	13.24	23.56	5.498	14.83	25.93	19.27

168	8.308	10.24	10.6	109.8	110.1	29.42	44.06	26.12	24.89	15.18	27.36	13.24	23.65	5.494	15.01	25.93	19.27
172	8.308	10.24	10.6	109.7	109.7	29.4	44.08	25.94	24.89	15.01	27.18	13.07	23.65	5.445	14.83	25.93	19.27
176	8.308	10.24	10.6	109.4	109	29.4	44.14	26.12	24.89	15.01	27.18	13.07	23.65	5.512	14.83	25.93	19.29
180	8.308	10.24	10.6	110.3	110.1	29.4	44.19	25.94	25.06	15.01	27.18	13.24	23.65	5.505	14.83	25.93	19.29
184	8.308	10.24	10.6	119.9	108.6	29.37	44.24	25.94	24.89	15.01	27.18	13.24	23.65	5.498	14.83	25.93	19.29
188	8.308	10.24	10.6	131.1	107.1	29.37	44.27	25.94	25.06	15.01	27.36	13.24	23.65	5.479	14.83	25.93	19.29
192	8.308	10.24	10.6	141.2	106.2	29.34	44.32	25.94	25.06	15.01	27.36	13.07	23.65	5.486	14.83	25.93	19.29
196	8.308	10.24	10.51	149.9	105.3	29.34	44.37	25.94	25.06	15.01	27.36	13.07	23.65	5.479	14.65	25.93	19.32
200	8.308	10.24	10.51	154.8	106.7	29.34	44.42	25.94	25.06	15.01	27.36	13.07	23.65	5.494	14.65	25.93	19.32
204	8.308	10.24	10.51	156.3	108	29.32	44.45	25.94	25.06	15.01	27.36	13.07	23.65	5.494	14.65	25.93	19.32
208	8.308	10.24	10.51	157	109	29.32	44.5	25.77	25.06	15.01	27.36	13.07	23.74	5.498	14.65	25.93	19.32
212	8.308	10.24	10.51	155.9	108.8	29.29	44.55	25.94	25.06	15.01	27.36	13.07	23.74	5.512	14.48	25.93	19.34
216	8.308	10.24	10.51	156.5	110	29.29	44.58	25.77	25.06	15.01	27.36	13.07	23.74	5.501	14.48	25.93	19.34
220	8.308	10.24	10.6	157.1	109.6	29.29	44.6	25.77	25.06	14.83	27.36	13.07	23.74	5.498	14.48	25.93	19.34
224	8.308	10.24	10.6	155.9	108.8	29.29	44.66	25.77	25.06	14.83	27.36	13.07	23.74	5.385	14.48	25.93	19.34
228	8.308	10.42	10.6	156.6	110.4	29.27	44.68	25.77	25.06	15.01	27.18	13.07	23.74	5.441	14.48	25.93	19.34
232	8.308	10.42	10.6	156.2	110.3	29.27	44.71	25.77	25.06	15.01	27.18	13.07	23.74	5.456	14.48	25.93	19.37
236	8.308	10.42	10.6	155.4	110.1	29.27	44.73	25.59	25.06	14.83	27.18	13.07	23.74	5.452	14.48	25.96	19.37
240	8.308	10.42	10.6	156.1	110.1	29.24	44.76	25.59	25.06	14.83	27.18	13.07	23.74	5.483	14.3	25.96	19.37
244	8.308	10.42	10.6	155.4	110	29.24	44.81	25.59	25.06	14.83	27.18	13.07	23.74	5.464	14.48	25.96	19.37
248	8.308	10.42	10.6	155.5	110.3	29.24	44.84	25.59	25.06	14.83	27	13.07	23.74	5.468	14.3	25.96	19.4
252	8.308	10.42	10.6	156.1	111	29.24	44.86	25.59	25.06	14.83	27.18	13.07	23.74	5.456	14.3	25.96	19.4
256	8.308	10.6	10.6	156.2	109	29.21	44.89	25.59	25.06	14.83	27	13.07	23.74	5.475	14.3	25.96	19.4
260	8.308	10.42	10.69	156.9	110.1	29.21	44.92	25.59	25.06	14.83	27	13.07	23.74	5.471	14.3	25.96	19.42
264	8.308	10.42	10.69	156.8	109.4	29.21	44.94	25.59	25.06	14.83	27	13.07	23.74	5.46	14.3	25.96	19.42
268	8.308	10.42	10.69	156.4	110.3	29.21	44.97	25.59	24.89	14.83	27	13.07	23.65	5.452	14.3	25.96	19.42
272	8.308	10.42	10.69	156.3	110.7	29.19	44.99	25.41	24.89	14.83	27	13.07	23.74	5.464	14.3	25.96	19.42
276	8.308	10.6	10.69	155.8	109.9	29.19	44.99	25.41	24.89	14.83	27	13.07	23.65	5.46	14.3	25.96	19.45
280	8.308	10.6	10.69	156.7	110.5	29.19	45.02	25.41	25.06	14.83	27	13.07	23.65	5.456	14.3	25.96	19.45
284	8.308	10.6	10.69	156.7	111	29.19	45.05	25.41	25.06	14.83	26.83	13.07	23.65	5.479	14.3	25.96	19.45
288	8.308	10.6	10.69	156.1	110.5	29.19	45.07	25.41	24.89	14.83	26.83	13.07	23.65	5.464	14.3	25.96	19.47

