AIR-CONDITIONING

SYSTEM DESIGN FOR OPTIMUM

CONTROL PERFORMANCE IN HONG KONG

by

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ABSTRACT

of a PhD Thesis on

AIR-CONDITIONING SYSTEM DESIGN FOR OPTIMUM CONTROL PERFORMANCE IN HONG KONG

by W. F. Ho

Studies on design for control optimization of air-conditioning (a/c) system for better performance in Hong Kong are reported in this thesis. Typical plant configuration data was collected from an in-depth survey of a/c systems and control used in Hong Kong. Control performance has been used for the first time as an objective for optimizing a/c system designs. The study investigates and illustrates that optimization of a/c systems for application in the Hong Kong by simulation is promising and flexible. The accuracy of simulation is enhanced by using the survey data. The survey shows that some a/c control systems and their control strategies are not well considered in the design stage and their operation and set-up are not properly addressed. Hence, there exists optimization opportunities in the a/c system design and control strategies for a/c systems used in Hong Kong. Parameters affecting the control performance of a/c systems were investigated by carrying out experiments. Identified parameters are the objective function of optimization, controller settings, control valve and drive and, in case of direct digital control, sampling rate. The influence of these factors on the control performance is an essential consideration for the entire optimization process. Strategies in applying the findings in optimizing an a/c system for control performance by simulation were developed and suggested. This study provides platform for further simulation study of optimization in both methodologies and control strategies for a/c system design and operation.

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Table of Contents

		Page
Abstract		
Certificate of Originality		
Table of Contents		**i**
List of Tables		**vii**
List of Figures		**xi**
Chapter 1	**Introduction**	**1**
1.1	Approaches	3
1.2	Outline of the thesis	7
Chapter 2	**Optimization of Air-conditioning System Design**	**9**
2.1	Overview of optimization of a/c systems	9
2.2	Importance of control performance in the optimization of a/c system design	14
2.3	Optimization problems and methodologies	15

Page

2.4	Design criteria - objective function or performance index	16
	2.4.1 Error performance indexes	17
	2.4.2 Objective function based on cost	19
	2.4.3 Performance index derived from fuzzy rule-based approach	20
Chapter 3	**Survey of Air-conditioning System Installed in Hong Kong**	**21**
3.1	The survey	21
3.2	Survey results on the building size and form	22
3.3	Surveyed results for the air-conditioning systems (primary plant)	23
3.4	Surveyed results for the air-conditioning systems (secondary plant)	27
	3.4.1 Air-conditioning air equipment and systems	27
	3.4.2 Details of a/c air system designs	29
	3.4.3 Control strategies	34
3.5	Survey results on a/c system performance issues	41
3.6	Conclusion of the survey	43

Page

Chapter 4 **System Simulation Program, HVACSIM+ as Example** **46**

4.1 The simulation program 46

4.2 Overview of MODSIM and simulation by HVACSIM+ 48

4.3 Limitations of MODSIM 50

4.4 Selected applications of HVACSIM+ 51

Chapter 5 **The Preliminary Simulation Study of a Simple A/C Sub-System Example for Control Performance** **54**

5.1 Sizing of 'realistic' a/c cooling coil for simulation 55

5.2 The objective function for the preliminary simulation study 55

5.3 The effect of the controller settings 55

5.4 The simulation study 56

5.4.1 The application of survey data from chapter 3 for simulation study of the a/c cooling coil sub-system

5.4.2 Definition of the a/c sub-system for the optimization study for weather conditions in Hong Kong from the data collected in the a/c survey in chapter 3 58

5.4.3 Preliminary simulation study of the simple a/c sub-system for optimum control performance 58

Page

Chapter 6	**Investigation of Parameters for Control Performance of A/C System by Experimental Studies**		**74**
6.1	The control valve characteristics and sizing		75
	6.1.1	The experimental study on the characteristics of control valve and actuator drive	76
	6.1.2	The application of experimental results in the HVACSIM+ simulation for optimization study of a/c system	90
6.2	The sampling rate		90
	6.2.1	The experimental study of sampling rate on the control performance of an air duct DDC temperature system	91
	6.2.2	Selection of sampling rate in relationship with the dynamic responses of the system and temperature sensor	104
6.3	The objective function		106
6.4	The controller settings		108
6.5	Conclusions		111
Chapter 7	**Detail Simulation Study for Optimum Control Performance of a Practical Single Zone CAV A/C System**		**112**
7.1	The system		113
7.2	The effect of control valve characteristics on the system control performance		115

Page

7.3 The controller tuning methods 115

7.4 The objective function for optimization study 116

7.5 Detail simulation study of control performance of a practical single zone CAV a/c cooling system 116

7.5.1 The application of the data collected in the a/c survey in chapter 3 for detailed simulation study for optimization of the practical a/c system 116

7.5.2 The simulation 119

7.6 The results 133

7.7 Analysis 159

7.7.1 The comparison of different controller tuning methods on the control performance of the a/c cooling system 159

7.7.2 The objective function for assessing the control performance of the a/c cooling system 161

7.7.3 Comparison of control performance at different time constants 161

7.7.4 Comparison of control performance at different valve resistances 162

7.7.5 The optimization of cooling coil design by step change test 162

7.8 Conclusion 169

Page

Chapter 8 Conclusions **170**

8.1 The flexibility of applying simulation study for optimum control of practical a/c system 172

8.2 Parameters affecting the controls performance of a/c system 173

8.3 The development of strategies for improving a/c design for control performance by simulation 175

8.4 Further study 176

References 178

Appendix 1 Questionnaire for survey on air-conditioning systems A-1

Appendix 2 Brief summary of HVAVSIM+ models used in the simulation study A-16

Appendix 3 Related journal and conference papers A-38

List of Tables

Page

Chapter 3

Table 3.1	The nature of buildings and their distribution	22
Table 3.2	Distribution of building height	22
Table 3.3	Distribution of the floor areas of buildings	23
Table 3.4	The % distribution of the number of chillers in the a/c plant against their cooling capacity range	24
Table 3.5	Distribution of different types of heat rejection system used in the a/c system	26
Table 3.6	Distribution of different types of chilled water distribution system	26
Table 3.7	Distribution of different control schemes for the chilled water distribution system	27
Table 3.8	Distribution of different types of air distribution systems in buildings	27
Table 3.9	Distribution of different combinations of a/c air side systems	28
Table 3.10	Frequency of application of different types of a/c system	29
Table 3.11	Control scheme for PAHU	34
Table 3.12	Control scheme for FCU	34
Table 3.13	Control schemes for central AHU (air handling unit) of VAV multizone	35
Table 3.14	Control schemes for zone VAV box of VAV multizone	35
Table 3.15	Control schemes for VAV single zone	36
Table 3.16	Control schemes for CAV multizone	36
Table 3.17	Control schemes for CAV single zone	37

Page

Table 3.18	Distribution of different controller actions used in a/c control	39
Table 3.19	Distribution of different parameters on the concerns of the a/c system performance	41
Table 3.20	Distribution of criteria for evaluating the performance of a/c system	42

Chapter 5

Table 5.1	Simulation configuration of the system	61
Table 5.2	Simulation configuration of superblock 1	63
Table 5.3	The five cooling coil designs	66
Table 5.4	The mean square errors and other error based objective functions of the five cooling coil designs	70
Table 5.5	Performance of design vs objective function	70

Chapter 6

Table 6.1	Time constants of control valve actuators	83
Table 6.2	Rangeabilities of the control valves	84
Table 6.3	Controller estimates based on Nichols and Ziegler method	85
Table 6.4	The mean square errors of the supply air temperature control of different control valve sizes and types	88
Table 6.5	Experimental results and mean square errors of step responses of an air duct DDC system	103

		Page
Table 6.6	Time constant of temperature sensor and, time constant and dead Time of the air duct temperature DDC system at air velocity 0.7 m/s and 1.1 m/s	105
Table 6.7	Comparison of mean square errors and energy consumption of step responses of an air duct temperature DDC system	107
Table 6.8	Valves of objective functions of air duct temperature DDC system at air velocities of 0.7 m/s and 1.1 m/s at sampling rate = 15 s and controller settings P = 54, I = 120 s and D = 30 s	109
Table 6.9	Valves of objective functions of air duct temperature DDC system at air velocity = 1.1 m/s and with Nichols and Ziegler estimated controller settings P = 51, I =120 s and D =30 s	110

Chapter 7

Table 7.1	Simulation configuration of superblock 1, 2, 3 & initial conditions	124
Table 7.2	The five cooling coil designs	131
Table 7.3	The estimated P+I values for the simulation tests with control Valve resistance = 0.015 kg^{-1} m^{-1}	131
Table 7.4	The estimated P+I values for the simulation tests with control Valve resistance = 0.03 kg^{-1} m^{-1}	132
Table 7.5	Energy consumption, initial and final steady state zone return air Temperature, control criteria of return air temperature step response for control valve resistance = 0.015 kg^{-1} m^{-1}	154
	i Control valve actuator time constant = 21.5 seconds	154
	ii Control valve actuator time constant = 10 seconds	155

Page

Table 7.6 Energy consumption, initial and final steady state zone return air Temperature, control criteria of return air temperature step response for control valve resistance = 0.03 $kg^{-1}\ m^{-1}$ 156

i Control valve actuator time constant = 21.5 seconds 156

ii Control valve actuator time constant = 10 seconds 157

Table 7.7 Comparison of cooling coil design against performance measures of the step response of zone return air temperature of the single zone a/c system, controller estimate based on Lopez et al minimizing integral absolute error 163

i Control valve resistance = 0.015 $kg^{-1}\ m^{-1}$, control valve actuator time constant = 21.5 seconds 163

ii Control valve resistance = 0.015 $kg^{-1}\ m^{-1}$, control valve actuator time constant = 10 seconds 164

iii Control valve resistance = 0.03 $kg^{-1}\ m^{-1}$, control valve actuator time constant = 21.5 seconds 165

iv Control valve resistance = 0.03 $kg^{-1}\ m^{-1}$, control valve actuator time constant = 10 seconds 166

Table 7.8 Performance of the five cooling coil designs based on the five point grading system against the corresponding mean square error under the specific values of the control valve resistance and actuator time constant 168

List of Figures

		Page
Chapter 4		
Figure 4.1	Complete architecture of the HVACSIM+	47
Figure 4.2	Configuration of the building emulator for control evaluation	53
Chapter 5		
Figure 5.1	Schematic for cooling coil temperature control system with components and types shown	60
Figure 5.2	Connection of components of cooling coil temperature control system	62
Figure 5.3	Cooling coil outlet air temperature control of CCD1	67
Figure 5.4	Cooling coil outlet air temperature control of CCD2	68
Figure 5.5	Cooling coil outlet air temperature control of CCD3	68
Figure 5.6	Cooling coil outlet air temperature control of CCD4	69
Figure 5.7	Cooling coil outlet air temperature control of CCD5	69
Figure 5.8	Cooling coil outlet air temperature control of CCD1 for 11 July 1989, Hong Kong	71
Figure 5.9	Cooling coil outlet air temperature control of CCD4 for 11 July 1989, Hong Kong	71

Page

Chapter 6

Figure 6.1	Experimental set-up for study of control valves in the supply air temperature control of air indoor aerodynamic chamber	77
Figure 6.2	Control valve characteristics, 1/2" pneumatic control valve	79
Figure 6.3	Control valve characteristics, 3/4" pneumatic control valve	80
Figure 6.4	Control valve characteristics, 1" pneumatic control valve	80
Figure 6.5	Control valve characteristics, 1 1/4" pneumatic control valve	81
Figure 6.6	Control valve characteristics, 3/4" electric control valve	81
Figure 6.7	Control valve characteristics, 1" electric control valve	82
Figure 6.8	Control valve characteristics, 1 1/4" electric control valve	82
Figure 6.9	Room supply air temperature control by (i) 1/2" pneumatic Control valve, (ii) 3/4" pneumatic control valve, (iii) 1" pneumatic control valve, (iv) 1 1/4" pneumatic control valve	86
Figure 6.10	Room supply air temperature control by (i) 3/4" electric control Valve, (ii) 1" electric control valve, (iii) 1 1/4" electric control valve	87
Figure 6.11	Digital control of an air duct temperature control system	93
Figure 6.12	PID tuning for the air duct temperature control system	94
Figure 6.13	Air duct air temperature control with sampling rate = 1 s	96
Figure 6.14	Air duct air temperature control with sampling rate = 2 s	96
Figure 6.15	Air duct air temperature control with sampling rate = 5 s	97
Figure 6.16	Air duct air temperature control with sampling rate = 8 s	97
Figure 6.17	Air duct air temperature control with sampling rate = 11 s	98
Figure 6.18	Air duct air temperature control with sampling rate = 15 s	98
Figure 6.19	Air duct air temperature control with sampling rate = 20 s	99
Figure 6.20	Air duct air temperature control with sampling rate = 25 s	99

Page

Figure 6.21 Air duct air temperature control with sampling rate = 30 s 100

Figure 6.22 Air duct air temperature control with sampling rate = 35 s 100

Figure 6.23 Air duct air temperature control with sampling rate = 40 s 101

Figure 6.24 The mean square error against energy consumption for each sampling time 108

Chapter 7

Figure 7.1 Schematic diagram of CAV single zone control system 114

Figure 7.2 Schematic for single zone CAV control system with components and types shown 120

Figure 7.3 Room return air temperature response for CCD1, control valve actuator time constant =21.5 s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 134

Figure 7.4 Room return air temperature response for CCD2, control valve actuator time constant =21.5 s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 135

Page

Figure 7.5	Room return air temperature response for CCD3, control valve actuator time constant =21.5 s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	136
Figure 7.6	Room return air temperature response for CCD4, control valve actuator time constant =21.5 s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	137
Figure 7.7	Room return air temperature response for CCD5, control valve actuator time constant =21.5 s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	138
Figure 7.8	Room return air temperature response for CCD1, control valve actuator time constant =10s, control valve resistance = 0.015 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	139

Page

Figure 7.9	Room return air temperature response for CCD2, control valve actuator time constant = 10 s, control valve resistance = 0.015 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	140
Figure 7.10	Room return air temperature response for CCD3, control valve actuator time constant =10 s, control valve resistance = 0.015 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	141
Figure 7.11	Room return air temperature response for CCD4, control valve actuator time constant =10 s, control valve resistance = 0.015 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	142
Figure 7.12	Room return air temperature response for CCD5, control valve actuator time constant =10 s, control valve resistance = 0.015 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	143

Page

Figure 7.13 Room return air temperature response for CCD1, control valve actuator time constant =21.5 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 144

Figure 7.14 Room return air temperature response for CCD2, control valve actuator time constant =21.5 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 145

Figure 7.15 Room return air temperature response for CCD3, control valve actuator time constant =21.5 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 146

Figure 7.16 Room return air temperature response for CCD4, control valve actuator time constant =21.5 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 147

Page

Figure 7.17	Room return air temperature response for CCD5, control valve actuator time constant =21.5 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	148
Figure 7.18	Room return air temperature response for CCD1, control valve actuator time constant = 10 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	149
Figure 7.19	Room return air temperature response for CCD2, control valve actuator time constant = 10 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	150
Figure 7.20	Room return air temperature response for CCD3, control valve actuator time constant = 10 s, control valve resistance = 0.03 $kg^{-1} m^{-1}$, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon	151

Page

Figure 7.21 Room return air temperature response for CCD4, control valve actuator time constant = 10 s, control valve resistance = 0.03 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 152

Figure 7.22 Room return air temperature response for CCD5, control valve actuator time constant =10 s, control valve resistance = 0.03 kg^{-1} m^{-1}, for PI estimates based on (I) Nichols & Ziegler, (ii) Lopez et al integral error square, (iii) Lopez et al integral absolute error, (iv) Lopez et al integral product of absolute error and time, (v) Cohen & Coon 153

CHAPTER 1 INTRODUCTION

The optimization studies of air-conditioning (a/c) systems for achieving energy saving, comfortable indoor environment, efficient building operation and maintenance have been reported in the literature [1 to 11]. Different approaches were proposed. Wright and Hanby[1], and Braun et al.[2] illustrated the approaches of component-based and simple system-based methods. Basically, these approaches use the concept of building system from component models. These models are developed and are readily available to be coupled to build the system under different design conditions. Manufacturers' data is used in building component models. Hence, in these methods, effort has to be spent in converting the manufacturers' data into the format usable by the model. Wright[3] and Albert[4] applied genetic algorithms for optimizing the a/c system designs. Genetic algorithms are clearly computational by computer and local maxims or minims during the optimization search can be avoided. Dexter and Trewhella used fuzzy logic[5] in studying the performance of a simple air-conditioning system. Athough fuzzy rule-based approach gives plausible assessment of the performance of the building control system, it requires care in interpreting the results. Besides, it is required that it should be capable of incorporating qualitative opinion and resolving conflicting points of view. Cascia[6] and Cumali[7] used adaptive methods in optimizing the energy consumption and plant operation of a/c systems. The features of direct digital control (DDC) were ultized in implementing the adaptive methods, yet Cascia did not report the actual energy savings upon the implementation of the adaptive schemes and the use of the schemes to specify equipment selection. Also only hourly simulation was used for optimizing the a/c plant operation. Ho[8], Ho and Ng[9] compared different direct search algorithms by direct computer search methods for tuning DDC controllers. The direct search algorithms are simple and can be computed easily. They do not require thorough knowledge of the system under studied. Nevertheless, the search requires a number of step-run tests and is time consuming thus causes inconvenience for commissioning of real systems. Massie et al.[10] used neural networks to predict the performance of a central a/c plant. The neural networks could be updated in response to change in the operation characteristics. However, the generality of the network increases for multi-network approach. So it is critical to determine the peritinent independent variables and use them for training the network. Gatton et al.[11] applied the technique of knowledge-based expert system to create energy-saving retrofit potential of public buildings. Such technique requires full understanding of expert knowledge and sometimes experiences difficulties with new change of plant characteristics.

Air-conditioning design based on static analysis is not sufficient to achieve optimum system performance. Traditionally, consultant engineer would design a/c systems according to his engineering knowledge, conventional practice and rule of thumb. Sometimes, the main objective of design focuses on satisfying the comfort of the occupancy without complaints from them. This focus is to be maintained provided that the design is within the project sum limit. Most of the time, a/c systems are oversized. Further optimization in the initial and maintenance cost as well as better comfort for the occupancy is often ignored. In fact, system stability and controllability are important issues to be addressed as they affect the comfort and hence productivity of the building occupancy. They are strongly related to system maintenance cost and energy required maintaining the indoor comfort. Therefore in studying the optimization design of a/c system, control performance of the system cannot be overlooked. Unfortunately, the current optimization researches of a/c systems do not or seldomly focus on the control performance. In practice, it is very difficult to predict the control performance of a/c systems in the design stage as the performance may be changed due to the shift in the design conditions, component as well as system characteristics after installtion. With the advance and development of simulation in a/c systems, it is now possible to simulate an a/c system and predict its control performance during the design stage. Optimization of the system design could be achieved by optimization of the physical and non-physical parameters of the system.

The research aims at studying, by simulation, the optimization for control performance of a/c system design for a/c installations in Hong Kong. The parameters affecting the control of a control system are investigated. These are the parameters that affect the control performance of a/c design. Both experimental study and simulation study using commercially available simulation software are carried out. To effect the simulation, a criterion that measures the control performance has to found. The criterion should be able to quantify the control performance and easily computed. In order that the simulation to be realistic and applicable in Hong Kong, weathers data and data of a/c systems used in Hong Kong are used. Hence an a/c system survey was carried out to collect data on the a/c systems and their controls used in Hong Kong. The effect of the control parameters for control performance was studied experimentally for Hong Kong local environmental conditions. The results of the experimental studies can be used and integrated with the simulation to improve and enforce the effectiveness of the simulation study. It is aimed that the results of the study will demonstrate and conclude the importance of control in the optimization of a/c design. It is also hoped that the study will help to formulate and show how to implement the optimization of a/c system design by simulation for control performance.

1.1 APPROACHES

In the research, a simulation study of an a/c sub-system and a practical constant air volume (CAV) a/c air handling unit supplying cold air to a single zone was performed. These simulations employ practical data collected from the a/c system and control survey for a/c installations in Hong Kong. In such a way, the results of the simulation can be applicable in Hong Kong. The preliminary simulation study of the a/c sub-system tests the feasibility of simulation and serves to identify the possible objective function to be used for optimization study of a/c cooling system design. In the detailed simulation, a CAV a/c cooling system supplying cold air to a single zone is studied. In the detailed study, the interaction between a CAV a/c cooling system and a zone is investigated for the control performance. This detailed simulation of a fundamental a/c system interacting with an air-controlled zone can be extended to cover an overall a/c design and should be considered for future study. In the simulation study, the performance of the a/c sub-system and the simple single zone a/c system were evaluated for its control performance against an appropriate objective function. The simulation study demonstrates how optimization of a/c system design can be studied and evaluated by control performance. Factors affecting the control performance of a/c system were studied experimentally and by simulation. These factors include objective function, controller settings and control valve characteristics. Although direct digital control (DDC) is not included in the simulation study, the effect of sampling rate for DDC was investigated experimentally. The results of which can be used when optimization of DDC a/c cooling system is considered in a future study. The approaches to the research are described as follow.

In optimization studies of a/c system, the first task is to formulate an objective function or performance index, which is a function of system parameters satisfying the requirements of different groups of people, including the occupants, building owner, maintenance engineer, designer etc. The objective function or performance index is a number, which is usually a function of error of the controlled variable, indicating the goodness of system performance. The system performance is considered optimal if the values of the system parameters are chosen such that the selected objective function is either a minimum or maximum. Using a selected objective function, results of different optimization methods for optimum design and performance of a/c systems can be evaluated and compared for the objective function. The objective function used for evaluating the performance of a system should be characterized for the system concerned. Different objective functions have been used and different optimization methods appeared in the literature. These objective functions are also discussed in the objective function section of the following chapter 2. In the performance analysis of a/c system, thermal comfort and energy consumption, both measures of the control performance are the prime criteria. From Fisk's argument[12], the objective function, mean square error when used as a criterion for temperature control systems was both one of easiest computed, and one that related to both the mean value of the Predicted Percentage Dissatisfied (PPD) in assessing thermal comfort suggested by Fanger[13] and to the probability of occupants' acceptance to change of their thermal environment. It was also argued that the criteria suitable for design with optimal control theory would at the same time have implications for predicting the energy saving that could be associated with conventional thermal controls. Fisk[12] made general conclusions on the use of mean square error as the control system criterion that despite its simplicity, it appeared to provide a very useful penalty function approximation for thermal control work.

In order to verify the use of the mean square error as an objective function in assessing the control performance of a/c system, a simulation study of a simple a/c sub-system was firstly carried out. The simulation program used for study is the HVACSIM+[14], which was developed at the National Bureau of Standards (NBS), U.S.A. In the study, different sizes of cooling coil were tested. The resulting mean square errors and other conventional error based objective functions for different coils were compared against their control performance. The test shows that mean square error seems to be a good objective function in thermal work. This argument is further confirmed in the detailed simulation of a CAV single zone a/c cooling system for optimum control performance design.

A survey on the a/c systems and their control strategies used in Hong Kong was conducted to collect data for the simulation study. The purpose of this survey exercise is to collect practical data and design conditions of a/c systems and their control strategies locally. The data collected from the a/c survey is important to be used in the simulation, as to simulate the real case and application in Hong Kong. The simulation is also aimed to evaluate the effectiveness of the objective function to be used aginst conventional control performance criteria in assessing the control performance of a/c control system.

In the surevy, the a/c systems and their control strategies are also analyzed and commented. One questionnaire was designed. The questionnaire requests detail information of a/c systems and their control as well as the building details. Three hundred and fifty copies were sent to different parties of buildings and the data were analyzed.

Working with the a/c data collected from the a/c system survey, a preliminary optimization study, with technical data collected, for a simple a/c sub-system used in Hong Kong was conducted by simulation. In the preliminary optimization study, different conventional error based objectivr functions were evaluated and compared for comtrol performance. It is found that mean square error objective function is good and appropriate for use in assessing the control performance of a/c sub-system. This study shows how control performance can be applied to the optimization study of a simple a/c sub-system. It also suggests that mean square error could be used as an objective function in the optimization study of a/c system design.

The accuracy and performance of the simulation study are greatly affected by the control parameters of the a/c control sub-system simulated. In the preliminary simulation study, each of the simulated a/c control sub-system designs was first tuned for controller settings by the open loop method suggested by Nichols and Ziegler[15]. Nichols and Ziegler open loop test was used in the preliminary simulation because of its simplicity of use as reported in (16).

The other major control parameters affecting practically the control performance and, hence the performance of simulation are, the control valve and its drive, and in the case of direct digital control, the sampling rate. The weather in Hong Kong for the cooling season from May to September is hot and humid (in the range of mean daily maximum dry bulb temperature of about 26 to 31°C, and mean daily relative humidity ratio of 75 to 85%). Although free cooling control or dry bulb economizer for a/c systems are possible in the marginal weather months from October to April, only a few commercial buildings in Hong Kong have their a/c systems installed with these types of energy control. Also only 15% of commercial buildings are installed with central heating a/c system (information from the survey discussed in chapter 3). In fact, over 80% of buildings employing central a/c system are installed with year-round cooling a/c systems. The control of these a/c systems focuses on the temperature control of supply, return and indoor air. Acceptable humidity level is designed to be achieved through the temperature control. The features of these a/c control systems and weather conditions need special consideration in the simulation study. The effect on the control performance of a/c control systems used in Hong Kong by the sampling rate, control valve and its drive has not been investigated and reported. To enhance the simulation for practical application in Hong Kong, their effects are studied experimentally for a/c control systems built against local Hong Kong weather conditions. The results of the experimental studies could be used in the simulation study so as to improve the accuracy and reality of the simulation for local practice.

In the experimental study on the effect of sampling rate, the sampling rate of DDC was studied for its influence on the control performance of an a/c system. An air duct temperature DDC control system was built to evaluate experimentally the effect of sampling rate on the control performance of the air duct temperature control system.

Both the size and characteristic of the control valve used affect the control performance of a control system. A correctly sized control valve is a prime factor in achieving a desired control for a control system. Also, the control valve characteristics will affect the dynamic behaviour of the control system. A test rig used to test the dynamic characteristics of control valve was set up. Both electric and pneumatic control valves were tested and compared.

Using the experimental results and survey data collected, a further detailed simulation study of a CAV a/c cooling system supplying cold air to a single zone was studied for control performance. Unfortunately, the simulation software HVACSIM+ simulates only analog control system and hence, the effect of sampling rate cannot be studied by the HVACSIM+. However, the experimental results on sampling rate will be useful and has to be considered when future study on optimization of DDC a/c system is carried out.

1.2 OUTLINE OF THE THESIS

Having defined the research objectives, the thesis explores and outlines in details the research approaches focusing to achieve the research objectives. In chapter 1, the research objectives and approaches to tackle the objectives are defined. In chapter 2, a full literature survey on the optimization of a/c systems is conducted. The survey reveals the initiatives of using control performance in assessing a/c system design. Chapter 3 outlines the need and the fieldwork in collecting technical design data and control strategies of a/c systems used in Hong Kong. The data collected is to be used in the simulation study of a/c systems. The simulation study aims to illustrate the usefulness of simulation in the optimization study of a/c systems and to demonstrate how objective function is related to and can be used to evaluate the control performance of an a/c system. The tool for the simulation study is the simulation program, HVACSIM+, which is described in detail in chapter 4. In chapter 5, a preliminary and test simulation study of a simple a/c sub-system example is described. The simulation uses practical a/c system and control data collected in the a/c and control survery. The simulation illustrates how control performance of a simple sub-a/c control system can be studied by simulation. It also compares different objective functions in assessing the control performance of the a/c system. Chapter 6 focuses on studying the major parameters of control system that affect the control performance of a/c systems. The parameters are examined experimentally and their effects on the control performance are fully investigated. In chapter 7, a detailed simulation of a single zone CAV a/c cooling system with the data collected in the a/c survey exercise was carried out. The simulation shows that the mean square error is a promising objective function to be used to study the control performance of a/c cooling systems. It also shows that simulation is a useful tool for optimization study of a/c cooling systems. The final chapter 8 concludes the results of the research from simulation and experimental studies in exploring the a/c system design for optimum control performance.

CHAPTER 2 OPTIMIZATION OF AIR-CONDITIONING SYSTEM DESIGN

2.1 OVERVIEW OF OPTIMIZATION OF A/C SYSTEMS

Optimization studies of a/c systems would enable engineers to achieve the best a/c systems under given constraint conditions. Usually, under the constraint conditions of initial cost, space and operation conditions, an a/c design engineer will try to optimize the a/c design such as to provide the desired internal thermal comfort conditions for least energy consumption or least cost of maintenance and operation of individual components. Traditionally, in the design process of a/c systems, engineers would prepare a set of system schematic diagrams for the a/c systems with full consideration on the space requirement for installation and maintenance. Sizing and matching of the individual components are determined by the design conditions such as the loading of the systems. Engineers would design the system based on their experience, and perhaps using standard charts, tables, etc. provided in the CIBSE guides, ASHRAE standards or handbooks. The actual performance of individual components cannot be estimated at the design stage, and optimization studies (if exists!) cannot be performed in a systematical fashion. This might be one reason why so many a/c systems are oversized.

With the development of system simulation on describing the performance of the individual components by mathematical models, it is possible to estimate how the entire system will work and perform upon changing the operation conditions of the components. Using the technique of system simulation, the effects of changing different components can also be studied. This would enable the optimization studies of the a/c systems to be performed at the design stage. In an optimization study, objective function of the system, which is the goal of optimization, has to be defined. This would give a numerical value on how the proposed design is deviated from the optimum. Typical candidates for objective function are the energy consumption, initial cost, operating cost and the payback period. Optimal search algorithm is then used to search the system variables and equipment ranges to get the minimum values of the objective function. Choice of the search methods depends on the form of the objective function. Various studies focusing on improving the accuracy and methodologies of simulation for optimization of a/c design have been investigated.

In the literature, Hanby and Wright (1989) used component models[1]. There are four parts in the component model. Firstly, the component performance equations are expressed in terms of the system variables only. The equations are written in residual form and the different component equations are linked to form a system model. The second part is the energy model, which would put in the energy consumption of each component to the entire system. The third part is the cost model, which includes the initial and maintenance costs. The last part is the constraint model describing the constraints imposed on the system. In this way, with a good library of component models, performance of the systems under different conditions can be simulated and optimization design is achieved with suitable searching algorithm. Demonstration of the technique on heat recovery system for minimum energy consumption was reported. The component model should give the same results as the product performance data supplied by manufacturers. But different manufacturers would prepare the product data in their own format. Therefore, the product performance data must be converted to a format suitable for use in preparing the component model. Wright[17] illustrated this on developing a steady-state fan model with example referred to the part-load simulation of an axial flow fan in a simplified variable air volume system. In a similar type of approach on applying the technique of system simulation, Braun et al. proposed two methodologies[2] for determining optimum values of the independent control variables that minimize the instantaneous cost of operating chilled water systems in response to the uncontrolled variables. They are the modular component-based optimization algorithm and the 'simple' system-based methodology for near-optimum control. The component-based approach takes the advantage of providing a true solution to the optimization problem, but detail performance data is required. In contrast, the system-based near-optimal control methodology utilizes an overall system cost function and the approach is simpler. However, both methodologies have to be tested as part of an energy management system for controlling the actual system. In addition to that, hour-by-hour building energy simulation programs had been used to study the dynamic response of buildings under different choices of material and building component for thermal insulation, system sizing and operating strategies.

Works reported by Thun and Witte[18] using the energy simulation program BLAST is an example adopting the hour-by-hour simulation approach. There, the optimization of building envelope, optimization of room temperature control and optimization of air handling system and plant equipment on eight buildings were illustrated. It seems that hourly energy simulation programs can be used in conjunction with appropipate system component models for simulating the dynamic loading. In the detailed study of energy use in a high-rise building complex reported by Cumali[7], application of the global optimization technique to determine control and operation strategies in real time on a large scale was demonstrated. The optimization software was developed with the aid of the hourly energy simulation program DOE-2. Different function-minimization methods including the projected and/or augmented Lagrangian multiplier methods were tested and found to be unsuitable. Instead, a specialized reduced gradient code was developed for the study. User graphics interface was identified to be an important part in the whole study. More important, the potential use of optimization technique for a/c systems in buildings was illustrated clearly, and there would be tremendous opportunities for engineers to use this technique.

Also Shelton and Weber[19] reported updated modelling of chiller/cooling tower systems for commercial building using high efficiency chillers. Based on the component data supplied by the manufacturers, mathematical models were developed to simulate the performance of the chillers. With the developed models, optimum condenser water flow rate could be found. Green and Roberts[20] used modeling techniques to find out the control performance of an air-coil design by varying the design parameters.

Kimbara et al.[21] applied autoregressive integrated moving average (ARIMA) model for building load monitoring and optimization. The ARIMA model was updated everyday to decrease the difference between the predicated load and the actual load. The use of actual chiller plant performance data collected from an energy management system (EMS) for changing plant operating conditions to minimize energy consumption in a direct digital controller (DDC) was reported by Cascia[6]. Advanced DDC features were applied for optimal control including optimizing the staging on or off of multiple chillers; optimizing condensed water temperature; and optimizing chilled water temperature when variable speed chilled water pumps were used. Those three features were described clearly but the energy savings upon implementation of the schemes and the use of them to specify the plant equipment had not been reported.

An empirical model of a chilled water coil for predicting the system responses to inputs from different control algorithms (proportional, proportional+integral, Proportional+ integral+derivative) DDC was reported by Maxwell et al.[22]. Transient behaviour of a chilled water coil was investigated experimentally. Results were then applied to understand the dynamic behaviour of the DDC under different control algorithms. Neural network technique had been used in the optimization study of a/c systems. Massie et al.[10] used neural network to predict the performance of an ice-storage air-conditioning system. The neural network can be updated in response to change of operation characteristics. Curtiss[23] showed examples of neural networks used as tools for predictive process control and supervisory optimization. Kawashima et al.[24] studied the performance of an ice-storage system by neural network controller and concluded that the accuracy of load prediction was a key element for optimization control of operation cost.

In the optimization of a/c control systems, the system performances are affected by the controller settings besides system designs. Classical and simulation methods have been used to find 'optimum' controller settings during design stages. Yet these 'optimum' controller settings found may shift due to change of characteristics of control systems after installation. Research works have being done to find 'optimum' controller settings during real time operation by adaptive and auto-tuning methods. These concepts seen promising for a/c systems that usually have non-linear system components. Loveday et al.[25] employed advanced stochastic model-based predictive approach control algorithms, which were then tested for comparison against conventional control techniques. Curtiss et al.[26] explored the adaptive control of a/c systems by using neural networks (ANNs). The ANNs controller was applied for a/c system optimization similarly to other predictive controllers. The performance of

ANNs controllers developed was tested against a conventional controller. Wallenborg[27] designed a general linear discrete time controller using pole placement based on input-output models. Bekker et al.[28] used theory of pole-zero cancellation and root locus techniques to design controllers for fast-acting actuators and electric actuators commonly used in a/c control systems. Ho[8] illustrated the application of computer in performing Hooke and Jeeves computer search algorithms for optimum PID controller settings for a simulated room temperature control system. Different computer search algorithms were also tested and compared for an air duct temperature control system[9].

Rule-based controllers (expert systems) based on a set of control rules attracted much attention in these years. There are two basic types of rule-based controller. The first type is fuzzy logic type, which operates on rules based on the experience of plant operators. Fuzzy logic controller is further developed into self-organizing controller where an extra learning level has been added that can change the rules on which the control is based. The other type of rule-based controller is causal controller, which incorporates a knowledge base to describe the behavior of the system under control, and is able to reason in a simple way. Rule-based controllers are found better cope with non-linear behaviours of components of lowest level of controls such as air-conditioning plants and actuators where time-varying non-linear characteristics are found. John and Dexter[29] discussed in details the performance of a self-adaptive optimum-start controller, a rule-based self-organizing regulator for three-port valves and a self-tuning control scheme for cooling coils. Huang and Nelson[30] developed a PID-law-combining fuzzy controller (PFC) for a/c applications. The PFC was tested and verified by computer simulation.

Some of the controllers discussed above, such as in[8],[9],[16] have the ability of self-tuning during real time operation for optimum system performance. Ho & Ng[16] verified the idea of self-setting concept according to the Nichols-Ziegler method through the use of ADDA (analog to digital and digital to analog) by implementing the auto-tuning software on a programmable digital controller. They further explored the idea in the development and application of a series of self-setting computer programs to optimize the P and I settings of a direct digital controller for a simple air duct heating system[9]. The computer software developed is a series of optimum searching algorithms with power consumption as the objective function for the optimization of P and I setting of the digital controller. Ho[8] developed a computer package for self-tuning of digital PID controllers using a computer searching technique for optimization. The software program developed was verified against a simulated room model.

2.2 IMPORTANCE OF CONTROL PERFORMANCE IN THE OPTIMIZATION OF A/C SYSTEM DESIGN

In the a/c system design, engineers will try to achieve the required internal conditions with the 'best' design focusing considerations to the initial cost, system flexibility including future expansion, space utilization, maintenance and operation cost. Control performance, which is a compromise of system response, stability, accuracy as well as controllability, after installing the a/c system is usually overlooked. The control performance of the a/c system will affect the controlled conditions of the building, which in term will affect the comfort and hence the productivity of the building occupancy. It affects also the stability of the control system and therefore the life and maintainability of the a/c system. The stability and controllability of a/c system influence the thermal comfort of the occupants. Dexter[5],[29] mentioned the criteria for control performance for a/c system. The conventional design approach, which bases on the static analysis of the system performance, is obviously insufficient for achieving optimum design. Consideration must be given to "control" in the design stage. In assessing the control performance, parameters of control of the system must be investigated. Such parameters are the controller settings, objective function used for control performance, characteristics and sizing of the control valve, sampling rate for direct digital control. These parameters will be investigated experimentally and by simulation studies with practical weather data and technical data of a/c systems used in Hong Kong.

2.3 OPTIMIZATION PROBLEMS AND METHODOLOGIES

Optimization problems can generally be divided into unconstrained and constrained problem[31]. In the unconstrained optimization problem, the objective is to find the minimum or maximum value of the objective function in its domain of search space. In constrained optimization, the constrains divide the search space of objective function into two parts. The points in one part of the search space satisfying the constraints are called feasible points and the region is known as the feasible region. The other points in the another part of search space (non-feasible region) are called non-feasible points. The usual approach for solving constrained optimization problems is to convert them into unconstrained optimization problems by adding a penalty to offset the constraints.

Optimization methods are well discussed in much literature and an overview is described following. In the traditional methods[31], unconstrained optimization often uses calculus-based methods. In the classic theory, the first partial derivatives of the objective function must exist. A necessary condition for a minimum objective function is that the first partial derivatives of the objective function with respect to the independent variables must all equal to zero. The sufficient condition for a point to be a minimum is that all the second partial derivatives exit at that point. The practical difficulties in using this classical method are firstly, local minimum points may be obtained instead of the true minimum. Secondly, it is often difficult to observe the number of stationary points of the objective function from the first partial derivative equations. Lastly, the theory is not readily applicable to function with discontinuous derivatives. In the constrained optimization, the popular tool for solving constrained optimization problems by mathematician is the method of Lagrangian multipliers. Yet for real problems, complex method is used. Traditional methods of using iterative technique require an initial point x_0 to be defined. From the initial point, a sequence of points x_i, $i = 1, 2, 3, \ldots$ is produced such that the objective function $f(x)$ is improving each time, i.e. $f(x_{i+1}) <$ or $= f(x_i)$ where $x_{i+1} = x_i + hi.di$, di is an one-dimensional direction vector, and hi is a distance moved along it. Iterative methods can be divided into direct search and gradient methods.

Direct search methods do not require the explicit evaluation of any partial derivatives of the function. They are dependent only on the values of the objective function and information obtained in the early iterations. The search moves in a direction of improved objective function until minimum or maximum is reached. Gradient methods select the direction di using values of partial derivatives of the objective function with respect to the independent variables and values of the objective function itself, together with information gained from earlier iterations.

In the non-traditional method[(3),(4),(32)], the search space of objective function can be considered to have regions of good objective function values and regions of poor objective function values. Traditional methods of optimization need long and considerable computational time for optimum search. The idea to first differentiate regions of good and bad objective function values, and then apply traditional method(s) to search for optimization in those regions of good values of objective function gives rise to varius techniques in solving optimization problems. A number of these types of optimization technique are being used successfully, such as the use of genetic algorithms (GAs) by Wright[(3)] in problem optimization and optimal supervisory control through application of the simulated annealing method by Flake et al.[(33)].

2.4 DESIGN CRITERIA - OBJECTIVE FUNCTION OR PERFORMANCE INDEX

Various objective functions or performance indices have been used for different optimization methods for different systems. There is no unique answer for the best objective function. Take for an example of an a/c system in a building, there are different objective functions for different perspectives on the problem. Obviously, the building occupants aim for optimum comfort. The building owner would like the energy consumption to be minimum. The maintenance and operation teams wish to optimize the system for trouble free and user friendliness. The control engineer will look for optimization of the system stability and controllability. The unique solution for optimization if workable will be a function of these objectives. The requirements for a performance index are that it must distinguish between optimal and nonoptimal adjustments of the system parameters and yield a single positive or zero number. It should be a function of the system parameters and should be easily computed analytically and experimentally. Optimization of a system performance involves the adjustments of the system parameters to give minimum or maximum objective function. There are different objective functions used, which can be summarized as follows:

2.4.1 Error performance indexes[34]

Numerous error performance indices have been proposed in the literature for control systems. These are defined through the error function e(t), which is the difference between the set value and the instantaneous value of a parameter (e.g. temperature). There are several examples:

i. Integral absolute -error criterion (IAE):

$$\mathrm{IAE} = \int_0^\infty |e(t)|\, dt \qquad \ldots (2.1)$$

where t is the time; and e(t) is the error at time t and equals to the setpoint minus the value of controlled variable at time t.

This performance index can be applied easily and is found to be particularly useful for systems with reasonable damping and satisfactory transient-response characteristic. However, this index does not give a good selectivity between optimal and nonoptimal system parameter adjustments.

ii. **Integral-of-time-multiplied absolute-error criterion (ITAE):**

$$ITAE = \int_0^\infty t \cdot |e(t)|\,dt \qquad \ldots (2.2)$$

This criterion weighs a large initial error lightly and imposes heavy penalty to errors occurring late in the transient response. That is to say a control system employing this criterion reduces overshoot and gives well damping to oscillations. Although it provides good selectivity, there is no simple method to evaluate it analytically.

iii. **Integral-of-time-multiplied square-error-criterion (ITSE):**

$$ITSE = \int_0^\infty t \cdot e(t)^2\,d(t) \qquad \ldots (2.3)$$

The characteristics of this criterion are similar to ITAE. However its selectivity, though better than IAE, is not as good as ITAE.

iv. **Integral square-error criterion (ISE):**

$$ISE = \int_0^\infty e(t)^2\,dt \qquad \ldots (2.4)$$

This index weighs large errors heavily and small errors lightly. A control system using this criterion yields rapid decrease in a large initial error, thus the response is fast and oscillatory. The system is relatively not stable. Yet the practical significance is that minimization of this criterion results in the minimization of power consumption for some systems[34], such as spacecraft systems..

v. **Mean square error:**

If the error caused by a disturbance of amplitude $y(\omega)$ at frequency ω is given by:

$$e(\omega) = G(\omega).y(\omega),$$

where $G(\omega)$ is the gain.

Then by Parseval's theorem, the mean square error is given by:

$$| e(\omega) |^2 = | G(\omega) |^2 \cdot | y(\omega) |^2$$

For discrete sampling intervals, the mean square error is given by

$$[\sum_{i=0}^{k} \text{error}(i)^2] / (k+1)] \qquad \text{... (2.5)}$$

where i and k are integers and k is the number of time intervals.

According to Fisk[12], the mean square error provides a very useful penalty function approximation for thermal control work.

2.4.2 Objective function based on cost

A simple cost function can be derived from simple weighting techniques for multi-criteria decision-making problem. There, each of the system criteria is multiplied by user-defined weighting factors and the results are summed up to form a single cost function[35] for comparison. The single cost function is an objective function in which there is minimization of the total cost is used. Hanby and Wright[1] illustrated this concept in an example where an a/c system was optimized for minimum total cost that composed of the energy consumption, primary energy consumption, first cost, annual operating cost, net present value and discounted payback period. The objective function was obtained by the sum of the operation costs of each component in the system.

Further, Braun et al.[2] illustrated that the operating cost of a/c components such as chiller, pump and fan might be expressed in a general form as a quadratic function of its continuous control and/or output stream variables. The optimization problem for a/c systems was stated as the minimization of the sum of the operation costs of each component with respect to all discrete and continuous controls.

2.4.3 Performance index derived from fuzzy rule-based approach

Dexter and Trewhella[5] illustrated the use of fuzzy rule-based approach to performance assessment of an example building fan-coil control system in which the opinions of occupants, building manager and control engineer were used for fuzzy inference. In their case study, a set of heuristic rules is used to infer the degree of membership of the fuzzy sets describing the opinions of occupants, building manager and control engineer and the overall measure of performance. The values of the performance criteria are computed from test results and fuzzified to determine the degree of membership of the fuzzy sets that describe each criterion. Defuzzification of the whole fuzzy sets gives single numercial value of performance index.

CHAPTER 3 SURVEY OF AIR-CONDITIONING SYSTEM INSTALLED IN HONG KONG

A survey was carried out to collect data and details of the types of buildings in Hong Kong and their air conditioning (a/c) systems used. The information collected include building height and construction, building shell details, technical data of a/c water and air handling equipment and systems, a/c control strategies and schemes, a/c design conditions as well as criteria used to evaluate the performance of the a/c systems by the building personnel and engineering team.

The target of the survey is to identify typical examples of a/c systems and their control schemes for applications in Hong Kong. Simulation study for optimum control performance of the typical system example was then carried out. Typical a/c example system is selected from Hong Kong a/c system installations because Hong Kong is a wet and humid city, air-conditioning is an essential installation for any commercial building and complex. Such particular situation in Hong Kong obviously enhances the optimization studies and it is also wished that the results of the studies would benefit the design for comfort and energy efficient a/c system in Hong Kong as well as in other countries.

3.1 THE SURVEY

A questionnaire was designed for this surveying exercise. The questionnaire aims at surveying the information of the building envelope, the a/c system installation and the associated control systems. A copy of the questionaire is attached as appendix 1. Three hundred and fifty questionnaires were sent to building owners, building maintenance and engineering teams, architects, developers in October 1995 and January 1996 of which 47 completed questionnaires were returned. The response % for questionnaire is 13.43 %.

The data collected are analyzed and reported in this chapter.

3.2 SURVEY RESULTS ON THE BUILDING SIZE AND FORM

Most of the buildings under survey are high rise commercial buildings and 35.5% of buildings have curtain walls. The building use can be classified as shown in table 3.1.

Table 3.1: The nature of buildings and their distribution

Nature of building	%
Commercial	68.9
Industrial	8.9
Institutional	6.7
Hostel	2.2
Composite	6.7
Domestic	2.2
Other	4.5

The height distribution of buildings is shown in table 3.2. It is worth to point out that more than 77% of the surveyed building are over 10 storeys in height. This reflects the high cost of land and limited space for construction in Hong Kong. Tall building has a high static pressure head. So in the design of building services systems in Hong Kong, care has to be given to the pressure rating of services equipment used.

Table 3.2: Distribution of building height

Height of building	%
Above 50 storeys	6.7
30-50 storeys	31.1
10-29 storeys	40.0
Below 10 storeys	22.2

About 95% of the buildings have flat roof and approximately 45% of the buildings have basement(s) mainly used as carpark areas. Over 24.4% of the buildings are facing either South or North. The orientation of the building affects the solar heat gain to the building. However, it is indicated from the survey that no significant effort is considered by building architects in this energy aspect. Most of the building design has top priority of high rental value and the orientation effect on energy is often ignored. In fact, there is no significant solar effort since most of buildings in Hong Kong are square or rectangular and glazed on all facades. In addition, solar heat gain effect is often ignored in some of the commercial buildings in Hong Kong because the building tenants share the air-conditioning fee.

From table 3.3, over three-quarters of the buildings have area in the ranges less than 100,000 m^2.

Table 3.3: Distribution of the floor areas of buildings

Area of building (m^2)	%
0-50000	59.0
50001-100000	20.5
100001-150000	2.6
150001-200000	10.3
300000	2.6
480000	2.6
2000000	2.6

Remark: The buildings under surveyed are almost high rise commercial buildings.

3.3 SURVEYED RESULTS FOR THE AIR-CONDITIONING SYSTEMS (PRIMARY PLANT)

The most important part of the survey is on the a/c systems used in buildings of Hong Kong. From the feedback, 71% of the a/c systems are air and water systems[36].

Table 3.4 gives the percentage distribution of the chiller number against the corresponding cooling load range in the a/c plant.

Table 3.4: The % distribution of the number of chillers in the a/c plant against their cooling capacity range

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
Chillers						Ton range										
1	2.3															
2	2.3															
3	16	2.3					2.3						2.3			
4	4.5	6.8										2.3				
5	2.3			4.5	2.3	2.3				2.3				2.3		2.3
6		2.3														
7			2.3		2.3		2.3									
8				2.3												
9						2.3										
10	2.3			2.3												
11					2.3											
12		2.3														
18						2.3										
36						2.3										
Unknown				2.3					2.3							

UNKNOWN cooling capacity and number of chillers 9.1%

Legend:

A:	0 - 283 kW	B:	284 - 567 kW
C:	568 - 851 kW	D:	852 - 1135 kW
E:	1136 - 1419 kW	F:	1420 - 1703 kW
G:	1704 - 1987 kW	H:	1988 - 2271 kW
I:	2272 - 2555 kW	J:	2556 - 2839 kW
K:	2840 - 3123 kW	L:	3124 - 3407 kW
M:	3408 - 3691 kW	N:	3692 - 3975 kW
O:	3976 - 4259 kW	P:	4260 - 4543 kW

About 77%, 64% and 57% of the buildings have cooling load in the ranges of 0 to 1987 kW, 0 to 1419 kW and 0 to 1135 kW respectively. The common range for the cooling load is 0 to 283 kW, accounting for 30% of the buildings. This figure agrees to the popular type of buildings that has area in the range 0 to 5000 m^2 in Hong Kong.

The number of chillers in the a/c plant varies across the cooling load ranges. The combinations of 3, 4 and 5 chillers are the popular ones and account a total of 54.8% of the buildings. The most popular combination is the three-chiller plant with a cooling load of 0 to 283 kW. This design is found in 16% of the buildings.

The direct air cool, indirect seawater cool and direct seawater cool[37] are widely used as the a/c heat rejection methods (50%, 29% and 12% respectively). Summary of heat rejection systems is given in table 3.5. Direct air cool heat rejection is used in 50% of the a/c systems. Seawater cool systems, in spite of their high coefficient of performance, are constrained by space, initial cost and as well as the availability of seawater. The uses of cooling tower are prohibited for hygienic and water source reasons.

Table 3.5: Distribution of different types of heat rejection system used in the a/c system

Cooling system	%
Direct seawater cool	11.9
Indirect seawater cool	28.6
Direct air cool	50
Indirect air cool	7.1
Indirect fresh water cool	0
Seawater + cooling tower	2.4

For the chilled water distribution system, primary-secondary decoupler bypass and direct return systems[37] are the most popular systems used (46% and 35% respectively). A minority of 19% employs the reverse-return open-loop system[37] for the chilled water distribution. In the primary-secondary decoupler system, a fairly constant chilled water flow is maintained in the primary circuit to prevent water from freezing in the evaporator. This favors partial load operation and sequence control of chiller plant as practiced by most a/c systems in Hong Kong. The reverse-return open loop system is not as common as the direct return system. It is because it requires the piping circuits to be almost equal in length that is difficult to achieve for the building nature in Hong Kong. There is seldom boiler to provide heating for buildings in Hong Kong because the winter season in Hong Kong is mild enough not to warrant heating. Decoupler sequence step control method and energy (BTU) measurements with differential pressure bypass control method[37] are almost equally used for the chilled water distribution control. Table 3.6 and table 3.7 show the distribution of different types of chilled water distribution systems and the control schemes.

Table 3.6: Distribution of different types of chilled water distribution system

Chilled water distribution system	%
Primary and secondary decoupler bypass	46.5
Direct return	34.9
Reverse-return open-loop	18.6

Table 3.7: Distribution of different control schemes for the chilled water distribution system

Control scheme of the chilled water distribution system	%
Decoupler sequence step control with constant water flow in the primary circuit & variable chilled water flow in the secondary circuit	50
BTU measurement with differential pressure bypass control	50

3.4 SURVEYED RESULTS FOR THE AIR-CONDITIONING SYSTEMS (SECONDARY PLANT)

3.4.1 Air-conditioning air equipment and systems

All building under surveyed employed primary air handling unit(s) (PAHU) which treats outside air before admitting to some of the air handling units in the building, From table 3.8, single air duct(36) is found in 79% of the a/c systems. Only 15% of the a/c systems are dual (hot and cold) duct(36). It is because heating is seldom needed, as the winter season is usually mild in Hong Kong. Only 5.9% of the a/c systems use window or split type direct expansion (DX) units. This is due to the fact that over 97% of the surveyed buildings are commercial and complex buildings. Only 2.2% are domestic buildings that use almost window or split type DX units. A majority of the a/c systems are single air duct systems. Hence single air duct a/c systems were focused in the survey.

Table 3.8: Distribution of different types of air distribution systems in buildings

Air distribution system	%
Central single duct (cold duct)	79.4
Central dual duct (cold & hot duct)	14.7
DX split type / individual window unit	5.9

As regard to the a/c air side systems, different combinations of a/c systems are used and are shown in table 3.9 and table 3.10. These include 22% of fan coil unit (FCU) alone; 11% of variable air volume (VAV) alone; 22% of FCU + VAV; 16% of FCU + constant air volume (CAV); and 20% of FCU + CAV + VAV. The FCU system has been employed in 80% of all combinations. CAV and VAV systems have been applied in 42% and 58% of all combinations respectively. Most buildings use a combination of air systems because there are usually different functions within the same building. Typically, commercial and composite buildings in Hong Kong may consist of shops, shopping arcade, coffee shops, restaurants, office, etc. Hence, FCU may be applied for shops, VAV for offices and CAV for restaurants, etc.

Table 3.9: Distribution of different combinations of a/c air side systems

Combination of a/c air side systems employed	%
FCU*	22.2
CAV*	4.5
VAV*	11.1
FCU + CAV	15.6
FCU + CAV + VAV	4.5
FCU + CAV + VAV with VAV boxes that installed with electric reheater	2.2
FCU + VAV	4.5
FCU + VAV with VAV boxes that installed with electric reheater	2.2
FCU + VAV with VAV boxes that installed without electric reheater	2.2
FCU + CAV + single zone VAV + VAV with VAV boxes that installed with electric reheater	4.5
FCU + CAV + single zone VAV + VAV with VAV boxes that installed without electric reheater	6.7
FCU + CAV + single zone VAV + VAV with VAV boxes that installed with and without electric reheater	2.2
CAV + VAV	2.2
Single zone VAV + VAV with VAV boxes that installed without electric reheater	2.2
FCU + single zone VAV + VAV with VAV boxes that installed with electric reheater	8.9
FCU + single zone VAV + VAV with VAV boxes that installed without electric reheater	4.5

* FCU - fan coil unit

CAV - constant air volume a/c system (single and/multizone)

VAV - variable air volume a/c system (single zone)

Table 3.10: Frequency of application of different types of a/c system

Type of air side a/c system	Frequency of application as % of all a/c combinations
FCU	80.1
CAV	42.2
VAV (single zone)	51.0
VAV with VAV boxes that installed with electric reheater	20.0
VAV with VAV boxes that installed without electric reheater	17.7

3.4.2 Details of a/c air system designs

From the above observation, the FCU system is the most important a/c system installation found in Hong Kong. With the detail information provided for PAHU, FCU, CAV and VAV air systems, the following typical system design of PAHU, FCU, CAV and VAV can therefore be identified.

Primary air handling unit

	Typical	Range
Tube/fin material:	Copper/Copper	
Number of rows:	6	6 and 8
Sensible load:	60 kW	15 kW to 390 kW
Total load:	156 kW	37 kW to 1260 kW
Entering air temperature:	33°C db, 28°C wb	28°C db, 18°C wb to 33.9°C db, 28°C wb
Chilled water flow:	6.56 kg/s	1.73 kg/s to 51.83 kg/s
Entering water temperature:	8°C	6°C to 8.2°C
Leaving water temperature:	13.7°C	12.7°C to 14.3°C
Water pressure drop:	21.6 kPa	8.1 kPa to 40 kPa
Working pressure:	depends on static water head	
Air coil face velocity:	2.5 m/s	2.5 m/s
Supply air flow:	2.42 m^3/s	0.8 m^3/s to 16.2 m^3/s
Motor rating:	4.1 kW	2.2 kW to 26.5 kW

Fan coil unit (FCU)

	Typical	Range
Tube/fin material:	Copper/Aluminum	
Number of rows:	3	2 to 6
Sensible load:	4.8 kW	1.14 kW to 16.7 kW
Total load:	6.3 kW	1.28 kW to 19.4 kW
Entering air temperature:	24°C db, 18°C wb	23°C db, 18°C wb to 30°C db, 23°C wb
Chilled water flow:	0.3 kg/s	0.05 kg/s to 0.83 kg/s
Entering water temperature:	7°C	5°C to 9.5°C
Leaving water temperature:	12°C	11°C to15.8°C
Water pressure drop:	28 kPa	4.05 kPa to 40 kPa
Working pressure:	depends on static water head	
Air coil face velocity:	2.5 m/s	2.5 m/s
Supply air flow:	0.38 m^3/s	0.148 m^3/s to 1.4 m^3/s
Motor rating:	180 W	80 W to 880 W

Constant air volume air handling unit (CAV)

	Typical	Typical wih direct outside air supply	Range
Tube/fin material:	Copper/ Aluminum	Copper/ Alumimum	Copper/copper, Copper/Aluminum
Sensible load:	90 kW	24 kW	8.1 kW to 133.8 kW
Total load :	110 kW	40 kW	12.4 kW to 203.6 kW
On coil temperature:	24 °C db, 18 °C wb	25.3°C, 0.19 kg/kg abs. humidity	22.2°C db, 17.2°C wb 34°C db, 28°C wb
Off coil temperature:	10.7 °C db, 10.6 °C wb	21.2°C, 0.0091 kg/kg abs. humidity	10.7°C db, 10.6°C wb to 17°C db, 15.6°C wb
Entering water temperature:	7 °C	7 °C	6°C to 9°C
Leaving water temperature:	12.5 °C	16.32 °C	12°C to 16.32°C
Chilled water flow:	4.0 kg/s	1.088 kg/s	0.44 kg/s to 8.9 kg/s
Water pressure drop:	25 kPa	5.4 kPa	5.2 kPa to 35.77 kPa
Air pressure drop:	180 kPa	35	22.5 kPa to 300.9 kPa
Working pressure:	depends on the static water head		
Air coil face velocity:	2.5 m/s	1.88 m/s	1.67 m/s to 2.53 m/s
Supply air flow:	5.5 m^3/s	2 m^3/s	0.33 m^3/s to 11.5 m^3/s
Motor rating:	11 kW	1.33 k w	0.375 kW to 18.5 kW

Variable air value air handling unit (VAV)

	Typical	Range
Tube/fin material:	Copper/Aluminum	Copper/copper, Copper/Aluminum
Sensible load:	170 kW	46.1 kW to 185.4 kW
Total load:	190 kW	57.2 kW to 272.7 kW
On coil temperature:	25 °C db, 17 °C wb	23.3°C db, 17.8°C wb to 26.3°C db to 18 °C wb
Off coil temperature:	11.9 °C db, 11.5 °C wb	10.7°C db, 10.6°C to 17°C db, 15.6°C wb
Entering water temperature:	6.5 °C	5.2°C to 8.2°C
Leaving water temperature:	13.5 °C	12.2°C to 14.3°C
Chilled water flow:	6.5 kg/s	2.6 kg/s to 9.4 kg/s
Water pressure drop:	20 kPa	9.8 kPa to29 kPa
Air pressure drop:	135 kPa	50 kPa to 168 kPa
Working pressure:	depends on the static water head	
Air coil face velocity:	2.5 m/s	2.32 m/s to 2.6 m/s
Supply air flow:	10.5 m^3/s	1.9 m^3/s to 10.5 m^3/s
Motor rating:	22 kW	4 kW to 22 kW

From the above typical air systems, the latent heat load for both CAV and VAV systems have small ratios of 11% and 18% respectively in the total heat load (except CAV with direct outside air supply). These figures agree with the on and off coil dry bulb and wet bulb temperatures. In fact, before entering the CAV and VAV air handling units, the outside air has been pre-treated in the primary air handling unit (PAHU) where most of the outside humidity has been removed. A typical PAHU has on coil temperatures of 33°C db, 28°C wb and off coil temperatures of 13.3°C db, 13.2°C wb (i.e., latent load to total load ratio=72%). The typical air velocity for PAHU, FCU, CAV and VAV is 2.5 m/s and this value is in line with the ASHRAE standard.

3.4.3 Control strategies

There are various schemes[38] for a/c control in buildings. The summary of schemes used on the control strategies for different a/c air side systems in Hong Kong are shown in table 3.11 to table 3.17.

Table 3.11: Control scheme for PAHU

Scheme	% of application
The supply air temperature is controlled by supply air temperature sensor and controller by changing chilled water flow through cooling coil in response to supply air temperature.	100%

Table 3.12: Control scheme for FCU

Scheme	% of application
The room air temperature is controlled by thermostat by changing chilled water flow through cooling coil in response to room air temperature.	100%

Table 3.13: Control schemes for central AHU (air handling unit) of VAV multizone

Scheme	% of application
1. AHU supply air temperature controlled by changing chilled water flow through cooling coil in response to supply air temperature.	100%
2. AHU supply air volume controlled by	
(i) frequency inverter	77.3%
(ii) inlet guide vane	22.7%
in response to	
(iii) air duct static pressure	95.5%
(iv) air flow measurement	4.6%

Table 3.14: Control schemes for zone VAV box of VAV multizone

Scheme	% of application
Individual zone temperature controlled by changing supply air volume in response to room air temperature only.	100%
AND	
(i) Box without reheater	52.6%
(ii) Box with reheater	47.4%

Table 3.15: Control schemes for VAV single zone

Scheme	% of application
1. Supply air temperature controlled by changing chilled water flow through cooling coil in response to supply air temperature.	100%
2. Supply air volume controlled by : (i) frequency inverter (ii) inlet guide vane in response to room return air temperature.	 60% 40%

Table 3.16: Control schemes for CAV multizone

Scheme	% of application
AHU supply air temperature controlled by changing chilled water flow through the cooling coil in response to return air temperature. AND (i) AHU without heater (ii) AHU with heater	100% 80% 20%

Table 3.17: Control schemes for CAV single zone

Scheme	% of application
AHU supply air temperature controlled by changing chilled water flow through the cooling coil	100%
in response to:	
(i) return air temperature	73.3%
(ii) supply air temperature	26.7%

The supply air temperature in a PAHU is controlled by adjusting the chilled water flow through the cooling coil in response to the supply air temperature change reference to the supply air temperature set point.

For FCU installations, the room air temperature is used as the control variable to control the chilled water flow through the cooling coil to achieve the setpoint room air temperature.

For the VAV multizone system, 100% of the central VAV AHUs have their supply air temperature controlled by varying the chilled water through the cooling coil in response to the supply air temperature. The supply air volume is being controlled almost in response to the change of air duct static pressure by use of either frequency inverter (77.3%) or inlet guide vane (22.7%). All the zone VAV boxes have their individual zone temperature being controlled by changing the supply volume in response to individual zone temperatures. More than a half (52.6%) of the VAV boxes has electric reheater.

For the VAV single zone system, again all systems have their supply air temperature being controlled by changing the chilled water flow through the cooling coil in response to the supply air temperature. All the systems use room return air temperature as the controlled variable to control the supply air volume by using frequency inverter (60%) or inlet guide vane (40%).

For the CAV multizone and single zone systems, all multizone systems have their supply air temperature being controlled by changing the chilled water flow through the cooling coil in response to the return air temperature. In the single zone CAV systems, 73.33 % and 26.66 % of the systems use the return air temperature and the supply air temperature as the controlled variable respectively. 80% of the multizone system have no electric air heaters.

In the above survey, all FCU and nearly all CAV systems use room air temperature or room return air temperature to control the cooling coil. This is expected as the room air temperature or room return air temperature measures directly the change of room cooling load. Hence, using the room air or return air temperature to control the cooling coil is a simple and direct approach. Return air temperature is used for cases of large room area or air being supplied to different rooms from the same CAV system. The use of supply air temperature to control the cooling coil should only be considered if the room has a fairly constant cooling load. In the VAV, basically, there are two control loops, i.e., the supply air temperature control loop and the supply airflow control loop. In the supply air temperature control loop, the supply air temperature is maintained by regulating the chilled water flow through the cooling coil in response to the supply air temperature. In the airflow control, the supply airflow is regulated to compensate cooling load change in the room. For single zone VAV, similar to the CAV, the room or room return air temperature is used to control the supply air flow. However, for VAV system supplying air to a number of VAV boxes, the system air flow is controlled by inlet guide vane or frequency inverter in response to the air static pressure in the downstream of the main supply air duct. The air static pressure has been used as the controlled variable instead of the air flow. It is because the actual supply air flow needs air velocity measurement at each VAV box and the instrumentation cost will be very expensive. Hence, direct air flow measurement is seldom used. Similar to FCU, individual VAV box damper control is achieved in response to the room air temperature. The survey supports the above argument.

For the controller action[39] being used, on/off control is popularly applied in FCU (63.7%), VAV box terminal reheater (55.6%) and CAV multizone AHU terminal reheater (50%). About 36.4% to 40% of the supply air temperature control of various systems use proportional (P) control. Proportional control is also applied to the supply air volume control of VAV systems (27% to 40%). Proportional + Integral (P+I) control is applied in 20 % to 40% of the supply air temperature control. The P+I control is also used in 20% to 36% of the supply air volume control and 22.2% to 50% of the terminal reheater control of CAV and VAV systems. Proportional + Integral + Derivative (P+I+D) has been used in 20% to 40% of various controls in CAV and VAV systems. However, it has not been used for control of the terminal reheater of CAV and VAV systems. Proportional + Derivative (P+D) has not been used for the a/c systems. The controller action used in PAHU is mainly proportional + integral (P+I) control (68%) and proportional (P) control (27%).

The distribution of controller actions in air side systems is shown in table 3.18.

Table 3.18: Distribution of different controller actions used in a/c control

Air side a/c system	on/off %	P* %	P+I* %	P+I+D* %	P+D %
1. FCU	63.7	18.2	12.1	6.1	0
2. VAV Multizone					
a) AHU					
i) AHU supply air temperature	0	36.4	31.9	31.9	0
ii) AHU supply air volume	0	27.3	36.4	36.4	0
b) Zone VAV box					
i) Zone temperature	0	31.6	31.6	36.9	0
ii) Terminal reheater	55.6	11.1	22.2	11.1	0
3. VAV single zone					
a) AHU supply air temperature	0	40	20	40	0
b) AHU air supply volume	0	40	20	40	0
4. CAV multizone					
a) AHU supply air temperature	10	40	30	20	0
b) AHU terminal reheater	50	0	50	0	0
5. CAV single zone					
a) AHU supply air temperature	0	40	40	20	0

* P - proportional control, I - integrate control, D - derivative control

On/off control has been applied greatly in the FCU and electric heater control. On/off controller is simple and involves only simple electro/mechanical devices. It is also considered adequate and cost saving for FCU and electric heater control. Besides, fan coil system is always under occupant's control. Complicated control is usually beyond the knowledge of the occupants. However, the problem of water hammer in the on/off control of FCU has always been an under estimated issue to be tackled. Proportional control alone is also popularly (approximately one-third) applied in all the air system control specially the FCU. Besides on/off control, the conventional electro/mechanical proportional controller costs less than controllers of other combinations. Nevertheless, proportional control has the disadvantages of causing offset and increasing hunting to the design set temperature. Hence, integral control has largely been added to the proportional control and accounts 20% to 50% of different air side systems. Derivative control has been applied to P+I control to give damping to the control and accounts 20% to 40 % of the air side systems. However, derivative control has not been combined to use with proportional control. It is because thermal control system does not usually involve a quick change of the air temperature and proportional control alone is not oscillatory. The addition of derivative control to proportional control has no practicable benefit.

To follow up the control, the engineering teams of 10 of the responded commercial or educational buildings have been interviewed. Eight of the above buildings have been installed with computerized building management system and direct digital control (DDC) is applied for air temperature control. From the interview, over 90% of the building engineering teams have given no particular attention to the controller settings and the sampling rate of the control systems. Most the controller settings and sampling rates are initially set by the control suppliers without further tuned after systems have been in operation. Further interview with three control suppliers indicated that system characteristics are often ignored in the controller and sampling time settings. In sizing the control valves, the value 50% valve authority is often used as the rule of thumb. No particular attention has been made to the impact of valve authority, characteristics and drive (electric or pneumatic) on the performance of a/c control systems.

3.5 SURVEY RESULTS ON A/C SYSTEM PERFORMANCE ISSUES

This part of survey tries to find out from building users the parameters that they consider in judging the performance of a/c systems. Basically, the survey is divided into three parts. The first part deals with the occupants. It surveys the users' complaints that come from system reliability and room thermal comfort that consists of room air temperature, humidity, diffusion and the air quality. The complaint also consists of noise arising from the fan section and air movement in the duct section. The second part of the survey aims to find out the a/c system performance from engineering points of view. In this part, survey on the system operation and design conditions such as chilled water supply temperature and balancing, air flow distribution in room and room air supply conditions of temperature, humidity and quality are done. The last part of the survey aims to find out the management and operation measures that can be used to evaluate the performance of a/c system.

Summary of the survey is given in table 3.19 and table 3.20.

Table 3.19: Distribution of different parameters on the concerns of the a/c system performance

Parameters	%
Room temperature	17.64
Room temperature and humidity	11.76
Air flow and distribution	11.76
Chilled water supply temperature	5.89
Water balance	5.89
Complaints from the users	17.64
Noise problem	5.89
Indoor air quality	11.76
Comfort	5.89
To the design considerations	5.89

Table 3.20: Distribution of criteria for evaluating the performance of a/c system

Criteria	%
Energy consumption	44.8
Efficiency of plant	7.9
Reliability of plant and systems	18.4
Easiness of operation	13.2
Maintenance	2.6
To the design specifications	13.2

From the survey, about 47 % of the responses indicate that room temperature is their major concern on the performance of the a/c system. The room air temperature is also the main cause of complaints from the occupants. A small number of responses consider humidity their major concerns.

The room air flow and distribution (12%), and indoor air quality (12%) are other concerns. A small amount of the responses also points out their concern on chilled water supply temperature (6%), noise (6%), design considerations (6%), comfort (6%) and water balance (6%).

In evaluating the performance, 45% of the responses use energy consumption as their major criterion. Other criteria are the system reliability (19%), the maintenance consideration of the a/c plant and system (2.6%) and efficiency of the a/c plant (8%). The easiness of plant operation (13%) and the fulfillment of design specifications (13%) are also important criteria being used.

From the above survey, it is obvious that energy consumption and room temperature that relates to the room thermal comfort are important elements in evaluating the performance of an a/c system. The performance indicator for a/c system should be a function of these two factors. It is believed that more energy is required to maintain comfort. However, most of the time energy is wasted without achieving the required comfort conditions. In practice, these elements are closely affected by how the system is designed and operated. An optimized system design and operation will result in a cost effective and energy efficient system in maintaining good thermal comfort in a building.

3.6 CONCLUSION OF THE SURVEY

From the above surveying study, the following can be concluded on the buildings and air conditioning systems:

Most of the buildings under surveyed are high rise commercial buildings. Of these surveyed buildings, about 35 % are curtain walled, 31% in the height range of 30 to 50 storeys and 45% have basement(s). About 80% of these buildings have floor areas in the range of 0 to 100000 m^2. More than half of these buildings is found facing either South or North.

Because of the low response rate on detail information of the building construction, it is difficult to conclude the typical building construction details.

The air conditioning systems in most of the buildings are single duct (cold) air and water systems. More than half of the buildings have cooling load in the range of 0 to 1135 kW. Three, four and five chiller plants are commonly in use. Direct air cool is the commonest method used for the heat rejection in a/c system installations. A possible reason is because seawater is not always readily available and the local authorities for health and water source reasons forbid the use of cooling tower. However, because of the excess fresh water supply from China, the relaxation in the use of cooling tower is under debate recently.

Primary-secondary decoupler bypass system and direct return system are the two most common systems for the chilled water distribution. Mostly either decoupler sequence step control or BTU measurement controls the distribution of chilled water with differential pressure bypass control.

The fan coil unit (FCU), variable air volume (VAV) and constant air volume (CAV) are the major a/c air-side systems used in the a/c installations. FCU and VAV, used alone or used with other a/c systems, are most commonly in use and represent 80% and 58% of frequencies of use respectively in buildings.

Typical examples of FCU, CAV AHU and VAV AHU systems can be defined from data provided in the feedback.

For the control strategies and controller action, the followings can be concluded.

- FCU

  Room air temperature is used in all applications for regulating the chilled water flow through the cooling coil for room air temperature control. On/off control is applied.

- CAV

  Return air temperature is commonly used in single and multizone CAV for regulating the chilled water flow through the cooling coil for the supply air temperature control. Most of the applications do not have reheaters. The controller actions of P, P+I and P+I+D are all used in the control. However, P control is the most common controller action used.

- VAV

  There are two control loops in VAV AHU (air handling unit). In the first loop, supply air temperature is controlled in all applications by regulating the chilled water flow through the cooling coil in response to the supply air temperature. The controller actions, P and P+I+D are adopted in most applications. In the second control loop, the supply air volume is controlled largely by frequency inverter for the multizone VAV in response to the air duct static pressure. P, P+I, P+I+D are about equally used for the control. For the single zone VAV, the supply air volume is controlled in all applications by frequency inverter (60%) and inlet guide vane (40%) in response to the room return air temperature. P and P+I+D are used equally in 80% of all applications. For the VAV box control, individual zone air temperature is controlled by changing the supply air volume in response to the room air temperature in all applications. Over 50% of the VAV boxes have reheaters. P, P+I and P+I+D controller actions are well applied to the supply air volume control. Over 50% of the terminal reheaters are controlled by on/off control.

Room air temperature and energy consumption are the major concerns for the performance of the a/c system. The thermal comfort control and energy consumption are strongly related to the system design and control strategies. An oversized or undersized system will produce unnecessary energy waste in trying to maintain the system design intents and sometimes makes the system hunting in case of oversized and sluggish in response when undersized. Even a system is properly designed, if the control strategies are not well considered, it will also cause bad comfort control and energy waste. Such control strategies include the controller settings, sampling time of the control system, control valve sizing and valve drive. These factors are vital in affecting the performance of a/c systems and hence should be investigated further by simulation and experimental or case studies.

Results and data collected on the a/c systems and control strategies from this a/c survey is used in the following chapters 5 and 7 for studying the optimzation of a/c system designs by simulation for Hong Kong application

CHAPTER 4 SYSTEM SIMULATION PROGRAM, HVACSIM+ AS EXAMPLE

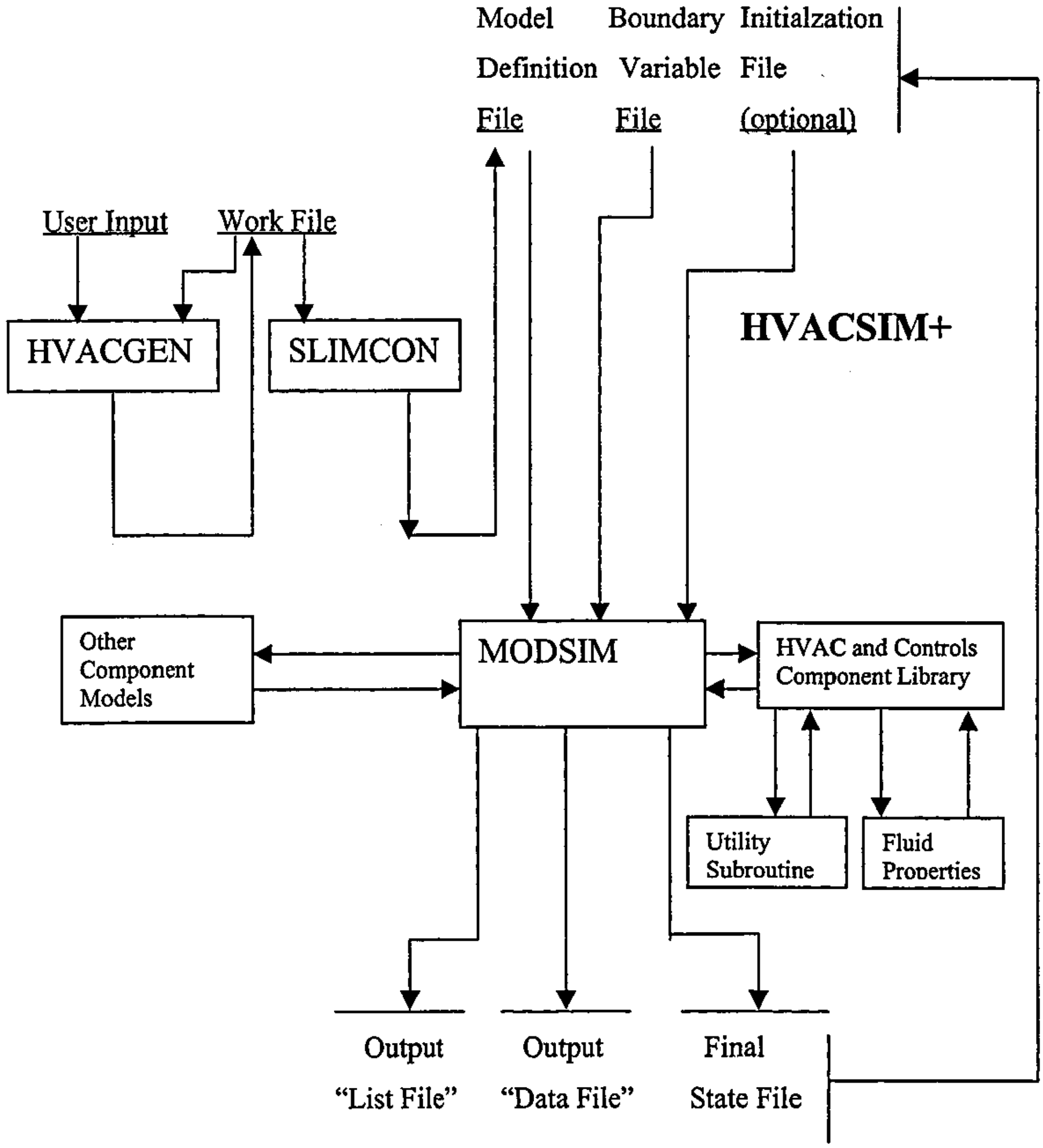
4.1 THE SIMULATION PROGRAM

The simulation program used for study is the HVACSIM+ developed at the National Bureau of Standards (NBS), U.S.A. It has been used for detailed simulation of entire building systems: the HVAC systems, the equipment control system, the building shell, the physical plant, and the dynamic interactions among these subsystems[14]. The program is widely used for HVAC control simulation. Although it is no longer supported, it is still being applied to simulation problems. Some of its application[40],[41],[42],[43],[44] are reported recently in the Sixth International IBPSA Conference, September 13-15, 1999 Kyoto, Japan.

The predicated component performance of most of the major component models, such as control valve, conduit, duct, pipe, cooling coil etc., is compared and validated against experimental results.

The HVACSIM+ package consists of a main program called MODSIM, which contains a set of mathematical models of building energy system component, an interactive 'front end' program called HVACGEN which is used to create a description of the system to be simulated, and a program called SLIMCON which processes the output of HVACGEN and converts it into the form required by MODSIM. The architecture of HVACSIM+ package is shown in figure 4.1.

Figure 4.1: Complete architecture of the HVACSIM+



4.2 OVERVIEW OF MODSIM AND SIMULATION BY HVACSIM+

MODSIM uses the concept of modular simulation. An entire simulation of a HVAC system is constructed from basic **UNITS** that represent different **TYPES** of HVAC components. Component models, even of the same type, distinct from each other by their unit number. The component model can be regarded as black box described by the component parameters and with a set of defined input data and output data. MODSIM uses a simultaneous nonlinear equation-solving package called SNSQ, which is based on modified Powell hybrid method. MODSIM partitions large sets of equations into smaller subsets. This results two levels between individual units and the entire simulation, i.e., **BLOCKS** and **SUPERBLOCKS.**

A block consists of a set of units, which maybe functional grouped together to represent a subsystem such as a control loop. A block can be viewed as a black box that takes a set of block inputs and produces a set of block outputs. The outputs are related to the inputs. An output of a unit can be one of the inputs to other units within the same block or in another block. Sometimes output is also one of the inputs to the same unit.

A superblock consists of a set of blocks. Blocks within a superblock are related to each other by a set of simultaneous equations to be solved by SNSQ. A simulation of a building system contains a number of superblocks, which are weakly coupled and are regarded as independent subsystems. Unlike blocks within a superblock, there is no simultaneous equations defined between superblocks.

With reference to figure 4.1, a simulation starts with the interactive program HVACGEN. HVACGEN requires user to define the type of each component in the simulation, to provide index numbers for each state variable, and to enter values for all parameters. The program then assigns unit number to each unit in the entire simulation so as to distinguish the system components. Initial values of all state variables, indices of time dependent boundary variables, reporting interval for each superblock and the variables to be reported have to been input to the HVACGEN. The output of HVACGEN is a "simulation work file".

To activate the simulation, the program SLIMCON must be run. SLIMCON reads the simulation work file produced by HVACGEN and converts it to a "model definition file" which can be used by the MODSIM. In the context of MODSIM, a boundary condition or variable is defined as a state variable which is external to the system being simulated, in the sense that it is not an output of any component of the system. Boundary variables may be constants or time dependent. Any number of boundary variables may be specified as time dependent, and their values are stored in a data file. MODSIM starts running the simulation by reading the data from the "boundary variable file" and fixes them any time to change the appropriate state variables. The boundary variable file contains state variables that are external to the system being simulated. The boundary variables maybe time dependent or time independent. Typical examples are the outside air temperature and relative humidity. MODSIM uses third order Lagrangian interpolation to find boundary variables at intermediate times. It begins simulation at time zero. Thus a long simulation has to be run in several pieces of simulation, a separate boundary variable file beginning at time zero will usually required for each piece. An optional initialization file is included in HVACSIM+ package that allows the final state of one simulation to be used as the initial states of another. Outputs of the simulation are reported as two kinds of report, i.e. a "data file" reports raw data at each time taken during the simulation and a "list file" that summarizes the simulation configuration, any diagnostics generated during the computation, and an interpolated listing of the reported variables at equal time intervals.

HVACSIM+ is a dynamic simulation package. The simulation of a system is carried out on a set of differenrtial equations describing different components of the system. Inaccuracy and instability may occur if too large a time step is used during the simulation[45]. This is particularly a problem for simulating a/c system, which has a wide range of componenet time constants. The set of differential equations, which describes the componenets of the a/c system, has widely varying time constants and is referred as a stiff set of equations. Because of the stiffness of ordinary differential equations encountered due to different time constants of components, the order of integration and the size of the time step are varied during calculation to minimize the computation required for a specified degree of accuracy. The algorithm used by HVACSIM+ is a modification of the algorithm described by Brayton[46]. The algorithm makes use of backward differentiation formulae. To save computing time, state variables are continuously monitored so that those state variables, which are steady, will be "frozen", i.e. such these steady state variables are removed from the set of simulatenous equations. The frozen state variables are being monitored so that they can be returned to the calculation, or unfrozen, if necessary.

4.3 LIMITATIONS OF MODSIM

There are various limitations for MODSIM. First is the convergence of the equation solver. This is a common problem with all equation solvers. The initial conditions and error tolerances used in the simulation affect convergence. HVACSIM+ provides the initialization option to minimize the problem. Secondly, there are limitations of block structure in the HVACSIM+. Convergence is particularly a problem at the beginning of a simulation, when the equation solver must work from the set of user supported initial conditions chosen somewhat arbitrarily to start the calculation. The equation solver may fail to converge if the initial conditions are too far from the final solution. Care has to taken to define the block structure to achieve good convergence properties. Thirdly, care has to be given to the coupling between superblocks. Superblocks should be independent and hence have to be weakly coupled. It is vital to ensure this when dividing the entire system into superblocks so that no simultaneous equations are defined between superblocks. HVACSIM+ has an input scanning option that scans all superblock inputs after each time step. If the inputs to a superblock are greater than a specified tolerance, the superblock will be called and its state will be calculated according to the new values of inputs. Fourthly, as mentioned previously, it is important to have time step control. The value of time step affects accuracy, stability as well as computational time of a simulation. MODSIM employs variable time step. The maximum and minimum time steps are defined by the user. A time step size selection is available in HVACSIM+. MODSIM also has limitation on the size of simulations. Such limitations include 15 inputs and 15 outputs per unit; 10 differential equations per unit; 20 units per block; 50 inputs and 50 outputs per block; 30 simultaneous equations between units in a block; 10 blocks per superblock; 20 differential equations per superblock; 20 simultaneous equations between blocks in a superblock; 30 reported variable per superblock; 200 units per simulation; 50 blocks per simulation; 10 superblocks per simulation; 50 differential equations per simulation; 600 state variables per simulation; 1000 parameters per simulation; 1000 saved variables per simulation and 30 time-dependent boundary variables per simulation.

4.4 SELECTED APPLICATIONS OF HVACSIM+

HVACSIM+ has been applied for various simulation studies. The following case studies are some of the selected applications:

Case 1: Simulation of a large office building system by Park et al.[47]

A typical floor of a large office building was selected and divided into four zones. Simulations using HVACSIM+ were performed using the building shell and zone models along with a/c equipment and control system models. The dynamic interactions and behaviours between the building zones, a/c system and control system for three control strategies were studied by simulation. The results of the simulation were compared and verified by experimental and site measurement tests. A series of simulations were performed in order to compare the simulation results with the experimental measurements. Because of the difficulties encountered in estimating some of the input parameters, repeated simulations with different input values were tested before the calculations agreed with the experimental results.

The simulation study concluded that HVACSIM+ was capable for simulating a large office building for studying the dynamic interaction between the building shell, office zones a/c systems and control systems. The necessary estimation of input constants and assignment of initial conditions for component models affected significantly the convergence of solutions. A large step change in any state variable would cause inaccurate solutions. Park et al[47] found that the preparation of input file was a very difficult task. However, HVACSIM+ was found a capable and effective tool for studying and comparing on energy consumption for different control strategies applied in the a/c control scheme.

Case 2: Application of HVACSIM+ for optimization study of a/c control system[5] by Dexter and Trewhella

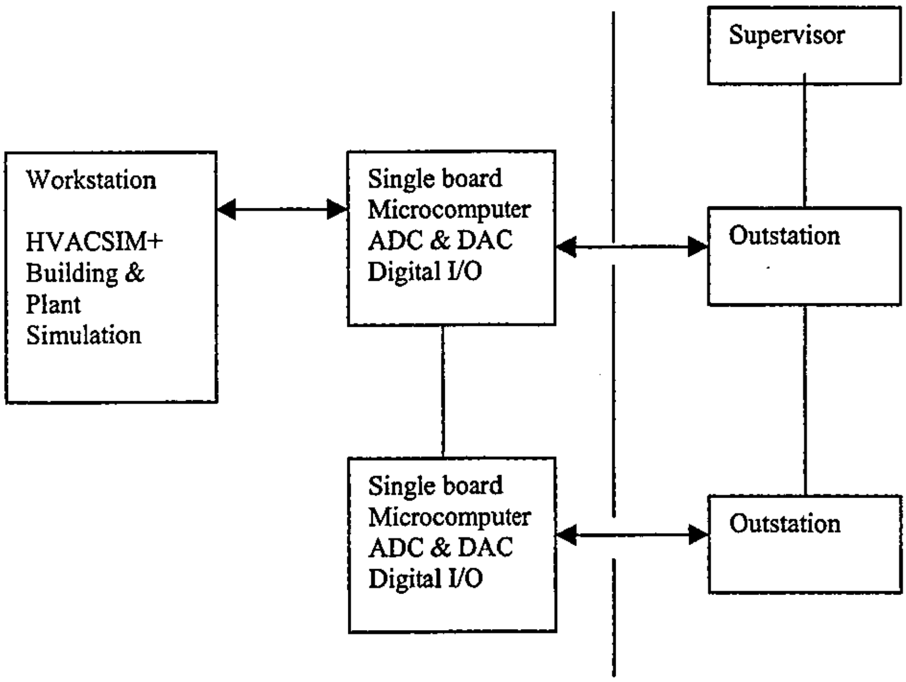
Dexter and Trewhella studied the application of fuzzy rule-based approach to performance assessment of the water temperature controller of a fan coil unit that was supplied with hot water from a central boiler. Standard HVACSIM+ a/c component models, building shell model and zone model were used to simulate and study the dynamic behaviour of the building control system. With the aid of HVACSIM+, the study concluded that the fuzzy rule-based approach gave a plausible assessment of the performance of a building control system.

Case 3: Application of HVACSIM+ in a building emulator for studying the performance of control systems[48] by Dexter and Haves

Dexter and Haves used a building emulator to evaluate the performance of a building VAV control system. HVACSIM+ was used to simulate a building and plant. The building had a VAV a/c system with a single AHU and three zones. Fuzzy rule-based approach was applied to study the performance of the a/c system under two cases, i.e. the effect of changing the strategy used to determine the zone temperature set points and the effect of changing the parameters of the control system. Test data were generated by the emulator consisting of a real-time simulation of the building shell and a/c plant, together with a hardware interface that connected the simulation to commercial control equipment. The application detail of HVACSIM+ in the emulator is shown in figure 4.2.

Figure 4.2: Configuration of the building emulator for control evaluation

Emulator **BEMS (Building Energy Management System)**



CHAPTER 5 THE PRELIMINARY SIMULATION STUDY OF A SIMPLE A/C SUB-SYSTEM EXAMPLE FOR CONTROL PERFORMANCE

This preliminary simulation study aims to test the application of the simulation package HVACSIM+ for the optimization of a/c system design. It tries to identify a suitable objective function that can be used in assessing the control performance of a/c system design. As this is a preliminary simulation study before a detailed simulation study is conducted, the a/c system selected is a simple a/c cooling sub-system, primary air handling unit, to facilitate the study. The sub-system is simple because it deals only with the air handling components. There is no interaction with any air zone. The sub-system is a primary air handling unit that will treat the outside air before admitting to different air handling units to provide conditioned air to various space zones. In fact, a primary air handling unit is an essential sub-system for a/c installations of commercial buildings in Hong Kong, the design of which in relationship to the control performance is important. From the previous survey, the control is a simple loop that controls the cooling coil outlet air temperature. The results and experience of the preliminary simulation is to be used for a following detailed simulation that interacts an air handling unit with an air-conditioned control zone for optimization of a/c system design for optimum control performance.

5.1 SIZING OF 'REALISTIC' A/C COOLING COIL FOR SIMULATION

In this preliminary simulation study of the a/c sub-system, the coil size arbitrarily chosen is related to a size derived from a conventional sizing method, the LMTD (Log mean temperature difference) method, such that the external surface area of the cooling coil to meet a 'peak' load on the cooling coil for a specified temperature difference across the cooling coil can be determined. From this designed cooling coil, larger and smaller cooling coils can be chosen for study by varying the fin surface area.

5.2 THE OBJECTIVE FUNCTION FOR THE PRELIMINARY SIMULATION STUDY

In this preliminary simulation study of the simple a/c sub-system, conventional error based objective functions including the mean square error are computed and compared for control performance. The comparison aims to identify the 'best' objective function in assessing the control performance of the a/c sub-system.

5.3 THE EFFECT OF THE CONTROLLER SETTINGS

For any control system, there exists a set of optimum controller settings, which gives the required control as intended. The values of the optimum controller settings will depend on the static and dynamic characteristics of the control system. Therefore, in this preliminary simulation study, the controller of the a/c control sub-system for each coil size used is first tuned for controller settings according to the Nichols & Ziegler open loop method[15] before the response test for each cooling coil is conducted. The Nichols and Ziegler method was used because of its simplicity.

Also in the study, only proportional plus integral control is used. This is because the derivative control action, which accounts for rate of change of controlled variable error, has little control effect for the control of an a/c system as the thermal response of an a/c system is rather slow and stable. In fact, P+I is a common control practice for HVAC[49], [50].

5.4 THE SIMULATION STUDY

5.4.1 The application of survey data from chapter 3 for simulation study of the a/c cooling coil sub-system

In this preliminary simulation study of a primary air handling unit control system, the HVACSIM+ simulation package is used to simulate the a/c sub-system. The sub-system simulation model is configured from model components available in the simulation package HVACSIM+. The typical primary air handling unit with data collected in the a/c survey as mentioned in page 30 is selected for study so as to reflect the application of simulation for practical installations in Hong Kong. Type 12 model from HVACSIM+ is used to model the cooling coil of the PAHU. The physical parameters required by the coil model come from the survey data of the typical PAHU. The survey also shows that about 68% of the control of primary air handling units are P+I control. The supply air temperature is maintained by regulating the chilled water flow through the cooling coil in response to the supply air temperature change from the set point. Hence type 8 P+I controller model is employed for modeling the controller. The controller setting is found by Nichols and Ziegler open loop method, which is a common practice in the control industries in Hong Kong. The controller time constant of the market controller was experimentally determined by the author and equals to 2 seconds. The temperature sensor measuring the supply air temperature is modeled by a simple first order type 7 temperature model. Again the time constant of a market temperature sensor was experimentally determined by the author to be 20 seconds. Most of the control valves found from the survey are pneumatic linear valves and therefore the control valve is modeled by a type 9 pneumatic linear valve model. The actuator time constant is not provided and is estimated. Experimental study on control valve in the following chapter 6 will provide this information. Nearly all the control valve used in Hong Kong found from the survey is normally opened type. Thus, a reverse relay model, type 26, is used to reverse the controller signal. The design conditions for the simulation are those collected from the survey. The initial operation conditions are based on 50 % normal load. With all these design conditions, component physical parameters and input data, the simulation model is configured. The coil size is the variable for study. The simulation makes use of the information collected in the survey and represents a realistic simulation study for application in Hong Kong. A summary of the details of the HVACSIM+ models used in the simulation throughout this research is attached in appendix 2. These design conditions and parameters are described below.

Outside air conditions: 33°C db, 28°C wb

Cooling coil supply air temperature: 19°C

Primary air handling unit:

Tube/fin material:	Copper/Copper
Number of rows:	6
Sensible load:	60 kW
Total load:	156 kW
Entering air temperature:	33°C db, 28°C wb
Chilled water flow:	6.56 kg/s
Entering water temperature:	8°C
Leaving water temperature:	13.7°C
Air coil face velocity:	2.5 m/s
Supply air flow:	2.42 m^3/s

Temperature sensor:

Sensor time constant: 20 s (experimental results for market sensor)

Controler:

Controller time constant:	2 s (experimental results for market controller)
Proportional gain:	determined by Nichols & Ziegler estimation
Integral gain:	determined by Nichols & Ziegler estimation

Pneumatic linear control valve:

Flow resistance:	0.03 kg^{-1} m^{-1}
Actuator time constant:	5 s (estimated)

Initial condidtion (based on 50% load):

Chilled water flow:	3.2782 kg/s
Air flow:	2.75 kg/s
Total cooling load:	78 kW
Total sensible load:	30 kW
Outside air conditions:	28°C db, 0.019 absolute humidity
Supply air conditions:	19°C db, 0.0092 absolute humidty
Control valve opening:	0.5

5.4.2 Definition of the a/c sub-system for the optimization study for weather conditions in Hong Kong from the data collected in the a/c survey in chapter 3

From the data collected in the a/c survey, real conditions and system details can be defined for the preliminary simulation study for Hong Kong application. The simulation model for the simple a/c sub-control system is defined and configured with the help of the data given in section **5.4.1.** The system data and initial conditions are given in the following Table 5.1 to Table 5.2. The typical Hong Kong summer weather conditions to be used are given in appendix 3.

5.4.3 Preliminary simulation study of the simple a/c sub-system for optimum control performance

In this simulation, five practical coil sizes were tested for controllability against different conventional error based objective functions including mean square error. For each coil size, an open loop test was carried out based on the Nichols and Ziegler method to find the P+I controller estimates to be used. The details of the preliminary simulation study are described as follows.

For each cooling coil design, the controller was tuned by an open loop test[15]. With the same initial variable values used, the performance of the resulting 'best' and 'worst' cooling coil designs obtained from the example study were tested by simulation against a typical summer day (11, July) of the Example Weather Year 1989[51] in Hong Kong. The cooling coil temperature control sub-system studied in this example is commonly used in the central air-conditioning systems for commercial buildings in Hong Kong.

In the simulation, eight components are identified for the system. They are represented by unit 1 to unit 8 as indicated in Figure 5.1 and consequently programmed (with the specific types) in the simulation. The configuration of the system components is summarized in Table 5.1. The interface and signal flow between components of the control system is shown in the information flow diagram of Figure 5.2. The primary (tube exterior) surface area, the secondary (fin) surface area and the internal surface area are taken as design variables for the cooling coil design. The initial values of state and boundary variables of the system components are typical of the cooling coil control sub-systems used in Hong Kong and are given Table 5.2. The simulation was tested against the time dependent boundary conditions of ambient dry bulb temperature and inlet air humidity ratio that had initial values of 28°C and 0.019 respectively for five different cooling coil designs CCD1 to CCD5 listed in Table 5.3. The P+I controller setting for each coil size was tuned and determined by the open loop test[15]. The coils chosen are with increasing distinctive sizes. Such distinctive sizes are designed to differentiate the performance of alternative designs.