292	8.308	10.6	10.69	156	110.1	29.19	45.1	25.41	24.89	14.83	26.83	13.07	23.65	5.456	14.13	25.98	19.47
296	8.308	10.6	10.69	154.7	109.4	29.16	45.1	25.41	24.89	14.83	26.83	13.07	23.65	5.464	14.13	25.98	19.47
300	8.308	10.6	10.69	156.4	110.5	29.16	45.13	25.24	24.89	14.83	26.83	13.07	23.65	5.46	14.13	25.98	19.5
304	8.308	10.6	10.69	157.2	111.1	29.16	45.15	25.24	24.89	14.83	26.83	13.07	23.65	5.464	14.13	25.98	19.5
308	8.308	10.6	10.69	156.4	109.6	29.16	45.18	25.24	24.89	14.83	26.83	13.07	23.65	5.468	14.3	25.98	19.5
312	8.308	10.6	10.69	156.1	110.3	29.16	45.2	25.41	24.89	14.83	26.83	13.07	23.65	5.449	14.3	25.98	19.53
316	8.308	10.6	10.69	157	110.3	29.16	45.2	25.24	24.89	14.83	26.83	13.07	23.65	5.434	14.13	25.98	19.53
320	8.308	10.6	10.69	155.9	110.8	29.16	45.23	25.24	24.89	14.83	26.65	13.07	23.65	5.43	14.13	25.98	19.53
324	8.308	10.6	10.69	156.6	110.5	29.16	45.26	25.06	24.89	14.83	26.83	13.07	23.65	5.449	14.13	25.98	19.55
328	8.308	10.6	10.69	156.4	110	29.16	45.26	25.24	24.89	14.83	26.65	13.07	23.65	5.434	14.13	25.98	19.55
332	8.308	10.6	10.77	156.3	110.4	29.16	45.28	25.06	24.89	14.83	26.65	13.07	23.65	5.43	14.13	25.98	19.55
336	8.308	10.6	10.77	156.4	110.3	29.16	45.31	25.06	24.89	14.83	26.65	13.07	23.65	5.438	14.13	25.98	19.58
340	8.308	10.6	10.77	157.2	110.1	29.14	45.31	25.06	24.71	14.83	26.65	13.07	23.65	5.445	14.13	25.98	19.58
344	8.308	10.6	10.77	156.7	109.3	29.14	45.33	25.06	24.71	14.83	26.65	13.07	23.65	5.438	14.13	25.98	19.58

There are 87 sets of data