Figure 5.1: Schematic for cooling coil temperature control system with components and types shown

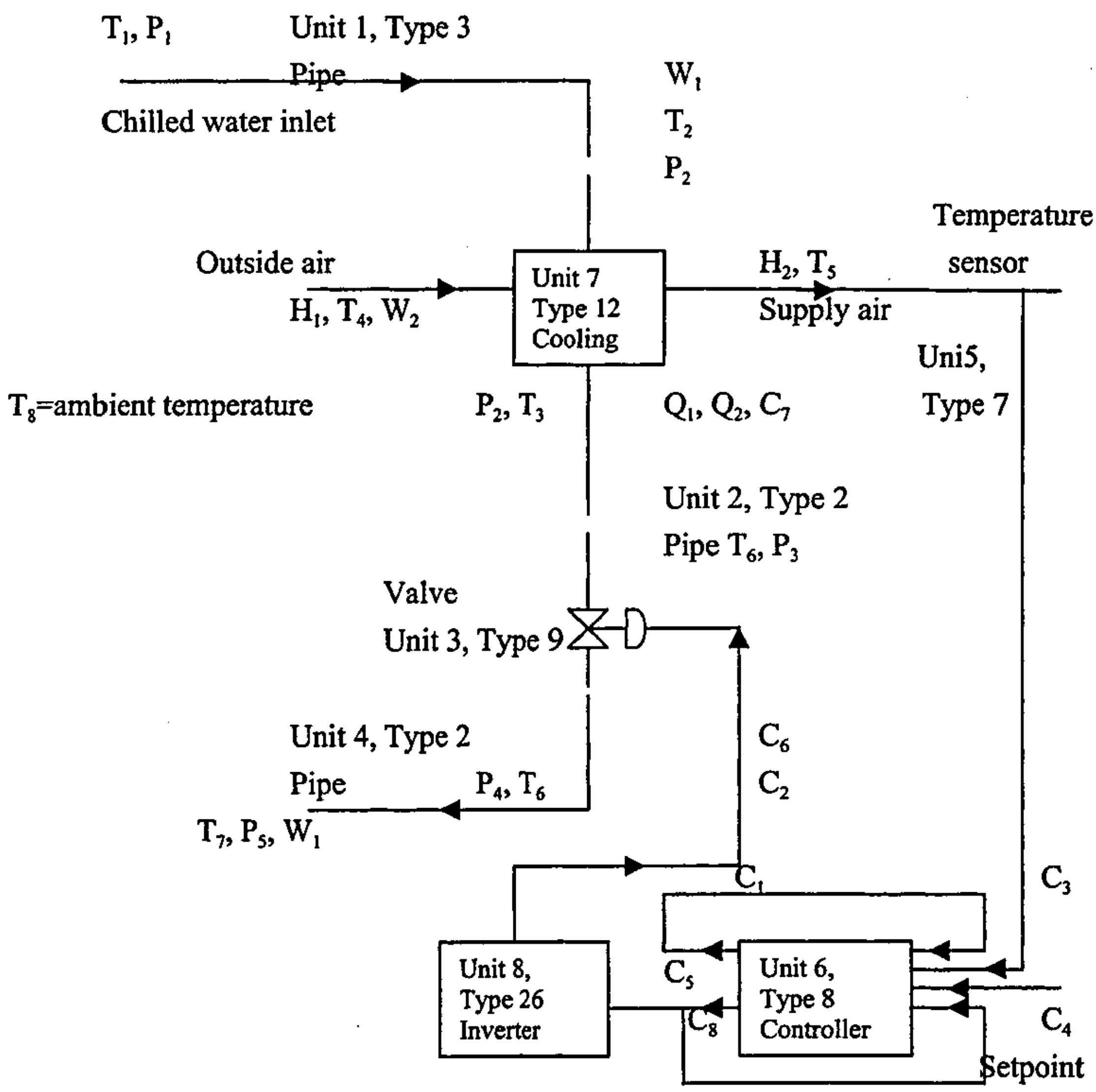


Table 5.1: Simulation configuration of the system

Unit	Type	Input Variable	Output Variable
No.1	Type 3, Inlet Pipe	P1-Inlet water pressure P2-Outlet water pressure T1-Inlet water temperature T8-Ambient air temperature T2-Outlet water temperature	T2-Outlet water temperature W1-Water mass flow rate
No.2	Type 2, Outlet Pipe	W1-Water mass flow rate P3-Outlet pressure T3-Inlet temperature T8-Ambient temperature T6-Outlet temperature	T6-Outlet temperature P2-Inlet pressure
No.3	Type 9, Linear Valve with Pneumatic Actuator	P4-Pressure at outlet W1-Water mass flow rate C1-Input control signal C2-Actuator relative position	C2-Actuator relative position P3-Pressure at inlet C6-Valve stem relative position
No.4	Type 2, Outlet Pipe	W1-Water mass flow rate P5-Outlet pressure T6-Inlet temperature T8-Ambient temperature T7-Outlet temperature	T7-Outlet temperature P4-Inlet pressure
No.5	Type 7, Temperature sensor	T5-Temperature input C3-Temperature sensor signal	C3-Temperature sensor output
No.6	Type 8, P+I Controller	C3-Controlled variable C4-Setpoint temperature C5-Integral of control signal C8-Output control signal	C5-Integral of control signal C8-Output control signa
No.7	Type 12, Cooling Coil	W1-Water mass flow rate T2-Inlet water temperature W2-Dry air mass flow rate T4-Inlet dry bulb air temperature H1-Inlet air humidity ratio T3-Outlet water temperature T5-Outlet air dry bulb temperature H2-Outlet air humidty ratio	T3-Outlet water temperature T5-Outlet dry bulb Temperature H2-Oulet air humidity ratio Q1-Total cooling load Q2-Sensible cooling load C7-Wet fraction of coil Surface area
No.8	Type 26, Inverter	C8-Input control signal	C1-Output control signal

Figure 5.2: Connection of components of cooling coil temperature control system

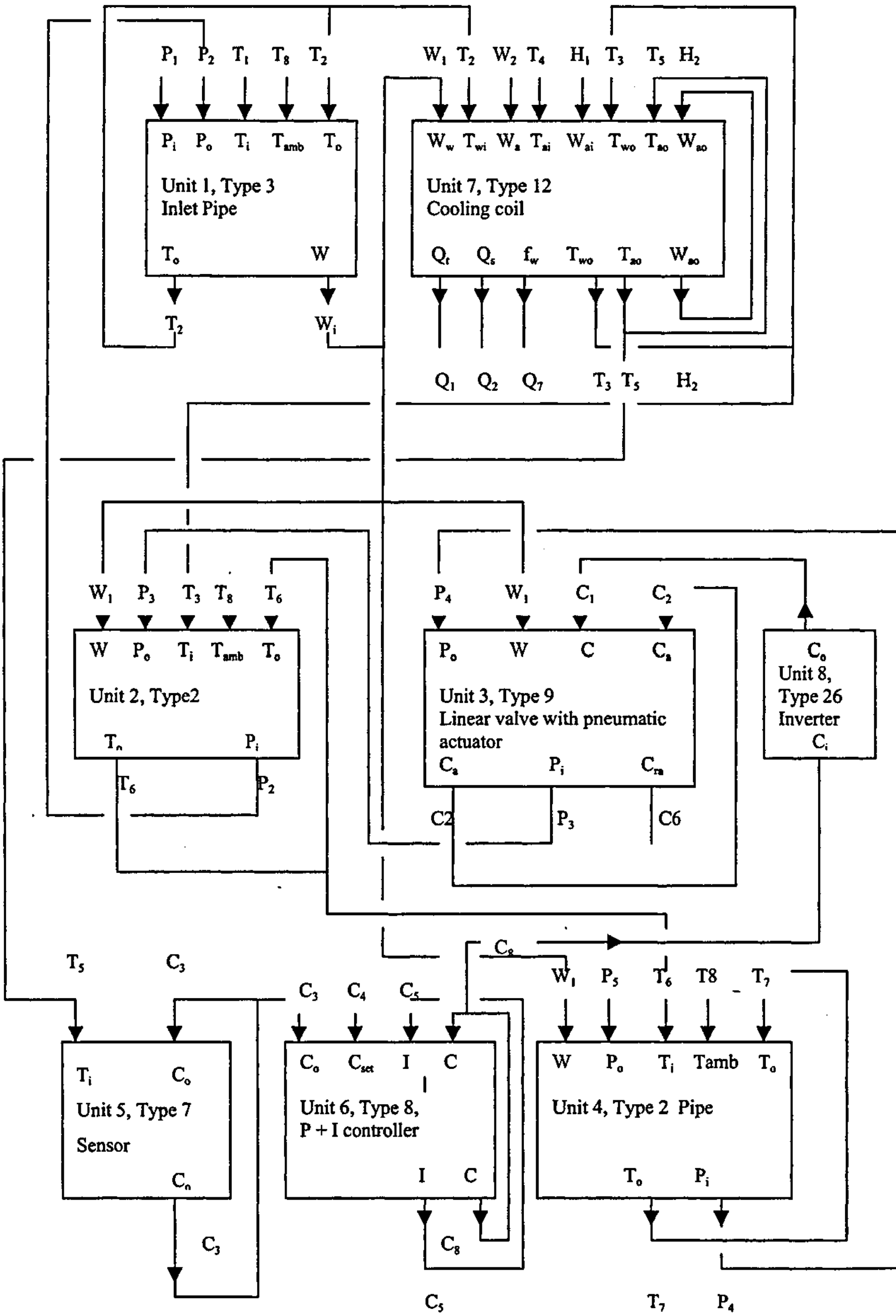


Table 5.2: Simulation Configuration of Superblock 1

Unit No	Type	Input Variable	Output Variable	Parameters	
Unit 1	Type 3 Inlet Pipe	P1 - Inlet water pressure P2 - Outlet water pressure T1 - Inlet water temperature T8 - Ambient air temperature T2 - Outlet water temperature (same as first output)	T2 - Outlet water temperature (same as fifth input) W1 - Water mass flow rate	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x are (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	2.97 0.0031 2.277 0.00251 21.36 -0.5588 2 (water)
Unit 2	Type 2 Outlet Pipe	W1 - Water mass flow rate P3 - Outlet pressure T3 - Inlet temperature T8 - Ambient temperature T6 - Outlet temperature	T6 - Outlet temperature P2 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	2.7 0.00281 2.07 0.002278 11.9 0 2 (water)
Unit 3	Type 9 Linear Valve with Pneumatic Actuator	P4 - Pressure at outlet W1 - Water mass flow rate C1 - Input control signal C2 - Actuator relative position (same as first output)	C2 - Actuator relative position (same as fourth input) P3 - Pressure at inlet C6 - Valve stem relative position	Flow resistance [(0.001/(kg m)] Actuator time constant (sec.) Leakage parameter Hysteresis parameter	30 5 0.00316 0.02
Unit 4	Type 2 Outlet Pipe	W1 - Water mass flow rate P5 - Outlet pressure T6 - Inlet water temperature T8 - Ambient temperature T7 - Outlet water temperature (same as first output)	T7 - Outlet water temperature (same as fifth input) P4 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	38 0.044 32.2 0.0355 11.9 0 2 (water)
Unit 5	Type 7 Temperature Sensor	T5 - Temperature input C3 - Temperature sensor (same as first output)	C3 - Temperature sensor output (same as second input)	Temperature sensor time constant (sec.) Offset (°C) Gain	20 0 100

Unit 6	Type 8 P+I Controller	C3 - Controlled variable C4 - Setpoint for controlled variable C5 - Integral portion of control signal (same as first output) C8 - Output control signal (same as second output)	C5 - Integral portion of control signal (same as third input) C8 - Output control signal (same as fourth input)	Proportional gain Integral gain Controller time constant (sec.)	? ? 2
Unit 7	Type 12 Cooling Coil	W1 - Water mass flow rate T2 - Inlet water temperature W2 - Dry air mass flow rate T4 - Inlet dry bulb air temperature H1 - Inlet air humidity ratio T3 - Outlet water temperature (same as first input) T5 - Outlet air dry bulb temperature (same as second output) H2 - Outlet air humidity ratio (same as third output)	T3 - Outlet water temperature (same as sixth input) T5 - Outlet dry bulb temperature (same as seventh input) H2 - Outlet air humidity ratio (same as eighth input) Q1 - Total cooling load Q2 - Sensible cooling load C7 - Wet fraction of coil surface area	Coil type Primary (tube exterior) surface area (m^2) Secondary (fin) surface area (m^2) Internal surface area (m^2) Ratio of minimum air flow area to face area Fin material thermal conductivity [kW/(m K)] Coil face area (m^2) Number of fins per centimeter Number of tubes per row Number of row Outside tube diameter (m) Inside tube diameter (m) Fin thickness (m) Thermal capacitance (mass x specific heat) of coil (kJ/°C) Tube row spacing in air flow direction (m) Fin diameter (m) Coil depth in air flow direction (m) Tube thermal conductivity [kW/(m K)]	0 (Flat) Ap As Ai ? 0.204 0.7226 ? 16 6 0.01588 0.0139 0.00015 32.08 0.0259 0.508 0.1651 0.386
Unit 8	Type 26 Reversing Relay (Inverter)	C8 - Input control signal	C1 - Output control signal		

INITIAL VARIABLE VALUES:					
PRESSURE (kPa) -	FLOW (kg/s) -	TEMPERATURE (° C) -	CONTROL -	POWER (kW) -	ABSOLUTE HUMIDITY [kg (water)/kg (air)] -
P1 = 250 P2 = 227.754 P3 = 212.307 P4 = 210.571 P5 = -250	W1 = 3.2782 W2 = 2.75	T1 = 8 T2 = 8.013 T3 = 13.7 T4 = 28 T5 = 19 T6 = 13.705 T7 = 13.9 T8 = 28	C1 = 0.5 C2 = 0.5 C3 = 0.19 C4 = 0.19 C5 = 0.5 C6 = 0.5 C7 = 1 C8 = 0.5	Q1 = 78 Q2 = 30	H1 = 0.019 H2 = 0.0092
TIME DEPENDENT BOUNDARY VARIABLES:					
Temperature T4 Inlet air humidity ratio H1					

Table 5.3: The five cooling coil designs

Cooling Coil Design	CCD1	CCD2	CCD3	CCD4	CCD5
Primary (tube exterior) surface area, Ap (m^2)	6.571	6.491	6.41	6.33	6.249
Secondary surface area, As (m^2)	43.647	58.196	72.745	87.294	101.844
Internal surface area, Ai (m^2)	6.199	6.199	6.199	6.199	6.199
Ratio of minimum air flow area to face area	0.562	0.555	0.548	0.542	0.535
Number of fins per unit length (per cm)	2.3622	3.1496	3.937	4.7244	5.5118
Optimum controller settings: Proportional gain Integral gain	 2.574 0.39	 1.4625 0.277	 0.858 0.13	 0.828 0.1255	 0.6773 0.1026

The performances of the coil designs in term of outlet air temperature against time were recorded for step change of inlet air at time 1500 seconds after the simulation started. The step change is from 28°C dry bulb and 0.019 humidity ratio to 30°C dry bulb and 0.022 humidity ratio respectively. To allow stability in the initial conditions, the comparison of performances between different coils started at time 1500 seconds. Different conventional error based objective functions had been used and compared in evaluating the performances of the five coil designs in addition to that using controllability and stability analysis.

The responses of the cooling coil outlet temperature against time for the five cooling coil designs CCD1 to CCD5 under step changes of ambient air temperature and humidity ratio at time 1500 seconds (taken as the reference time 0 second) are shown in Figure 5.3 to Figure 5.7. Five conventional error based objective functions including the mean square error for the five cooling coil designs from the reference time 0 second for a period of 1500 seconds are given in Table 5.4. The performance of design basing on objective function is given in Table 5.5. The analysis basing on the control performance shown in the following paragraph reveals the performance of design is in the order of CCD1>CCD2>CCD3>CCD5=>CCD4. From Table 5.5, the assessment using the integral square-error and the mean sqaure error is in good agreement with the control analysis. Design CCD1 is considered the best design and design CCD4 the worst design. With the same initial values as used in the coil performance study, these two designs were further verified against a typical summer day (11, July 1989) in Hong Kong. During the typical day, the time dependent boundary conditions of ambient temperature and inlet air humidity ratio vary from 27.65°C to 30.65°C, and 0.01845 to 0.02175 respectively. The responses of the cooling coil outlet air temperature against time of the day, 11 July 1989 for cooling coil designs CCD1 and CCD4 are shown in Figure 5.8 and Figure 5.9.

Figure 5.3: Cooling coil outlet air temperature control of CCD1

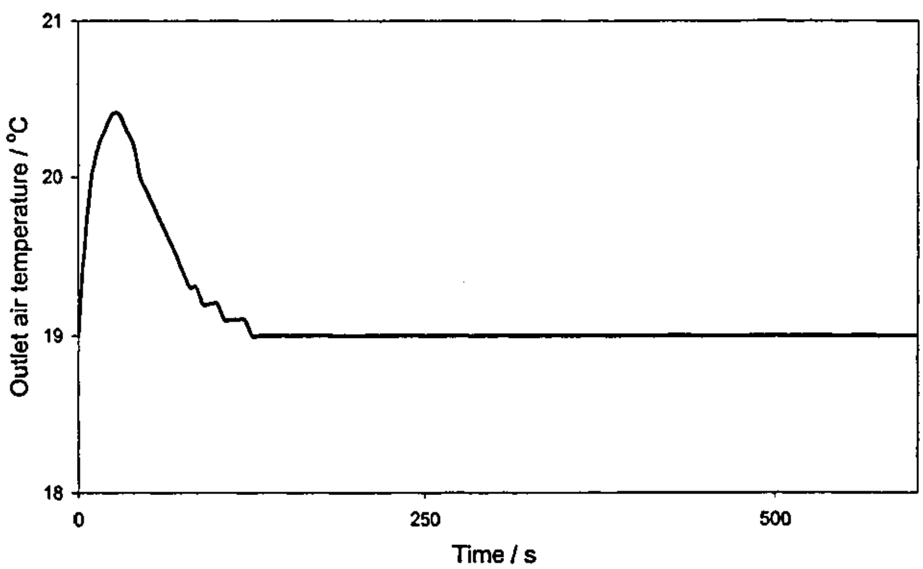


Figure 5.4: Cooling coil outlet air temperature control of CCD2

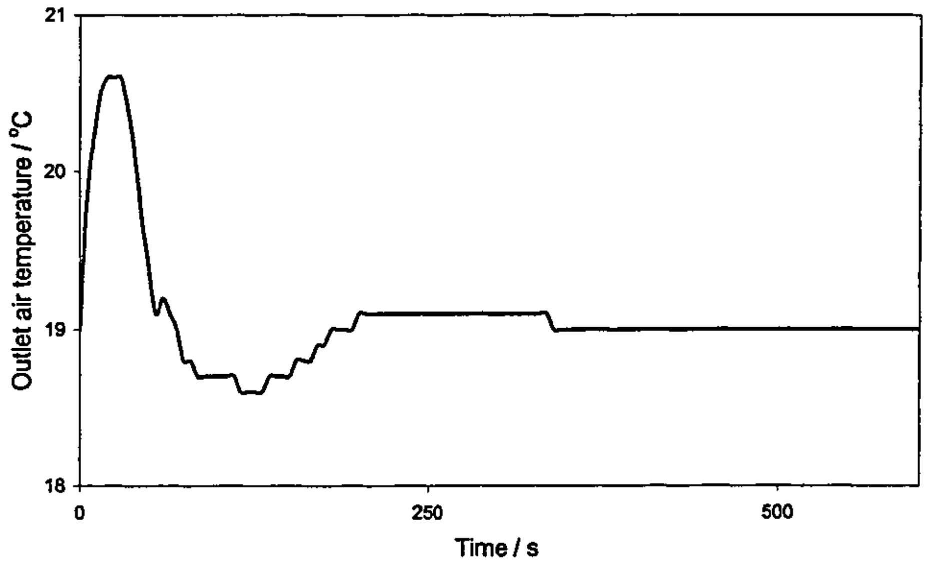


Figure 5.5: Cooling coil outlet air temperature control of CCD3

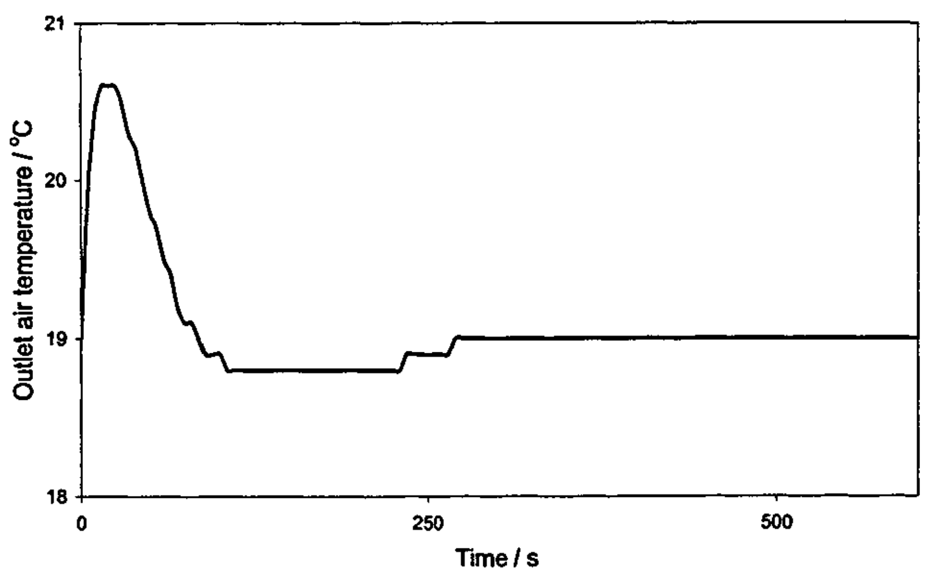


Figure 5.6: Cooling coil outlet air temperature control of CCD4

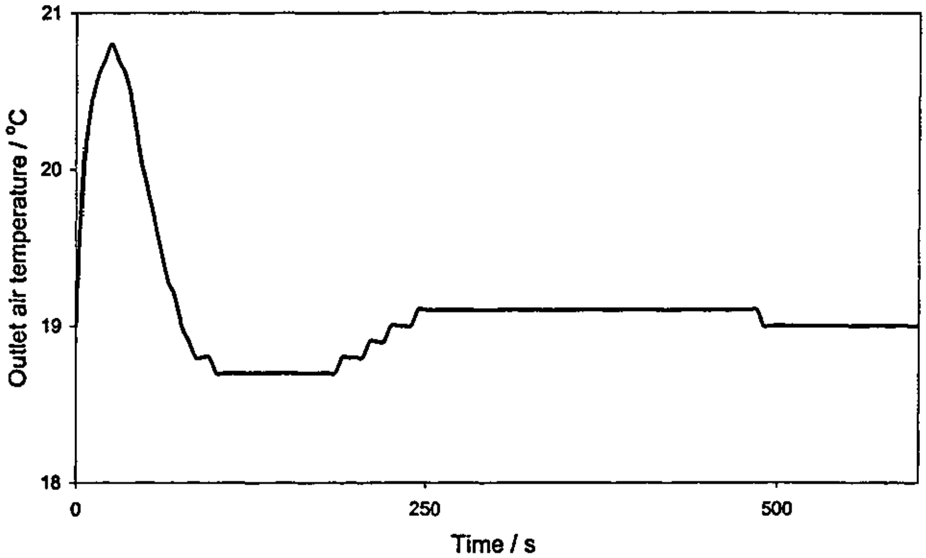


Figure 5.7: Cooling coil outlet air temperature control of CCD5

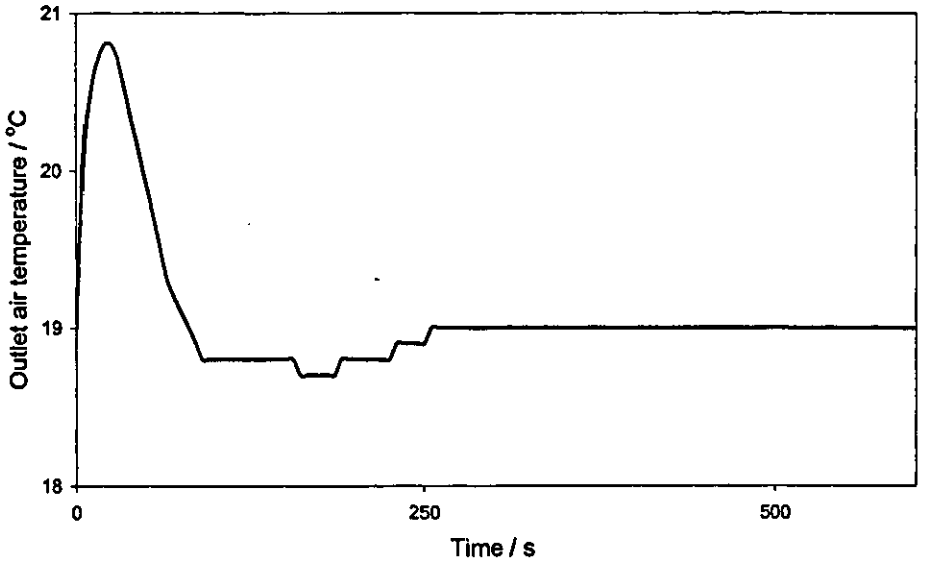


Table 5.4: The mean square errors and other error based objective functions of the five cooling coil designs

Cooling Coil Design	CCD1	CCD2	CCD3	CCD4	CCD5
ISE over an integrating period from 0 to 1500 seconds.	15.59	17.93	17.98	24.38	24.08
ITSE over an integrating period from 0-1500 seconds.	525.50	680.10	2630.20	1061.40	931.75
IAE over an integrating period from 0-1500 seconds.	15.90	20.90	36.40	28.00	22.20
ITAE over an integrating period from 0-1500 seconds.	659	1780	11752	3324	1124
Mean square error for time period 1500 seconds from the reference time 0 second ($°C^2$)	0.08709	0.10017	0.10045	0.13620	0.13453

Table 5.5: Performance of design vs objective function

	Performance of design in order
ISE over an integrating period from 0 to 1500 seconds.	CCD1>CCD2>CCD3>CCD5>CCD4
ITSE over an integrating period from 0-1500 seconds.	CCD1>CCD2>CCD5>CCD4>CCD3
IAE over an integrating period from 0-1500 seconds.	CCD1>CCD2>CCD5>CCD4>CCD3
ITAE over an integrating period from 0-1500 seconds.	CCD1>CCD5>CCD2>CCD4>CCD3
Mean square error for time error time period 1500 seconds from the reference time 0 second ($°C^2$)	CCD1>CCD2>CCD3>CCD5>CCD4

Figure 5.8: Cooling coil outlet air temperature control of CCD1 for 11 July 1989, Hong Kong

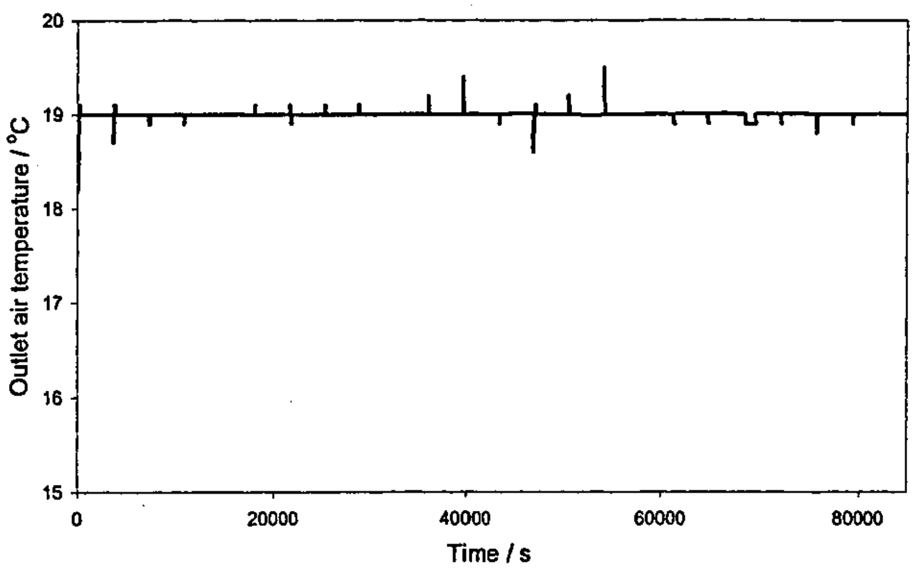
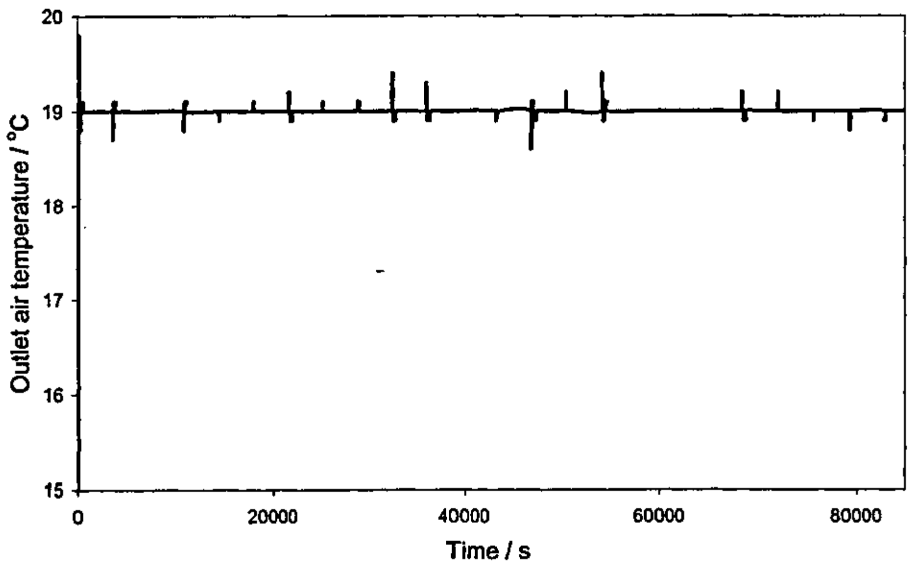


Figure 5.9: Cooling coil outlet air temperature control of CCD4 for 11 July 1989, Hong Kong



In the simulation test, the cooling coil control is under a step change at time zero and settles before the second step change at time 1500 seconds. The time at 1500 seconds can be considered as the reference time 0 second because the system is stable at that time. Analysis is thus made after the step change at the reference time 0 (that is, 1500 seconds) for a period of 1500 seconds. From the response curves of designs, CCD1 to CCD5, shown in Figure 5.3 to Figure 5.7, the followings are observed:

- The cooling coil outlet temperature for all five designs, when under the step change in the boundary variables, settles and can be controlled at the setpoint temperature of 19°C with only little hunting.
- The magnitude of mean square error is in the order:
  CCD1 < CCD2 < CCD3 < CCD5 < CCD4
- The maximum overshoot is in the order:
  CCD1 < CCD2 and CCD3 < CCD4 and CCD5
  Except CCD1, all other designs have undershoots.
- The settling time to the setpoint temperature of 19°C is in the order:
  CCD1 < CCD5 < CCD3 < CCD4 < CCD2
  If a reasonable tolerance of +/- 0.1°C is allowed in the final steady state value of the supply air temperature, the settling time will be in the order:
  CCD1 < CCD2 < CCD4 < CCD5 < CCD3
- The response is in the order: CCD3 > CCD2 and CCD5 > CCD1 and CCD4.

Considering the above, CCD1 has the less overshoot when under the step change. The response is reasonable and stable. The system settles at the setpoint temperature at a short time period without much overshoot. CCD2 and CCD3 have more or less the same overshoot magnitudes when under the step change. However, CCD2 is better in stability and settles within +0.1°C faster. CCD4 and CCD5 have about the same amount of overshoots and theirs overshoots are larger than the other designs when under the same step change. The settling time for CCD4, when a tolerance of 0.1°C is allowed, is a bit shorter than CCD5. Their responses are more or less the same. However CCD5 has less undershoot period than CCD4. In conclusion, the performance of designs, from control point of view, is in the order: CCD1 > CCD2 > CCD3 > CCD5 >= CCD4. This order is in good agreement with the criterion of using the mean square error (the mean square error of CCD4 is marginally larger than that of CCD5) and the integral square-error. Hence, it can be concluded that mean square error (or the equivalence, integral square error) is a good and effective objective function or indicator for evaluating the performance of the temperature control system.

The best and the worst designs, CCD1 and CCD4 were verified against a typical summer day in Hong Kong. From their response curves shown in Figure 5.8 and Figure 5.9, the followings are observed:

- The cooling coil outlet air temperature is within 14.9 to 19.8°C and can be controlled at 19°C without much hunting.
- The cooling coil outlet air temperature response is more stable in CCD1 than in CCD4. CCD4 has more hunting and longer hunting period than CCD1. Also CCD4 has larger overshoot magnitudes. CCD1 has shorter settling time.

Both cooling coil designs can fulfill the requirements of the coil outlet air temperature control. Yet CCD1 is more energy efficient and cost effective than the CCD4 (less hunting, smaller overshoot and shorter settling time). Moreover, the system response is also found more stable.

From the above simulation study, it can be concluded that for the simple a/c cooling coil control sub-system, simulation study using HVACSIM+ indicates that there exist good and bad designs. The performances of different designs are evaluated against the performance index 'mean square error'. The performances of different designs are also evaluated and compared by control considerations of stability and response. It is found that the performance evaluation using mean square error is generally in good agreement with those using control considerations. The mean square error is a good performance index for the simple a/c cooling coil control sub-system. The resulting coil designs are also verified against a typical summer weather day data of Hong Kong. The verification indicates good agreement with results from simulation. Hence, the mean square error should be considered for use in evaluating the performance of an entire a/c system.

CHAPTER 6 INVESTIGATION OF PARAMETERS FOR CONTROL PERFORMANCE OF A/C SYSTEM BY EXPERIMENTAL STUDIES

The preliminary simulation study in chapter 5 reveals that simulation can help to optimize the design of simple a/c cooling sub-systems. It also illustrates that mean square error is a good objective function in assessing the control performance of a simple a/c cooling sub-system. It is also believed that such findings could be useful for the optimization of an overall a/c system. However, in an a/c system, besides the physical coil design, the control performance of the a/c system is affected by the control valve, sampling rate (in DDC control), and controller settings. Of course, the objective function used in assessing the control performance is also an important consideration. The characteristic parameters of control valve and its actuator drive, as well as the controller settings of controller are input parameters of the simulation study. In addition, the sampling rate, which measures the control system feedback delay, also affects the control performance of a simulated a/c system. Hence, it is very important to understand how these factors will practically affect the control performance of an a/c system in order to improve the effectiveness and accuracy of applying simulation for optimizing a/c system design in control performance. The practical influence of these factors in affecting the control performance of a/c systems has never been reported in detail. Experimental studies of their influence will show their practical impact on current commercially used a/c systems and help to provide practical data for developing future theoretical study. Therefore, The results of the experimental studies basing on practical a/c systems under Hong Kong ambient conditions will definitely enhance and improve the accuracy of the simulation study for Hong Kong applications. Hence, experimental studies to investigate the impacts by these factors on the control performance of a/c systems were conducted.

Two experimental rigs were designed and built for studying and investigating the impacts on the control performance of a/c systems due to control valve, controller settings, sampling rate (in DDC control) and different objective functions. The first set-up aims to study the impacts due to size and actuator drive of control valve. In the set-up, the supply air temperature of an environmental chamber is controlled. The response of the supply air temperature is analyzed for different sizes and actuator drives of control valve used. The second experimental set-up aims to study and investigate the impacts of sampling rate, controller settings and objective function used on the control performance of an a/c air duct temperature control system. The air duct supply air temperature is being controlled and its response due to different sampling rates and controller settings are studied. Different objective functions are analyzed in relation to control performance. Details of these experiments and their experimental set-ups are described in the following sections.

6.1 THE CONTROL VALVE CHARACTERISTICS AND SIZING

Ho[52] discusses in details the engineering approaches in the selection of control valve for a/c applications. The critical engineering considerations are concluded as followings:

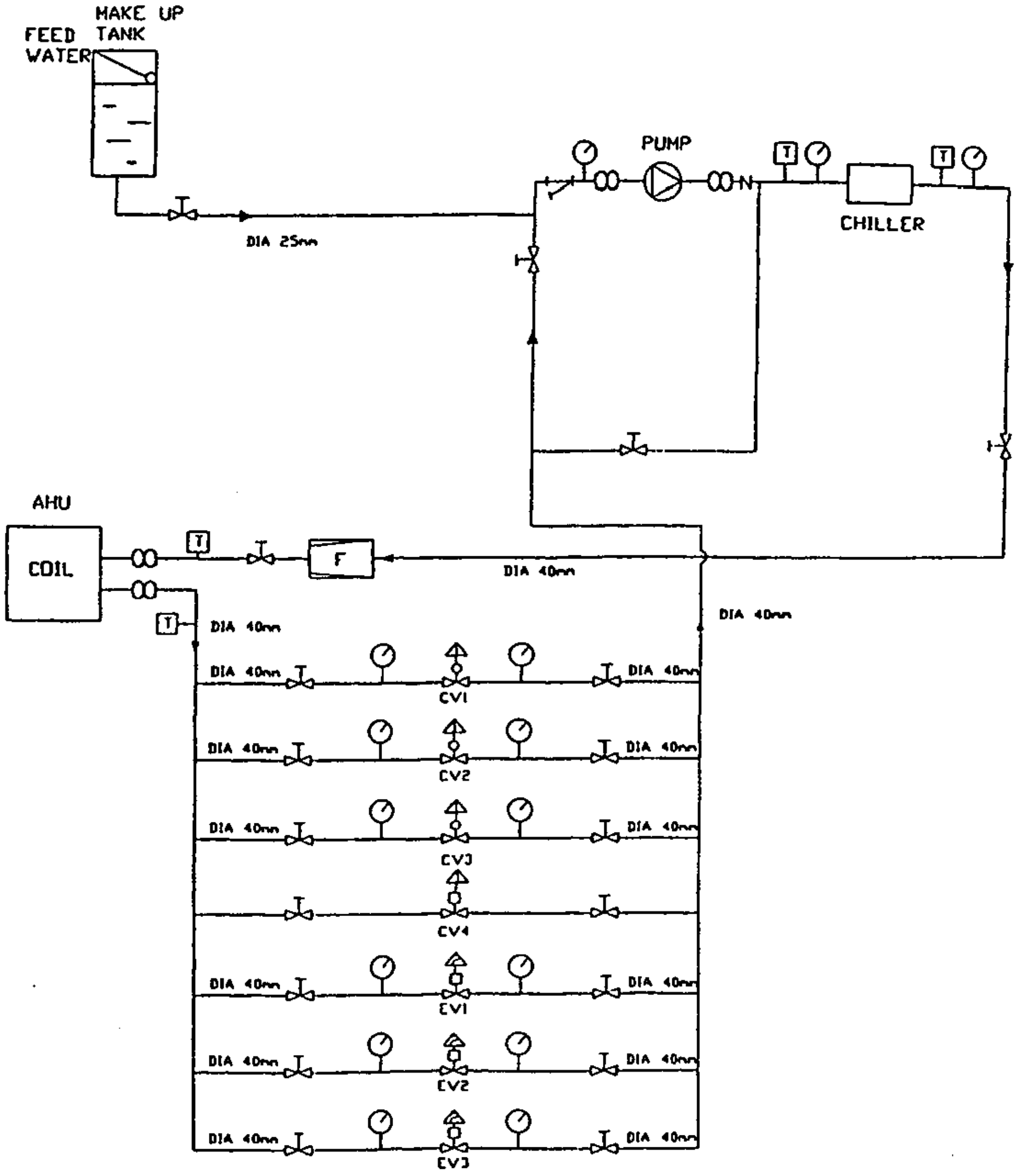
- Static pressure has to be considered for high rise buildings.
- Pump head has to be compromised between pump energy, control and valve cost.
- Close-off is important for no leakage flow requirement.
- Rangeability and hence valve characteristics have to be considered to suit the nature of control system.

Yet the impacts of control valve characteristics, size and drive on the control performance are not often considered. In this research study, the characteristics of different size control valves and their actuator drives were studied experimentally for an a/c control system. The aims of the study focus on, firstly the flow and opening relationship for each control valve and the response rate (time constant) of the associated valve actuator, secondly the impact of control valve size and the nature of control valve actuator drive in the control performance of a/c application.

6.1.1 The experimental study on the characteristics of control valve and actuator drive

The experimental set-up is an indoor aerodynamic chamber shown in Figure 6.1. There are a package chiller of 21 kW, an air handling unit (AHU) and an air distribution system. The total head of the system pump is 96 kPa. The supply air temperature is controlled by regulating the chilled water flow through the cooling coil in the AHU. The chilled water flowing in the 1½" pipe is measured by an electromagnetic indicating flowmeter. The flow signal is received by the analogue controller. There are eight control valves arranged in parallel. In the experiment, only one control valve is used to control the chilled water flow and the other control valves are completely closed. The control valves belong to two families of four control valves each. One set is pneumatic control valves of sizes ½", ¾", 1" and 1¼" and the other set is electric valves of ½", ¾", 1" and 1¼". However, the ½" electric was out of function and could not be tested. It is considered no problem for testing, as there are still different sizes of electric valve in the family. Each control valve is tested in sequence. The minimum and maximum chilled water flows in the coil circuit are 0 l/s and 1.04 l/s.

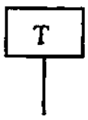
Figure 6.1: Experimental set-up for study of control valves in the supply air temperature control of air indoor aerodynamic chamber

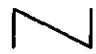


Legend for Figure 7.1

 Pressure gauge (0-300 Kpa)

 Flexible joint

 Temperature sensor

 Check valve

 Gate valve (Diameter 40 mm)

Pneumatic control signal from electric/pneumatic converter

 Electric control valve

Electric control signal from controller

 Pneumatic control valve

AHU Air handling unit

 Globe valve

 Strainer

F	Electromagnetic flowmeter, range 0-1.5 L/s
CV1	Pneumatic control valve, diameter 13 mm (1/2")
CV2	Pneumatic control valve, diameter 20 mm (3/4")
CV3	Pneumatic control valve, diameter 25 mm (1")
CV4	Pneumatic control valve, diameter 32 mm (1 1/4")
EV1	Electric control valve, diameter 20 mm (3/4")
EV2	Electric control valve, diameter 25 mm (1")
EV3	Electric control valve, diameter 32 mm (1 1/4")

The experiments aim on finding, firstly, the flow and opening relationship of control valves and their associated actuator time constants, and secondly the control performances of control valves in the supply air temperature control of the indoor aerodynamic chamber. These experiments were conducted for each control valve in sequence. The other control valves in parallel were completely closed during the experiments.

In the first test, the automatic control was disabled and the control valve was opened in small increment from fully closed to fully opened and then in small decrement from fully opened to fully closed. The corresponding chilled water flows were recorded. The response and hence the time constant of the control valve actuator was measured. The experimental results of the relationship between the water flow and the valve opening for the control valves are shown in Figures 6.2 to 6.8. The time constants of the control valve actuators are given in Table 6.1. The time constant for the temperature sensor was experimentally determined by an open loop test by inserting the temperature sensor at the ambient temperature of 28°C dry bulb into the supply air duct at 12°C when the 1" electric control valve was fully opened. The time constant of the temperature sensor was found to be 20 seconds.

Figure 6.2: Control valve characteristics, 1/2" pneumatic control valve

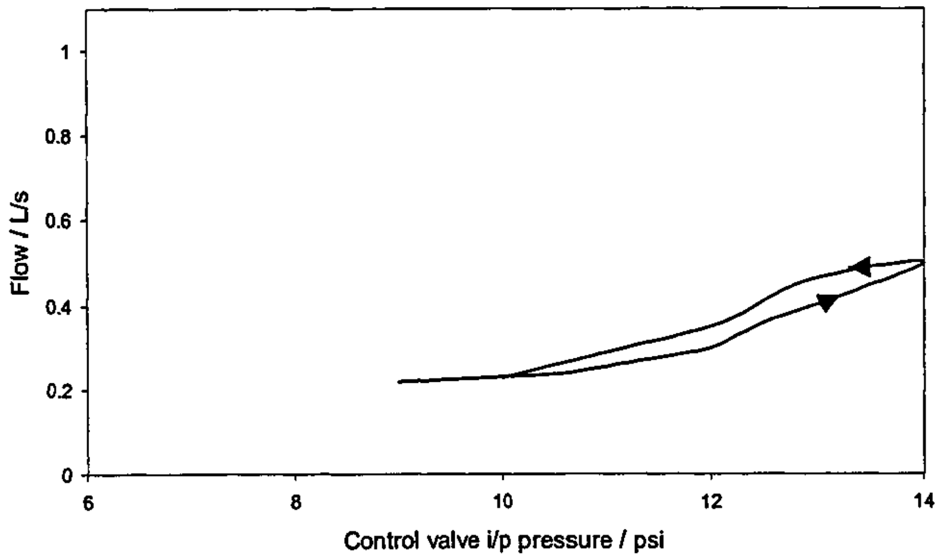


Figure 6.3: Control valve characteristics, ¾" pneumatic control valve

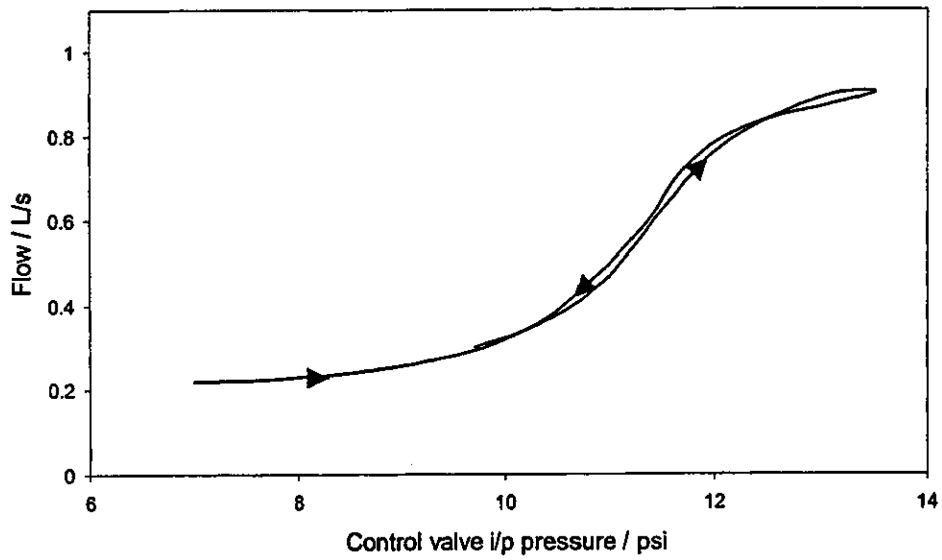


Figure 6.4: Control valve characteristics, 1" pneumatic control valve

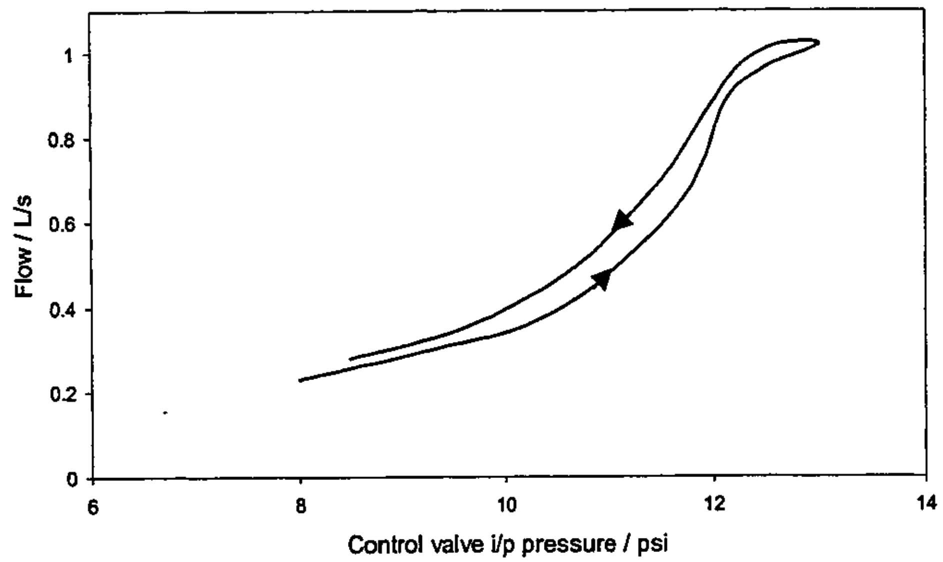


Figure 6.5: Control valve characteristics, 1 1/4" pneumatic control valve

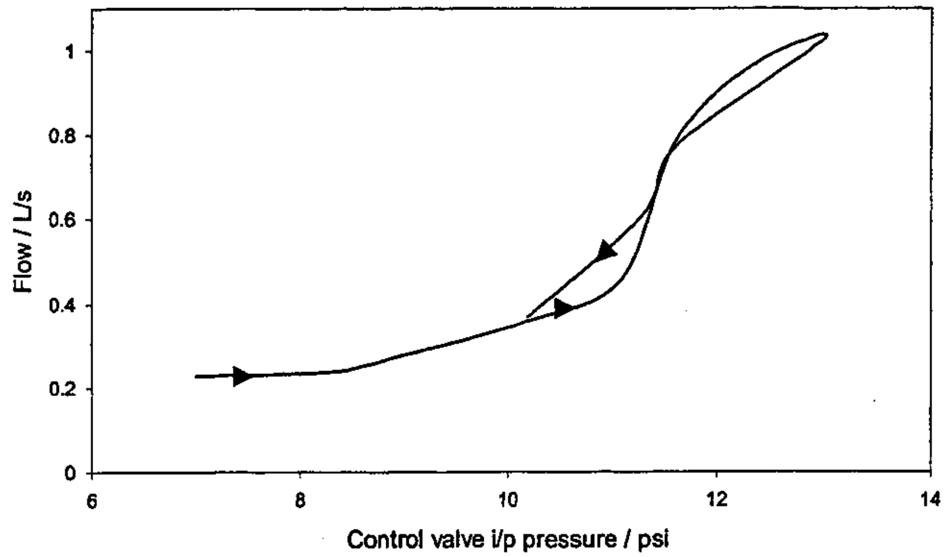


Figure 6.6: Control valve characteristics, 3/4" electric control valve

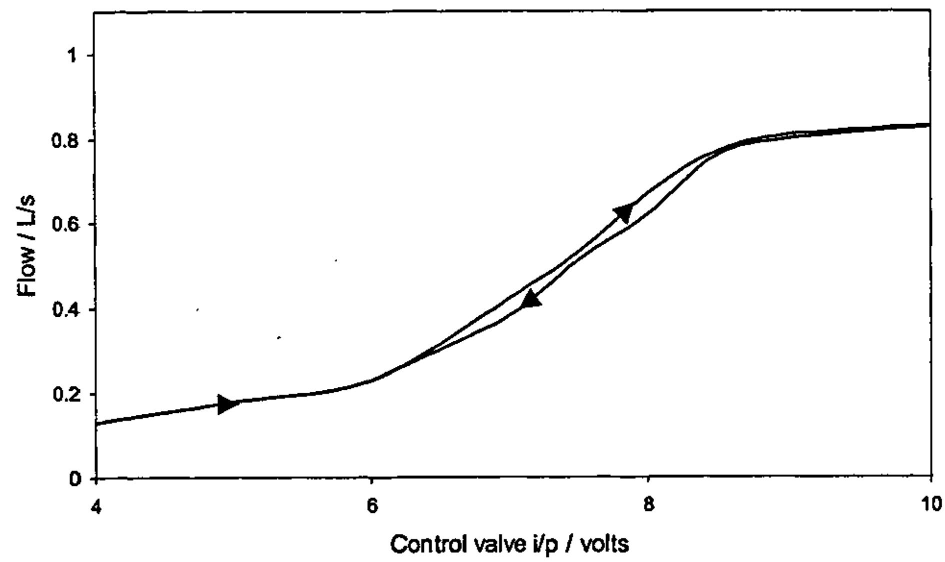


Figure 6.7: Control valve characteristics, 1" electric control valve

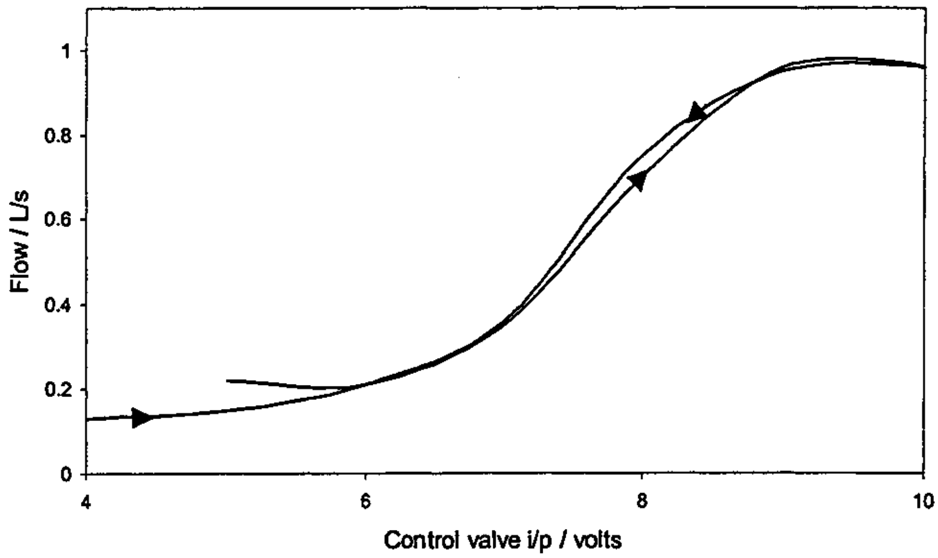


Figure 6.8: Control valve characteristics, 1 1/4" electric control valve

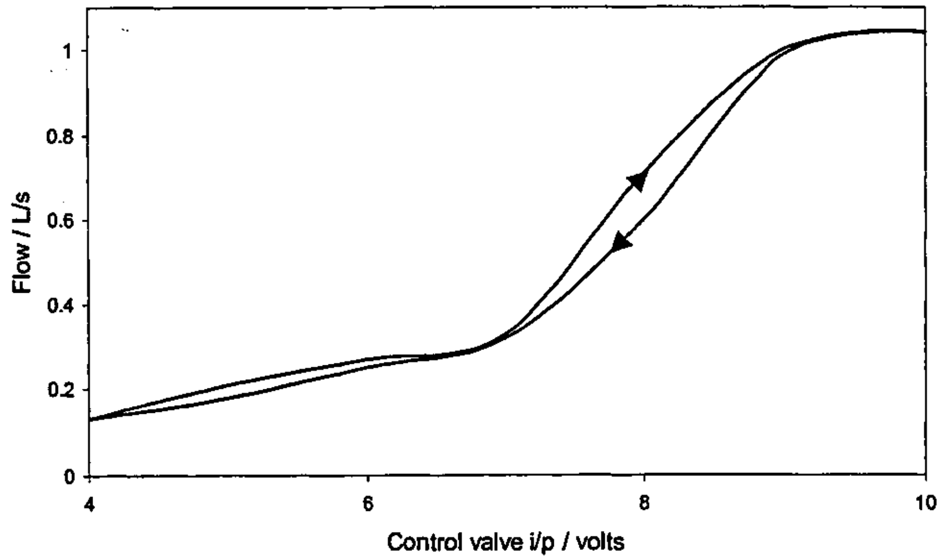


Table 6.1: Time constants of control valve actuators

Control valve actuator & type	Averaged time constant over flow range (s)
3/4 " electric actuator	9.5
1 " electric actuator	10
1 1/4 " electric actuator	10.6
Average	**10.0**
1/2 " pneumatic actuator	20
3/4 " pneumatic actuator	21
1 " pneumatic actuator	22.1
1 1/4 " pneumatic actuator	22.7
Average	**21.5**

The spring range of the pneumatic control valve is 9 psi (fully closed) to 13 psi (fully opened) and the electrical range of the electric control valve is 4 volts (fully closed) to 10 volts (fully opened). Analysis from Figures 6.2 to 6.8 and Table 6.1 shows the followings.

- There is leakage for all control valves. Electric control valves have smaller leakage (approximately 0.13 l/s) than the pneumatic valves (approximately 0.22 l/s). Hence the close-off rating for electric valve is better than the pneumatic valve.
- The performance in repeatability for pneumatic control valves is in the order of: ¾" > 1" > ½" > 1¼". In the electric control valves, the performance of repeatability is in the order of: ¾" > 1" > 1¼". The ¾" pneumatic control valve has the best repeatability. However, in general, electric control valve has better repeatability than the pneumatic control valve.
- The rangeabilities of the control valves are tabulated in Table 6.2 as shown. The rangeability of a control valve is defined as the ratio of maximum controllable flow to minimum controllable flow of a control valve.

Table 6.2: Rangeabilities of the control valves

Control valve size & type	Minimum flow (l/s)	Maximum flow (l/s)	Turndown ratio	Rangeability	Kv (from Manufacturer)
½", pneumatic	0.22	0.5	2.27	4.73	4.62
¾", pneumatic	0.22	0.5	2.27	4.73	7.28
1", pneumatic	0.23	1.02	4.43	4.52	11.56
1¼", pneumatic	0.23	1.03	4.48	4.52	18.5
¾", electric	0.13	0.83	6.38	8	7.28
1", electric	0.13	0.96	7.38	8	11.56
1¼", electric	0.13	1.04	8	8	18.5

- The rangeability of the control valves is rather low and no tight control is expected with such low ranges of rangeability. It can be seen that the rangeability of electric control valve is better than the equivalent pneumatic control valve. In fact, most control valve manufacturers do not provide data on the rangeability of commercial control valves. Therefore, the rangeability of control valve for HVAC applications should be specified when tight control is required.
- The time constant of the control valve actuator will affect the control system response. In fact, it is one of the input parameter that affects the dynamic response of the control valve model described in HVACSIM+. The average time constant of the electric valve actuators under study is approximately 10 seconds and that of the pneumatic valve actuators is 21.5 seconds.

The second experiment tested experimentally the control performance of control valve in the supply air temperature control of the indoor aerodynamic chamber. Before the control performance for each control valve was tested, the controller settings for supply air temperature control using that control valve was found by Nichols and Ziegler method[15]. The Nichols and Ziegler method was used because of its simplicity. So each control valve was tested with Nichols and Ziegler PID estimates for that control valve. During the experiment, the supply air temperature was first controlled at 17°C (by setting the setpoint of the supply air temperature at 17°C and allowing the control system to run for an hour). A step increase of setpoint temperature to 22°C was applied. The response was recorded for 50 minutes with recording interval of 5 seconds. The ambient conditions when the experiments were conducted were also recorded. There is experimental difficulty encountered during the tests. The chiller provides more cooling than the requirement of the air handling unit. As a result, the compressor of the chiller often cuts off and on. Therefore the chilled water temperature fluctuates within a range of 2 to 3°C around the set chilled water temperature and is not constant.

The controller PID estimates found and used for the control valves are tabulated in Table 6.3. The ambient conditions are also shown in Table 6.3. The step responses of supply air temperature against time of the control valves for setpoint supply air temperature increased from 17°C to 22°C were recorded and shown in Figures 6.9 to 6.10. Although the D control has little effect on the control, the control system is built with PID control for further future study. The D control was also used in the experiment so as to increase slightly the damping of the control system.

Table 6.3: Controller estimates based on Nichols and Ziegler method

Control valve type & size (ambient conditions dry bulb, wet bulb)	Proportional band (%)	Integral action time (s)	Derivative action time (s)
Pneumatic, ½" (27°C, 25°C)	25	34	5
Pneumatic, ¾" (32°C, 27°C)	14	12	2
Pneumatic, 1" (31°C, 26°C)	42	10	2
Pneumatic, 1¼" (31°C, 26°C)	31	29	4
Electric, ¾" (31°C, 26°C)	42	13	3
Electric, 1" (31°C, 26°C)	50	12	3
Electric, 1¼" (31°C, 26°C)	67	40	5

Figure 6.9: Room supply air temperature control by (i) 1/2" pneumatic control valve, (ii) 3/4" pneumatic control valve, (iii) 1" pneumatic control valve, (iv) 1 1/4" pneumatic control valve

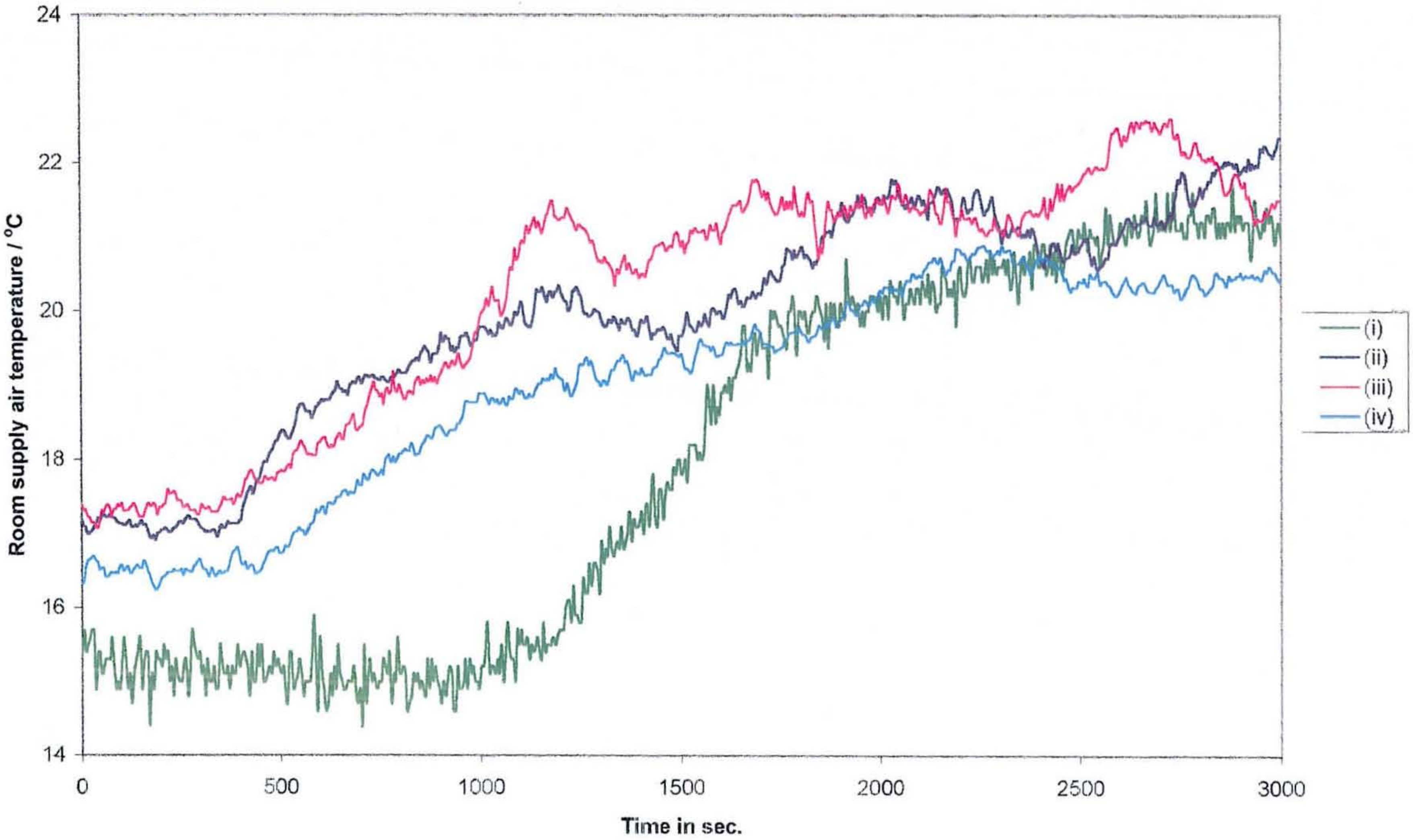
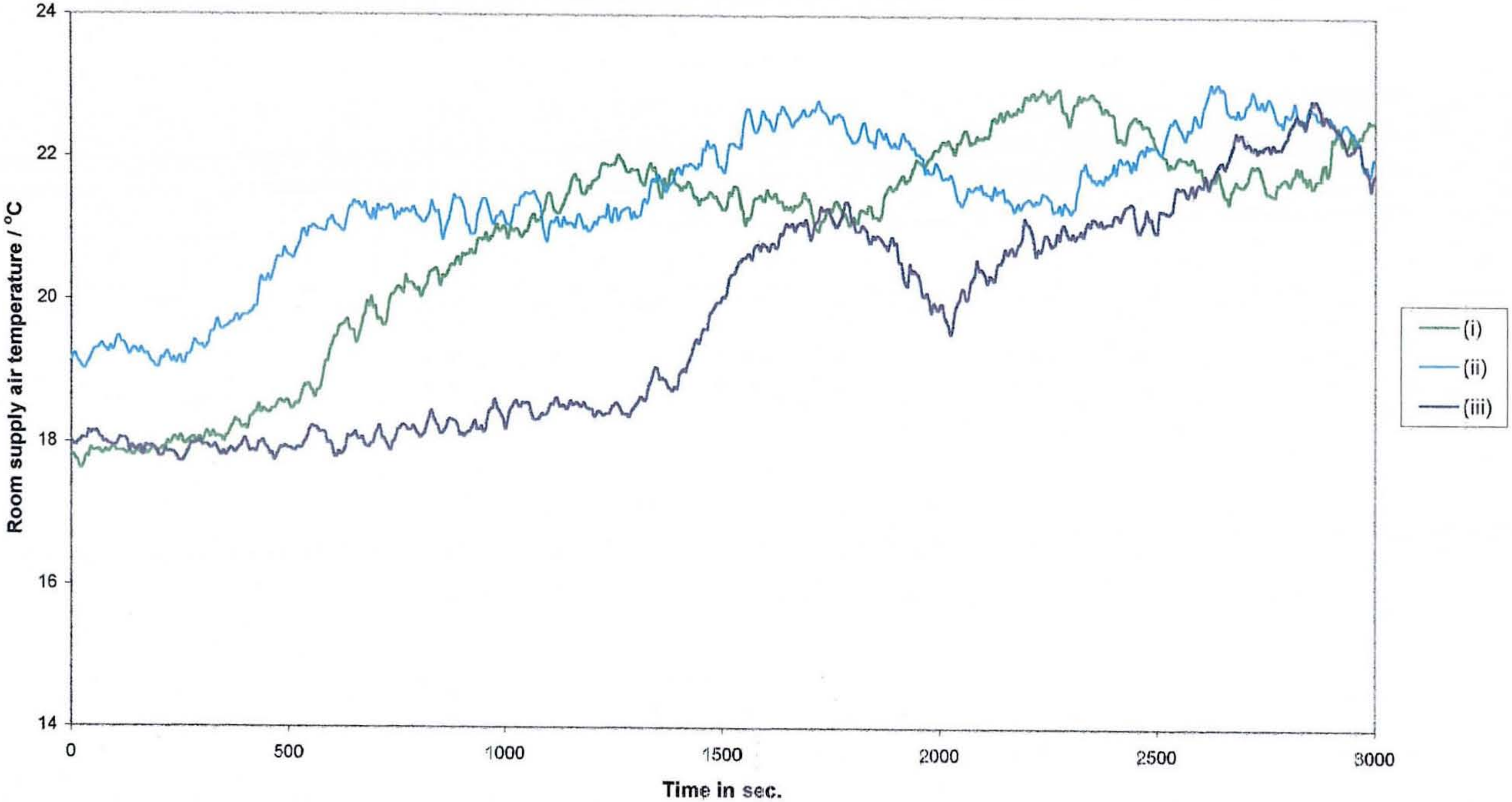


Figure 6.10: Room supply air temperature control by (i) 3/4" electric control valve, (ii) 1" electric control valve, (iii) 1 1/4" electric control valve



The responses of the control valves in the control of supply air temperature can be evaluated in terms of defined control objective function, offset, rate of response and settling time.

Based on the previous study[53] and the simulation study in chapters 5, mean square error is used as the objective function for study. The mean square errors for the supply air temperature control of the control valves are tabulated in Table 6.4.

Table 6.4: The mean square errors of the supply air temperature control of different control valve sizes and types

Control valve size & type	Mean square error
½", pneumatic	14.20
¾", pneumatic	6.5
1", pneumatic	5.75
1¼", pneumatic	10.99
¾", electric	3.58
1", electric	1.40
1¼", electric	7.607

The value of mean square error for the pneumatic control valves is 1" < ¾" < 1¼" < ½" and for the electric control valves is 1" < ¾" < 1¼". The 1" electric control valve has the smallest mean square error.

In making the analysis on response and stability, the response rate of the control valves in the pneumatic family is in the order of: 1"> ¾" > 1¼" > ½". For the electric family, the response rate is in the order of: 1" > ¾" > 1¼". This is the same as the pneumatic family. In fact, this response rate agrees with the control evaluation using the objective function mean square error discussed above. The initial response of pneumatic control valve is, in general, faster than the electric control valve.

However, afterwards, the electric control valve responses quite fast to the setpoint and becomes oscillatory around the setpoint. The oscillation is +/- 1°C for sizes ¾" and 1". For the pneumatic control valves, the control is stable and becomes settle. There are offsets for sizes ½" (approximately -1°C) and 1¼" (approximately -1½ °C). Both electric and pneumatic control can be considered acceptable. Nevertheless, as the oscillation of the electric control valves is within +/- 1°C, which is well within the comfort range suggested by Fanger(13), the control performance can be considered better than the pneumatic valves because of shorter time constants and also the mean square error is less for the same size.

Considering the effect of size of control valve on the control performance, both the 1" pneumatic and electric control valves give best control performance for the drive family concerned. It is obvious that sizing the proper control valve is important for the control of a thermal control system.

From the above experimental studies, the following could be concluded for the impacts on the control performance of a/c system due to the characteristics, size and actuator drive of control valves.

- Pneumatic control valve has the advantages of faster initial response, stable operation and settling. However, for the same size, electric control valve has shorter time constant, better objective function (less value of mean square error), better repeatability and close-off than pneumatic control valve. Through it gives more oscillatory response, the oscillation is within the comfort temperature range. It is appropriate for control of a/c systems, as the thermal response in a/c systems is relatively slow. The control performance is therefore better than pneumatic control valve.
- Sizing of control valve is important for achieving good control performance.

6.1.2 The application of the experimental results in the HVACSIM+ simulation for optimization study of a/c system

The experimental results provide important and useful data for the control valve model in the following detailed HVACSIM+ simulation study of a/c cooling system. The data will improve the accuracy of the control valve model and, therefore the accuracy of the simulation. The experimental results show that electric valve has a better control than pneumatic valve in a/c cooling control after the a/c system has started up for a period of time. Therefore electric valve should be chosen for a/c system installation. Electric control valve should therefore be specified in a/c control system simulation. The approximately electric actuator time constant to be used in the simulation is 10 seconds, which is an average value for most electric control valves used in HVAC systems. Besides, the time constant of the temperature sensor, which was found equals to 20 seconds in the experiment, will also be used in the detailed simulation study. The experimental results also indicate that the control performance of an a/c system is affected significantly by proper sizing of control valve. Proper control valve size for a particular a/c system can be achieved by tuning the control valve resistance parameter in the control valve model of the HVAVSIM+ simulation for that a/c system against a desired control response. The impact of these two issues on the control performance of a single zone CAV a/c cooling system will be tested in the following detailed simulation study in chapter 7.

6.2 THE SAMPLING RATE

Computerized building management system (BMS)[(54),(55)] is popularly in use for large commercial building and complexes. One of the main functions of BMS is to control building thermal environment by applying the technique of direct digital control (DDC) in the HVAC (heating, ventilating and air-conditioning) systems. There are two system concepts for control and monitoring in BMS i.e. centralized and distributed. In centralized BMS, sub-systems are controlled and monitored by a main computer located centrally. Centralized BMS was very common in the 70's. However, with the decrease in the cost of electronics, distributed BMS, which consists of self-functioning control and monitoring sub-systems, has become popular.

One of the control features in distributed BMS of building complexes is faster sampling rate for the HVAC system control. With a centralized BMS, the system has a number of monitoring and control functions. The technique of scanning functions with override priorities is applied. Hence, the DDC takes a much longer sampling rate for HVAC control than the HVAC control in distributed BMS. It is believed that the shorter the sampling rate, the better the control. Krakow[(56),(57)] concluded that the system response characteristics of a DDC control system are related to controller settings, sampling rates, performance parameters of the refrigeration system and the volume of the conditioned space. It is also suggested that the sampling rate should be smaller than the system response. The aim of this research is to find out experimentally the impact of sampling rate on the control performance of a/c systems. In the experiment, air duct temperature control is experimentally studied by an air duct temperature control rig, which consists of a short section of air duct, a fan, an electric heater and DDC control system.

Here, it needs to emphasize that the results of this experimental study will not contribute to the following detailed simulation for optimization of a/c system design. The detailed simulation focuses only on analogue a/c control system. Nevertheless, this study on the impact of sampling rate on the control performance of digital control system will be significantly important when optimization of DDC a/c system design is considered by digital simulation. It is a supplementary study to the main research program.

6.2.1 The experimental study of sampling rate on the control performance of an air duct DDC temperature system

The experiment set-up is shown in Figure 6.11. The air velocity is set at 0.7 m/s. The temperature sensor measures the air duct supply air temperature and sends this temperature signal to the controller. The digital controller, which is a personal computer, is programmed with a PID (proportional, integral and derivative) control in digital form[(16)]. On receiving the air duct supply air temperature signal, the digital controller will implement the PID control and send a digital control signal to the variac. The variac, based on the control signal, will regulate the amount of electrical power to the heater to affect the control of the supply air temperature to the set point temperature. Since both temperature sensor and the varic are analogue equipment, an ADDA (analogue to digital and digital to analogue) device is used to integrate and interface these equipment with the digital controller.

The PID control subroutine was programmed with the PID controller output equation as shown below.

Controller output,

$$c(t) = K_p[e(t) + T_d.(de/dt) + (1/T_i).\int e(t)dt] \quad \dots (6.1)$$

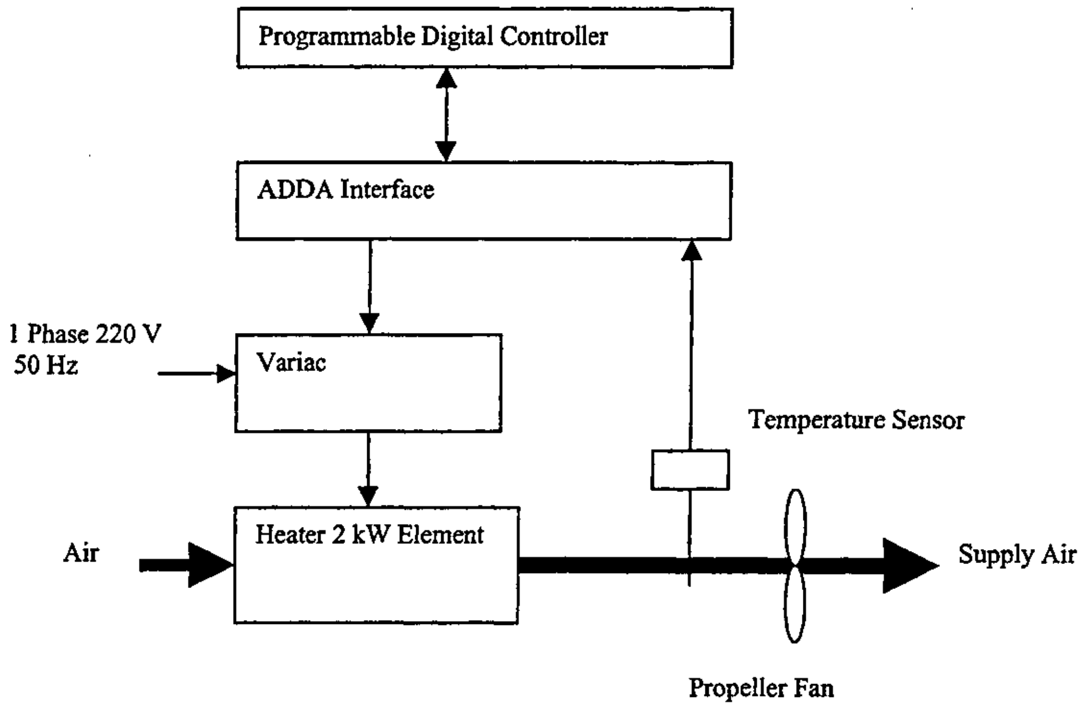
where

$c(t)$	=	controller output (V)
K_p	=	proportional constant (V/°C)
T_d	=	derivative action time (s)
T_i	=	integral action time (s)
$e(t)$	=	error = (control variable – set point)

and de/dt at $t = kT$ can be approximated by $[e(k) - e(k-1)]/T$

Where T is the sampling time in seconds, and $\int e(t)dt$ at $t = kT$ can be approximated by the mid-ordinate rule as: $T[0.5e(0)) + e(1) + \dots\dots + e(k-1) + 0.5e(k)]$.

Figure 6.11: Digital control of an air duct temperature control system



The experiment consists of two parts. The first part of the experiment is to find the estimates of PID controller settings for the air duct temperature DDC system based on the Nichols and Ziegler method(15). The second part of the experiment is the tests to find the impact of sampling rate on the control performance of an air duct temperature DDC system. The controller settings determined in the first part of the experiment were used for the experiments in the second part.

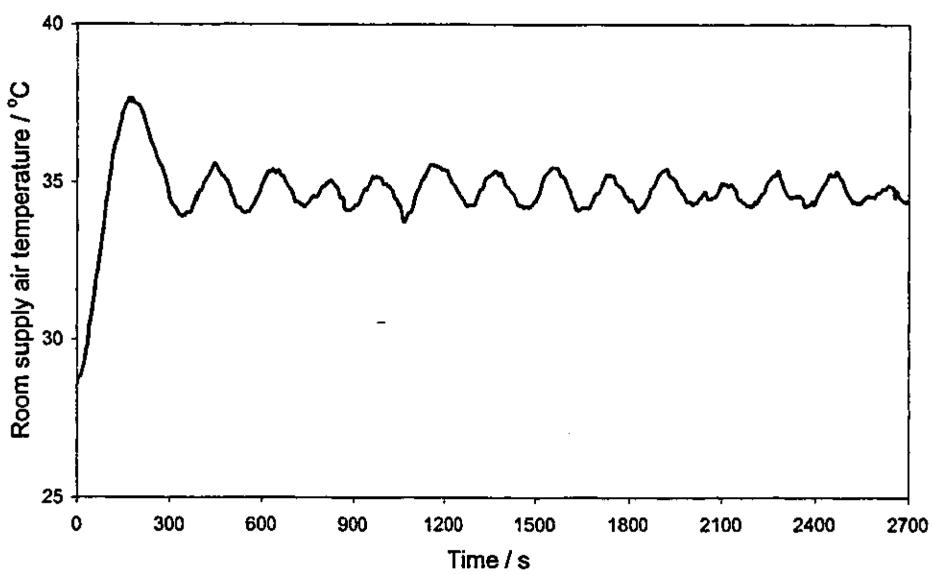
In the first part of the experiment, the sampling time (rate) was set at 5 seconds. Only proportional control was applied to the control system and the value of proportional constant was set very large in the initial experiment. The initial air duct supply temperature was 28.6°C. A step change in the set point of the supply air temperature to 35°C was applied. The air duct supply air temperature was recorded against time for 45 minutes. The test was repeated with different values of proportional constant until the condition of sustained oscillation was achieved in the supply air temperature. The proportional constant and the period recorded at sustained condition were used to calculate the PID values according to Nichols and Ziegler(15).

In the second part of the experiment, all the experiments were conducted at an initial temperature of 28.6°C and the estimated values of PID were set on the digital controller for each of the experiments. The following test was repeated for sampling time equals to 1 second, 2 seconds, 5 seconds, 8 seconds, 11 seconds, 15 seconds, 20 seconds, 25 seconds, 30 seconds, 35 seconds and 40 seconds.

A step input to set the setpoint air duct temperature to 35°C was applied to the digital control system with initial supply air temperature at around 28.6°C. The air duct supply air temperature was recorded against time for 20 minutes.

In the first part of the experiment for estimating the controller settings, the response curve of the air duct supply air temperature against time was shown in Figure 6.12. The proportional constant and period at the sustained oscillation condition were recorded and measured from the curve. Based on these values, the controller settings were calculated as proportional constant = 45, integral action time = 120 seconds and derivative action time = 30 seconds.

Figure 6.12: PID tuning for the air temperature control system



With the estimated PID controller settings achieved in the first part of the experiment, the digital control system was tested in the second part of the experiment for sampling time equals to 1 second, 2 seconds, 5 seconds, 8 seconds, 11 seconds, 15 seconds, 20 seconds, 25 seconds, 30 seconds, 35 seconds and 40 seconds. The response curves were shown in Figures 6.13 to 6.23.

Figure 6.13: Air duct air temperature control with sampling rate = 1 s

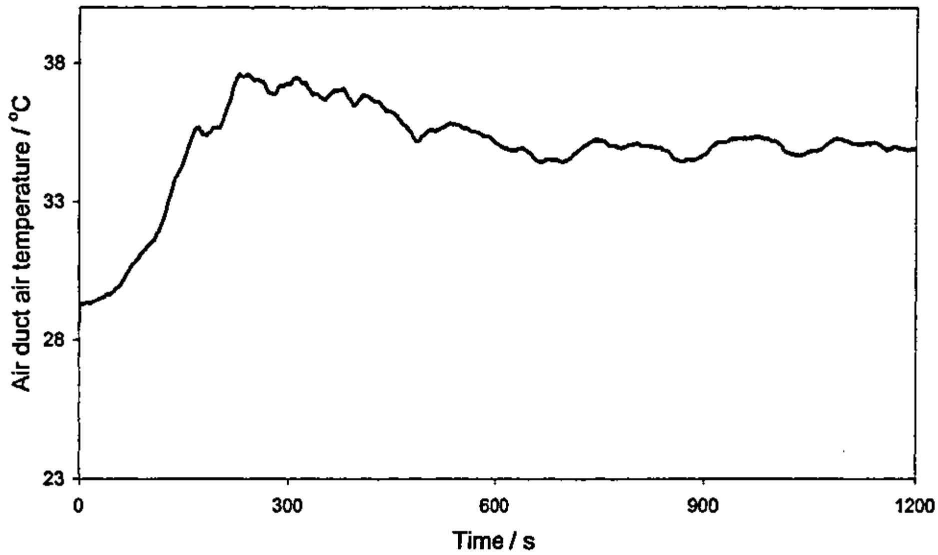


Figure 6.14: Air duct air temperature control with sampling rate = 2 s

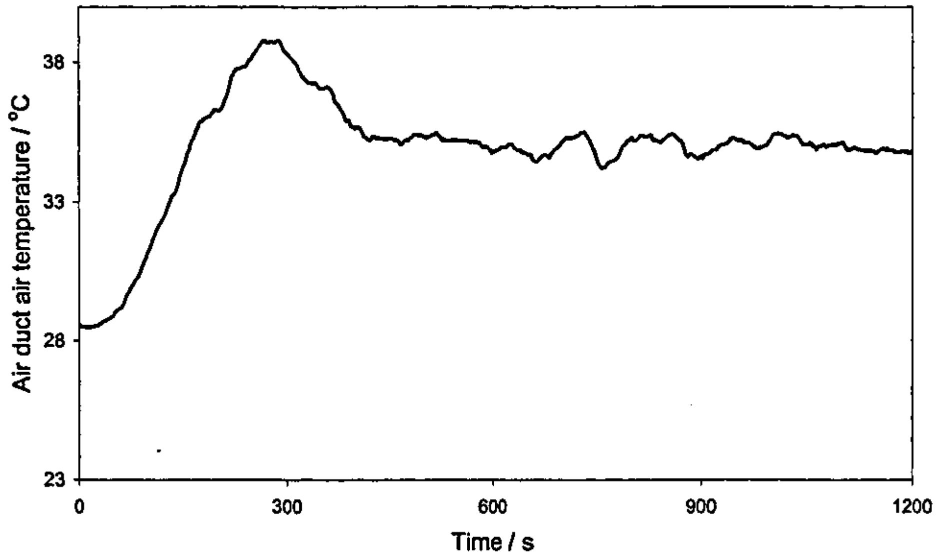


Figure 6.15: Air duct air temperature control with sampling rate = 5 s

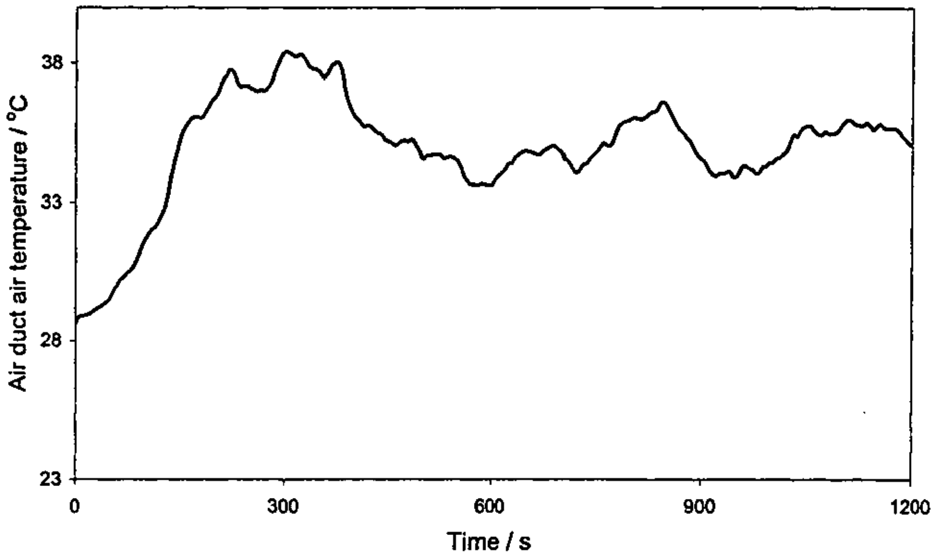


Figure 6.16: Air duct air temperature control with sampling rate = 8 s

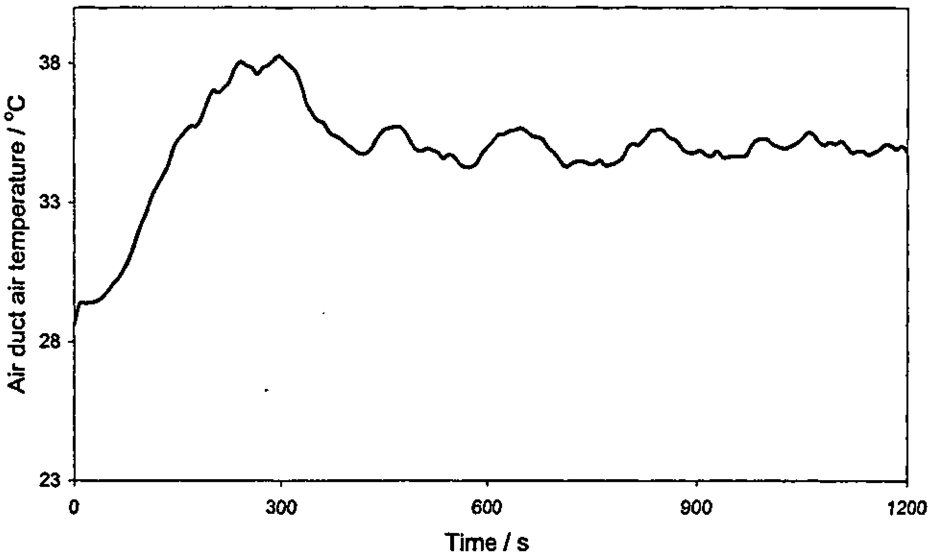


Figure 6.17: Air duct air temperature control with sampling rate = 11 s

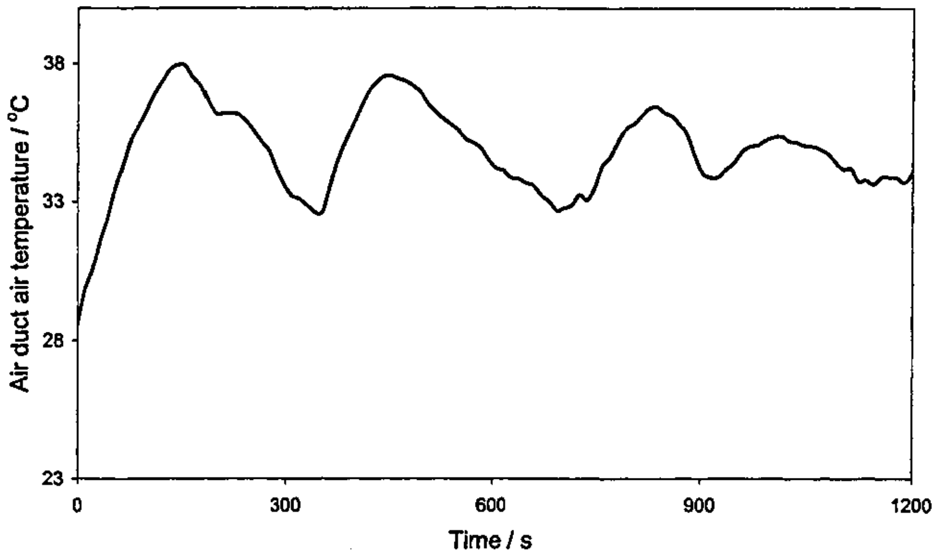


Figure 6.18: Air duct air temperature control with sampling rate = 15 s

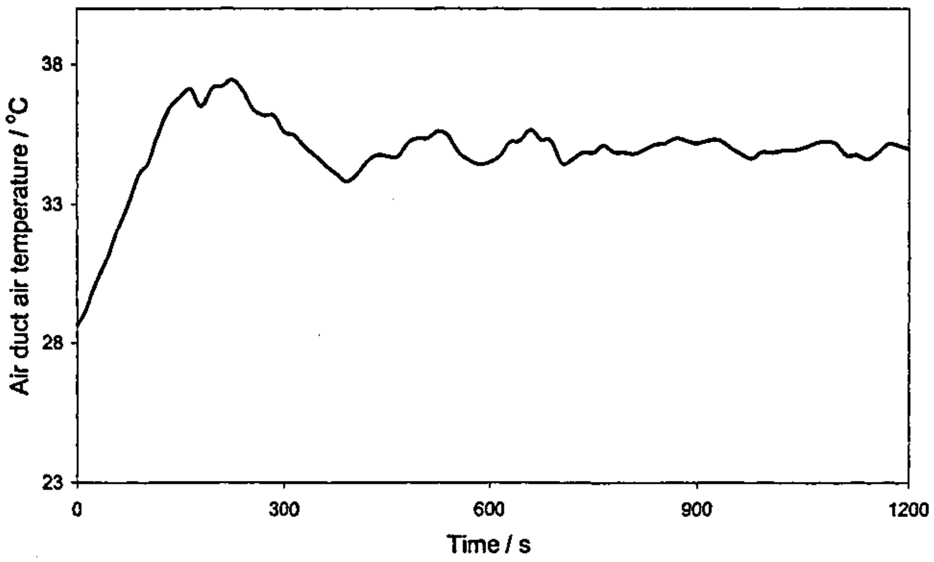


Figure 6.19: Air duct air temperature control with sampling rate = 20 s

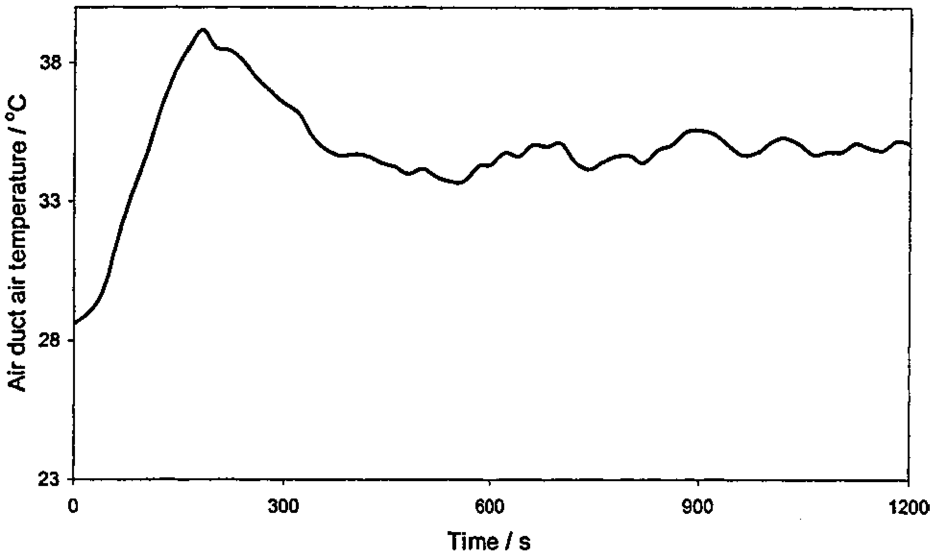


Figure 6.20: Air duct air temperature control with sampling rate = 25 s

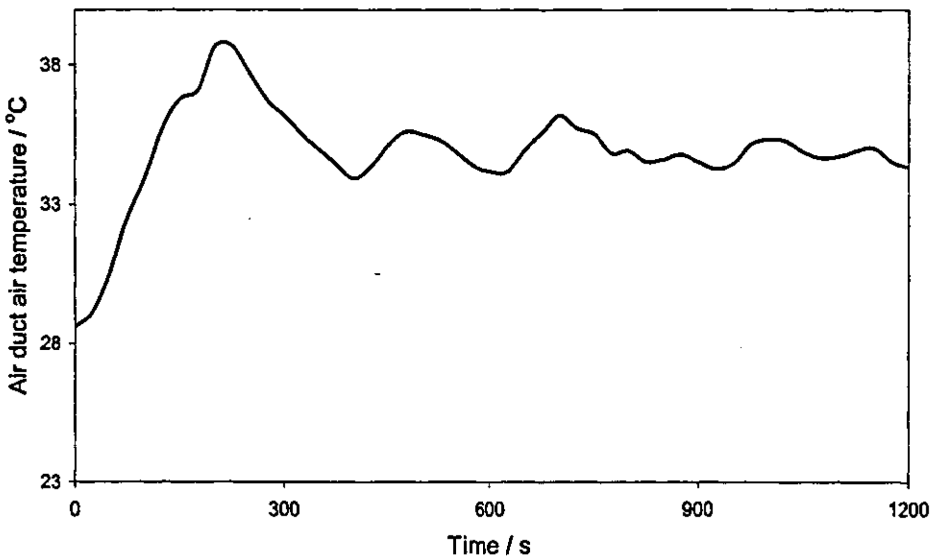


Figure 6.21: Air duct air temperature control with sampling rate = 30 s

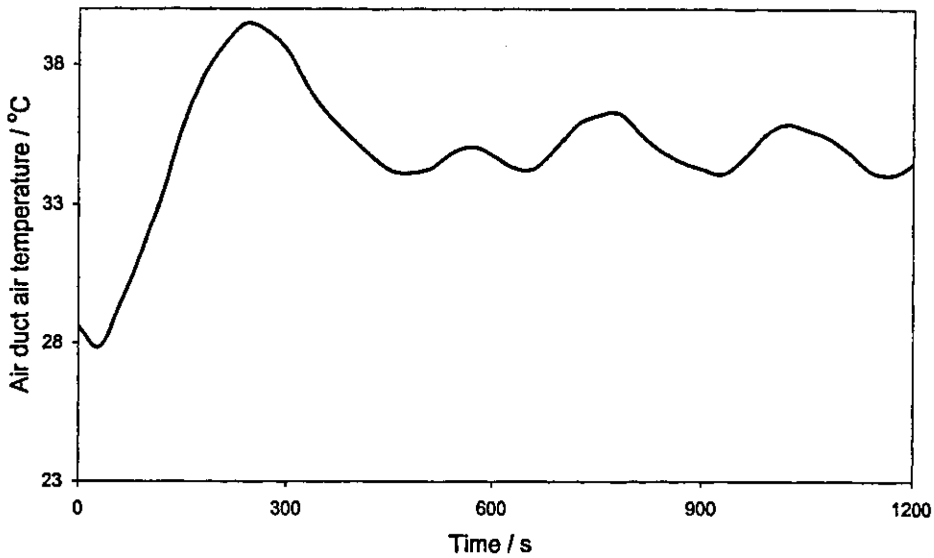


Figure 6.22: Air duct air temperature control with sampling rate = 35 s

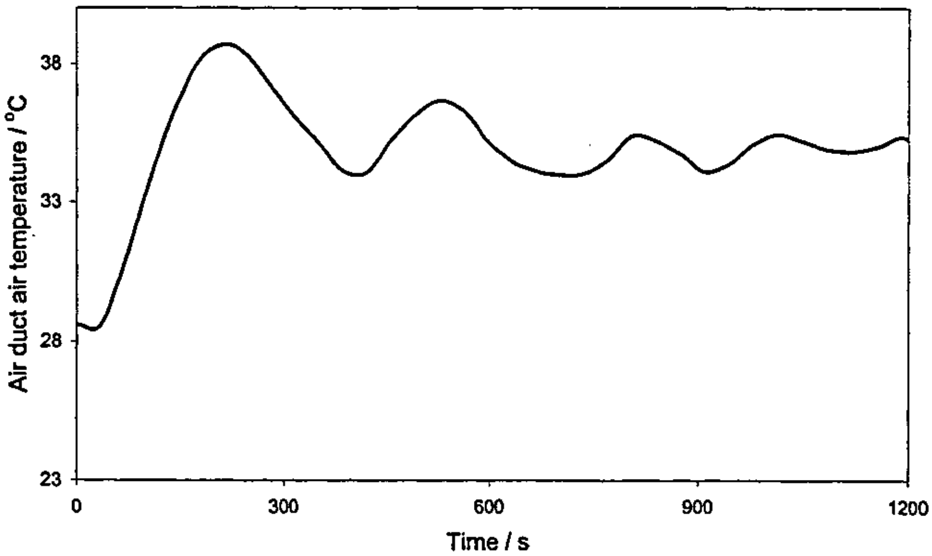
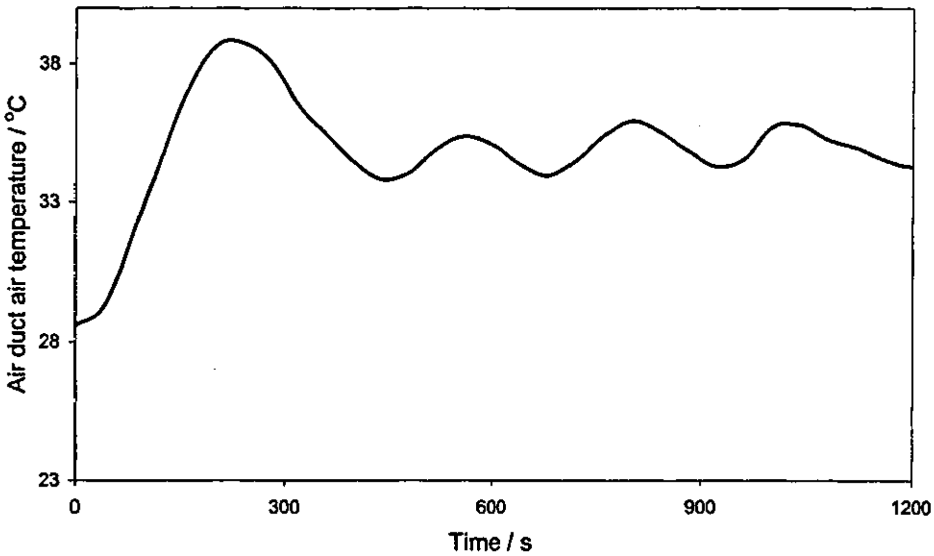


Figure 6.23: Air duct air temperature control with sampling rate = 40 s



The air duct temperature response for different sampling rates is analyzed as the transient response to a step input and characterized by the following[34].

1. Delay time:

   It is the time required for the response to reach half the final value of the very first time

2. Rise time (for overdamped systems):

   It is the time required for the response to rise from 10 to 90 % of its final value.

3. Peak time:

   It is the time for the response to reach the first peak of the overshoot.

4. Maximum (%) overshoot:

$$= \frac{(\text{Maximum overshoot value - final steady state value})}{\text{Final steady state value}} \times 100\%$$

The amount of maximum (%) overshoot directly indicates the relative stability of the control system.

5. Settling time:

It is time required for the response curve to reach and stay within a range about the final value as defined. It is related to the largest time constant of the system. In the experiments, the range about the final value (set point temperature) for settling time calculation is taken as ±½°C.

In addition, the control performance of the control system is also analyzed in term of the objective function mean square error. Other objective functions that based on integrating functions of time and error will not be appropriate as the number of samples taken over the specific period of time will be different for different sampling rates used. This gives irrelevant results for comparison between different sampling rates. From the response curve Figures 6.13 to 6.23, the above criteria for analyzing the response and control performance of the air duct temperature DDC system are tabulated in Table 6.5.

Table 6.5: Experimental results and mean square errors of step responses of an air duct temperature DDC system

Experiment	Sampling rate (s)	Delay time (s)	Rise time (s)	Peak time (s)	Max. % overshoot	Settling time (s)	Mean square error (°C²)
(1)	1	112	141	246	7.08	882	3.3
(2)	2	110	89	274	10.6	776	4.373
(3)	5	104	108	310	9.43	1436	4.127
(4)	8	91	129	296	9.31	850	3.242
(5)	11	35	65	154	8.34	>1200	2.84
(6)	15	57	85	225	6.97	710	2.120
(7)	20	65	70	180	11.94	920	3.572
(8)	25	65	34	200	10.49	951	3.245
(9)	30	100	70	241	12.9	≈1200	5.65
(10)	35	80	60	211	10.4	950	4.313
(11)	40	80	75	222	10.9	>1200	4.427

From the control point of view, the transient response of a control system should be sufficiently fast and sufficiently damped. These two criteria are quite conflicting with each other. A fast response control system is relatively unstable. However, a very stable control system as a result of heavy damping is often very sluggish in response. Hence, how the response terms are compromised with overshoot and settling time depend on the application and engineering judgement. From Table 6.5 test (9) and test (11) are both sluggish in response and have considerable overshoot. Their settling times are very long. Their control performances are marginally acceptable. Test (7), test (8) and test (10) response quite fast, but their overshoots are large. The settling time for these tests is also relatively long. Their responses conflict with their stability. Test (1), test (3) and test (4) are quite slow in response. However, they have small overshoots, except test (3), the settling times are considerable short. Test (2) responses relatively slow and has large overshoot. Nevertheless, the response settles very fast and stable. Test (5) is fast in response and has small overshoot, but the response is fairly oscillatory and takes long time to settle. Test (6) responses fast with minimum overshoot among all the tests. It settles and becomes stable in a very short time. It is considered the best among all in control performance. In terms of the criteria of stability and response rate, the goodness of control performance of the tests are in the order of:

Test (6) > Test (4) > Test (8) > Test (1) > Test (7) > Test (10) > Test (2) > Test (5) > Test (3) > Test (11) > Test (9)

When the objective function "mean square error" is used to assess the control performance, the order is as follow.

Test (6) > Test (5) > Test (4) > Test (8) > Test (1) > Test (7) > Test (3) > Test (10) > Test (2) > Test (11) > Test (9)

This is quite strongly consistent with the control analysis from the temperature responses of the tests.

6.2.2 Selection of sampling rate in relationship with the dynamic responses of the system and temperature sensor

The time constant of the thermister temperature sensor and, the time constant and dead time of the air duct temperature control system were measured at air velocity 0.7 m/s and 1.1 m/s. The air duct velocity was adjusted to the specified velocity by adjusting the fan speed through a variac. The air velocity was measured by a flowhood system with accuracy of +/- 3% reading. The time measurements were tabulated in Table 6.6.

Table 6.6: Time constant of temperature sensor and, time constant and dead time of the air duct temperature DDC system at air velocity 0.7m/s and 1.1 m/s

Air velocity (m/s)	Time constant of the therminster temperature sensor (s)	Time constant of the whole air duct temperature DDC system, T_τ (s)	Dead time of the whole air duct temperature DDC system, T_d (s)
0.7	19	140	45
1.1	20.5	265	86

The frequency of sampling affects the control performance of a HVAC digital control system[50]. According to Leigh[58], decrease in the sampling time may causes loss of information if the frequency of sampling is higher than the dynamics of the feedback system. For practical choice of sampling rate (T), Bennett[59] suggested, for a digital control system, the following.

- $T < T_\tau/10$
- $0.05 < T/T_d < 0.3$
- $T < 0.2\pi/$(the natural frequency)

At air velocity equals to 0.7 m/s, the optimum sampling rate (T) obtained from the above experiment is 15 seconds. From Table 6.6, $T_\tau/10$ = 140/10 seconds = 14 seconds, 0.3 T_d = 13.5 seconds. The time constant of the sensor is 19 seconds. Hence the 'optimum' sampling rate of 15 seconds found seems to satisfy the Bennett's and Leigh's requirements. Similarly, at the air velocity of 1.1 m/s, the `optimum' sampling rate of 30 seconds found from Table 6.9 (for experiment conducted in section 6.4), also satisfies both requirements. The sampling rate of a DDC HVAC control system can be tuned in the design stage by simulation technique for best control performance.

Since the control performance can be measured quantitatively by the objective function mean square error (proven in chapter 7), the tuning of sampling rate in the digital simulation is simply a computation of the mean square error instead of a complicated analysis of control performance criteria.

From the above analysis, it can be concluded that the sampling rate has a significant effect on the control performance of a/c digital control systems. There exists an optimum sampling rate that can achieve the desired control performance. It is not true that the shorter the sampling rate, the better the control performance[50]. The `optimum' sampling rate depends on the desired control performance. The control assessment from the control system transient responses strongly agrees with the use of the objective function mean square. Ho[53] concludes and demonstrates that mean square error is a good control performance index for assessing the control performance of a/c systems. The experimental results should be considered when simulation study on DDC a/c system is conducted.

6.3 THE OBJECTIVE FUNCTION

In the previous studies, mean square error has been proven to be an effective objective function for assessing the control performance of a/c systems. According to the investigation from Ho[9], minimum energy consumption would be an appropriate objective function to be used in a/c systems. Hence, the mean square error used as objective function in the current study of the effect of sampling rate on control performance is taken as an example to compare with that using the energy consumption. The purpose of such comparison is to prove the usefulness of mean square error in thermal studies. In the experiments conducted in 6.3, a power meter was used to measure the power consumption for a period of 1200 seconds for each experiment. The power consumption and the mean square error for the experiments for different sampling rates are tabulated in Table 6.7.

Table 6.7: Comparison of mean square errors and energy consumption of step responses of an air duct temperature control system

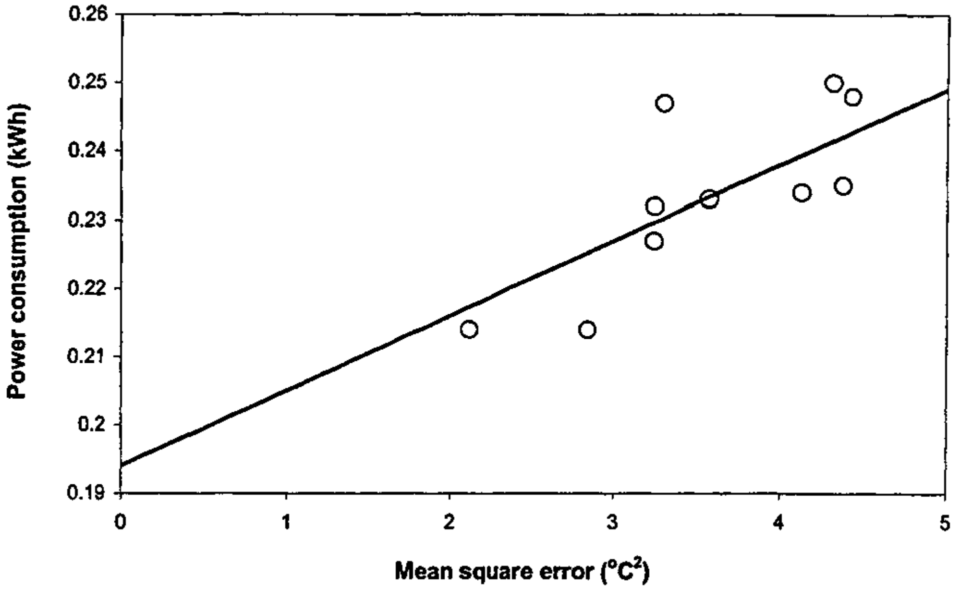
Experiment	Test for sampling rate (s)	Mean square error ($^\circ C^2$)	Power consumption (kWh)
(1)	1	3.3	0.247
(2)	2	4.373	0.235
(3)	5	4.127	0.234
(4)	8	3.242	0.227
(5)	11	2.84	0.214
(6)	15	2.120	0.214
(7)	20	3.572	0.233
(8)	25	3.245	0.232
(9)	30	5.65	0.250
(10)	35	4.313	0.250
(11)	40	4.427	0.248

From Table 6.7, the control performance of the air duct temperature control system when assessed by the mean square error is in the order of: Test (6) > Test (5) > Test (4) > Test (8) > Test (1) > Test (7) > Test (3) > Test (10) > Test (2) > Test (11) > Test (9). The mean square error is plotted against the power consumption for each sampling time and is shown in Figure 6.24. Based on the figure, the following equation is derived.

Power consumption = 0.194 + 0.011 x (Mean square error)

The coefficient of correlation, R, is 0.7905 which shows that the mean square error is related quite closely to the power consumption. However, the relationship only works on a narrow range of power consumption (the maximum power consumption is only 16.3% higher than the minimum power consumption) for a rather large mean range (the maximum is 167% higher than the minimum mean square error).

Figure 6.24: The mean square error against the energy consumption for each sampling time



6.4 THE CONTROLLER SETTINGS

The effect of controller settings on the control performance of a/c systems was examined experimentally by employing the same air duct temperature control system as used in **6.3**. In the experiments, the operation condition of the air duct temperature control rig was changed with new air velocity set at 1.1 m/s. For comparison, the previous experiment of optimum sampling rate was repeated at the new air velocity with the same estimated controller settings. The comparison is given in Table 6.8. The estimated controller settings for the new air velocity was then found by the same method as in the previous experiment. With the controller settings found, the previous set of experiments was repeated and the objective functions were computed. The results are given in Table 6.9.

Table 6.8: Values of objective function of the air duct temperature DDC system at air velocities of 0.7 m/s and 1.1 m/s at sampling rate = 15 s and controller settings P = 54, I = 120 s and D = 30 s

	Mean square error (° C^2)	Energy consumption (kWh)
Air duct temperature control with air velocity = 0.7 m/s at optimum sampling rate with estimated controller settings P = 54, I = 120 s and D = 30 s.	2.12	0.218
Air duct temperature control with new air velocity = 1.1 m/s at sampling rate with controller settings, P = 54, I = 120 s and D = 30 s.	3.39	0.252

Table. 6.9: Values of objective function of the air duct temperature DDC system at air velocity=1.1 m/s and with Nichols and Ziegler estimated controller settings P= 51, I= 120 s and D= 30 s

Experiment with sampling rate set at	Mean square error (° C^2)	Energy consumption (kWh)
1s	2.747	0.277
2s	2.771	0.252
5s	3.793	0.247
8s	3.801	0.243
11s	3.24	0.248
15s	3.64	0.244
20s	2.99	0.26
25s	5.367	0.283
30s	2.723	0.229
35s	4.05	0.24
40s	5.161	0.236

From Table 6.9, at air velocity 1.1 m/s, the optimum sampling rate is 30 seconds as the mean square error is minimum. So it can be seen that the optimum sampling rate changes with change in the air velocity. Also from Tables 6.8 and 6.9, experiment carried at the new air velocity 1.1 m/s but using the same Nichols and Ziegler estimated controller settings for air velocity 0.7 m/s does not give the minimum mean square error. There is a new Nichols and Ziegler estimated set of controller settings for new air velocity or new operation conditions. Hence, it can be concluded that:

- There is a set of estimated controller settings for each operation condition, i.e. controller settings affect the control performance of an a/c system; for new operation condition, estimated controller settings must be found and used to achieve the best control performance, and

- There is change in the optimum sampling rate when the operation condition changes.

Also from Table 6.9, the mean square error increases with the sampling time from 1 second to 8 seconds. For sampling times higher than 8 seconds, the mean square error oscillates with the sampling tome. Although it is stated in section 6.22 that it is not true that the shorter the sampling time, the better the control, it is obviously that the control is better guaranteed in the short sampling time range.

6.5 CONCLUSIONS

The experimental study of control valve shows that electric valve has better control performance than pneumatic valve due to better repeatability, close-off rating and control response. It also reveals that proper sizing of control valve is important in achieving good control.

The experiments on the DDC air duct system indicate that there exists an optimum sampling time on the control performance of the DDC a/c system under each of the operation condition. It is not true that the shorter the sampling time, the better the control. This may due to the information loss effect. An example is that a sensor for DDC system may need two seconds to construct a signal accurately from analogue to digital form whilst the controller is sampling at one second; hence information is lost between sampling intervals. Also noises or transients may throttle the control of a DDC system and cause energy waste if a very short sampling time is used. On the other hand, a reasonable longer sampling time may smooth out the transient effect provided that the Shannon's sampling theorem[50] is obeyed. However, control is better guaranteed in the short sampling time range.

CHAPTER 7 DETAILED SIMULATION STUDY FOR OPTIMUM CONTROL PERFORMANCE OF A PRACTICAL SINGLE ZONE CAV A/C SYSTEM

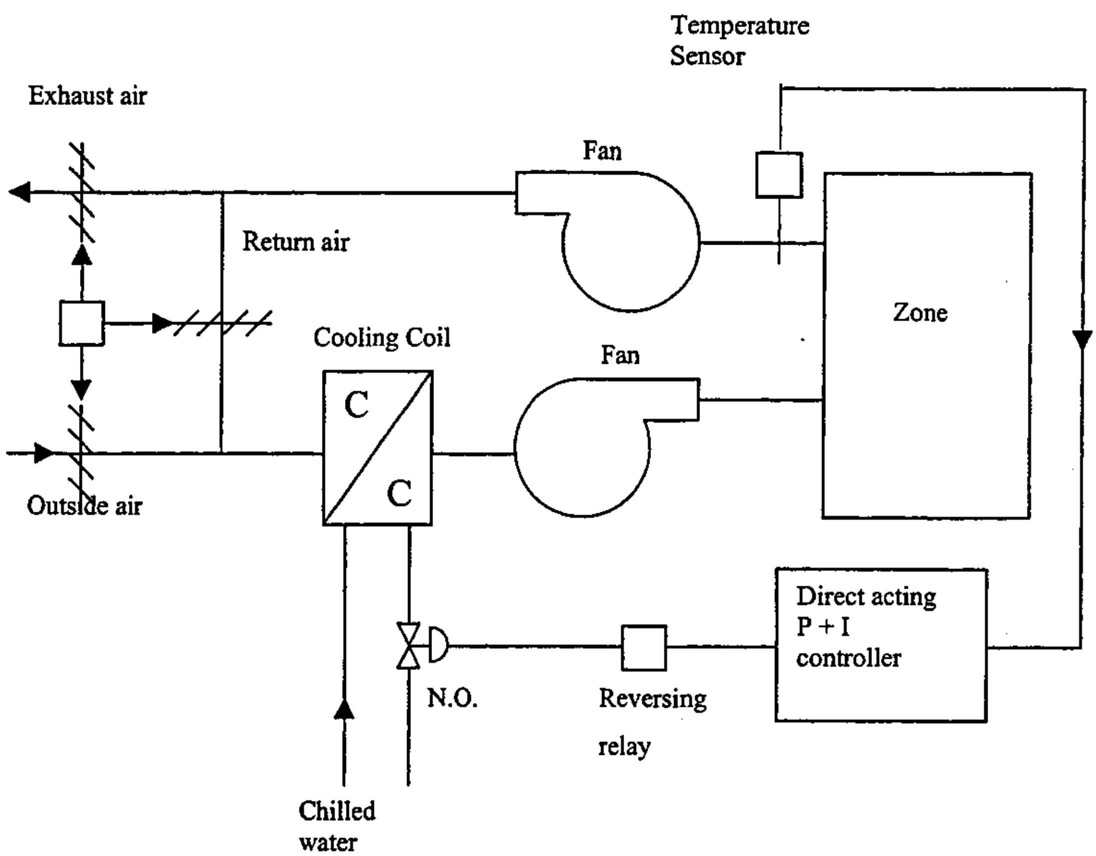
In chapter 5, a PAHU was used to study the concept of optimization of a/c system using the simulation program HVACSIM+. The simulation study indicated that there existed good and bad cooling coil designs for the simple a/c sub-system. The control performances, in terms of system stability and response, of different designs were evaluated against different conventional error based objective functions. It is found that the performance evaluation using mean square error is generally in good agreement with those using normal control considerations. Hence, the mean square error is feasible in use for optimization of simple a/c sub-system.

However, the preliminary study only aims for a conceptual study in using simulation technique for optimization of a/c system design. The simulation is based only on a simple a/c cooling sub-system and only the Nichols and Ziegler controller tuning method[15] was used. For a complete optimization study of an a/c system design, the interaction between the a/c system with the a/c controlled zone space(s) and, the impact of controller tuning method used have to be addressed. The present simulation aims for a detailed study for optimization a/c cooling system design for applications in Hong Kong. To achieve a simple but complete study, the detailed simulation is extended to investigate the control performance of a single zone CAV (constant air volume) a/c cooling system under different controller tuning methods. Conventional error based objective functions integrating with some of the controller tuning methods under study as well as energy consumption are also investigated for use as objective function in a/c system simulation. The detailed simulation aims to identify and conclude the approach in the use of simulation and its requirements in optimizing the control performance of a single zone a/c system.

7.1 THE SYSTEM

The a/c system under study is a typical single zone CAV cooling system, which is commonly found in Hong Kong a/c installations. The system is shown in Figure 7.1. From the figure, there is an intake of fresh air through the outside air damper. The amount of fresh air admitted is determined by the ventilation reqirement of the zone. The fresh air mixes with the return air from the zone. The mixed air passes through the cooling coil to supply conditioned air to the zone. The amount of air exhausted from the system equals to the volume of fresh air admitted. The corresponding air dampers control all the volumes of the fresh air, return air and exhausted air. The zone return air temperature, which is a measure of cooling load variation in the zone, is measured and used to control the chilled water flow through the cooling coil. The set point temperature of the zone return air is 24°C. Proportional + integral control is used for the reason as stated in chapter 5.

Figure 7.1 Schematic diagram of CAV single zone control system



7.2 THE EFFECT OF CONTROL VALVE CHARACTERISTICS ON THE SYSTEM CONTROL PERFORMANCE

The dynamic response of the system is affected by the time constants of the components in the control systems. From the experimental study in chapter 6, the time constant of control valve actuator contributes a large significant ratio in the overall control system time constant and hence will greatly affect the system control performance. Since equivalent pneumatic and electric control valve actuators have significant different time constants, the use of pneumatic and electric dives will affect differently the system control performance. The valve resistance also determines the controllability of the control valve as this changes the control valve authority. These factors will be investigated in the following simulation study.

7.3 THE CONTROLLER TUNING METHODS

The study in chapter 5 shown the importance of controller tuning in optimizing the control performance of the cooling coils design. Nichols and Ziegler tuning rule was used as the tuning method for the controller setting in the simulation study. However, there are some other researchers who investigated various controller tuning estimates based also on the open loop process-reactive methods. Cohen and Coon [(60)] considered the dead-time effects and arrived alternative controller setting estimate. Lopez et al.[(61)] estimated controller settings with objective in minimizing various conventional error-integral criteria. Hence, analysis in the control response when Lopez et al estimates are used in the controller settings also evaluates various error-integral criteria for optimization of a/c system. These estimates were given in Ref.(50).

In the present detail simulation study, these controller setting estimate approaches were investigated and compared against their control responses.

7.4 THE OBJECTIVE FUNCTION FOR OPTIMIZATION STUDY

In the preliminary optimization study of an a/c cooling sub-system in chapter 5, the mean square error is found and proven to be a good and satisfactory objective function in assessing the control performance of the a/c cooling sub-system. In the present detailed simulation study, the control performance of a single zone CAV a/c cooling system when various open loop process-reactive controller tuning estimates are used are compared. This includes the Lopez et al. controller setting estimates that seek to minimize various conventional error based objective functions. The energy consumption for each controller tuning estimate used is computed such as to compare the effectiveness of different tuning methods together with the control performance. In this way, the use of energy consumption and various error based objective functions for optimizing a/c system control performance can be evaluated. The comparison aims to identify the best objective function to be used in optimizing of a/c cooling system.

7.5 DETAILED SIMULATION STUDY FOR CONTROL PERFORMANCE OF A PRACTICAL SINGLE ZONE CAV A/C COOLING SYSTEM

7.5.1 The application of the data collected in the a/c survey in chapter 3 for detailed simulation study for optimization of the practical a/c system

The preliminary simulation study in chapter 5 deals with the optimization conceptual issue for a simple a/c cooling sub-system that only treats the outside air. A complete and useful study of the air side a/c cannot be achieved if the interaction of the a/c systems with air-conditioned zones is overlooked. Hence in optimizing a/c design, the performance of the a/c system must be analyzed together with the response of the air-conditioned zone(s). The detailed simulation mentioned in this chapter aims to optimize a/c system design by control performance.

The a/c system simulated in this detailed simulation is a single zone CAV cooling and control system. The zone return air temperature is used to regulate the chilled water flow through the cooling coil to maintain the room temperature at the setpoint temperature of 24°C. This control strategy for single zone CAV is found most common from the a/c survey in chapter 3. The control response of the zone return air temperature is used as the criterion for optimizing the CAV cooling and control system design. The data collected from the survey in chapter 3 is utilized as described in the following. The zone to be simulated is a typical zone found in the survey. The zone has a floor area of 211 m^2 and an internal volume of 630 m^3. The fresh air requirement and the total air flow for the zone are calculated by applying the population density of 8 m^2/person. The relevant ventilation, internal and external loads are calculated with reference to the design conditions. The type 15 zone model in the HVACSIM+ is found appropriate to simulate the typical zone and thus is used in the simulation model. The requirements of physical parameters of the model are satisfied with the survey data. The type 12 cooling coil model is employed to simulate the cooling coil in the simulation. The model matches with the typical CAV cooling coil in characteristics and hence physical data and parameters of the typical cooling coil obtained in the survey are used in the model. As the coil model does not provide information of the air resistance drop across the coil, a type 5 damper model is used to give the resistance drop. Similarly, the air filter is modeled by a type 5 damper model, as the HVACSIM+ does not have filter model. The fan models in the simulation model are input with data collected. In the control loop, the accuracy and realistic of the control valve model and temperature sensor model are supported and improved with the experimental results from chapter 6. The control valve authority and time constant which represents the actuator drive are the variables to be changed to investigate the control performance of different coil sizes. The controller setting from various tuning methods other than Nichols and Ziegler is tested against different coil sizes for control performance. The initial load is approximately 50 %. The major physical data and parameters of models, design and initial conditions are given as following. The details of these data are summarized in Table 7.1.

Outdoor conditions: 33°C db, 28°C wb

Indoor conditions: 24°C db, 50% RH

Zone:

External load: 14.576 kW

Internal load:	12.505 kW
Ventilation load:	12.739 kW
Room volume:	630 m^3
Room area:	211 m^2
Population density:	8 m^2/person
Fresh air requirement:	10 l/s per person

CAV Cooling coil (with direct outside air supply):

Coil rating:	40 kW
Tube/fin material:	Copper/Aluminum
Sensible load:	24 kW
Total load :	40 kW
On coil temperature:	25.3°C, 0.19 kg/kg absolute humidity
Off coil temperature:	21.2°C, 0.0091 kg/kg absolute humidity
Entering water temperature:	7 °C
Leaving water temperature:	16.32 °C
Chilled water flow:	1.088 kg/s
Water pressure drop:	5.4 kPa
Air pressure drop:	35 kPa
Working pressure:	depends on the static water head
Air coil face velocity:	1.88 m/s
Supply air flow:	2 m^3/s
Motor rating:	1.33 k W

Fan:

Diameter:	319 mm

Damper:

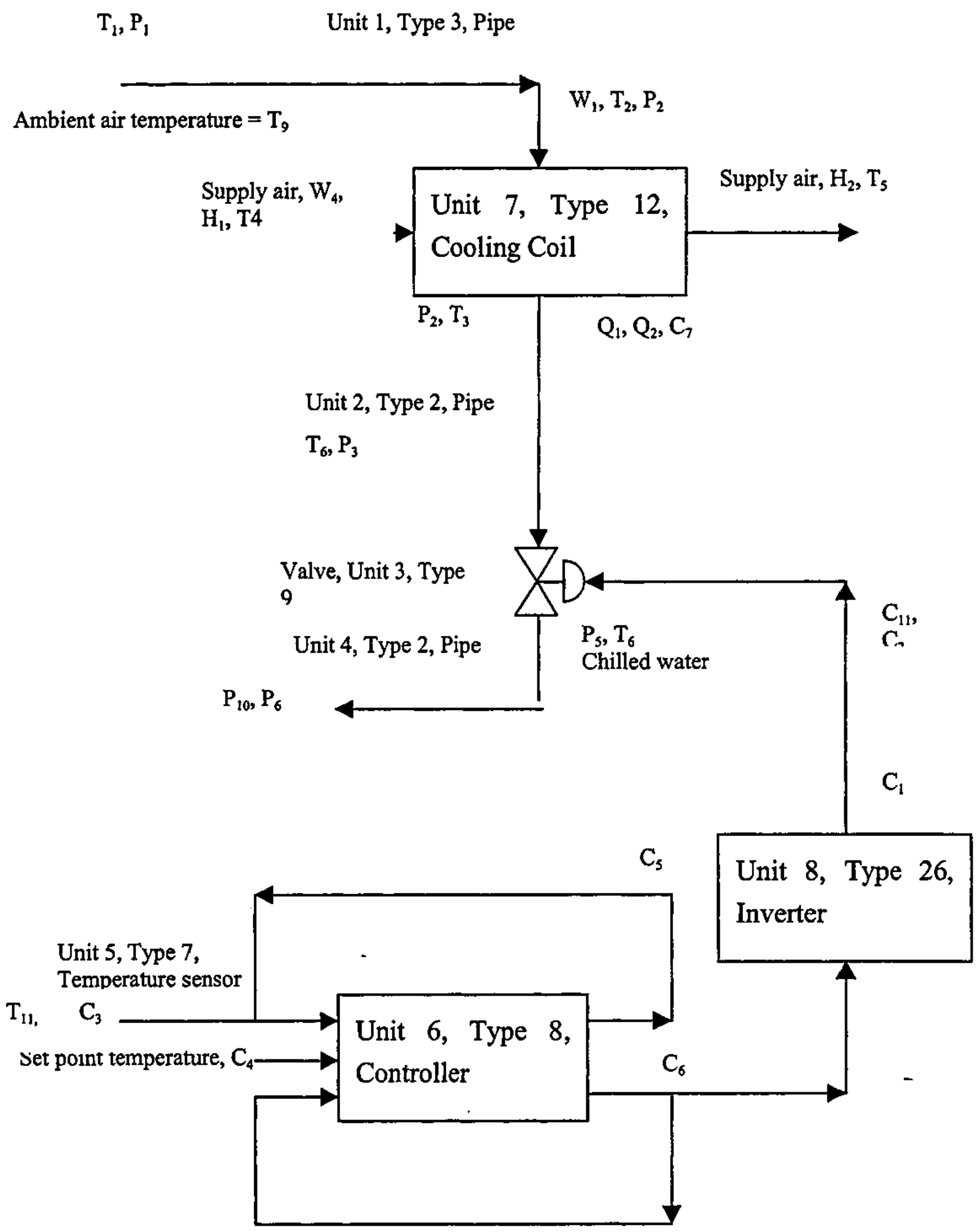
Flow resistance:	1×10^{-6} kg^{-1} m^{-1} (for outside, return and exhaust air dampers)
	4×10^{-6} kg^{-1} m^{-1} (for air filter)
	1.2×10^{-6} kg^{-1} m^{-1} (for cooling coil)

7.5.2 The simulation

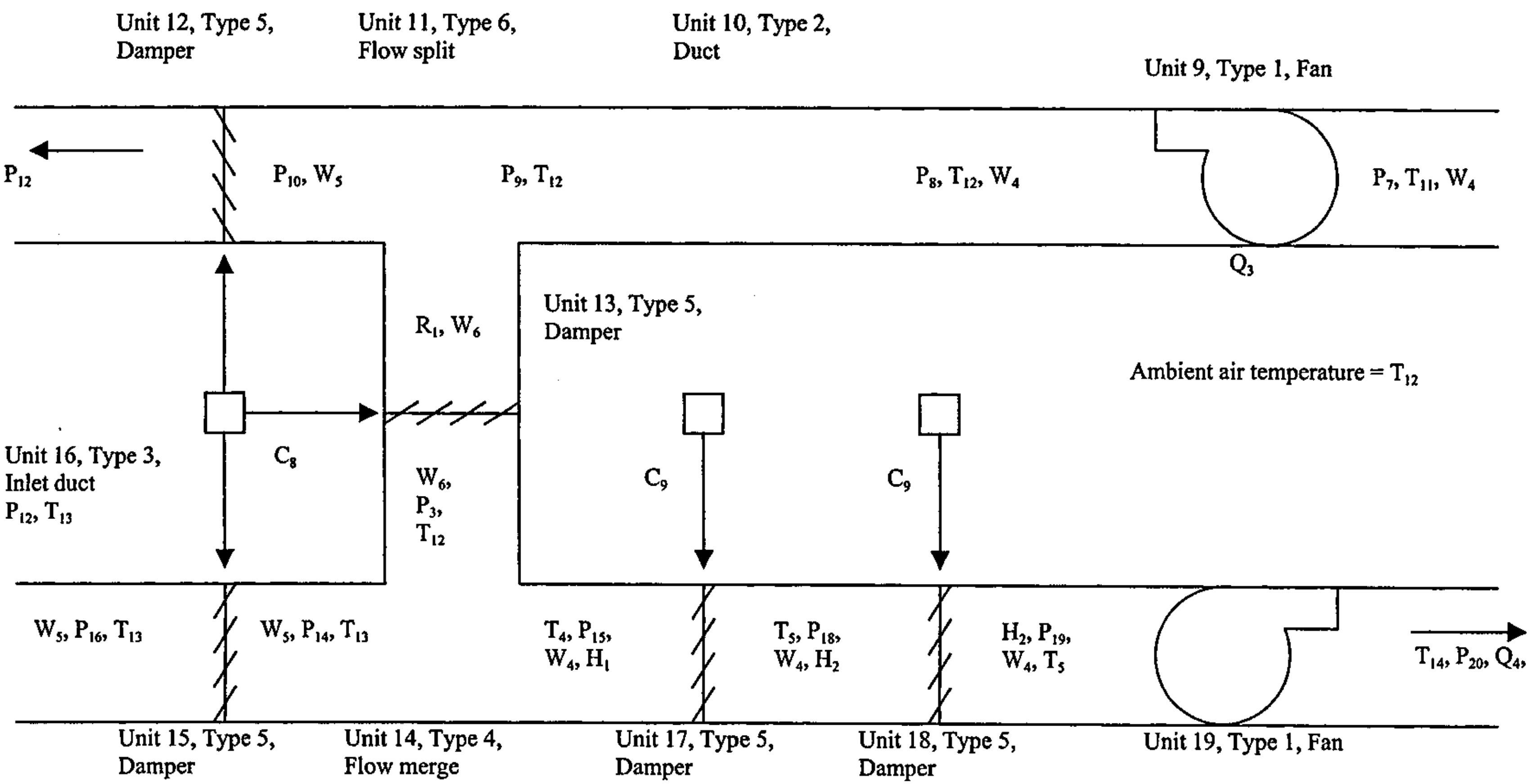
In the simulation, the same five practical coil sizes as used in chapter 5 were tested for controllability against the following five controller tuning estimates, i.e. Nichols & Ziegler, Lopez et al. integral absolute error, Lopez et al. integral error square, Lopez et al. integral absolute error times time, and Cohen & Coon. The design conditions of the system are given below and the system configuration of the a/c system is shown in Figure 7.2.

Figure 7.2 Schematic for single zone CAV control system with components and types shown

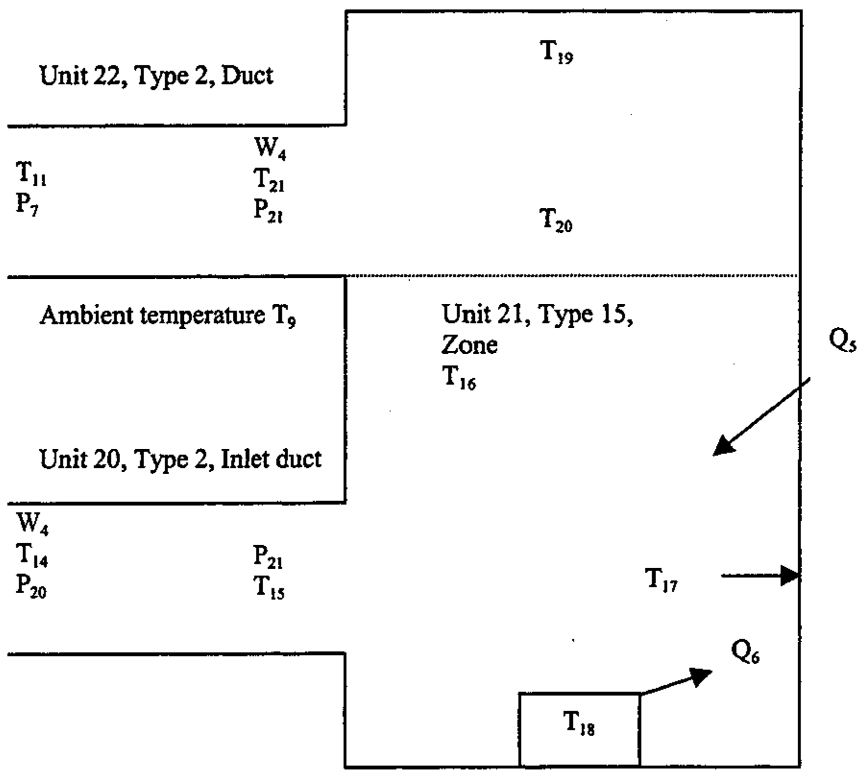
Block 1:



Block 2:


Unit 12, Type 5, Damper
Unit 11, Type 6, Flow split
Unit 10, Type 2, Duct
Unit 9, Type 1, Fan
P12
P10, W5
P9, T12
P8, T12, W4
P7, T11, W4
Q3
R1, W6
Unit 13, Type 5, Damper
Ambient air temperature = T12
Unit 16, Type 3, Inlet duct P12, T13
C8
W6, P3, T12
C9
C9
W5, P16, T13
W5, P14, T13
T4, P15, W4, H1
T5, P18, W4, H2
H2, P19, W4, T5
T14, P20, Q4,
Unit 15, Type 5, Damper
Unit 14, Type 4, Flow merge
Unit 17, Type 5, Damper
Unit 18, Type 5, Damper
Unit 19, Type 1, Fan

Block 3:



Unit 22, Type 2, Duct
T_{19}
T_{11}
P_7
W_4
T_{21}
P_{21}
T_{20}
Ambient temperature T_9
Unit 21, Type 15,
Zone
T_{16}
Q_5
Unit 20, Type 2, Inlet duct
W_4
T_{14}
P_{20}
P_{21}
T_{15}
T_{17}
Q_6
T_{18}

In the simulation, twenty two components are identified for the system. They are represented by unit 1 to unit 22 as indicated in Figure 7.2 and consequently programmed (with the specific types) in the simulation. The configuration of the system components is summarized in Table 7.1. The initial values of state and boundary variables of the system components are typical of the cooling coil control systems used in the Hong Kong and are also given in Table 7.1. A time period of 50000 seconds of zero internal occupant load condition was allowed for steady state starting conditions. The simulation was tested against a step increase of internal occupant cooling load from zero kW to 12.505 kW at time 50000 seconds for an eight hour working period for each of the five cooling coil designs under test as listed in Table 7.2.

The P+I controller settings achieved from the five tuning methods for control valve resistances equal to 0.15 kg^{-1} m^{-1} and 0.03 kg^{-1} m^{-1} are also given in Table 7.3 and Table 7.4 respectively.

Table 7.1 Simulation configuration of Superblock 1, 2, 3 & initial conditions

Unit No.	Type	Input Variable	Output Variable	Parameters	
Unit 1	Type 3 Inlet Pipe	P1 - Inlet water pressure P2 - Outlet water pressure T1 - Inlet water temperature T9 - ambient air temperature T2 - Outlet water temperature (same as first output)	T2 - Outlet water temperature (same as fifth output) W1 - Water mass flow rate	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume(m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	2.97 0.0031 2.277 0.00251 21.36 -0.5588 2 (water)
Unit 2	Type 2 Outlet Pipe	W1 - Water mass flow rate P3 - Outlet pressure T3 - Inlet pressure T9 - Ambient temperature T6 - Outlet temperature	T6 - Outlet temperature P2 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume(m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	2.7 0.0281 2.07 0.002278 11.9 0 2 (water)
Unit 3	Type 9 Linear Valve with Pneumatic Actuator	P5 - Pressure at outlet W1 - Water mass flow rate C1 - Input control signal C2 - Actuator relative position (same as first output)	C2 - Actuator relative position (same as fourth output) P3 - Pressure at inlet C11 - Valve stem relative position	Flow resistance [0.001/(kg m)] Actuator time constant (sec.) Leakage parameter Hysteresis parameter	V_r A_T 0.005 0.58
Unit 4	Type 2 Outlet Pipe	W1 - Water mass flow rate P6 - Outlet pressure T6 - Inlet water temperature T9 - Ambient temperature T10 - Outlet water temperature (same as first output)	T10 - Outlet water temperature (same as fifth input) P5 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of pipe material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	2.736 0.03168 23.13 0.02556 47.8368 0 2 (water)
Unit 5	Type 7 Temperature Sensor	T11 - Temperature input C3 - Temperature sensor (same as first output)	C3 - Temperature sensor output (same as second input)	Temperature sensor time constant (sec.) Offset (°C) Gain	20 0 100

Unit 6	Type 8 P + I Controller	C3 - Controller variable C4 - Setpoint for controlled variable C5 - Integral portion of control signal (same as first output) C6 - Output control signal (same as second output)	C5 - Integral portion of control signal (same as third input) C6 - Output control signal (same as fourth input)	Proportional gain Integral gain Controller time constant (sec.)	0.25411 0.001122 1
Unit 7	Type 12 Cooling coil	W1 - Water mass flow rate T2 - Inlet water temperature W4 - Dry air mass flow rate T4 - Inlet dry bulb air temperature H1 - Inlet air humidity ratio T3 - Outlet water temperature (same as first input) T5 - Outlet air dry bulb temperature (same as second output) H2 - Outlet air humidity ratio (same as third output)	T3 - Outlet water temperature (same as sixth input) T5 - Outlet dry bulb temperature (same as seventh input) H2 - Outlet air humidity ratio (same as eighth input) Q1 - Total cooling load Q2 - Sensible cooling load C7 - Wet fraction of coil surface area	Coil type Primary (tube exterior) surface area (m^2) Secondary (fin) surface area (m^2) Internal surface area (m^2) Ratio of minimum air flow area to face area Fin material thermal conductivity [kW/(m K)] Coil face area (m^2) Number of fins per centimeter Number of tubes per row Number of row Outside tube diameter (m) Inside tube diameter (m) Fin thickness (m) Thermal capacitance (mass x specific heat) of coil (kJ/°C) Tube row spacing in air flow direction (m) Fin diameter (m) Coil depth in air flow direction (m) Tube thermal conductivity [kW/(m K)]	0 (flat) A_p A_s A_i 0.562 0.204 0.7726 2.3622 16 6 0.01588 0.01445 0.00015 34.669 0.0259 0.508 0.1651 0.386
Unit 8	Type 26 Reversing Relay (Inverter)	C6 - Input control signal	C1 - Output control signal		1

Unit 9	Type 1 Fan	W4 - Air mass flow rate P8 - Outlet pressure R1 - Fan rotational speed T11 - Inlet air temperature	P7 - Inlet presure T12 - Outlet air temperature Q3 - Power consumption	1st pressure coefficient 2nd pressure coefficient 3rd pressure coefficient 4th pressure coefficient 5th pressure coefficient 1st efficiency coefficient 2nd efficiency coefficient 3rd efficiency coefficient 4th efficiency coefficient 5th efficiency coefficient Diameter (M) Mode	3.64 0.801 -0.19 -0.00445 0 0 0.564 -0.0862 0 0 0.3192 1 (air)
Unit 10	Type 2 Duct	W4 - Air mass flow rate P9 - Outlet pressure T12 - Inlet pressure T12 - Ambient temperature T12 - Outlet temperature (same as first output)	T0 - Outlet temperature (same as fifth output) P8 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of duct material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	0 0 0 0 0.01 0 1 (air)
Unit 11	Type 6 Flow Split	W4 - Inlet air mass flow rate P10 - Outlet pressure 1 P11 - Outlet pressure 2	W0 - Outlet mass flow rate 1 W6 - Outlet mass flow rate 2 P9 - Inlet pressure	Flow resistance [0.001/(kg m)]	0.011
Unit 12	Type 5 Damper	W5 - Air mass flow rate P12 - Outlet pressure C8 - Relative position of damper	P10 - Inlet pressure	Flow resistance, damper open [0.001/(kg m)] Leakage parameter (Dimensionless) Characteristic Mode (0 - closed when control = 0; 1 - closed when control = 1)	0.001 0.012 0.87 0
Unit 13	Type 5 Damper	W6 - Air mass flow rate P13 - Outlet pressure C8 - Relative position of damper	P11 - Inlet pressure	Flow resistance, damper open [0.001/(kg m)] Leakage parameter (Dimensionless) Characteristic Mode (0 - closed when control = 0; 1 - closed when control = 1)	0.001 0.012 0.87 1

Unit 14	Type 4 Flow merge	W6 - Inlet mass flow rate 1 W5 - Inlet mass flow rate 2 P15 - Outlet pressure T12 - Inlet temperature 1 T13 - Inlet temperature 2	W4 - Outlet mass flow rate P13 - Inlet pressure 1 P14 - Inlet pressure 2 T4 - Outlet temperature	Flow resistance [0.001/(kg m)]	0.011
Unit 15	Type 5 Damper	W5 - Air mass flow rate P14 - Outlet pressure C8 - Relative position of damper	P16 - Inlet pressure	Flow resistance, damper open [0.001/(kg m)] Leakage parameter (Dimensionless) Characteristic Mode (0 - closed when control = 0; 1 - closed when control = 1)	0.001 0.012 0.87 0
Unit 16	Type 3 Inlet duct	P12 - Inlet pressure P16 - Outlet pressure T13 - Inlet air temperature T13 - ambient air temperature T13 - Outlet air temperature (same as first output)	T0 - Outlet air temperature (same as fifth input) W5 - Air mass flow rate	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of duct material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	0 0 0 0 0.01 0 1 (air)
Unit 17	Type 5 Damper	W4 - Air mass flow rate P18 - Outlet pressure C9 - Relative position of damper	P15 - Inlet pressure	Flow resistance, damper open [0.001/(kg m)] Leakage parameter (Dimensionless) Characteristic Mode (0 - closed when control = 0; 1 - closed when control = 1)	0.04 0.00004 1 0
Unit 18	Type 5 Damper	W4 - Air mass flow rate P19 - Outlet pressure C9 - Relative position of damper	P18 - Inlet pressure	Flow resistance, damper open [0.001/(kg m)] Leakage parameter (Dimensionless) Characteristic Mode (0 - closed when control = 0; 1 - closed when control = 1)	0.012 0.000012 1 0

Unit 19	Type 1	W4 - Air mass flow rate	P19 - Inlet presure	1st pressure coefficient	3.64
		P20 - Outlet pressure	T14 - Outlet air temperature	2nd pressure coefficient	0.801
	Fan	R2 - Fan rotational speed	Q4 - Power consumption	3rd pressure coefficient	-0.19
		T5 - Inlet air temperature		4th pressure coefficient	-0.00445
				5th pressure coefficient	0
				1st efficiency coefficient	0
				2nd efficiency coefficient	0.564
				3rd efficiency coefficient	-0.0862
				4th efficiency coefficient	0
				5th efficiency coefficient	0
				Diameter (m)	0.3192
				Mode	1 (air)

Unit 20	Type 2 Inlet Duct	W4 - Air mass flow rate P21 - Outlet pressure T14 - Inlet pressure T9 - Ambient temperature T15 - Outlet air temperature (same as first output)	T15 - Outlet air temperature (same as fifth output) P20 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of duct material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	0 0 0 0 0.02 0 1 (air)
Unit 21	Type 15 Zone	W4 - Air mass flow rate T15 - Inlet air temperature T16 - Temperature, fully mixed part of air mass (same as first output) T17 - Wall mass temperature (same as second output) T18 - Interior mass temperature (same as third output) T19 - Spatial avg. temp., piston flow portion of air mass (same as fourth output) Q5 - Conduction heat flow into wall mass Q6 - Heat flow due to internal gains	T16 - Temperature of fully mixed portion of air mass T17 - Wall mass temperature T18 - Interior mass temperature T19 - Spatial avg. temp., piston flow portion of air mass T20 - Average room air temperature T21 - Exhaust air temperature	Volume of room air mass (m^3) Thermal capacitance of walls (kJ/°C) Thermal capacitance of interior mass (kJ/°C) Heat transfer coefficient x area for wall mass (kW/°C) Heat transfer coefficient x area for interior mass (kW/°C) Fraction of air mass which is fully mixed	630 17500 1750 0.4375 0.0875 0.9
Unit 22	Type 2 Duct	W4 - Air mass flow rate P7 - Outlet pressure T21 - Inlet pressure T9 - Ambient temperature T11 - Outlet air temperature (same as first output)	T11 - Outlet air temperature (same as fifth output) P21 - Inlet pressure	Inside surface heat transfer coefficient x area (kW/°C) Outside surface heat transfer coefficient x area (kW/°C) Thermal capacitance of duct material (kJ/°C) Volume (m^3) Flow resistance [0.001/(kg m)] Height of outlet above inlet (m) Mode	0 0 0 0 0.02 0 1 (air)

INITIAL VARIABLE VALUES:

PRESSURE (kPa) -		FLOW (kg/s) -		TEMPERATURE (°C) -		CONTROL -		REVOLUTION SPEED (rev/s) -		POWER (kW) -		ABSOLUTE HUMIDITY [kg (water)/kg (air)] -	
P1 =	250	W1 =	0.522	T1 =	7	C1 =	0.52	R1 =	30	Q1 =	20.3	H1 =	0.011
P2 =	227.6	W4 =	2.83	T2 =	7.03	C2 =	0.52	R2 =	30	Q2 =	11.9	H2 =	0.009089
P3 =	246	W5 =	0.1919	T3 =	16.32	C3 =	0.24			Q3 =	1.33		
P5 =	-236	W6 =	2.12	T4 =	25.29	C4 =	0.24			Q4 =	1.33		
P6 =	-250			T5 =	21.2	C5 =	0.42			Q5 =	7.288		
P7 =	-0.0506			T6 =	12.6	C6 =	0.42			Q6 =	0		
P8 =	0.01154			T9 =	25	C7 =	1						
P9 =	0.005683			T10 =	16.46	C8 =	0.32						
P10 =	0.002257			T11 =	22.6	C9 =	1						
P11 =	0.0006503			T12 =	22.64	C11 =	0						
P12 =	0			T13 =	33								
P13 =	-0.0006871			T14 =	21.2								
P14 =	-0.002294			T15 =	21.2								
P15 =	-0.005719			T16 =	24								
P16 =	-0.00003681			T17 =	24								
P18 =	-0.029157			T18 =	24								
P19 =	-0.04322			T19 =	24								
P20 =	0.018966			T20 =	24								
P21 =	-0.124			T21 =	22.6								

TIME DEPENDENT BOUNDARY VARIABLES :

Internal gain Q5

Table 7.2: The five cooling coil designs

Cooling Coil Design	1	2	3	4	5
Primary (tube exterior) surface area, Ap (m^2)	6.571	6.491	6.41	6.33	6.249
Secondary surface area, As (m^2)	43.647	58.196	72.745	87.294	101.844
Internal surface area, Ai (m^2)	6.199	6.199	6.199	6.199	6.199
Ratio of minimum air flow area to face area	0.562	0.555	0.548	0.542	0.535
Number of fins per unit length (per cm)	2.3622	3.1496	3.937	4.7244	5.5118

Table 7.3 The estimated P+I values for the simulation tests with control valve resistance = 0.015 $kg^{-1}\ m^{-1}$

Cooling Coil Design	CCD1	CCD2	CCD3	CCD4	CCD5
Nichols & Ziegler open loop estimate					
Proportion gain	0.2797	0.2020	0.2616	0.2944	0.2291
Integral gain	0.0013	0.00079	0.00129	0.00164	0.00105
Lopez et al. estimate, integral absolute error					
Proportional gain	0.2963	0.2148	0.2771	0.3112	0.2429
Integral gain	0.00131	0.00086	0.00128	0.00155	0.00107
Lopez et al. estimate, integral error square					
Proportional gain	0.3707	0.2707	0.3463	0.3874	0.3042
Integral gain	0.00142	0.00093	0.00139	0.00169	0.00116
Lopez et al. estimate, integral absolute error times time					
Proportional gain	0.2535	0.1842	0.2370	0.2658	0.2079
Integral gain	0.00117	0.00077	0.00114	0.00137	0.00096
Cohen & Coon estimate					
Proportional gain	0.2848	0.2046	0.2641	0.2968	0.2314
Integral gain	0.00145	0.00093	0.00143	0.00176	0.00118

Table 7.4 The estimated P+I values for the simulation tests with control valve resistance = 0.03 $kg^{-1}\ m^{-1}$

Cooling Coil Design	CCD1	CCD2	CCD3	CCD4	CCD5
Nichols & Ziegler open loop estimate					
Proportion gain	0.2515	0.2636	0.2546	0.2467	0.24457
Integral gain	0.001	0.00117	0.00115	0.00111	0.00112
Lopez et al. estimate, integral absolute error					
Proportional gain	0.2666	0.27913	0.26959	0.26123	0.25891
Integral gain	0.0010	0.00116	0.00114	0.0011	0.0011
Lopez et al. estimate, integral error square					
Proportional gain	0.3647	0.3488	0.33687	0.3264	0.3234
Integral gain	0.00121	0.00126	0.00124	0.00120	0.00120
Lopez et al. estimate, integral absolute error times time					
Proportional gain	0.2283	0.23871	0.23054	0.22340	0.22138
Integral gain	0.00091	0.00104	0.00101	0.00098	0.00098
Cohen & Coon estimate					
Proportional gain	0.25411	0.26608	0.25699	0.24902	0.24683
Integral gain	0.00112	0.00130	0.00127	0.00123	0.00123

The performances of the coil designs in term of return air temperature against time were recorded for the step change of internal cooling load. The five controller tuning estimates are evaluated against their control performance and energy consumption..The control performances of the five coil designs are then evaluated with respective to the objective function achieved.

The experimental results from chapter 6 shows that the time constant for an average pneumatic actuator is 21.5 seconds and that for an electric actuator is 10 seconds. The test was first conducted for the five coil designs with control valve resistance equals to 0.015 $kg^{-1}\ m^{-1}$ for both pneumatic actuator and electric actuator. The test was repeated at control valve resistance of 0.03 $kg^{-1}\ m^{-1}$. A total of 100 simulations were conducted.

7.6 RESULTS

The responses of the zone return air temperature against time for a period of 28800 seconds (8 hours) for the five cooling coil designs under step change of internal occupant load from zero kW to 12.505 kW at time 50000 seconds are shown in Figure 7.3 to Figure 7.22. From these response curves, the energy consumption, initial and final steady state zone return air temperature as well as the control criteria used to analyze the control performance are tabulated in Table 7.5 and Table 7.6 for control valve resistance equals to 0.015 kg^{-1} m^{-1} and 0.03 $kg^{-1}m^{-1}$ respectively.

Figure 7.3: Room return air temperature response for CCD1, control valve actuator time constant = 21.5s, control valve resistance= 0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

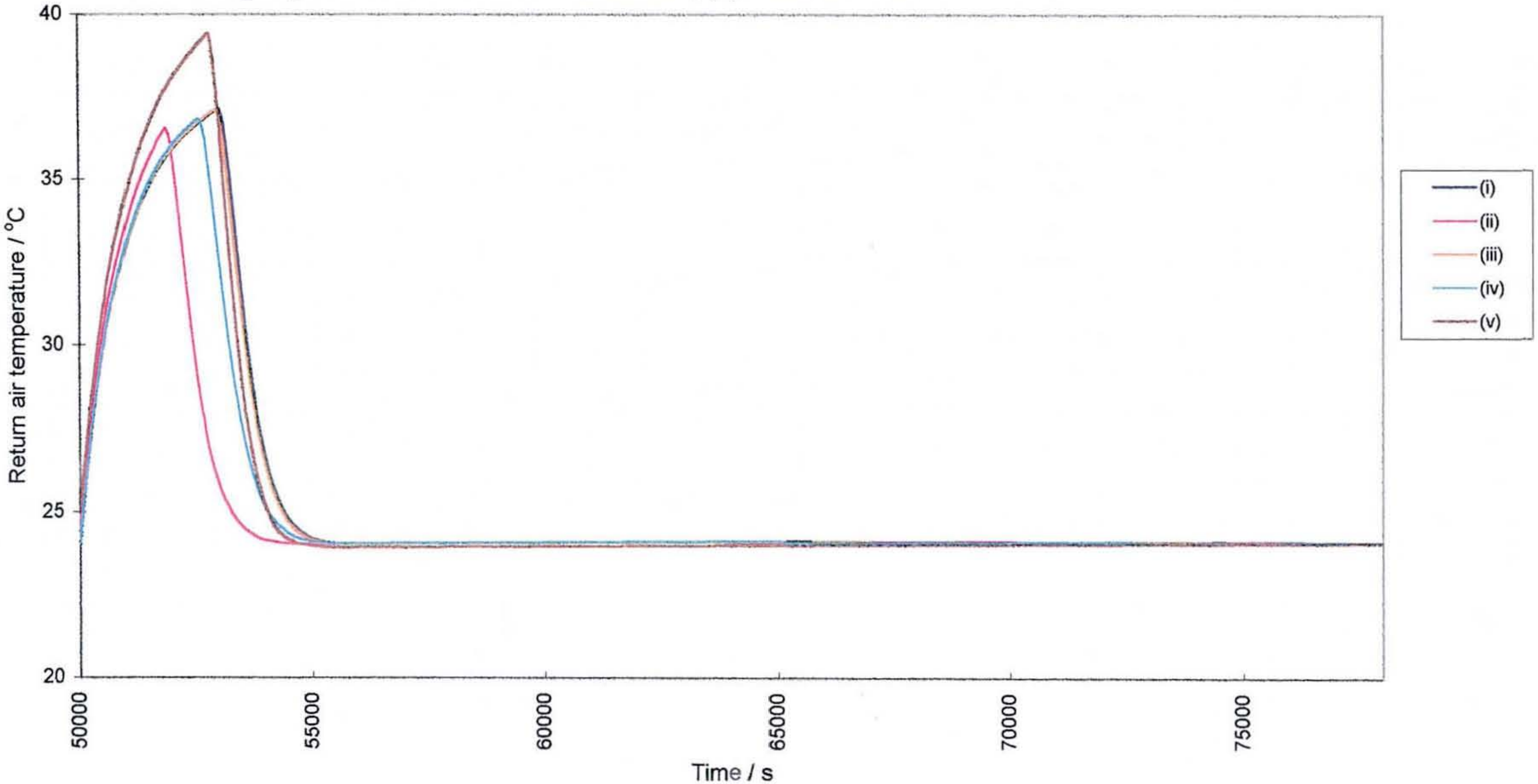


Figure 7.4: **Room return air temperature response for CCD2, control valve actuator time constant = 21.5s, control valve resistance= 0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon**

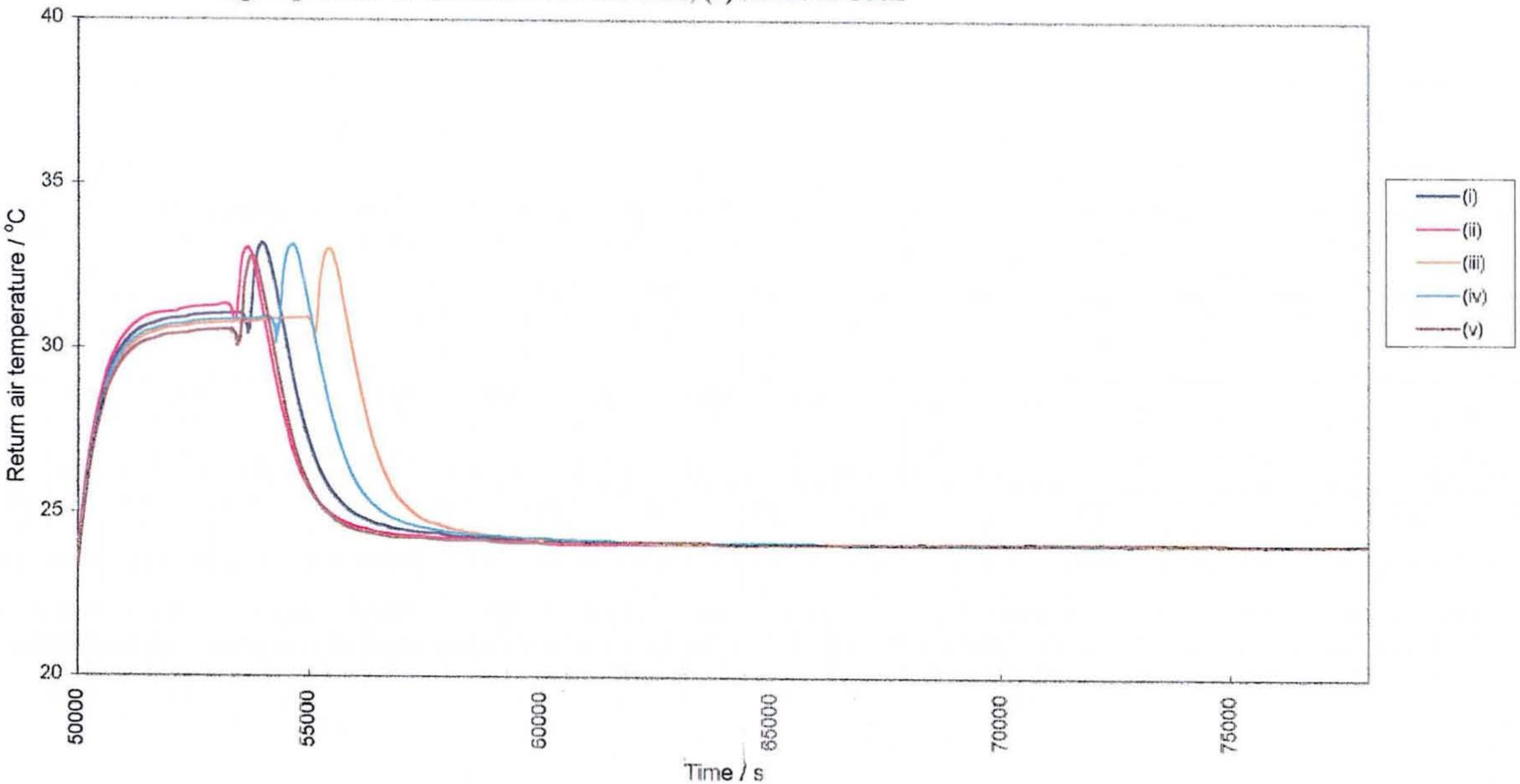


Figure 7.5: Room return air temperature response for CCD3, control valve actuator time constant = 21.5s, control valve resistance=0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

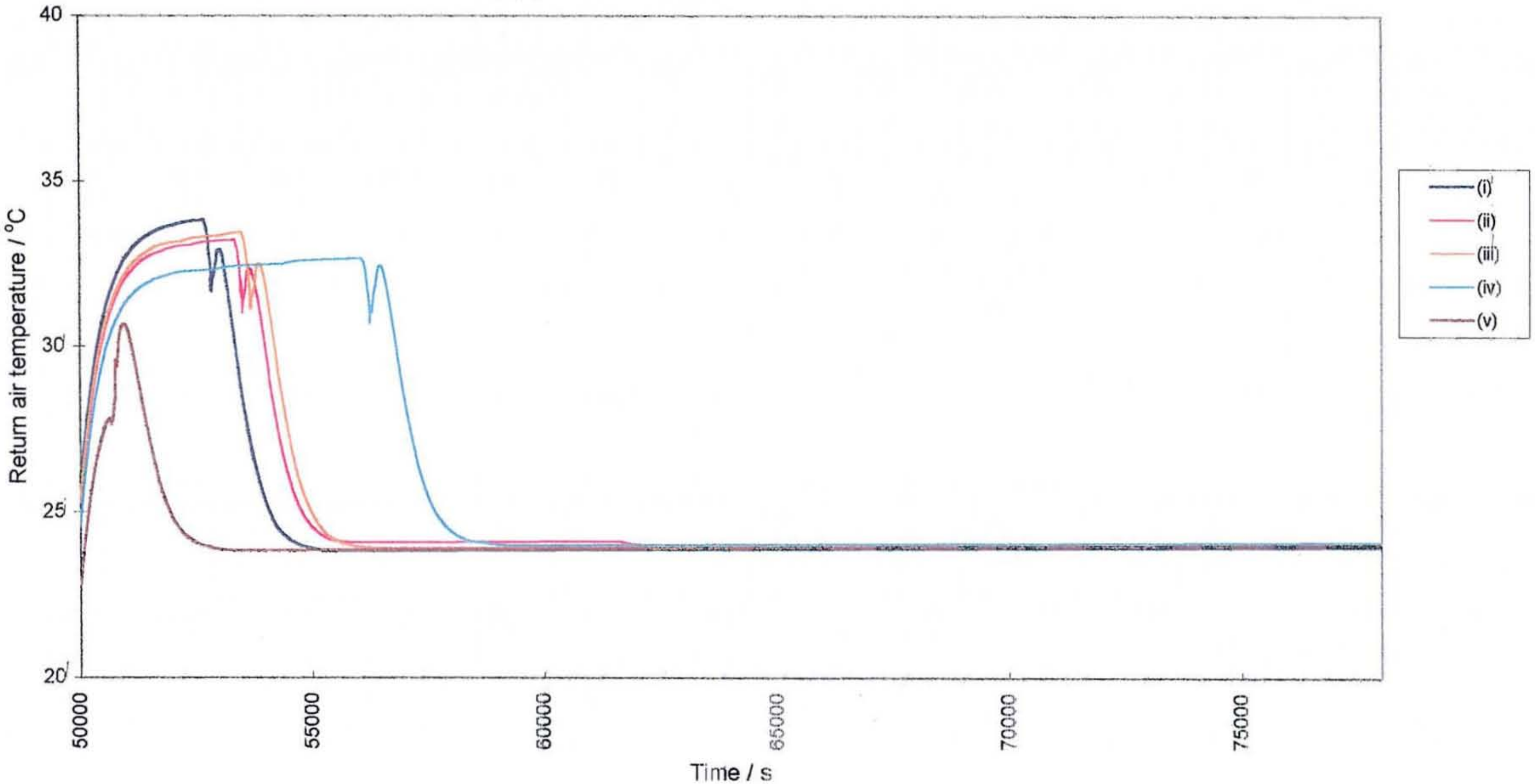


Figure 7.6: Room return air temperature response for CCD4, control valve actuator time constant = 21.5s, control valve resistance= 15 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

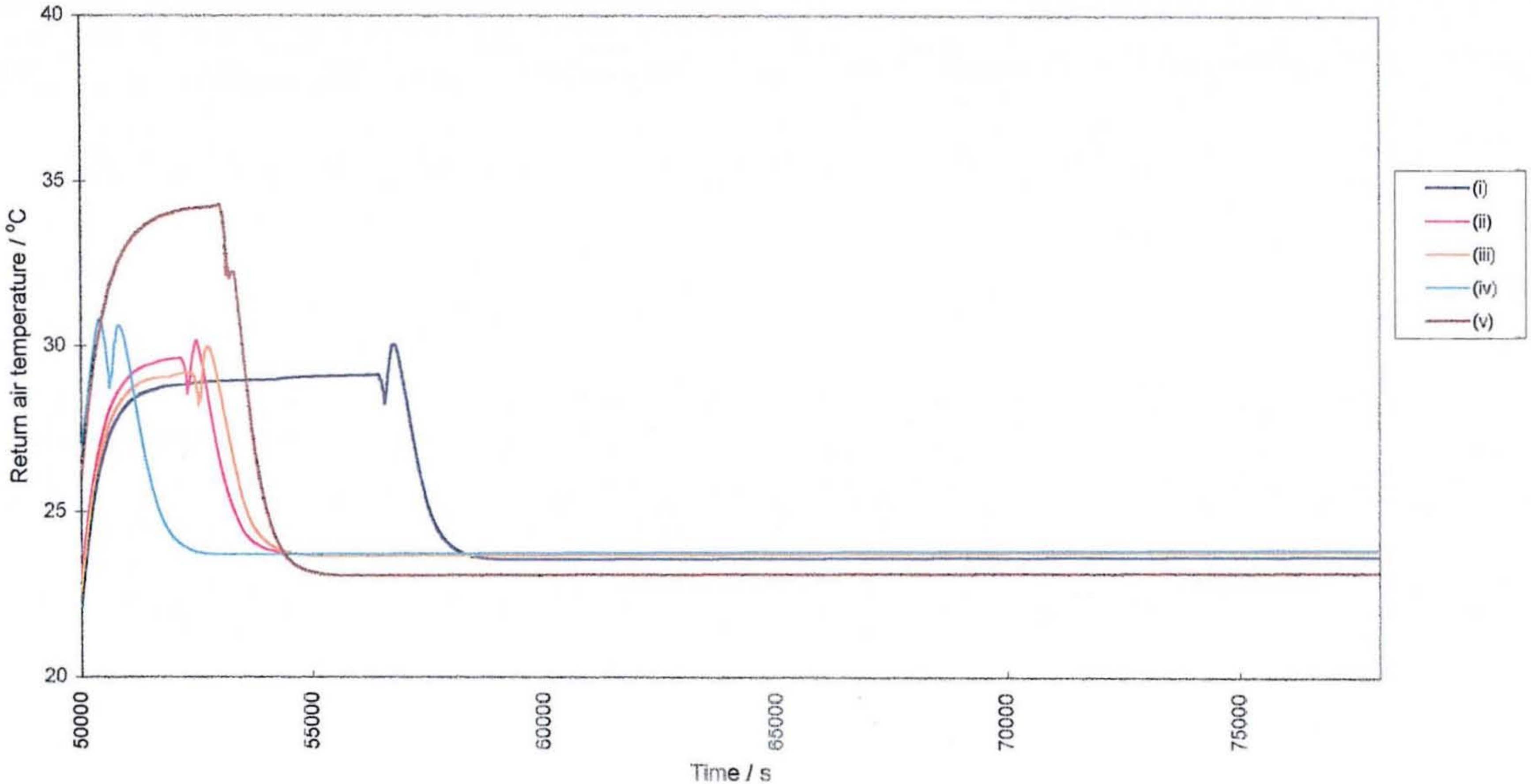


Figure 7.7: **Room return air temperature response for CCD5, control valve actuator time constant = 21.5s, control valve resistance= 0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon**

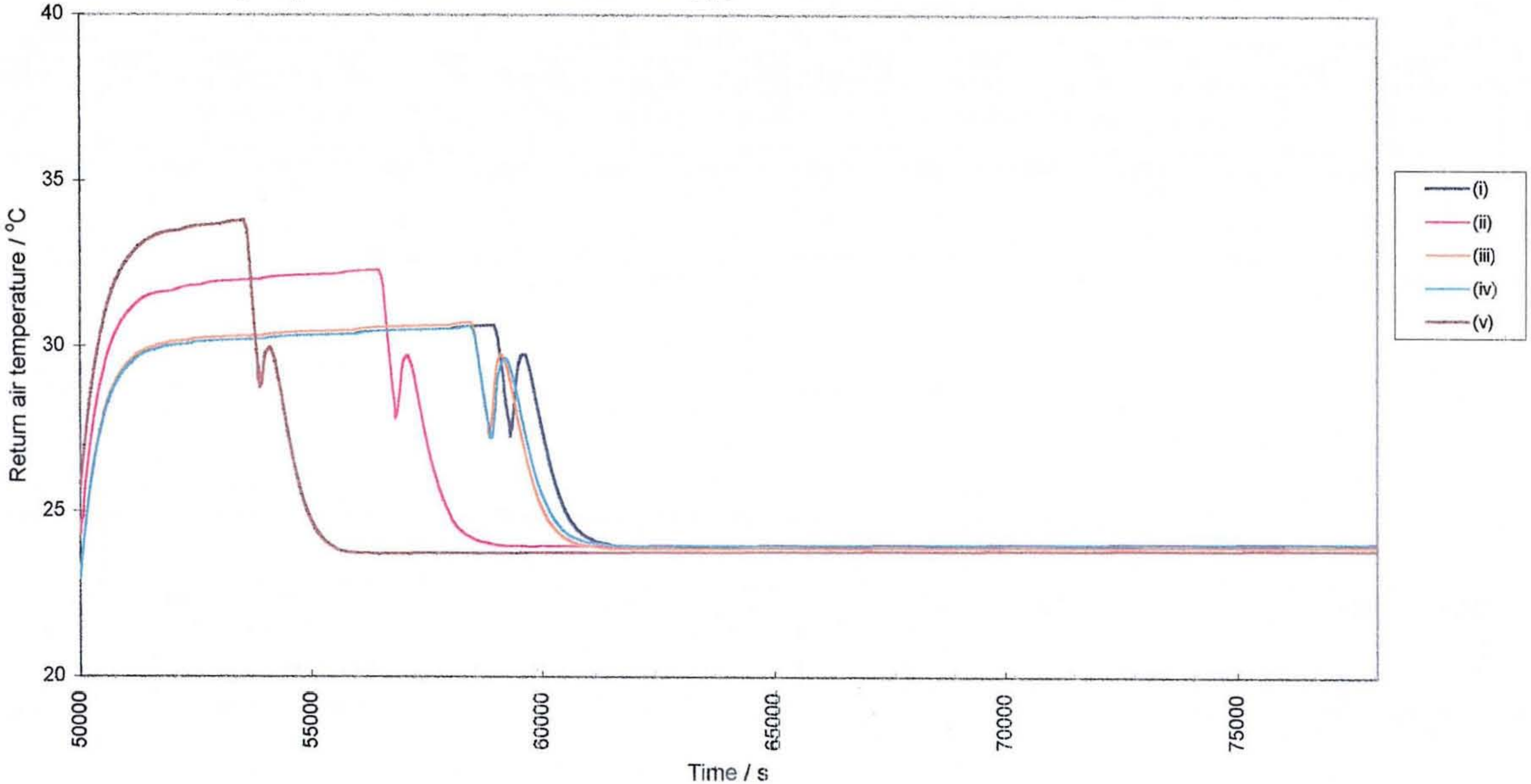


Figure 7.8: Room return air temperature response for CCD1, control valve actuator time constant = 10s, control valve resistance= 15 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

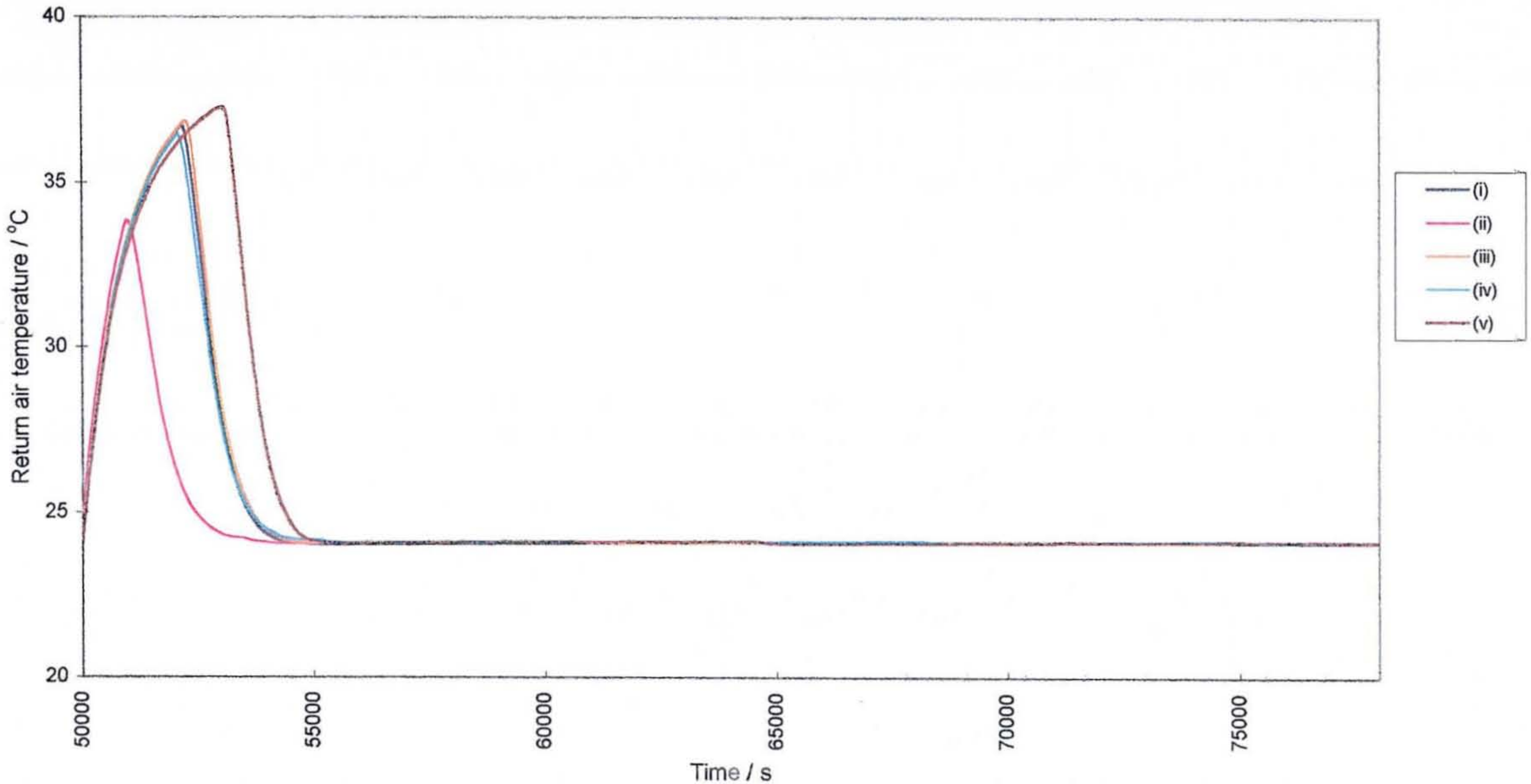


Figure 7.9: Room return air temperature response for CCD2, control valve actuator time constant = 10s, control valve resistance= 0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

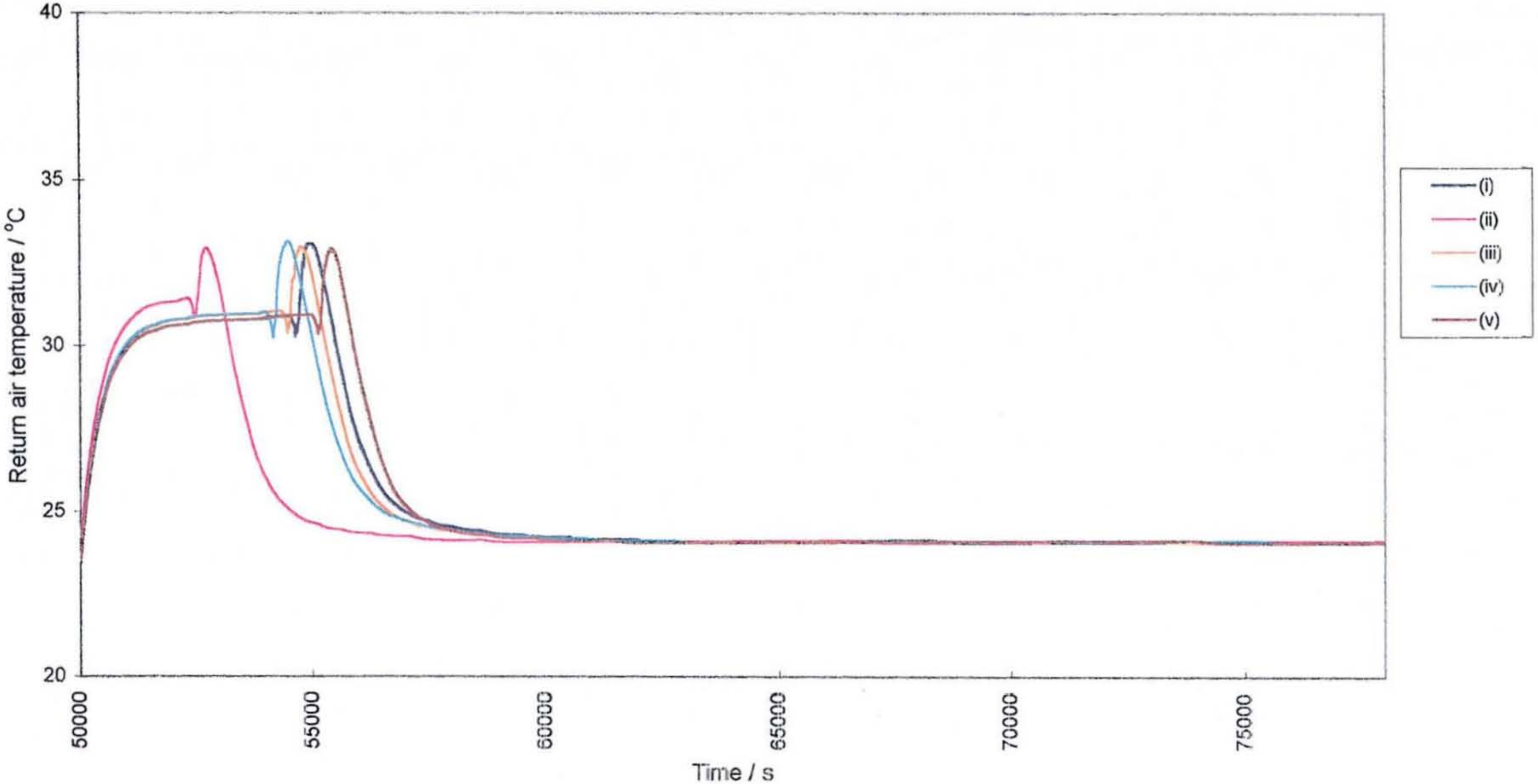


Figure 7.10: Room return air temperature response for CCD3, control valve actuator time constant = 10s, control valve resistance= 0.015 kg^{-1} m^{-1}, for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

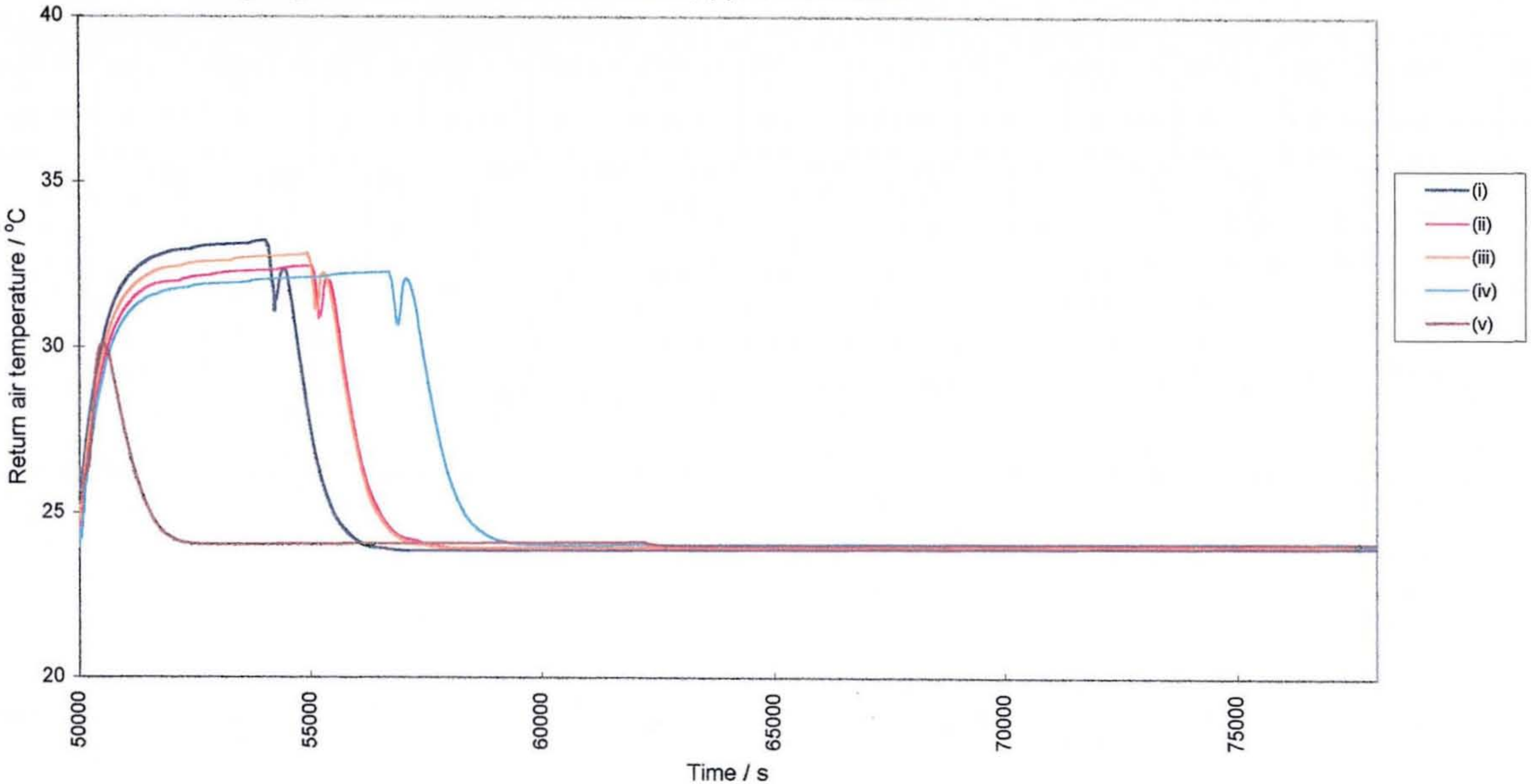


Figure 7.11: Room return air temperature response for CCD4, control valve actuator time constant = 10s, control valve resistance= 15 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

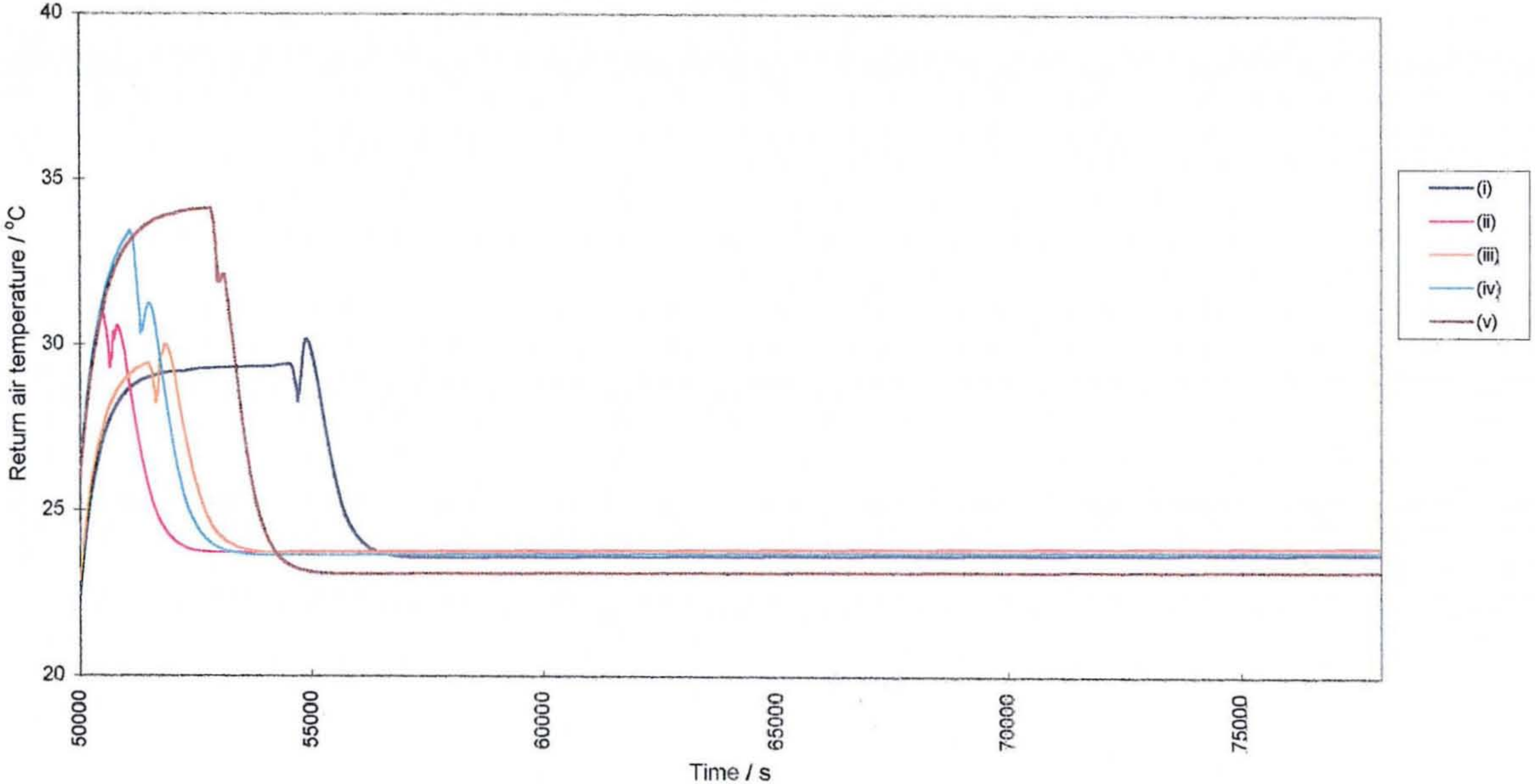


Figure 7.12: Room return air temperature response for CCD5, control valve actuator time constant = 10s, control valve resistance= 15 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

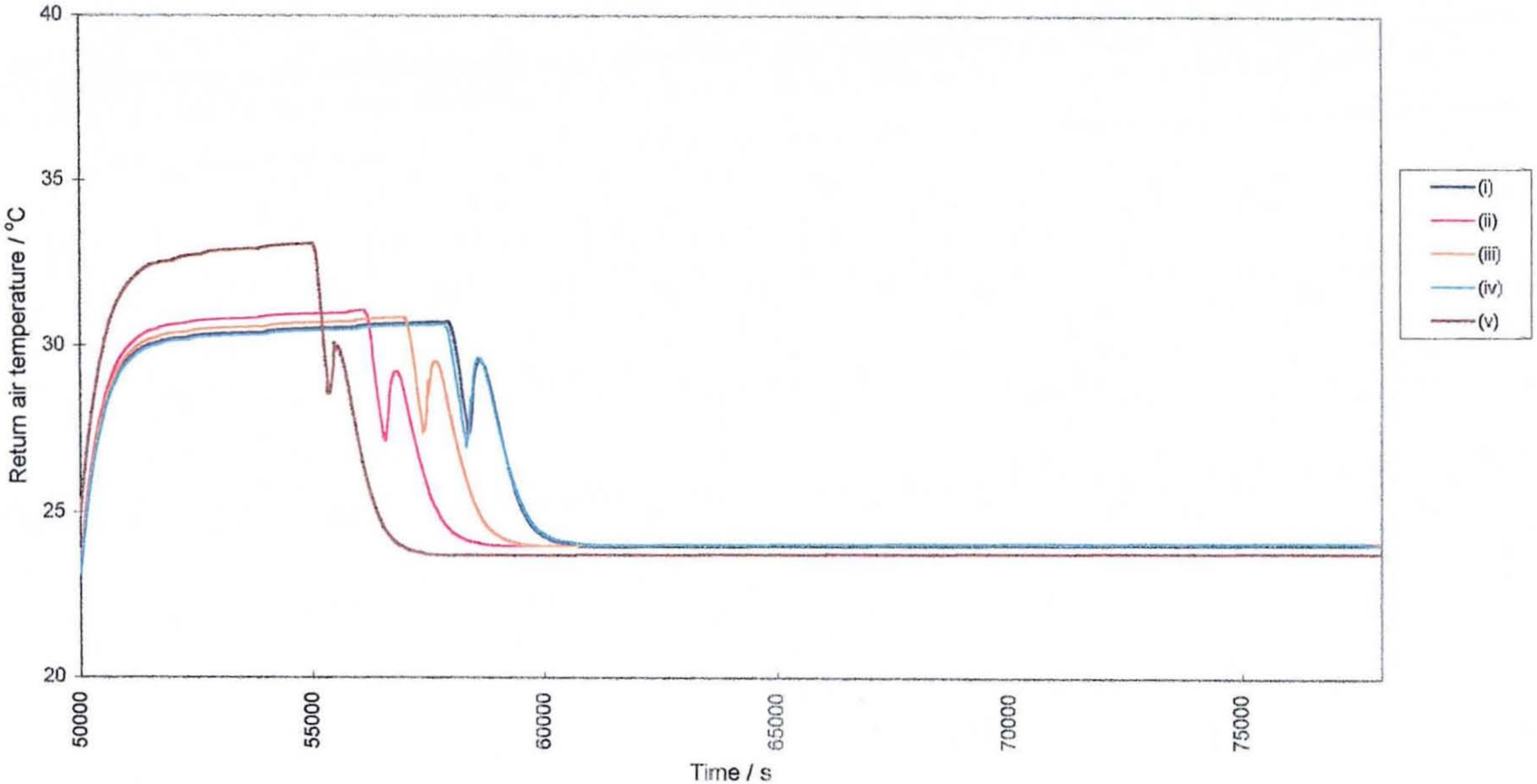


Figure 7.13: Room return air temperature response for CCD1, control valve actuator time constant = 21.5s, control valve resistance=0.03 kg^{-1} m^{-1}, for PI estimates based on (i) Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

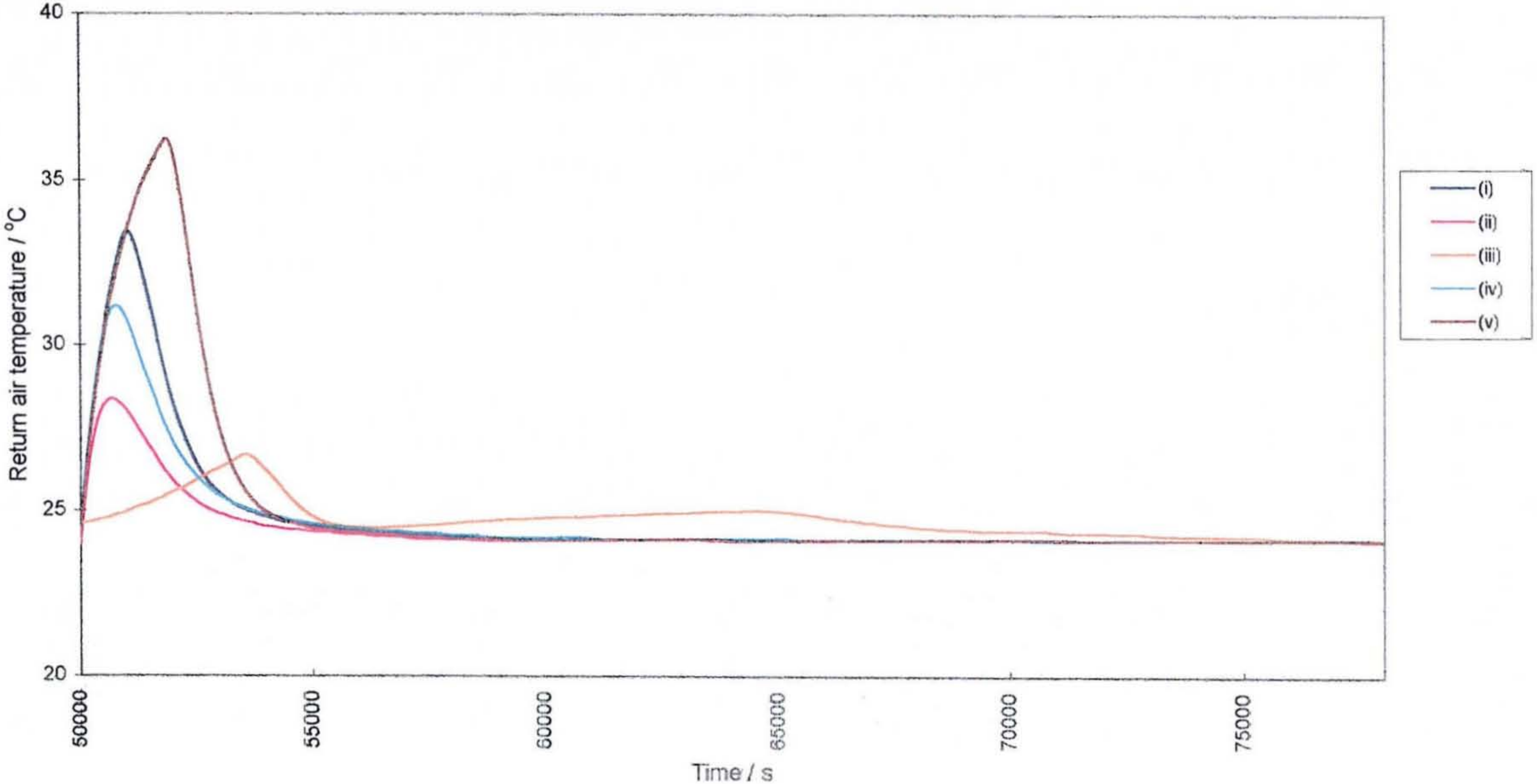


Figure 7.14: Room return air temperature response for CCD2, control valve actuator time constant = 21.5s, control valve resistance= 0.03 kg^{-1} m^{-1}, for PI estimates based on (i) Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

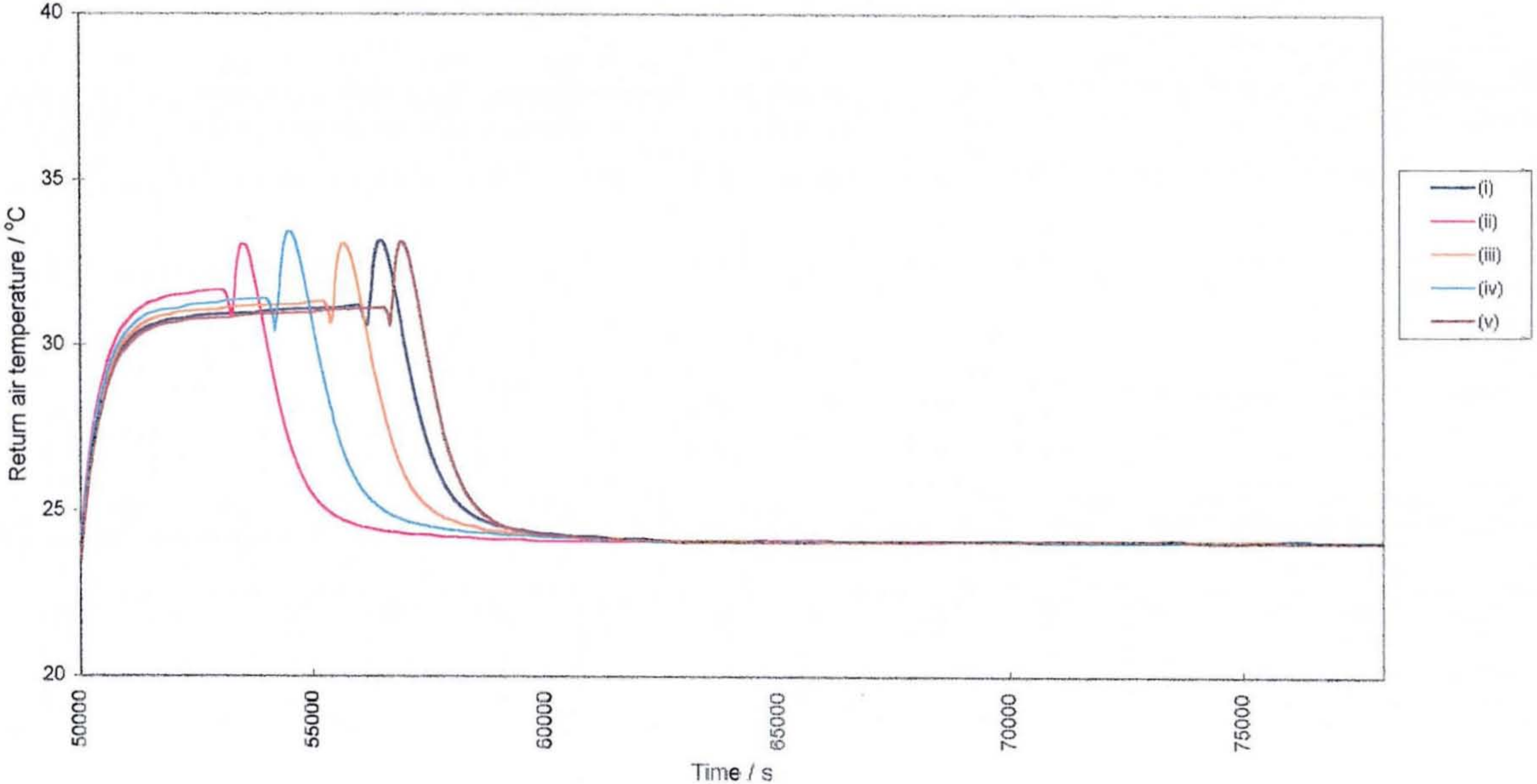


Figure 7.15: Room return air temperature response for CCD3, control valve actuator time constant = 21.5s, control valve resistance= 0.03 $kg^{-1}\ m^{-1}$, for PI estimates based on (i) Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

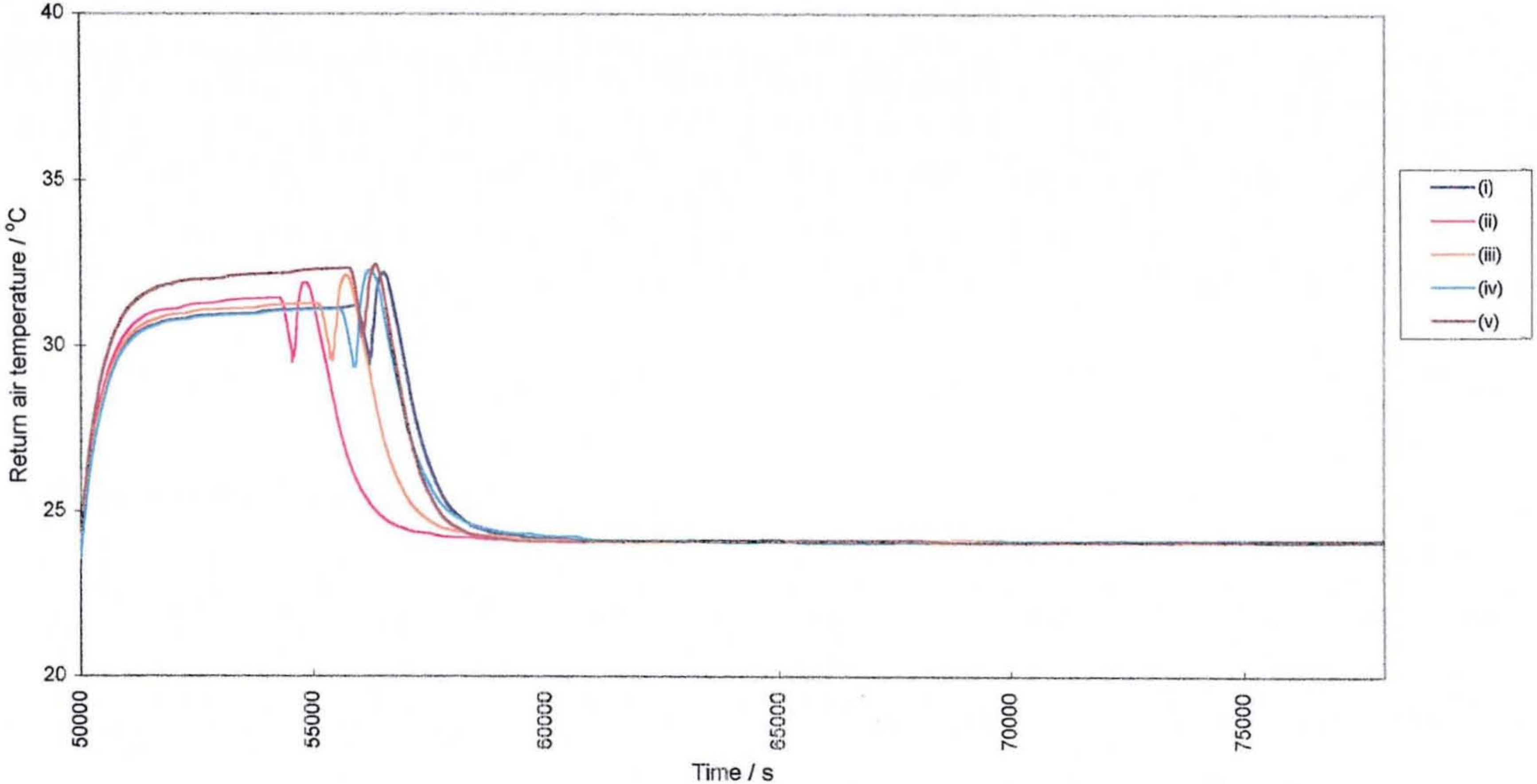


Figure 7.16: Room return air temperature response for CCD4, control valve actuator time constant = 21.5s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i) Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

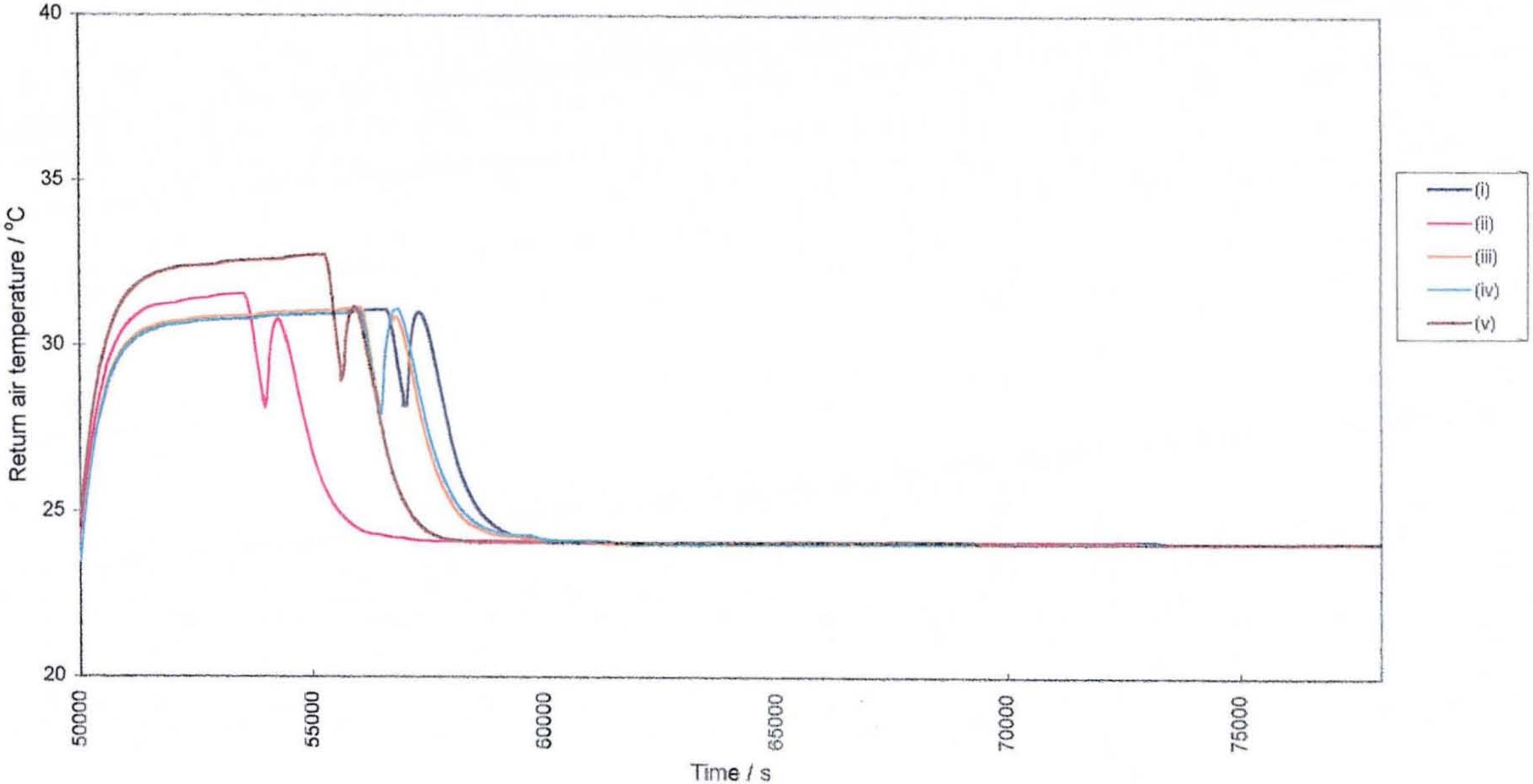


Figure 7.17: Room return air temperature response for CCD5, control valve actuator time constant = 21.5s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i) Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

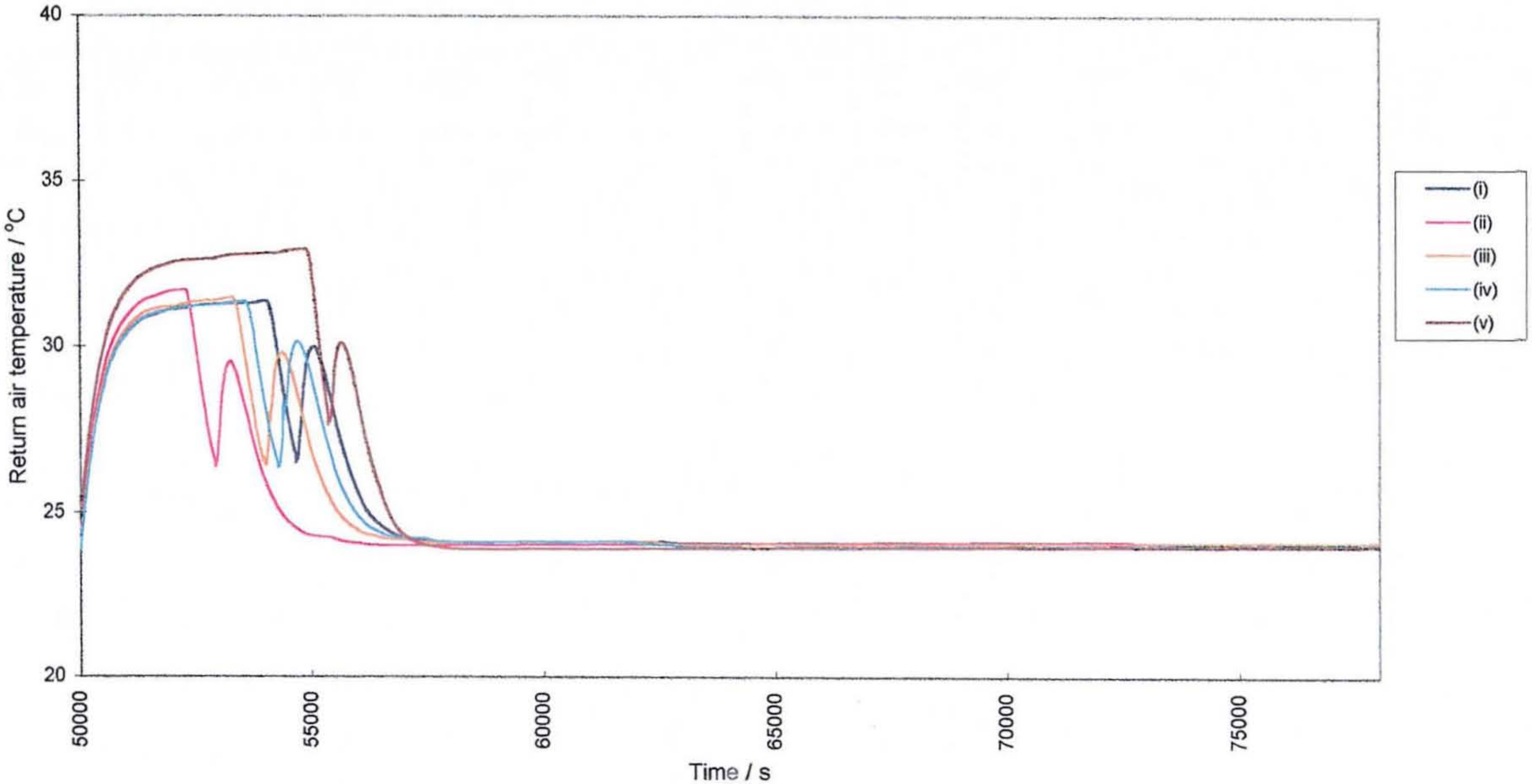


Figure 7.18: Room return air temperature response for CCD1, control valve actuator time constant = 10s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

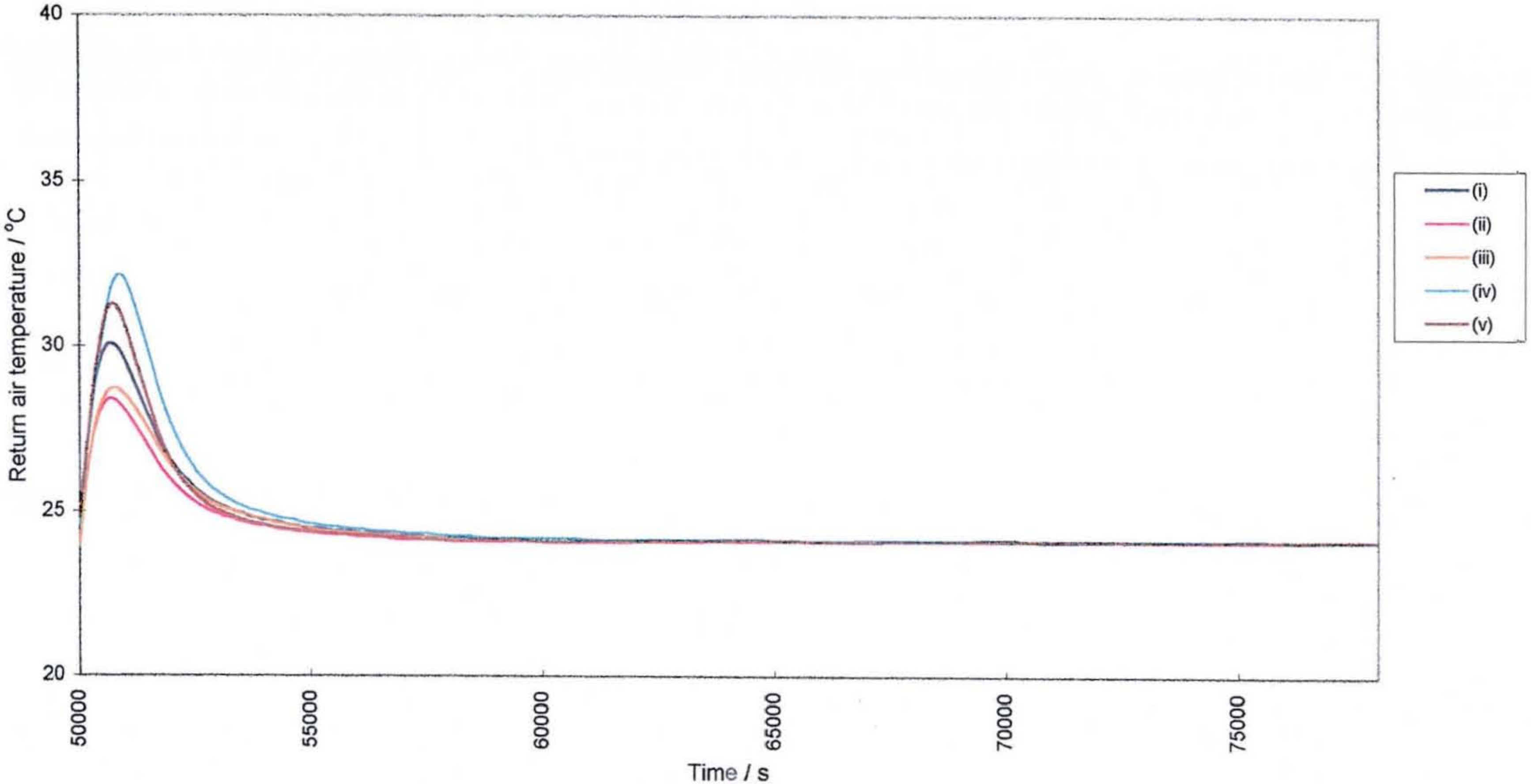


Figure 7.19: Room return air temperature response for CCD2, control valve actuator time constant = 10s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

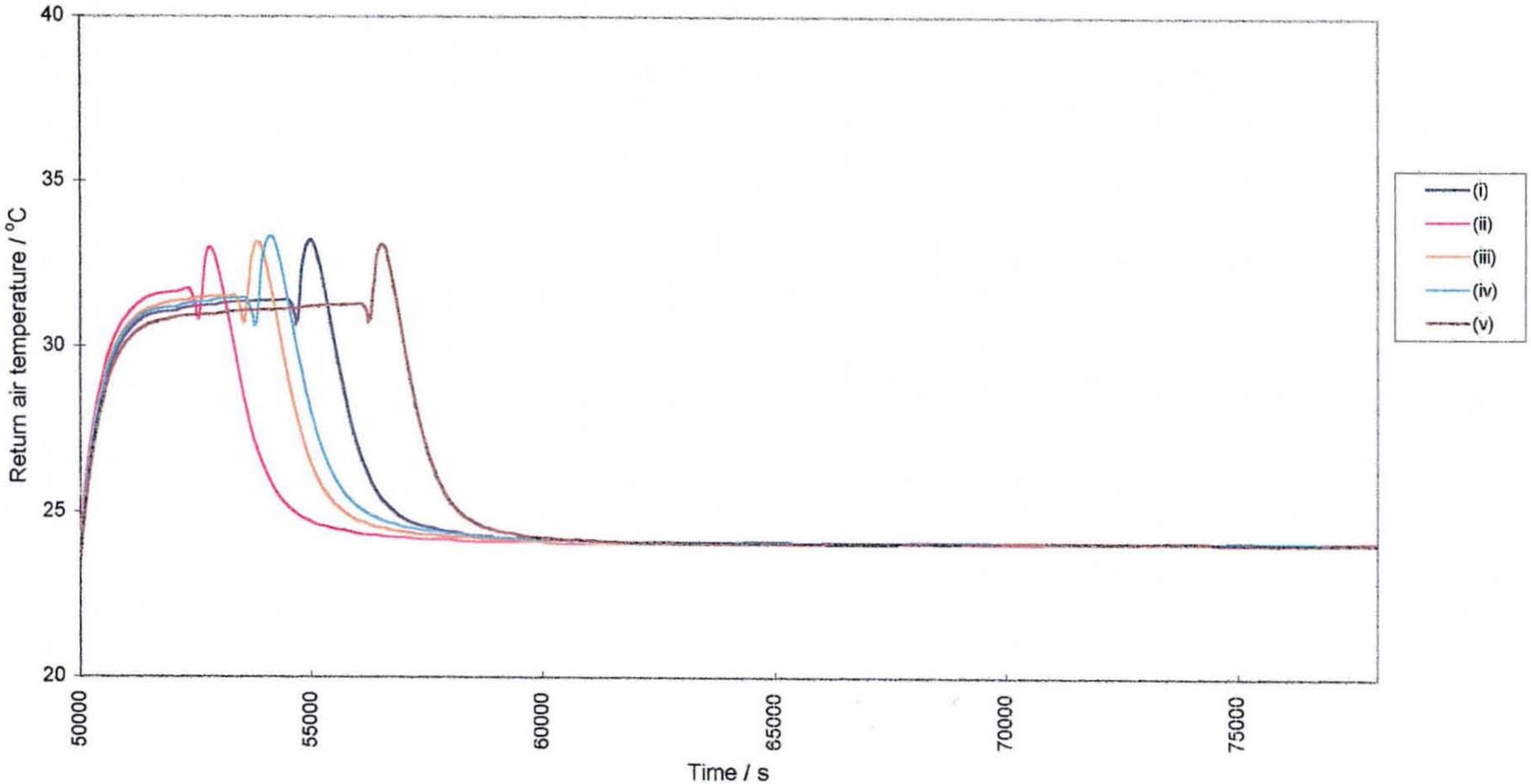


Figure 7.20: Room return air temperature response for CCD3, control valve actuator time constant = 10s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

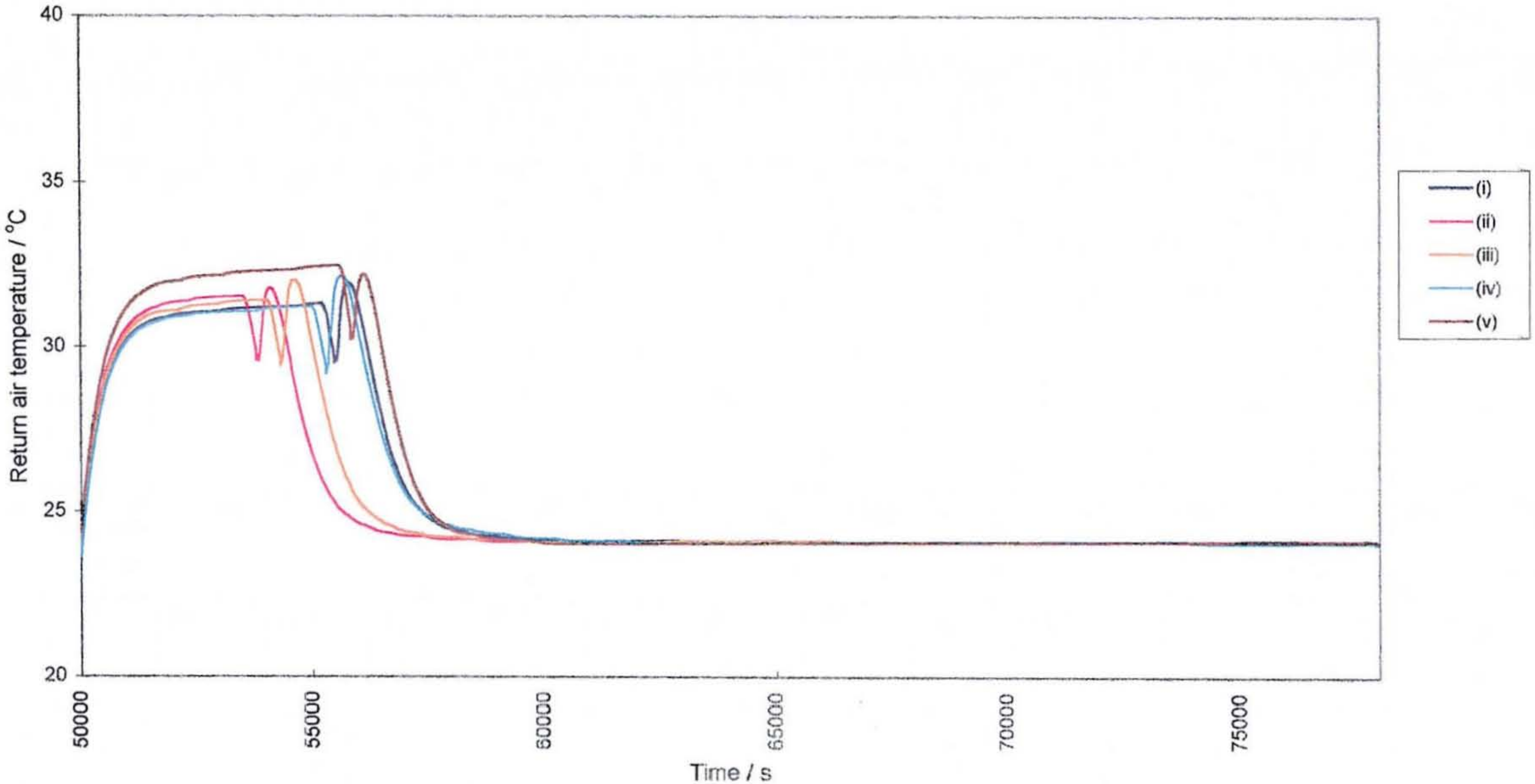


Figure 7.21: Room return air temperature response for CCD4, control valve actuator time constant = 10s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

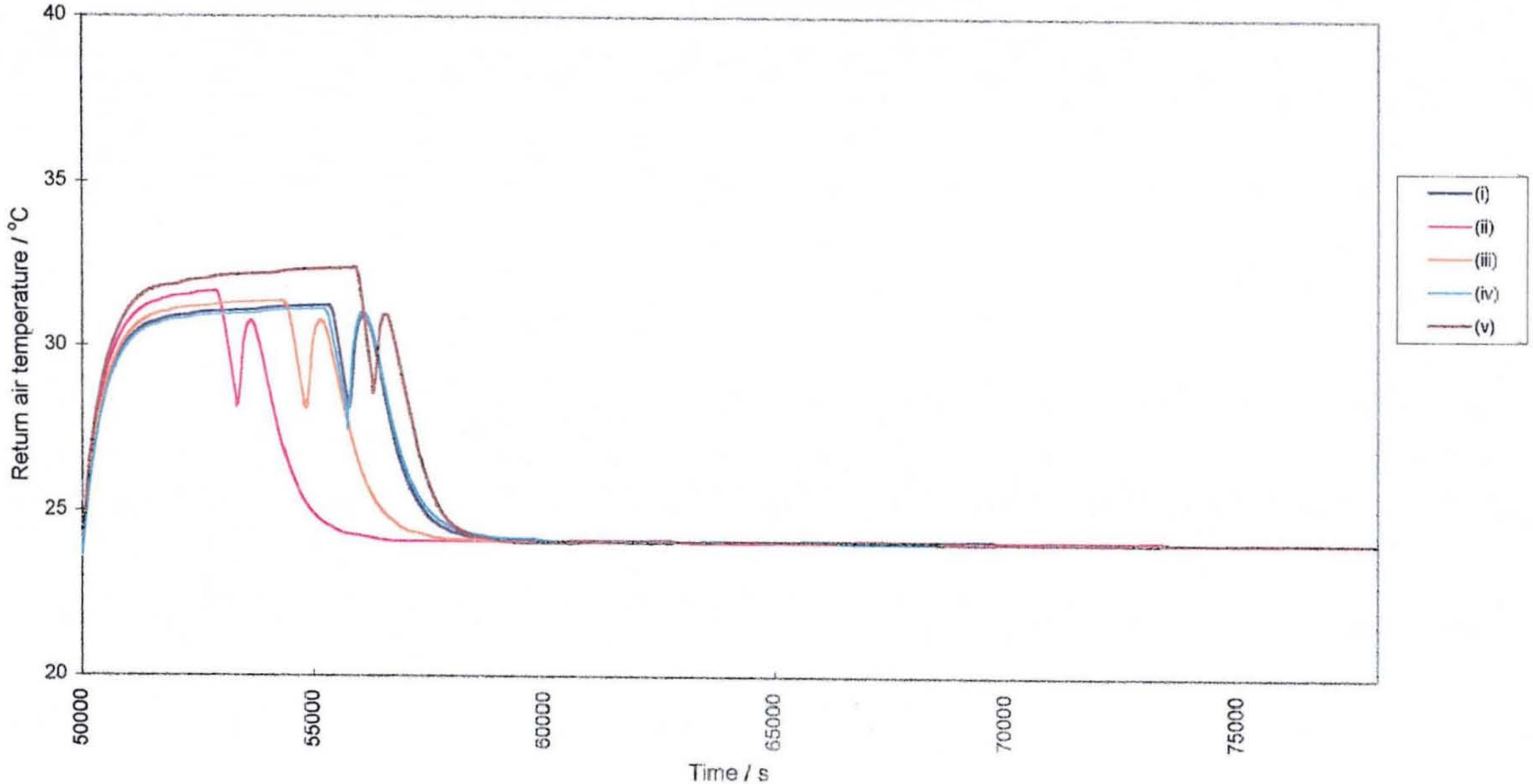


Figure 7.22: Room return air temperature response for CCD5, control valve actuator time constant = 10s, control valve resistance= 30 (0.001/kg M), for PI estimates based on (i)Nichols & Ziegler, (ii)Lopez et al integral error square, (iii)Lopez et al integral absolute error, (iv)Lopez et al integral product of absolute error and time, (v)Cohen & Coon

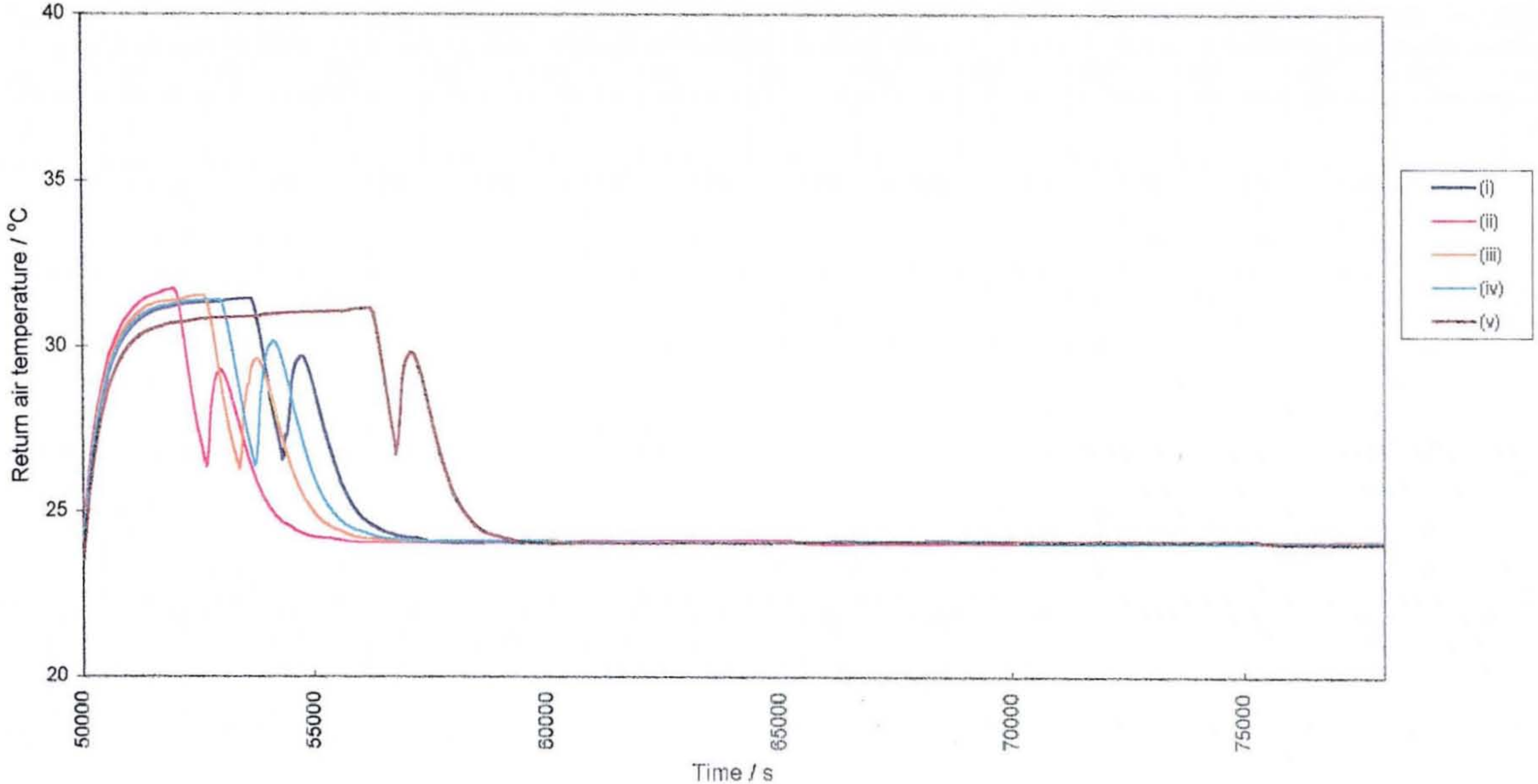


Table 7.5 Energy consumption, initial and final steady state zone return air temperature, control criteria of return air temperature step response for control valve resistance = 0.015 kg^{-1} m^{-1}

i. Control valve actuator time constant = 21.5 seconds

	1	2	3	4	5	6	7	8	9
CCD1									
A	24.013	24.122	280.94	589.23	3000.1	4582.4	53.92	37.129	13.21
B	24.499	24.111	281.29	511.30	1859.9	3469.4	51.56	36.543	7.538
C	24.052	24.126	281.02	587.14	2940.1	4527.8	53.83	37.113	12.90
D	24.086	24.137	280.79	562.53	2550.1	4259.0	52.30	36.761	10.97
E	25.111	24.099	283.86	639.97	2760.1	4180.2	63.57	39.419	16.27
CCD2									
A	23.689	24.089	275.77	316.05	3480.1	6721.4	28.94	31.060	7.873
B	23.874	24.112	276.58	441.96	3647.1	5959.3	37.13	33.065	7.543
C	23.433	24.087	268.00	483.15	5397.4	7958.9	37.17	33.040	9.587
D	23.499	24.125	272.80	485.11	4609.4	7301.5	37.40	33.148	8.957
E	23.374	24.100	276.97	480.10	3746.4	5810.5	36.13	32.807	6.495
CCD3									
A	25.805	23.961	286.98	341.5	2670.1	4301.0	41.16	33.823	9.605
B	25.283	24.107	282.79	342.6	3330.1	4952.0	37.83	33.227	10.65
C	25.421	24.040	280.99	349.1	3480.1	5178.4	39.22	33.468	11.08
D	24.645	24.104	264.79	352.4	6060.1	7859.9	35.52	32.666	15.70
E	22.766	24.028	300.20	372.3	931.2	2135.3	27.64	30.669	1.582
CCD4									
A	22.102	23.665	273.95	365.49	6764.2	7863.4	26.92	30.036	5.914
B	22.921	23.869	294.49	288.68	2160.0	3530.7	24.17	29.638	2.741
C	22.503	23.821	293.83	336.46	2736.3	3843.1	25.89	29.988	2.718
D	26.603	23.871	305.90	151.62	390.1	1912.3	29.03	30.801	1.851
E	26.063	23.176	292.46	355.68	3000.1	4418.6	47.94	34.287	12.08
CCD5									
A	22.951	24.015	253.69	335.14	8940.0	10726	27.60	30.643	12.40
B	24.289	24.052	263.99	345.84	6450.4	8166.4	34.42	32.331	14.13
C	23.026	23.959	256.12	334.97	8460.1	10276	28.28	30.735	12.10
D	22.945	24.041	255.79	332.46	8460.0	10381	27.33	30.611	11.72
E	25.713	23.850	283.71	348.97	3540.1	5153.3	41.81	33.822	11.09

ii. **Control valve actuator time constant = 10 seconds**

	1	2	3	4	5	6	7	8	9
CCD1									
A	24.309	24.119	280.92	538.47	2160.1	3776.8	52.24	36.719	9.083
B	24.863	24.108	281.62	348.50	960.1	2713.2	40.37	33.840	3.134
C	24.343	24.140	281.93	546.13	2220.1	3853.6	52.68	36.857	9.513
D	24.275	24.126	281.73	527.40	2070.1	3874.6	51.13	36.462	8.562
E	24.096	24.099	282.38	595.35	3060.1	4530.5	54.75	37.293	13.82
CCD2									
A	23.437	24.111	270.41	486.99	4995.8	7727.7	37.18	33.075	9.641
B	24.077	24.096	281.08	414.51	2708.9	5175.4	36.69	32.937	6.364
C	23.545	24.085	271.14	465.26	4757.9	7315.8	33.78	32.221	9.331
D	23.579	24.073	273.06	476.81	4452.9	7446.4	37.66	33.139	8.967
E	23.459	24.080	267.40	473.69	5407.0	7635.1	36.75	32.929	9.603
CCD3									
A	25.190	23.973	277.84	350.25	4020.1	5763.6	38.64	33.236	12.09
B	24.544	24.031	271.82	356.26	5006.3	6730.6	35.10	32.466	12.53
C	24.804	24.058	271.78	350.44	4920.1	6653.5	36.27	32.784	13.58
D	23.940	24.053	261.88	352.68	6688.1	8596.6	34.18	32.274	15.68
E	25.300	24.116	303.06	268.01	501.5	1702.3	24.88	30.116	0.912
CCD4									
A	22.414	23.721	282.72	351.03	4879.2	5989.5	27.16	30.164	4.781
B	26.051	23.932	304.34	177.71	510.0	1835.5	29.07	30.889	2.194
C	22.856	23.913	297.67	312.35	1815.1	2910.4	25.45	29.999	2.075
D	26.258	23.809	300.17	291.32	1080.1	2619.1	40.42	33.433	4.740
E	26.015	23.210	293.30	349.81	2817.7	4245.8	47.02	34.123	10.96
CCD5									
A	23.093	24.072	258.32	329.74	7950.1	9764.8	27.65	30.728	11.58
B	23.465	24.112	267.23	328.71	6120.1	7902.3	28.90	31.080	9.805
C	23.262	24.108	263.25	328.85	6990.1	8770.2	28.02	30.863	10.69
D	23.078	24.037	258.56	328.01	7860.1	9848.3	27.50	30.647	11.23
E	25.006	23.813	273.59	350.43	4980.1	6674.1	38.95	33.088	13.55

Table 7.6 **Energy consumption, initial and final steady state zone return air temperature, control criteria of return air temperature step response for control valve resistance = 0.03 $kg^{-1}\ m^{-1}$**

i. Control valve actuator time constant = 21.5 seconds

	1	2	3	4	5	6	7	8	9
CCD1									
A	24.589	24.119	281.86	343.19	1020.1	4688.5	38.67	33.446	3.677
B	23.994	24.118	281.85	160.42	690.1	3798.5	17.64	28.372	0.878
C	24.627.	24.097	254.85	2157.0	3540.2	5444.6	10.75	26.687	0.812
D	24.652.	24.123	282.45	237.32	780.1	5010.5	29.29	31.189	2.267
E	24.401	24.116	281.18	504.03	1860.1	4661.6	50.28	36.242	8.068
CCD2									
A	23.571	24.075	261.24	332.25	5970.1	8964.9	29.62	31.206	12.24
B	24.135	24.101	276.42	418.95	3458.8	5894.2	37.17	33.059	7.856
C	23.674	24.079	265.43	457.77	5682.9	8302.6	37.35	33.073	11.39
D	23.876	24.109	271.84	473.63	4475.3	7296.0	38.64	33.425	9.725
E	23.469	24.085	258.75	490.11	6927.0	9024.0	27.31	30.663	12.63
CCD3									
A	23.565	24.082	267.80	401.28	6520.8	8415.1	33.82	32.227	11.28
B	23.910	24.110	276.66	354.04	4775.8	6735.0	32.37	31.914	9.119
C	23.724	24.104	272.25	384.56	5692.7	7643.6	33.39	32.152	10.38
D	23.553	24.075	269.84	408.42	6250.0	8487.9	34.13	32.292	10.83
E	24.423	24.108	266.63	354.70	6332.5	7988.2	34.66	32.464	14.60
CCD4									
A	23.453	24.078	265.78	331.10	6600.1	8919.7	29.15	31.097	11.45
B	24.025	24.077	282.99	322.41	3540.1	5891.4	31.15	31.577	7.398
C	23.540	24.088	269.21	329.65	6030.1	8470.9	29.38	31.165	10.73
D	23.471	24.107	269.22	327.90	6000.1	8590.9	28.85	31.062	10.61
E	24.752	24.094	270.85	350.03	5280.1	7324.1	35.97	32.761	13.70
CCD5									
A	23.854	24.071	278.66	321.63	4020.1	6521.4	30.28	31.360	7.581
B	24.295	24.077	287.51	313.08	2280.1	4622.9	31.76	31.724	4.711
C	23.993	24.043	283.17	319.46	3300.1	5910.0	30.98	31.492	6.499
D	23.841	24.045	281.51	319.69	3570.1	6282.9	30.43	31.362	7.075
E	24.904	24.002	272.50	347.4	4890.1	6878.3	37.28	32.950	12.76

ii. Control valve actuator time constant = 10 seconds

	1	2	3	4	5	6	7	8	9
CCD1									
A	24.822	24.123	282.42	186.61	660.1	4529.5	24.72	30.086	1.644
B	23.942	24.144	281.62	161.72	660.1	3746.6	17.73	28.425	3.134
C	23.956	24.118	281.90	175.29	750.1	4535.2	19.18	28.744	1.134
D	24.587	24.135	282.39	284.17	870.1	5107.7	33.30	32.172	2.880
E	24.846	24.127	282.16	234.24	720.1	4020.9	29.66	31.283	2.050
CCD2									
A	23.852	24.103	268.71	458.82	4990.7	7421.5	37.84	33.223	10.23
B	24.318	24.099	279.99	400.46	2810.0	5326.5	36.92	32.996	6.805
C	24.019	24.111	274.96	438.22	3831.3	6330.4	37.55	33.165	8.484
D	23.955	24.100	273.60	458.85	4089.6	6991.1	38.32	33.335	9.007
E	23.603	24.087	260.64	469.15	6539.1	8710.9	37.34	33.081	15.54
CCD3									
A	23.682	24.005	271.90	371.97	5773.2	7961.8	33.12	31.955	10.48
B	23.993	24.139	280.95	337.02	4102.0	6033.9	31.66	31.781	7.763
C	23.889	24.128	278.27	362.54	4645.3	6604.8	32.73	32.025	8.607
D	23.653	24.081	273.15	390.79	5645.7	8006.5	33.65	32.184	10.09
E	24.473	24.087	267.62	347.21	5580.1	7881.7	34.80	32.469	14.31
CCD4									
A	23.658	24.097	272.58	325.80	5340.1	7710.9	29.65	31.242	9.895
B	24.145	24.098	285.90	322.99	2910.1	5292.4	31.44	31.674	6.422
C	23.813	24.107	278.29	322.80	4350.1	6819.4	30.14	31.373	8.647
D	23.605	24.111	273.86	323.74	5220.1	7891.3	29.14	31.137	9.662
E	24.425	24.099	267.83	355.75	5906.3	8016.1	34.41	32.270	13.93
CCD5									
A	23.932	24.093	280.49	320.62	3630.1	6248.0	30.50	31.441	7.088
B	24.361	24.105	289.91	311.61	1980.1	4412.6	31.71	31.749	4.189
C	24.110	24.087	285.76	314.31	2610.1	5295.0	30.97	31.547	5.341
D	23.994	24.085	284.42	314.42	2970.1	5701.7	30.41	31.409	6.009
E	23.502	24.038	267.18	330.10	6210.1	8518.1	29.61	31.156	10.42

Legend:

Column 1- Starting initial zone return air temperature at time 50000 seconds, °C

Column 2- Final steady state zone return air temperature at time 78800 seconds, °C

Column 3- Energy consumption of the cooling coil of the step response of the zone return air temperature for the time period from 50000 seconds to 78800 seconds after the step increase of internal load at time 50000 seconds, kWh

Column 4- Rise time, time for the step response of the zone return air temperature to reach half of its peak value measured from the static value at time 50000 seconds, seconds.

Column 5- Peak time, time for the step response of the zone return air temperature to reach first peak value from time 50000 seconds, seconds.

Column 6- Settling time, time for the step response of the zone return air temperature to reach and stay within +/- 2 % of its final static value, seconds.

Column 7- The % of maximum overshoot from the final steady state zone returns air temperature.

Column 8- The maximum step response zone return air temperature, °C.

Column 9- The mean square error, $°C^2$

Row A- Nichols and Ziegler P+I controller estimates.

Row B- Lopez et al P+I controller estimates based on integral of square error.

Row C- Lopez et al P+I controller estimates based on integral of absolute error.

Row D- Lopez et al P+I controller estimates based on integral of absolute error times time

Row E- Cohen and Coon P+I controller estimates.

7.7 ANALYSIS

7.7.1 The comparison of different controller tuning methods on the control performance of the a/c cooling system

The five common controller tuning methods used in the simulation are compared in relationship to their control performances and energy consumption of the single zone a/c cooling system under step load increase of the internal load. The following are observed from the simulation results.

Accuracy of control

Each of the simulation starts from the same initial conditions and with zero internal loads. The control is allowed to settle for a time period of 50000 seconds before the step increase of the internal load from zero kW to 12.505 kW is imposed. From the simulation results, the initial starting return air temperature, except a few cases, is well controlled within +/- 1°C around the set point temperature of 24°C. Control using the Lopez et al. seeking minimum square error gives the most accurate steady starting return air temperature around the setpoint 24 °C. Cohen and Coon estimate gives relatively inaccurate starting initial return air temperature. Nichols and Ziegler estimate is, in general, not as accurate as than Lopez et al. estimate methods.

As regard to the final steady state value of the return air temperature after a time period of 28800 seconds from the step load increase is imposed, except one case, all simulations show accurate control within +/- 0.5°C. All the five tuning methods give similar results.

Response of control

Lopez et al. controller tuning method seeking minimum square error shows remarkable better and faster response than other tuning methods. The rise time, the peak time and the settling time are faster than those produced by other tuning approaches and the response also shows reasonable stability.

Maximum overshoot and maximum response temperature

Among the tuning methods under study, Lopez et al. estimate seeking minimum square error gives, on average, least maximum overshoot, and hence the least response temperature. On the other hand, Cohen & Coon estimate gives average larger maximum overshoot and response temperature. The other estimates give fairly averages in maximum overshoot and temperature response.

Energy consumption

The energy consumption of the cooling coil for all the simulations is calculated from the following equation for the test period of 28800 seconds after the step increase is imposed at time 50000 seconds.

Energy consumption of the cooling coil =

Σcoil chilled water flow x water specific heat x (coil outlet temperature - coil inlet temperature) x sampling interval ...(7.1)

For each design with control valve resistance equals to 0.015 $kg^{-1}\ m^{-1}$, among the five tuning methods, the maximum energy consumption is higher than the minimum energy consumption by 1 to 15.7%. At control valve resistance equals to 0.03 $kg^{-1}\ m^{-1}$, the range is from 2.8 to 10.6%. With the control valve resistance equals to 0.15 $kg^{-1}\ m^{-1}$, Nichols & Ziegler estimate gives, in general, least energy consumption. Cohen & Coon estimate requires relatively higher energy consumption. The estimates suggested by Lopez et al. give average energy consumption. When the control valve resistance is changed to 0.03 $kg^{-1}\ m^{-1}$, the simulation results show that Cohen & Coon estimate requires less energy than other estimates. This conflicts quite differently from that when the control valve resistance is 0.015 $kg^{-1}\ m^{-1}$. Nichols & Ziegler estimate and Lopez et al. estimate seeking minimum absolute error times time are marginally better than the other two Lopez et al. estimates.in energy consumption.

The results of the analysis in the above show that Lopez et al. estimate seeking the minimum square error is the best controller tuning method to be used for the conditions studied. It has the advantages of better control accuracy, stable and faster response, less maximum overshoot and response temperature, and the energy consumption for control is acceptable.

7.7.2 The objective function for assessing the control performance of the a/c cooling system

From the analysis in 7.7.1, the Lopez et al. controller tuning estimate seeking for minimum error square is the best tuning method to be used for the optimization study of a/c cooling system for the applied conditions. Since the method is based on seeking the minimum square error, the minimum square error and hence the mean square error, should be an appropriate and suitable objective function to be used in assessing the control performance of the a/c cooling system in Hong Kong.

7.7.3 Comparison of control performance at different time constants

As concluded in 7.7.1 and 7.7.2, Lopez et al. controller estimate seeking the minimum square error and hence the mean square error are both the best tuning method and the objective function to be used respectively in the simulation study of optimization in the a/c cooling system in Hong Kong. Therefore, the control performance of different coil sizes at different control valve actuator time constants and valve resistancs will be compared for simulation using Lopez et al. estimate seeking minimum square error.

From Table 7.5, for each cooling coil design, the control performance using Lopez et al. minimum integral square error estimate at control valve actuator time constants equal to 21.5 seconds and 10 seconds shows the following.

- The step response for each cooling coil design with time constant equal to 10 seconds is more reactive and faster than that at time constant equal to 21.5 seconds.

- However, for designs 1, 2, 4 and 5, the energy consumption of the cooling coil when the time constant equals to 10 seconds, is a little bit higher (0.12 to 3.34% higher) than that with time constant equals to 21.5 seconds.. For design 3, the energy consumption, when the actuator time constant equals 10 seconds, is 3.88% less than the one with time constant equals to 21.5 seconds. The control performance is therefore better for a shorter control valve actuator time constant. This is, however, at the average expense of approximately 1.5% higher energy consumption.

This result is further confirmed from the results in Table 7.6 when the control valve resistance is changed to 0.03 $kg^{-1}\ m^{-1}$.

7.7.4 Comparison of control performance at different valve resistances

For each of the cooling coil sizes, at the same control valve time constant and Lopez et al. minimum integral square error controller estimate, the control performance of the a/c cooling system when a control valve resistance of 0.015 $kg^{-1}\ m^{-1}$ is compared with one of 0.03 $kg^{-1}\ m^{-1}$. From Table 7.5 and Table 7.6, the results show that both cases give more or less control accuracy. However, except for the cooling coil design 4, the control performance improves when the control valve resistance is increased to 0.03 $kg^{-1}\ m^{-1}$. This is at the expense of only a few percent higher energy consumption.

7.7.5 The optimization of cooling coil design by step change test

In evaluating the cooling coil design from the control performance of the corresponding single zone a/c cooling control system, the performance measures of the step response when the controller is set with the Lopez et al. minimizing integral square error estimate are compared. The comparison of these measures from Table 7.5 and Table 7.6 are tabulated in the following Table 7.7

Table 7.7: Comparison of cooling coil design against performance measures of the step response of zone return air temperature of the single zone a/c cooling control system, controller estimate based on Lopez et al minimizing integral absolute error

i Control valve resistance= 0.015 kg^{-1} m^{-1}, control valve actuator time constant= 21.5 seconds

Performance measures of the step response	**Control performance**
Accuracy of initial starting temperature at time 50000 s	CCD2>=CCD5>=CCD1>CCD4>CCD3
Accuracy of final steady state temperature at time 78800 s	CCD5>=CCD3>=CCD1>=CCD2>=CCD4
Energy consumption from 50000 to 78800 s	CCD5>CCD2>CCD1>CCD3>CCD4
Rise time	CCD4>CCD3>=CCD5>CCD2>CCD1
Peak time	CCD1>CCD4>CCD3>CCD2>CCD5
Settling time	CCD1>=CCD4>CCD3>CCD2>CCD5
% of maximum overshoot	CCD4>CCD5>CCD2>=CCD3>CCD1
Maximum response temperature	CCD4>CCD5>CCD2>=CCD3>CCD1
Mean square error	CCD4>CD1>=CCD2>CCD3>CCD5

ii Control valve resistance 0.015 kg^{-1} m^{-1}, control valve actuator time constant= 10 seconds

Performance measures of the step response	**Control performance**
Accuracy of initial starting temperature at time 50000 s	CCD2>CCD5>CCD3>CCD1>CCD4
Accuracy of final steady state temperature at time 78800 s	CCD3>=CCD4>=CCD2>=CCD1>=CCD5
Energy consumption from 50000 to 78800 s	CCD5>=CCD3>CCD2>=CCD1>CCD4
Rise time	CCD4>CCD5>CCD1>CCD3>CCD2
Peak time	CCD4>CCD1>CCD2>CCD3>CCD5
Settling time	CCD4>CCD1>CCD2>CCD3>CCD5
% of maximum overshoot	CCD5>=CCD4>CCD3>CCD2>CCD1
Maximum response temperature	CCD4>=CCD5>CCD3>=CCD2>CCD1
Mean square error	CCD4>CD1>CCD2>CCD5>CCD3

iii Control valve resistance 0.03 kg-1 m^{-1}, control valve actuator time constant= 21.5 seconds

Performance measures of the step response	Control performance
Accuracy of initial starting temperature at time 50000 s	CCD1>=CCD4>=CCD3>=CCD2>=CCD5
Accuracy of final steady state temperature at time 78800 s	CCD5=CCD4>=CCD2>=CCD3>=CCD1
Energy consumption from 50000 to 78800 s	CCD2>=CCD3>CCD1>=CCD4>CCD5
Rise time	CCD1>CCD5>=CCD4>CCD3>CCD2
Peak time	CCD1>CCD5>CCD2>=CCD4>CCD3
Settling time	CCD1>CCD5>CCD4>=CCD2>CCD3
% of maximum overshoot	CCD1>CCD4>=CCD5>CCD3>CCD2
Maximum response temperature	CCD1>CCD4>=CCD5>=CCD3>CCD2
Mean square error	CCD1>CD5>CCD4>CCD2>CCD3

iv Control valve resistance 0.03 kg-1 m^{-1}, control valve actuator time constant= 10 seconds

Performance measures of the step response	**Control performance**
Accuracy of initial starting temperature at time 50000 s	CCD3>=CCD1>CCD4>CCD2>=CCD5
Accuracy of final steady state temperature at time 78800 s	CCD4>=CCD2>=CCD5>=CCD3>=CCD1
Energy consumption from 50000 to 78800 s	CCD2>CCD3>CCD1>CCD4>CCD5
Rise time	CCD1>CCD5>CCD4>CCD3>CCD2
Peak time	CCD1>CCD5>CCD2>CCD4>CCD3
Settling time	CCD1>CCD5>CCD4>CCD2>=CCD3
% of maximum overshoot	CCD1>CCD4>=CCD3>=CCD5>CCD2
Maximum response temperature	CCD1>CCD4>=CCD5>=CCD3>CCD2
Mean square error	CCD1>CD5>CCD4>CCD2>CCD3

In assessing the control performance the a/c system, the performance measures of the step response have to be weighed. It will be a function of control accuracy (controllability), response rate and settling (stability), and energy consumption. From Table 7.7, (i) to (iv), the initial and final steady state zone return air temperature under the four conditions of simulation show reasonable and acceptable accuracy. In fact, they are almost within +/- 1°C around the setpoint temperature of 24°C. Hence, the accuracy of control will not be considered in the control assessment. With the control valve resistance equals to 0.015 kg^{-1} m^{-1}, when the control valve actuator time constant equals to 21.5 seconds, the performance from Table 7.7, (i) is analyzed. If equal weighting is assigned to the rise time, peak time, settling time and the maximum overshoot temperature, and also a five point grading system is assigned to the respective performance, the control performance can be quantified. The mean square error as well as the grade points and hence the performance for each of the designs under the specific values of control valve resistance and control valve actuator time constant are tabulated in Table 7.8.

From Table 7.8, at control valve resistance equals to 0.03 kg^{-1} m^{-1}, the control performance based on the five point system agrees exactly with that using the mean square error as the objective function. There is no obvious effect of the control valve time constant. At a control valve resistance equal to 0.015 kg^{-1} m^{-1}, the agreement deviates slightly. The order of control performance among the five cooling coil designs alters with the control valve resistance. The control valve actuator time constant has only slight effect on the agreement.

The control performance of the practical a/c cooling system cannot be measured by the energy consumption. Unless a sluggish control is desired, a fast reactive response requires more energy consumption. In fact, the control performance quite conflicts with the energy consumption.

Table 7.8: Performance of the five cooling coil designs based on the five point grading system against the corresponding mean square error under the specific values of control valve resistance and actuator time constant

Case	Grade point for CCD1	Grade point for CCD2	Grade point for CCD3	Grade point for CCD4	Grade point for CCD5	Performance based on the five point system	Performance based on the mean square error
Control valve resistance =0.015 kg^{-1} m^{-1}& actuator time constant= 21.5 seconds	12	9	12	18	9	CCD4>CCD1= CCD3>CCD2= CCD5	CCD4>CCD1> CCD2>CCD3> CCD5
Control valve resistance =0.015 kg^{-1} m^{-1}& actuator time constant= 10 seconds	12	9	9	20	10	CCD4>CCD1> CCD5>CCD2= CCD3	CCD4>CCD1> CCD2>CCD5> CCD3
Control valve resistance =0.03 kg^{-1} m^{-1}& actuator time constant= 21.5 seconds	20	7	6	11	15	CCD1>CCD5> CCD4>CCD2> CCD3	CCD1>CCD5> CCD4>CCD2> CCD3
Control valve resistance =0.03 kg^{-1} m^{-1}& actuator time constant= 10 seconds	20	7	6	12	15	CCD1>CCD5> CCD4>CCD2> CCD3	CCD1>CCD5> CCD4>CCD2> CCD3

7.8 CONCLUSION

The results of the detailed simulation show that it is capable to improve a/c cooling system design used in Hong Kong by simulation. The a/c system design can be assessed and improved by the control performance of the a/c control system under a simple step test. Besides the size of the cooling coil, the control performance of an a/c system is also affected by the controller settings, control valve resistance and control valve actuator time constant. The simulation study simulates the control performance of five coil sizes for different conventional controller tuning methods. Lopez et al. tuning method seeking minimum integral square error is found the most effective tuning method. It gave the best control performance for a/c cooling control system for typical Hong Kong conditions. The control performance is improved by increasing the control valve resistance, hence the valve authority, at moderate increase of cooling coil energy consumption. When the control valve actuator time constant is decreased, the control response is faster and reactive. The increase in response rate is desirable for a/c cooling system. The increase in the energy consumption is only a few percent for a considerable improvement in the control performance. The selection of cooling coil from the five cooling coil sizes under study by assessing their control performance agrees well with that using their mean square errors resulted from the step test. The order of control performance of the five cooling coils is affected when different value of control valve resistance is used. However, it is not affected by the value of control valve time constant used. Based on these findings, strategies for optimization of a/c system design by simulation are developed and are described in the Conlusions chapter 8.

CHAPTER 8 CONCLUSIONS

The main objective of the research is to highlight the importance of control as a design influence for the sub-tropical environment of Hong Kong. The initiatives of the research focus on control performance and its application in Hong Kong. The use of control performance for assessing the performance of the a/c is innovative and workable. The main concern for an optimum a/c system is that the system should be able to provide the best thermal comfort conditions for the building occupancy at the possible minimum initial and running cost. Control performance measures and weights importantly these factors. The study is based on a/c applications in Hong Kong. The hot and wet weather in Hong Kong defines the application, but the approach can be well applied in other similar countries that require year-round air conditioning.

In an optimization study of an a/c system design, it is necessary to have a tool that can be used to study the control performance of the a/c design during the design stage. The tool for the current study is simulation. The other important area of optimization to be studied is to study and find out the parameters that affect the control performance of the system design before optimization strategies are applied. These objectives have been studied, and are concluded in the following sections.

The use of control performance as a criterion in optimizing the design of a/c system has been studied by simulation. How effectively the simulation can be applied in the optimization study of a/c system design for practical applications was assessed by using actual data of a/c systems installed in Hong Kong buildings in the simulation. This data was collected by a detailed survey of a/c systems and their control used in local commercial buildings.

The simulation package employed in the study is HVACSIM+, a dynamic simulation package for a/c system and control. The mean square error of zone temperature was found to be a suitable objective function for the optimization study because it gives a good measure of control performance. The control performance of an a/c system is affected by characteristics of both the physical parameters (e.g. coil size, control valve) and non-physical parameters (e.g. controller setting, sampling rate) in the system. The understanding of these parameters qualitatively and quantitatively increases the accuracy and hence the robustness of simulation for optimizing a/c system and control design. These control parameters included the control valve and its actuator drive, controller setting and sampling rate in DDC control. The characteristics of control valve and its actuator drive were studied experimentally.

The study gives important and useful data for a/c simulation study at the design stage. The experimental results indicate that an electrical valve is considerably better than pneumatic control valve in a/c system control and this is different from the conventional belief. Therefore, the electric valve, which has an average actuator time constant of approximately ten seconds, should be used in practical installations. The study also shows that the control performance of an a/c system is affected by control valve size. The proper size can be found by adjusting the control valve resistance factor of the control valve model in the simulation for minimum objective function.

With regard to the controller settings, an experimental study shows that for each design of a/c control system, there exists an optimum set of controller settings for that design. Hence, the controller settings for each design should be tuned for optimum control performance. There are various methods available for tuning controllers. They are tested and compared in the detailed simulation. There are also self-tuning controller. The experimental and simulation studies on the algorithms for self-tuning controller were reported in Ref.(8, 9 and 16). Similarly, the experimental study also shows that in DDC, the control performance of the a/c system is affected by the sampling rate used and there is an optimum sampling rate for control performance for each a/c system design. It is not true that the shorter the sampling rate, the better the control. However, the control is normally guaranteed in the short sampling time range. The flexibility and applicability of different objective functions in assessing the control of an a/c system were studied and compared, and it was concluded that mean square error is a good objective function to use.

8.1 THE FLEXIBILITY OF APPLYING SIMULATION STUDY FOR OPTIMUM CONTROL OF PRACTICAL A/C SYSTEM

In the research, the flexibility of applying simulation and the approach to find and evaluate the criterion for optimum a/c design were tested and verified. The simulation study is conducted by using the simulation package HVACSIM+ that provides dynamic simulation for a/c systems. The simulation study starts first with a simple a/c sub-system study and is then finalized by applying the simulation for a typical a/c application in Hong Kong. The data of the typical a/c system is collected by a survey on a/c systems and control used in Hong Kong.

The results of the simulation study reveal that the mean square error is a good and promising objective function in assessing the control performance of a/c systems. It correlates well the control performance of the a/c system. The results also show that it is flexible to study optimization problem of a/c system design by simulation. The simulation is robust and verified for practical a/c system application in Hong Kong against typical summer day in Hong Kong. However, the convergence and effectiveness of simulation depends on how good is the simulation package. In other word, the accuracy of the a/c component models, which depend on the system assumptions made, solving equation and technique as well as the inherent characteristics of the components, of the simulation package affect the accuracy of a simulation. The accuracy of the simulation can be greatly improved if a detailed understanding of the component model, especially the components of the control loop, can be obtained. This addresses why the following study on the parameters that affect the control performance of an a/c design is important. Therefore, the use of practical data is also important to ensure the accuracy of the simulation. That is why, the current research seeks the practical a/c system data by a detailed surrey for the simulation study. The survey on the a/c systems and control used in Hong Kong enhances the simulation study and explores the opportunity of optimization via the design and control strategies. The survey provides detailed information of some typical a/c systems used in Hong Kong. In order to make simulation applicable for use locally and in other countries in the South East Asia, the use of the practical data and conditions collected is important. The data also shows that if the simulation can be used for real applications.

8.2 PARAMETERS AFFECTING THE CONTROL PERFORMANCE OF A/C SYSTEM

The control performance of an a/c system is affected by the physical and non-physical components of the control loop employed in the a/c system. It is vital that the influence of these parameters is understood before optimization strategies and schemes can be applied in the a/c system design. Experimental studies on these parameters were conducted and are described as follows.

The effect of objective function on the control performance of a/c systems was investigated in the simulation studies of a simple a/c cooling sub-system and a practical single zone CAV a/c cooling system mentioned in chapters 5 and 7. In the simulation study of the practical a/c system mentioned in chapter 7, different conventional error based objective functions used in assessing control performance were evaluated and compared. From the comparison, mean square error is found an effective objective function in evaluating the control performance of the air duct system. It strongly agrees with the control response. The goodness of mean square error is further confirmed in the experimental study on the temperature response of the air duct air temperature control system mentioned in chapter 6. It is found in good agreement with control. Hence, it is concluded that mean square error can be used as the objective function for assessing the control performance of a/c systems.

A digital air duct air temperature control rig was built to study experimentally the influence due to sampling rate and controller settings on a/c system. In the study, the controller was first tuned by traditional Nichols and Ziegler open loop test method. With such estimated controller settings set, a step test was applied and the air duct temperature response was measured.

In modern control, electronic digital control is commonly applied in a/c control. Most of the time, a digital controller will control more than a control loop. The number of control loops that can be controlled by a digital controller depends on the sampling rate (time) used. The sampling rate for use in digital control a/c system should be one that can provide an appropriate control without suffering discomfort. The effect of sampling rate on the supply air temperature control was studied experimentally on the digital air duct air temperature control system. The experimental study shows that there exists an optimum sampling rate that can give the best control response for different a/c systems at different working conditions. It is not true, for a/c system, the shorter the sampling rate the better the control response. The optimum sampling rate will depend on the characteristics of system being controlled.

The engineering approaches in the selection of control valve for a/c application has been investigated by the author and reported in Ref.(52). The current research focuses on the experimental study for the effect of control valve and its actuator drive on the control performance of an a/c system. The experimental set-up is an indoor aerodynamic chamber. From the experimental study, it could be concluded that pneumatic control valve has the advantages of faster initial response, stable operation and settling. However, for the same size, the electric control valve has better control performance, better repeatability and close-off than the pneumatic control valve. Though electric control valve gives more oscillatory results in the controlled variable; the oscillation is within the comfort temperature range. It is suitable for control of a/c systems, as the thermal response is relatively stable in thermal a/c systems. The control performance is therefore better than pneumatic control valve because it has a lower mean square error. The study provides useful data on the time constants of control valve actuators to be used in a simulation work. It also indicated that sizing of control valve can be done by adjusting the control valve resistance for minimum mean square error in simulating the a/c system.

The experiments therefore conclude that in optimizing an a/c system design for control performance, electric type control valve should be used for better control performance and the selection of proper valve sizing is prime important for achieving good control performance. Controller settings have to be tuned for optimum performance for any change of design conditions. The experimental study also confirms the usefulness of mean square error.

8.3 THE DEVELOPMENT OF STRATEGIES FOR IMPROVING A/C DESIGN FOR CONTROL PERFORMANCE BY SIMULATION

The research investigates the influence of different parameters in affecting the control performance of a/c systems and the flexibility of applying simulation to study optimization problem of a/c system design. The study concludes that simulation is a promising tool for optimization study and parameters affecting the control performance can be quantified. The optimization of a/c system design comprises of optimization study of the system major physical component designs i.e., cooling coil, control valve and actuator, and the system major 'software-based' components, the controller settings and sampling rate in DDC. Simulation provides good mean to improve an a/c system design during the design stage. From this research study, strategies are developed for improving a/c design based on control performance. It is suggested that simulation package for HVAC systems can be used for improving a/c system designs during design for Hong Kong applications. The strategies for optimizing an a/c system by simulation technique are summarized in step as following.

i. For each of the possible coil sizes to be selected, find the corresponding controller settings by Lopez et al open loop reactive method minimizing integral square error.

ii. Configure the simulation model using the simulation package for HVAC systems for each coil size with all given/estimated initial conditions and physical data. The controller is set with the estimated values obtained from (i) and electric control valve with actuator time constant equal to ten seconds is to be used in the simulation.

iii. Run the simulation for a coil size and compute the mean square error for a specific time period that allows the controlled variable to be stable and steady within a defined limited range.

iv. Repeat the simulation run for other coil sizes set with their own estimated controller settings. Compute the resulting mean square error for the same period of time as mentioned in iii.

v. Compare the mean square errors of all coil sizes. The coil that gives the minimum mean square error will be the selected coil size.

vi. For the selected coil, tune the control valve size by adjusting the control valve resistance (hence the control valve authority) until minimum mean square error is achieved within the specific period of time mentioned in iii.

vii. For the control valve resistance found, obtain the control valve size from the manufacturer's data.

The effect of sampling rate is not considered in the above strategies because only analogue control is considered in this research work. However, the same conceptual strategies could be applied in DDC when the impact of sampling rate is considered.

Regarding the simulation, the accuracy of simulation is greatly affected by system data and state variables of the system input. The accuracy is improved with practical and experimental data. Further work can be extended to focus on improving the accuracy of simulation models by evaluating and tuning models with practical system data and performance.

The simulation also provides platform for studying different optimization methodologies and search techniques used in optimizing those parameters that affect the control performance of a/c systems. Optimization of system component selection and system variable tuning, coupled with system optimization strategies, will provide a global optimum control performance a/c system design.

8.4 FURTHER STUDY

The research in this thesis highlights the importance of control for optimum design of a/c systems for Hong Kong conditions. The influence of different control parameters affecting the control performance of a/c system design was examined experimentally and by simulation one by one. As the analysis of these parameters is one-dimensional, the interaction and influence between these parameters has not been investigated. Optimization of these parameters for optimum a/c design based on control performance is a vigorous mathematical procedure involving multi-dimensional analysis of these parameters. This entire optimization, based on different optimization methodologies and algorithms as mentioned in Chapter 2, such as GAs(3) & (4) , fuzzy rule-based approach[5] and computer search algorithms[8], [9] & [16], would be the main task for next stage study.

The a/c and control survey for Hong Kong buildings as mentioned in chapter 3 revealed that some of the automatic control a/c systems of commercial buildings have been operated for by manual strategies. "Expert" control, which is based on long year operational experience, has not been studied in detail. It is believed that such expert control should be valuable and enhance the design and installation of a/c systems. This kind of artificial intelligence could be useful in giving initial feasible point of optimization search.

Furthermore, the survey showed that most of the a/c control and system designs in Hong Kong are not well considered and properly designed. The treated air from most PAHUs supplying to different AHUs is usually not well balanced. As results, AHUs are overloaded or underloaded with the treated fresh air. The performance of cooling coils is thus affected. In addition, most of the buildings built more than ten years ago did not care about the fresh air requirement of the occupancy. Minimum fresh air in the range of 10 to 15 % of zone supply air is supplied. This gives indoor quality problem. Also only a few percent of a/c systems are equipped and designed for free cooling provision. Hence, outside air is not fully ultized economically for energy saving Therefore; there are plenty of work that can be done to improve the control strategies and design for control performance. These tasks should also be considered for further study.

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APPENDIX 1: QUESTIONNAIRE FOR SURVEY ON AIR-CONDITIONING SYSTEMS

**(Prof. W.K. Chow and Mr. W.F. Ho,
Department of Building Services Engineering,
The Hong Kong Polytechnic University)**

Name of the building: ______________________________

Location of the building: ______________________________

____

Please tick one:

1. The building is a:

☐ curtain walled building
☐ high rise building (height higher than 30 m)
☐ low rise building (height not higher than 30 m)

2. The building is used as a:

☐ commercial building
☐ domestic building
☐ industrial building
☐ institutional building
☐ hotel
☐ composite building

3. The building is of ____________ storeys and each with height ________ m; and ____________ basements each with height __________ m.

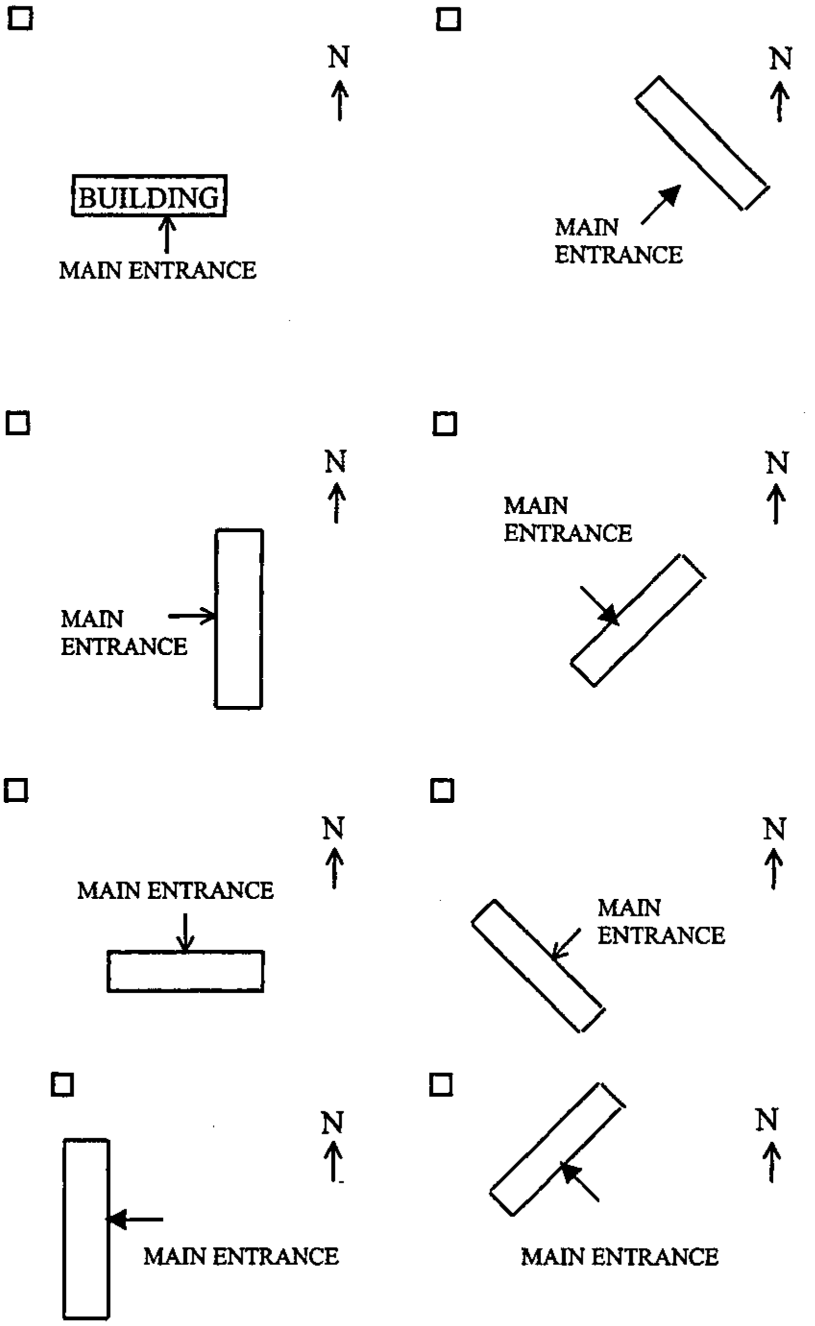
4. The total floor area of the building is _________ m^2.

5. Total floor area of air conditioned space is __________ m^2.

6. The air conditioning system of the building is an:

☐ all- air system
☐ air-and-water system
☐ all- water system
☐ DX split type system (no need to answer the following questions please)
☐ individual window unit system (no need to answer the following questions please)

7. The orientation of the building is:



8. The construction details of the walls are:

	Front view wall	Left wall	Right wall	Back side	Roof	Floor	Ceiling of a typical floor
(a) The exterior surface material is:							
brick, rough plaster	☐	☐	☐	☐	☐	☐	☐
concrete	☐	☐	☐	☐	☐	☐	☐
smooth plaster	☐	☐	☐	☐	☐	☐	☐
glass	☐	☐	☐	☐	☐	☐	☐
(b) Details of the exterior surface is: (please fill the number in Appendix)							
Material	____	____	____	____	____	____	____
Paint	____	____	____	____	____	____	____
(c) Window:							
without window	☐	☐	☐	☐	☐	☐	☐
with window of area	__m^2	__m^2	__m^2	__m^2	__m^2	__m^2	__m^2
(d) The wall surface area: (including window area if applicable):	__m^2	__m^2	__m^2	__m^2	__m^2	__m^2	__m^2
(e) Other parameters: (if applicable)							
Ground reflectivity	____	____	____	____	____	____	____
Solar absorptance of the outer surface	____	____	____	____	____	____	____
Short wave absorptance of the inner surface	____	____	____	____	____	____	____
Emissivity of the inner surface	_	_	_	_	_	_	____
Transmittance of the glass windows	_	_	_	_	_	_	____
Shading coefficient of the glass window	_	_	_	_	_	_	____
(f) Wall construction:							
less than 22 kg/m^2	☐	☐	☐	☐	☐	☐	☐
between 23-199 kg/m^2	☐	☐	☐	☐	☐	☐	☐
between 200-379 kg/m^2	☐	☐	☐	☐	☐	☐	☐
between 380-569 kg/m^2	☐	☐	☐	☐	☐	☐	☐
570 kg/m^2 or above	☐	☐	☐	☐	☐	☐	☐
(g) Air space for wall:							
5 mm	☐	☐	☐	☐	☐	☐	☐
10 mm	☐	☐	☐	☐	☐	☐	☐
20 mm	☐	☐	☐	☐	☐	☐	☐
50 mm	☐	☐	☐	☐	☐	☐	☐
75 mm	☐	☐	☐	☐	☐	☐	☐
100 mm	☐	☐	☐	☐	☐	☐	☐

9. The roof is:

- ☐ flat roof
- ☐ sloped roof 22.5°
- ☐ sloped roof 45°

10. If the air conditioning system is an all-air system (please go to Q12 if it is NOT), it is a:

(a) ☐ Single duct

- ☐ Constant volume
  - ☐ single zone
  - ☐ multiple zoned reheated
  - ☐ bypass
- ☐ Variable air volume
  - ☐ reheat
  - ☐ induction
  - ☐ fan powered
  - ☐ dual conduit
  - ☐ variable diffusers

(b) ☐ Duct duct

- ☐ Dual duct
  - ☐ constant volume
  - ☐ variable volume
- ☐ Multizone
  - ☐ constant volume
  - ☐ variable volume
  - ☐ three deck or Texas multizone

11. Data of air handling systems (A complete schedule if possible, otherwise a typical one).

(a) COIL SECTION

Total load (kW): ____________
Sensible load (kW): ____________
On coil db temp. (°C): ____________
On coil wb temp.(°C): ____________
Water in temp. (°C): ____________
Water out temp. (°C): ____________
Water flow (l/s): ____________
Water pressure drop (kPa): ____________
Air pressure drop (Pa): ____________
No. of passes: ____________
No. of rows: ____________
Fin pitch (mm): ____________
Tube/fin material: ____________/
Air coil face velocity (m/s): ____________
Coil working pressure (kPa): ____________

Type:
- ☐ VDT
  - ☐ side discharge
  - ☐ top discharge
- ☐ HDT
  - ☐ side discharge
  - ☐ top discharge

Insulation material: ____________
Insulation thickness (mm): ____________

(b) FAN SECTION

Air volume (l/s): ____________
Fan static pressure (Pa): ____________
Fan efficiency (%): ____________
Fan diameter (mm): ____________
Fan speed (rpm): ____________
Fan brake power (kW): ____________
Motor rating (kW): ____________
Type of drive: ____________

(c) SYSTEM

☐ VAV SYSTEM
☐ CAV SYSTEM

Number of VAV/CAV boxes in the system: ____________
Inlet diameter (mm) of the system: ____________
Total design air delivery (l/s): ____________
Total pressure drop at design air delivery / at wide open (Pa): ____________
Outlet size, W x H (mm): ____________
Insulation materials: ____________
Insulation thickness/density (mm/kg/cu.m): ____________

12. If the air conditioning system is an air-and-water system (please go to Q16 if this is not), it is a:

☐ air-water induction system
☐ air-water fan coil system

13. Data of air handling units (a complete schedule if possible, otherwise a typical unit).

(a) COIL SECTION

Total load (kW): ____________
Sensible load (kW): ____________
On coil db temp. (°C): ____________
On coil wb temp.(°C): ____________
Water in temp. (°C): ____________
Water out temp. (°C): ____________
Water flow (l/s): ____________
Water pressure drop (kPa): ____________

Air pressure drop (Pa): ____________
No. of passes: ____________
No. of rows: ____________
Fin pitch (mm): ____________
Tube/fin material: ____________/____________

Air coil face velocity (m/s): ____________

Coil working pressure (kPa): ____________

Type: ☐ VDT
☐ side discharge
☐ top discharge
☐ HDT
☐ side discharge
☐ top discharge

Insulation material: ____________
Insulation thickness (mm): ____________

(b) FAN SECTION

Air volume (l/s): ____________
Fan static pressure (Pa): ____________
Fan efficiency (%): ____________
Fan diameter (mm): ____________
Fan speed (rpm): ____________
Fan brake power (kW): ____________
Motor rating (kW): ____________
Type of drive: ____________

14. Induction system (Applicable to induction system only), the coil section data are:

(a) Tube/fin material: ____________
(b) No. of row: ____________
(c) Sensible cooling capacity (kW): ____________
(d) Total cooling capacity (kW): ____________
(e) Air entering temperature db (°C): ____________
wb (°C): ____________
(f) Chilled water flow rate (l/s): ____________
(g) Entering water temperature (°C): ____________
(h) Leaving water temperature (°C): ____________

(i) Working pressure (kPa): ______________
(j) Pressure drop (kPa): ______________

15. Fan coil unit (Applicable to fan coil system only).

(a) COIL SECTION

Tube/fin material: ______________
No. of rows: ______________
Sensible cooling capacity (kW): ______________
Total cooling capacity (kW): ______________
Air entering temperature db (°C): ________
wb (°C): ________
Chilled water flow rate (l/s): ______________
Entering water temperature (°C): ______________
Leaving water temperature (°C): ______________
Working pressure (kPa): ______________
Pressure drop (kPa): ______________

(b) FAN SECTION

Type: ______________
No. of blower ______________
Supply air flow rate ______________
Fan speed at high (rpm) ______________
External static pressure (Pa) ______________

c) MOTOR

Type: ______________
Power supply (v/Ph/Hz): ______________
Power input (w): ______________

16. If the air conditioning system is an all-water system, the data of fan coil unit (a complete schedule if possible, otherwise a typical unit):

(a) COIL SECTION

Tube/fin material: ______________
No. of rows: ______________
Sensible cooling capacity (kW): ______________
Total cooling capacity (kW): ______________
Air entering temperature db (°C): ________
wb (°C): ________
Chilled water flow rate (l/s): ______________
Entering water temperature (°C): ______________
Leaving water temperature (°C): ______________
Working pressure (kPa): ______________
Pressure drop (kPa): ______________

(b) FAN SECTION

Type: ______________
No. of blower ______________
Supply air flow rate ______________
Fan speed at high (rpm) ______________
External static pressure (Pa) ______________

(c) MOTOR

Type: ______________

Power supply (v/Ph/Hz): ____________
Power input (w): ____________

17. The total cooling capacity of the chiller plant is ________________ tons or kW.

    Detail arrangement of chillers are:
    ________ (No.) chillers of ___________tons or kW each.
    ________ (No.) chillers of ___________tons or kW each.
    ________ (No.) chillers of ___________tons or kW each.

18. The heat rejection system for the chiller plant is

    - ☐ Direct seawater cool
    - ☐ Indirect seawater cool
    - ☐ Direct air cool
    - ☐ Indirect air cool
    - ☐ Indirect fresh water cool
    - ☐ Others.Please specify:____

19. The chilled water distribution system is a:

    - ☐ primary-secondary decoupler system (primary circuit of constant chilled water flow and secondary circuit of variable chilled water flow)
    - ☐ direct-return system
    - ☐ reverse-return system
    - ☐ open-loop system
    - ☐ others, please specify

      ____
      ____
      ____

20. The control scheme of the chilled water distribution system is:

    - ☐ decoupler sequence step control with constant chilled water flow in the primary circuit and variable chilled water flow in the secondary circuit.
    - ☐ BTU measurement with differential pressure bypass control.
    - ☐ others, please specify ____________________________________

      ____
      ____

21. The control scheme of the air side of the air-conditioning system is:

	on/off	proportional (P)	proportional + integral (P + I)	proportional + integral + derivative (P+ I +D)	proportional + derivative (P + D)
☐ FCU- room air temperature controlled by thermostat and controller action is:	☐	☐	☐	☐	☐
☐ VAV multizone - (A) Central air handling unit: (i) air handling unit supply air temperature controlled by changing chilled water flow through cooling coil in response to supply air temperature with controller action:	☐	☐	☐	☐	☐
(ii) air handling unit supply air volume controlled by: ☐ frequency inverter ☐ inlet guide vane in response to: ☐ air duct static pressure ☐ air flow measurement ☐ others (please specify) ________________ and controller action is:	☐	☐	☐	☐	☐
(B) Zone VAV box: individual zone temperature controlled by changing supply air volume in response to: ☐ room temperature only ☐ room temperature with humidity override and with controller action:	☐	☐	☐	☐	☐
AND there is: ☐ no terminal reheater ☐ terminal reheater with controller action:	☐	☐	☐	☐	☐

	on/off	proportional (P)	proportional + integral (P + I)	proportional + integral + derivative (P + I +D)	proportional + derivative (P + D)
☐ VAV single zone - (A) Central air handling unit: (i) air handling unit supply air temperature controlled by changing chilled water flow through cooling coil in response to supply air temperature with controller action:	☐	☐	☐	☐	☐
(ii) air handling unit supply air volume controlled by: ☐ frequency inverter ☐ inlet guide vane in response to: ☐ room return air temperature ☐ others, please specify ________ and controller action is:	☐	☐	☐	☐	☐
☐ CAV multizones - air handling unit supply air temperature controlled by changing chilled water flow through the cooling coil in response to: ☐ return air temperature ☐ others, please specify ________ with controller action:	☐	☐	☐	☐	☐
AND there is: ☐ no individual reheater for individual room temperature control ☐ individual reheater for individual room temperature control and the controller action is:	☐	☐	☐	☐	☐
☐ CAV single zone - air handling unit supply air temperature controlled by changing chilled water flow through the cooling coil in response to: ☐ return air temperature ☐ others, please specify ________ with controller action:	☐	☐	☐	☐	☐

22. What are your major concerns about the performance of the air-conditioning system and explain what criteria you are using to evaluate the performance.

APPENDIX of Questionnaire: Materials Number

Material:-

M1	black glass	M9	black concrete	M16	stafford blue brick
M2	red brick	M10	bituminous felt	M17	blue grey slate
M3	roofing, green	M11	brown concrete	M18	asphalt pavement, weathered
M4	wood, smooth	M12	uncoloured concrete	M19	white marble
M5	white mosaic tiles	M13	light buff brick	M20	built-up roof,white
M6	bituminous felt, aluminized	M14	gravel	M21	white on galvanized iron
M7	white glazed brick	M15	polished aluminium reflector sheet	M22	aluminized mylar film
M8	tinned surface				

Paint:-

P1	optical flat black paint	P11	flat black paint	P20	black lacquer
P2	dark grey paint	P12	dark blue lacquer	P21	black oil paint
P3	dark olive drab paint	P13	azure blue or dark green lacquer	P22	dark brown paint
P4	dark blue-grey paint	P14	medium brown paint	P23	medium light brown paint
P5	brown or green lacquer	P15	medium rust paint	P24	light grey oil paint
P6	red oil paint	P16	medium dull green paint	P25	medium orange paint
P7	medium yellow paint	P17	medium blue paint	P26	medium kelly green paint
P8	light green paint	P18	alumimium paint	P27	white semi-gloss paint
P9	white gloss paint	P19	silver paint	P28	white lacquer
P10	laboratory vapour deposited coatings				

Explanation:

1. 'Curtain walled building' means a building which has curtain walls. A curtain wall is a non load bearing wall primarily fixed in front of the structural frame with its own dead weight and wind loads transferred to the structural frame through anchorages.

2. 'High rise building' means any building of which the floor of the uppermost storey exceeds 30 m above the point of staircase discharge at ground floor level.

3. 'Low rise building' means any building of which the floor of the uppermost storey does not exceeds 30 m above the point of staircase discharge at ground floor level.

4. 'Commercial building' means a building, or that part of the building, constructed or intended to be used for business, trade or entertainment.

5. 'Domestic building' means a building constructed or intended to be used for habitation.

6. 'Industrial building' means any building used wholly or in part in any process for or incidental to any of the following purposes, namely:

   (a) the making of any article or part of any article; or
   (b) the altering, repairing, ornamenting, finishing, cleaning or washing or breaking up or demolition of any article; or
   (c) the adapting for sale of any article being a building in which work is carried out by way of trade or for purposes of gain.

7. 'Institutional building' means any building used wholly or in part for the purposes of the following:

   (a) Club premises
   (b) Educational establishings
   (c) Hostels
   (d) Hospitals including mental institutions and clinics
   (e) Prisons and similar corrective institutions
   (f) Sanatoria

8. 'Hotel' means any building used wholly or in part primarily for the purposes of accommodation on a commercial basis.

9. 'Composite building' means any building which is constructed or intended to be used for a combination of any two or more of the following purposes, and in respect of each of these purposes, separate sections of this Code shall apply:

   (a) Domestic
   (b) Commerrcial
   (c) Institutional
   (d) Hotel

10. 'Ground reflectivity' the portion of the radiation striking unit area of a surface ground that is not absorbed or transmitted by the surface.

11. 'Solar absorptance' is the absorbed portion of the solar radiant energy striking a surface.

12. 'Short wave absorptance' is the absorbed portion of short wave radiation striking a surface.

13. 'Emissivity' is the emitted portion of radiation striking a surface. It is a radiation property of a material, evaluated with its surface optically smooth and clean, and of sufficient thickness to be opaque.

14. 'Transmittance' (thermal) is the portion of thermal radiation incident on a surface that is transmitted through the surface.

15. 'Shading coefficient ' is the ratio of the solar heat gain through a particular type of glass under a specific set of conditions to the solar heat gain through that of double strength sheet clear glass under the same conditions.

16. 'An all-air' system provides complete sensible and latent cooling, preheating, and dehumidification capacity in the air supplied by the system. No additional cooling or humidification is required at the zone. Heating may be accomplished by the same airstream, either in the central system or at a particular zone.

17. 'An air-and-water' system conditions spaces by distributing air and water sources to terminal units installed in habitable space throughout a building.

18. 'An all-water' system provides cooling and/ heating of a space by direct heat transfer between water and circulating air.

19. 'A constant volume single zone system' is a supply unit serving a single - temperature control zone.

20. 'A constant volume multiple zoned reheated system' provides (1) zone or space control for areas of unequal loading; (2) simultaneous heating or cooling of perimeter areas with different exposures; and (3) close tolerance of control for process or comfort applications.

21. 'A constant volume bypass system' uses a bypass box in lieu of reheat. The quantity of supply air is varied according to the load variation by bypassing the room.

22. 'A variable volume reheat system' integrates heating at the terminal unit.

23. 'A variable volume induction system' uses a terminal unit to reduce cooling capacity by simultaneously reducing primary air and inducing room or ceiling air (relpaces the reheat coil) to maintain a relatively constant room supply volume.

24. 'A variable volume fan-powered system' is often selected to maintain a higher level of air circulation through a room at low loads while still retaining the advantages of VAV systems.

25. 'A variable volume dual conduit system' is designed to provide two air supply paths, one to offset exterior transmission cooling or heating loads, and the other where cooling is required through-out the year.

26. 'A variable volume variable diffuser system' keeps the discharge velocity relatively constant while reducing the conditioned supply airflow by reducing the discharge aperature of the diffuser.

27. 'A dual duct constant volume system' has two types, namely
    (i) single fan-no reheat: this type of system contains a face-and-bypass damper at the cooling coil, which is arranged to bypass a mixture of outdoor and recirculated air as the internal heat load fluctuates in response to a zone thermostat.
    (ii) single fan-reheat: reheat is applied at the central point.

28. 'A dual duct variable volume system' blends cold and warm air in various volume combinations. These systems may include single-duct VAV units connected to the cold deck for cooling only of interior spaces.

29. 'The multizone system' supplies several zones from a single, centrally located air-handling unit. Different zone requirements are met by mixing cold and warm air through zone dampers at the central air handler in response to zone thermostats. The mixed, conditioned air is distributed throughout the building by a system of single-zone ducts.

30. 'A Texas multizone or three-deck multizone' has the hot deck heating coil removed from the air handler and replaced with an air resistance plate matching the pressure drop of the cooling coil. Individual heating coils are then placed in each perimeter zone duct only.

31. 'VDT' means vertically drawn through.

32. 'HDT' means horizontal drawn through.

33. 'An air-water induction system' is a system in which centrally conditioned primary air is supplied to the unit plenum at medium to high pressure. The air then flows through the induction nozzles and induces secondary air from the room through the secondary coil. This secondary air is either heated or cooled at the coil.

34. 'An air-water fan coil system' may use recirculated air or sometimes up to 100% outside air. The cooling coil is usually selected to provide primary air at a dew point low enough to dehumidify the system totally.

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APPENDIX 2: BRIEF SUMMARY OF HVACSIM+ MODELS USED IN THE SIMULATION STUDY

MODELS

The units used in the models are S.I. units unless otherwise stated.

Type 1 model: Fan or Pump

Brief description

The model computes a pressure rise and efficiency as function of a mass flow rate, using dimensionless performance polynomial curves, with empirically determined coefficients supplied by the user. The efficiency is used to calculate the temperature rise across the fan or pump, as well as the power consumption.

Nomenclature

C_f-	dimensionless flow coefficient
C_h-	dimensionless pressure head coefficient
C_p-	specific heat of fluid (kJ kg^{-1} C^{-1})
D-	diameter of blade or impeller (m)
E-	power consumption (kW)
N-	rotational speed (rps)
P_i-	inlet pressure (kPa)
P_o-	outlet pressure (kPa)
T_i-	inlet temperature (C)
T_o-	outlet temperature (C)
w-	mass flow rate (kg s^{-1})
η-	efficiency
ρ-	fluid density (kg m^{-3})

Mathematical description

$C_f = w / (\rho N D^3)$

$Ch = 1000\Delta P / (\rho N^2 D^2)$

Where ΔP is the pressure rise, Po – Pi, in kPa. The model determines the head coefficient, C_h, and the efficiency, η, as functions of the flow coefficient, C_f:

$C_h = a_0 + a_1 C_f + a_2 C_f^2 + a_3 C_f^3 + a_4 C_f^4$

$\eta = e_0 + e_1 C_f + e_2 C_f^2 + e_3 C_f^3 + e_4 C_f^4$

$P_i = P_0 - 0.001 C_h \rho N^2 D^2$

$E = w\Delta P / \eta\rho$

$T_o = T_i + \Delta P(1/\eta - 1) / \rho C_p$

Component configuration

Inputs	Description	
1	w	mass flow rate
2	P_o	outlet pressure
3	N	rotational speed
4	T_i	inlet temperature

Outputs	Description	
1	P_i	inlet pressure
2	T_o	outlet temperature
3	E	power consumption

Parameters	Description	
1-5	a_i	coefficients for head against flow curve
6-10	e_I	coefficients for efficiency curve
11	D	diameter (m)
12	Mode	air = 1, water = 0

Type 2 model: Conduit (duct or pipe)

Brief description

The model accounts for thermal losses to ambient conditions, transport delays, and dynamics due to thermal capacitance.

Nomenclature

A-	surface area of conduit
C_m-	thermal capacitance (mass*specific heat) of conduit
C_p-	specific heat of fluid
h-	heat transfer coefficient
K-	flow resistance coefficient
P-	pressure
T-	temperature
t-	time
U-	overall heat transfer coefficient
V-	volume of conduit
w-	mass flow rate of fluid
ρ-	density of fluid
τ-	time constant
τ_c-	capacitance term of time constant
τ_x-	transport time

Subscripts

amb-	ambient
i-	inlet or inside
o-	outlet or outside
ss-	steady state

Mathematical description

$P_i = P_0 - \mathrm{sign}(w)Kw^2$

$T_{ss} = T_{amb} + (T_i - T_{amb})\exp(-\tau)$

Where $\Upsilon = UA/wC_p$

If MODE is positive, the temperature dynamics are represented by the following equations:

$dT_0/dt = (T_{ss} - T_0)/\tau$

Where $\tau = [h_i/(h_i + h_0)]C_m/wC_p$

$T_0+ = T_{ss} + (T_0^- - T_{ss})\exp(-\Delta t/\tau)$

This result is used by the DELAY subroutine, which returns the current outlet temperature:

$T_0 = \mathrm{DELAY}(T_0^+, \tau_x)$

$\tau_x = \rho V/w$

If MODE is negative, the temperature dynamics are represented by the following equation:

$dT_0/dt = -\exp(-\alpha)/\tau_c * [T_0 - \mathrm{DELAY}\{T_{ss} + \tau_c\exp(-\Upsilon - \alpha)dT_i/dt\}]$

Where $\tau_c = C_m/h_iA$, $\alpha = \Upsilon h_i/2h_0$

<u>Component configuration</u>

Inputs	Description	
1	w	fluid mass flow rate
2	P_0	outlet pressure
3	T_i	inlet temperature
4	T_{amb}	ambient temperature
5	T_0	outlet temperature

Outputs	Description	
1	T_0	outlet temperature
2	P_i	inlet pressure

Parameters	Description	
1	h_iA	inside surface heat transfere coefficient * area (kW C^{-1})
2	h_0A	outside surface heat transfere coefficient * area (kW C^{-1})
3	C_m	thermal capacitance (mass*specific heat) of conduit (kJ C^{-1})
4	V	volume of conduit (m^3)
5	K	flow resistance coefficient (0.001 $kg^{-1} m^{-1}$)
6	H	height of outlet above inlet (m)
7	Mode	-2 = water, detailed dynamics (one D.E.)
		-1 = air, detailed dynamics (one D.E.)
		1 = air, simple dynamics (one D.E.)
		2 = water, simple dynamics (one D.E.)

Type 3 model: Inlet conduit (duct or pipe)

Brief description

This model calculates the mass flow rate from the inlet and outlet pressures.

Nomenclature

K-	flow resistance coefficient
ΔP-	inlet pressure minus outlet pressure
w-	mass flow rate

Mathematical description

$w = \text{sign}(\Delta P)(\Delta P/K)^{0.5}$

Temperatures are calculated as described for the TYPE 2 conduit model.

Component configuration

Inputs	Description	
1	P_i	inlet pressure
2	P_o	outlet pressure
3	T_i	inlet temperature
4	Tamb	ambient temperature
5	T_o	outlet temperature

Outputs	Descritption	
1	T_0	outlet temperature
2	w	mass flow rate

Parameters	Description	
1	h_iA	inside surface heat transfer coefficient * area (kW C-1)
2	h_0A	outside surface heat transfer coefficient * area (kW C-1)
3	C_m	thermal capacitance (mass*specific heat) of conduit (kJ C^{-1})
4	V	volume of conduit (m^3)
5	K	flow resistance coefficient (0.001 kg^{-1} m^{-1})
6	H	height of outlet above inlet (m)
7	Mode	-2 = water, detailed dynamics (one D.E.)
		-1 = air, detailed dynamics (one D.E.)
		1 = air, simple dynamics (one D.E.)
		2 = water, simple dynamics (one D.E.)

Type 4 model: **Flow merge**

Brief description

This model calculates the mass flow rate and temperature of the single outlet flow stream, and the pressures at the two inlets. The model assumes that the flow resistance parameter is the same for all three branches.

Nomenclature

K-	flow resistance coefficient
P-	pressure
T-	temperature
w-	mass flow rate

Subscripts

1	first inlet
2	second inlet
3	outlet

Mathematical description

$w_3 = w_1 + w_2$

$T_3 = (w_1T_1 + w_2T_2)/w_3$

$P_1 = P_3 + K/2 * \{sign(w_1)w_1^2 + sign(w_3)w_3^2\}$

$P_2 = P_3 + K/2 * \{sign(w_2)w_2^2 + sign(w_3)w_3^2\}$

Component configuration

Inputs	Description	
1	w_1	flow rate at first inlet
2	w_2	flow rate at second inlet
3	P_3	pressure at outlet
4	T_1	temperature at first inlet
5	T_2	temperature at second inlet

Outputs	Description	
1	w_3	flow rate at outlet
2	P_1	pressure at first inlet
3	P_2	pressure at second inlet
4	T_3	temperature at outlet

Parameters	Description	
1	K	flow resistance coefficient (0.001 $kg^{-1} m^{-1}$)

Type 5 model: **Damper or valve**

Brief description

This model represents dampers or valves having inherent characteristics which are linear, exponential, or intermediate between linear and exponential. The characteristic is determined by parameter W_f. When W_f is zero, the model represents an exponential valve or damper. When it is one, a linear valve or damper is modeled. Intermediate values of W_f produce intermediate characteristics. The inlet pressure is a function of mass flow and relative damper or valve position. A non-zero leakage parameter is required to prevent infinite flow resistance when the damper or valve is fully closed. A valve with low authority has little effect on the flow rate over most of its operation range. The effect of authority is accurately calculated by TYPE 5 model.

Nomenclature

C-	relative position
K-	flow resistance coefficient (0.001 $kg^{-1} m^{-1}$)
M-	mode: M = 0 ----damper closed when C = 0
	M = 1 ----damper closed when C = 1
w-	mass flow rate ($kg\ s^{-1}$)
W_f-	weighting factor for linear term of flow resistance coefficient
λ-	leakage parameter: fractional leakage when ΔP = 1kPa

Subscripts

0	full open
1	inlet
2	outlet

Mathematical description

If M is not equal to 0, then C = 1-C

$K = W_f K_0 / [(1-\lambda)C + \lambda]^2 + (1-W_f)K_0 \lambda^{(2C-2)}$.

$P_1 = P_2 + \mathrm{sign}(w)Kw^2$

Component configuration

Inputs	Description	
1	w	flow rate at first inlet (kg s^{-1})
2	P_2	outlet pressure (kPa)
3	C	relative position of damper or valve

Outputs	Description	
1	P_1	inlet pressure (kPa)

Parameters	Description	
1	K_0	full open resistance (0.001 $kg^{-1} m^{-1}$)
2	λ	leakage parameter
3	W_f	weighting factor for linear term of flow resistance coefficient
4	M-	mode: M = 0 ----damper closed when C = 0
		M = 1 ----damper closed when C = 1

Type 6 model: Flow split

Brief description

This model calculates the mass flow rates at the two outlets, and the pressure at the single inlet. The model assumes that the flow resistance parameter is the same for all three branches.

Nomenclature

K- flow resistance coefficient

P- pressure

w- mass flow rate

Subscripts

0	centre of component
1	inlet
2	first outlet
3	second outlet

Mathematical description

$P_1 - P_0 = 0.5K\text{sign}(w_1)w_1^2$

$P_0 -- P_2 = 0.5K\text{sign}(w_2)w_2^2$

$P_0 - P_3 = 0.5K\text{sign}(w_3)w_3^2$

$w_1 = w_2 + w_3$

$w_2 = w_1/2 + [\text{sign}(w_1)]\ \Delta P/Kw_1$ if $S \geq 0$

$w_2 = w_1/2 + [\text{sign}(\Delta P)]\ [\text{abs}(\Delta P)/K - w_1^2/4]^{0.5}$ if $S < 0$

Where $\Delta P = P_3 - P_2$

And $S = w_1^2 - 2\text{abs}(\Delta P) / K$

$w_3 = w_1 - w_2$

When S is positive, all three flow rates have the same sign, and w_1 represents the single inlet (positive) or outlet (negative) flow rate of a split or merge. When S is negative, w_2 and w_3 do not have the same sign, and w1 represents one of two inlets (w_1 positive) or outlets (w_1 negative).

The pressure at the nominal inlet, P_1, is then calculated:

$P_1 = P_1 + K/2 * [\text{sign}(w_1)w_1^2 + \text{sign}(w_3)w_3^2]$

Component configuration

Inputs	Description	
1	w_1	flow rate at inlet
2	P_2	pressure at first outlet
3	P_3	pressure at second outlet

Outputs	Description	
1	w_2	flow rate at first outlet
2	w_3	pressure at second inlet
3	P_1	pressure at inlet

Parameters	Description	
1	K	flow resistance coefficient (0.001 $kg^{-1}\,m^{-1}$)

Type 7 model: Temperature sensor

Brief description

A simple first-order differential equation with a single time constant is used to model a temperature sensor. If a Celsius temperature output is desired, an offset of zero and a gain of one are used. If a control variable between 0 and 1 is required for use as controller input, the offset is the minimum allowable temperature and the gain is the maximum allowable temperature minus the minimum temperature.

Nomenclature

C_i-	temperature input modified by gain and offset
C_0-	output signal (e.g. sensed temperature)
T_i-	temperature input
T_g-	gain
T_0-	offset
Δt-	time step
τ-	time constant

Mathematical description

$C_i = (T_i - T_o)/ T_g$

$dC_0/dt = (C_i - C_0) / \tau$

If τ is less than one second, the differential equation is solved within the subroutine:

IF $(\tau / \Delta t) < 0.05$ OR absolute value of $(C_i - C_0^-) < 10^{-10}$ THEN $C_0 = C_i$

ELSE $C_0 = C_i - (C_i - C_0^-)\exp(-\tau / \Delta t)$

Where C_0^- is the value of C_0 one time step ago.

Component configuration

Inputs	Description	
1	T_i	temperature input
2	C_0	temperature sensor output

Outputs	Description	
1	C_0	temperature sensor output

Parameters	Description	
1	τ	temperature sensor time constant (sec.)
2	T_0	offset (C)
3	T_g	gain

Type 8 model: Proportional-integral controller

Brief description

This model represents an analog proportional-integral controller.

Nomenclature

C-	controller output (0 <= C =<1)
C_f-	feedback signal (controlled variable)
C_{set}-	setpoint
C_{ss}-	steady state controller output
E-	error signal
I-	integral portion of controller output
Ki-	integral gain

K_p- proportional gain

τ- time constant associated with controller response

Mathematical description

$E = C_{se} - C_f$

$dI/dt = K_i E$

$dC/dt = (C_{ss} - C) / \tau$

where $C_{ss} = K_p E + I$

If τ is less than one second, the differential equation is solved within the subroutine:

IF $(\tau / \Delta t) < 0.05$ OR absolute value of $(C_{ss} - C^-) < 10^{-10}$ THEN $C = C_{ss}$

ELSE $C = C_{ss} - (C_{ss} - C^-)\exp(-\tau / \Delta t)$

Where C^- is the value of C one time step ago.

Component configuration

Inputs	Description	
1	C_f	controlled variable
2	C_{set}	setpoint for controlled variable
3	I	integral portion of control signal (from first output)
4	C	output control signal (from second output)

Outputs	Description	
1	I	integral portion of control signal ($1 \geq I \geq 0$)
2	C	output control signal ($1 \geq C \geq 0$)

Parameters	Description	
1	K_p	proportional gain
2	K_i	integral gain
3	τ	controller time constant (sec.)

Type 9 model: **Linear valve with pneumatic actuator**

Brief description

For a linear valve, the flow rate is directly proportional to the valve position when the pressure drop across the valve is constant. The installed characteristic of an inherently linear valve is strictly linear only when the authority is one. A valve with low authority has little effect on the flow rate over most of its operation range. The effect of authority is accurately calculated by the TYPE 9 model. The dynamic response of the actuator is modeled by a first order differential equation. The actuator can be removed by setting time constant and the hysteresis parameter to zero. A non-zero leakage parameter is required to prevent indetermination of valve pressure drop when there is no flow through the valve.

Nomenclature

C-	requested relative valve position ($0 <= C =< 1$)
C_a-	relative actuator position ($0 <= C_a =< 1$)
C_h-	relative valve position ($0 <= C_h =< 1$)
Hys-	fraction of actuator's range over which C_h remains constant when actuator's direction of travel reverses
K-	flow resistance parameter when valve is open ($C_h = 1$)
w	mass flow rate
λ-	leakage parameter; fractional leakage when $\Delta P = 1$
τ-	actuator time constant

Mathematical description

$dC_a/dt = (C - C_a) / \tau$

This differential equation is solved by MODSIM unless τ is less than one second, in which case the differential equation is solved within the subroutine:

IF $(\tau / \Delta t) < 0.05$ OR the absolute value of $(C_{ss} - C) < 10^{-10}$ THEN $C_a = C$

ELSE $C_a = C - (C - C_a^{-})\exp(-\tau / \Delta t)$

Where C^{-} is the value of C one time step ago.

$C_h = \text{HYSTER}(C_a, \text{hys})$

$$P_i = P_0 + \mathrm{sign}(w)Kw^2[(1-\lambda)C_h + \lambda]^{-2}$$

Component configuration

Inputs	Description	
1	P_0	pressure at outlet
2	w	flow rate at first inlet
3	C	input control signal (0 <=C =< 1)
4	C_a	actuator relative position

Outputs	Description	
1	C_a	actuator relative position
2	P_1	inlet pressure

Parameters	Description	
1	K	flow resistance when valve is open (0.001 $kg^{-1} m^{-1}$)
2	τ	actuator time constant (sec.)
3	λ	leakage parameter
4	hys	hysteresis parameter

Type 12 model: Cooling or dehumidifying coil

Brief description

This model represents a circular or continuous finned serpentine heat exchanger with four or more rows in counterflow crossflow configuration. Equations for log mean temperature difference (LNTD) and log mean enthalpy difference (LMHD) which are strictly correct only for counterflow heat exchangers are used. The model accounts for condensation on the outside surface of the coil. There are three possible conditions of the coil: all wet, partially wet, or all dry. The subroutine determines which of these conditions applies, and treats each case separately. Fin efficiencies, heat transfer coefficients, and time constants are calculated within the subroutine from this information.

Nomenclature

A_m-	minimum area for air flow (m^2)
A_0-	external surface area (m^2)
b-	slope of enthalpy vs. temperature line at saturation, $\Delta H/ \Delta T$ ($kJ\ kg^{-1}\ C^{-2}$)
C_f-	correction factor (dimensionless)
C_p-	specific heat ($kJ\ kg^{-1}\ C^{-1}$)
C_m-	total thermal capacitance of coil material ($kJ\ C^{-1}$)
D_i-	tube inside diameter (m)
G-	maximum air mass flux, w_a/A_m ($kg\ m^{-2}\ s^{-1}$)
h-	heat transfer coefficient ($kJ\ kg^{-1}\ C^{-1}$)
H-	enthalpy ($kJ\ kg^{-1}$)
J-	Colburn heat transfer J-factor (dimensionless)
Pr-	Prandtl number (dimensionless)
Q-	heat transfer rate (kW)
Re-	Reynolds number (dimensionless)
T_w-	bulb water temperature (C)
U-	overall heat transfer coefficient, temperature basis ($kW\ m^{-2}\ C^{-1}$)
U_c-	overall heat transfer coefficient, enthalpy basis ($kg\ m^{-2}\ s^{-1}$)
V-	velocity (m/s)
w-	mass flow rate (kg/s)
η-	fin efficiency (dimensionless)
τ-	time constant (s)
ω-	humidity ratio (kg water / kg dry air)

Subscripts

a	air
d	dry
i	inside, inlet
0	outside, outlet
w	water, wet

Mathematical description

$h_w = 1.429[1 + 0.0146T_w]\ V_w^{0.8} D_i^{-0.2}$

$J = C_1 Re^{C2}$

$h_{ad} = J\ G\ C_{p,a}\ Pr^{-2/3}$

C_1 and C_2 are constants for a particular coil, and are calculated as functions of the coil geometry. Under wet coil conditions, the dry coil heat transfer function is modified as follows:

$h_{aw} = C_f h_{ad}$

C_f is a function of the Reynolds number.

Separate correlations are used to determine the dry fin efficiency, η_d and the wet fin efficiency, η_w. The overall heat transfer coefficient includes terms representing the thermal resistance of the tubes, and a constant fouling factor of 5 x 10^{-5}C m^2 / w(3 x 10^{-4}F ft^2/Btu). Under dry coil conditions, the steady state air and water outlet temperature are found by simultaneous solution of the following three equations:

$Q = w_a C_{p,a} (T_{ai} - T_{a0})$

$Q = U A_0 (LMTD)$

$Q = w_w C_{p,w} (T_{w0} - T_{wi})$

$LMTD = [(T_{ai} - T_{w0}) - (T_{a0} - T_{wi})] / [\ln(T_{ai} - T_{w0}) - \ln(T_{a0} - T_{wi})]$

When the coil is wet, enthalpies are used in place of temperatures:

$Q = w_a (H_{ai} - H_{a0})$

$Q = w_w C_{p,w} (H_{w0} - H_{wi}) / b$

$Q = U_c A_0 (LMHD)$

Where $LMHD = [(H_{ai} - H_{w0}) - (H_{a0} - H_{wi})] / [\ln(H_{ai} - H_{w0}) - \ln(H_{a0} - H_{wi})]$

H_{wi} and H_{w0} are saturated air enthalpies calculated at the water temperatures T_{wi} and T_{wo}, respectively, by:

$H_w = H_0 + bT_w$

The quantities H_0 and b are determined iteratively.

Dynamics have been added to the steady state model in a very simple and artificial way:

$dT'_{a0}/dt = (T_{a0} - T'_{a0}) / \tau$

$dT'_{w0}/dt = (T_{w0} - T'_{w0}) / \tau$

$d\omega'_{a0}/dt = (\omega_{a0} - \omega'_{a0}) / \tau$

where $\tau = C_m/U A_0$

Note that the dynamic variables T'_{a0}, T'_{w0}, and ω'_{a0} are used only in the three differential equations given above. Steady state variables are used in determining if the coil is wet or dry.

Component configuration

Inputs	Description	
1	w_w	water mass flow rate
2	T_{wi}	inlet water temperature
3	w_a	dry air mass flow rate
4	T_{ai}	inlet air dry bulb temperature
5	ω_{ai}	inlet air humidity ratio
6	T'_{wo}	dynamic outlet water temperature
7	T'_{ao}	dynamic outlet air temperature
8	ω'_{ao}	dynamic outlet air humidity ratio

Outputs	Description	
1	T'_{wo}	dynamic outlet water temperature C)
2	T'_{ao}	dynamic outlet air temperature (C)
3	ω'_{ao}	dynamic outlet air humidity ratio
4	Q_t	total cooling load at steady state (kW)
5	Q_s	sensible cooling load at steady state (kW)
6	f_w	wet fraction of total surface area

Parameters	Description	
1	I_c	coil type: 0 = flat continuous fins, 1 = circular fins
2	A_p	primary (tube exterior) surface area (m^2)
3	A_s	secondary (fin) surface area (m^2)
4	A_i	internal surface area (m^2)
5	A_m/A_f	ratio of minimum flow area to face area
6	k_f	thermal conductivity of fin material (kW m^{-1} C^{-1})

7	A_f	coil face area (m^2)
8	N_f	number of fins per centimeter
9	N_p	number of tubes per row
10	N_r	number of rows
11	D_0	tube outside diameter (m)
12	D_i	tube inside diameter (m)
13	δ	fin thickness (m)
14	C_m	total thermal capacitance of coil material (kJ C^{-1})
15	S_l	tube longitudinal spacing (in air flow direction) (m)
16	D_f	fin diameter (if $I_c = 1$) or fin length (if $I_c = 0$) (m)
17	D_c	coil depth in air flow direction (m)
18	k_t	thermal conductivity of tube material (kW m^{-1} C^{-1})

Type 15 model: **Room model**

Brief description

In this model, the air mass is divided into two parts: a fully mixed part near the ventilation supply, and a piston flow part near the ventilation exhaust. The fraction of the air mass in these two parts is adjustable. Each part of the air mass exchanges heat with two kinds of room mass: an interior mass and a wall mass. The temperature of each room mass is assumed to be uniform. Two heat flows are also defined: a heat flow into the wall mass, representing conduction through the walls, and a direct flow into the wall mass, representing conduction through the walls, and a direct heat flow into the fully mixed air mass, representing internal gains.

Nomenclature

C_a-	specific heat of air
C_i-	specific heat of interior mass
C_w-	specific heat of wall mass
f-	fraction of air mass which is fully mixed
H_i-	heat transfer coefficient times area of interior mass
H_w-	heat transfer coefficient times area of wall mass

M_i-	interior mass
M_w-	wall mass
Q_i-	heat flow due to internal gains
Q_w-	conduction heat flow due into wall mass
T_{ai}-	ventilation air inlet temperature
T_{a0}-	exhaust air temperature
T_i-	interior mass temperature
T_m-	temperature of fully mixed portion of air mass
T_p-	temperature of fully piston flow portion of air mass
T_r-	average room air temperature
T_w-	wall mass temperature
V-	volume of room air mass
w-	mass flow rate of ventilation air
ρ-	density of air

Mathematical description

$f\rho VC_a\, dT_m/dt \;=\; Q_i + wC_a(T_{ai} - T_m) + fH_w(T_w - T_m) + fH_i(T_i - T_m)$

The piston flow part of the air mass exchanges heat with the interior mass and the wall mass as it moves through the room by the following equation:

$wLC_a\, dT_p(x)/dt \;=\; (1 - f)[H_w\{T_w - T_p(x)\}) + H_I\{T_i - T_p(x)\}$

where L is the total distance moved by the air mass. The solution to this equation is:

$T_p(x) \;=\; T_s \;-\; (T_s - T_m)\exp(-ax/L)$

where $T_s \;=\; (H_wT_w + H_iT_i) / (H_w \;+\; H_i)$

$a \;=\; (H_w + H_i)(1 \;-\; f) / wC_a$

and $T_p(x)$ is the temperature of the air as a function of the distance traveled. The spatial average air mass temperature is approximated by:

$dT'_p / dt = \; w(T'_s - T'_p) / \rho V(1 - f)$

where $T_s \;=\; T_s \;-\; (T_s - T_m)[1 - \exp(-a)] / a$

The room mass temperatures are given by:

$M_wC_w dT_w/dt = Q_w + H_w(T_r - T_w)$

$M_iC_i dT_i/dt = H_i(T_r - T_i)$

$T_r = fT_m + (1-f)T'_p$

The exhausted air temperature is equal to the temperature of the piston flow portion of the air mass at x = L:

$T_{a0} = T_s - (T_s - T_m)\exp(-a)$

A transport delay is applied to the exhaust air temperature using the utility subroutine DELAY with a delay time equal to the flush time for the piston flow fraction of the room volume.

Component configuration

Inputs	Description	
1	w	ventilation air mass flow rate
2	T_{ai}	ventilation air inlet temperature
3	T_m	temperature of fully mixed portion of air mass
4	T_w	wall mass temperature
5	T_i	interior mass temperature
6	T'_p	spatial average temperature of piston flow portion of air mass
7	Q_w	conduction heat flow into wall mass
8	Q_i	conduction heat due to internal gains

Outputs	Description	
1	T_m	temperature of fully mixed portion of air mass
2	T_w	wall mass temperature
3	T_i	interior mass temperature
4	T'_p	spatial average temperature of piston flow portion of air mass
5	T_r	average room air temperature
6	T_{a0}	exhaust air temperature

Parameters	Description	
1	V	volume of room air mass (m^3)
2	M_wC_w	thermal capacitance of walls (kJ C^{-1})

3	M_iC_i	thermal capacitance of interior mass (kJ C^{-1})
4	H_w	heat transfer coefficient times area for the wall mass (kW C^{-1})
5	H_i	heat transfer coefficient times area for the interior mass (kW C^{-1})
6	f	fraction of the air mass which is fully mixed

Type 26 model: Control signal inverter

Brief description

This models an inverting relay. Its output is equal to one minus the input, which has a value between zero and one.

Nomenclature

C_{1}- input control signal ($0 <= C =< 1$)

C_0- output control signal (equals to one minus C_1)

Mathematical description

$C_0 = 1 - C_i$

$1 \geq C_0 \geq 1$

Component configuration

Inputs	Description	
1	C_i	input control signal

Outputs	Description	
1	C_0	output control signal

Parameters	Description
None	

APPENDIX 3: RELATED JOURNAL AND CONFERENCE PAPERS

1. Ho, W.F. & Ng, W.H.

   "Self-setting programmable digital controller"
   Journal of Measurement and Control, The Institute of Measurement and Control, Vol. 22, No. 10, December/January, p. 295-297 (1989/90).

2. Ho, W.F. & Ng, W.H.

   "A self-tuning software package for direct digital controllers"
   ASHRAE Journal, May, p. 40-49 (1990).

3. Ho, W.F.

   "Development and evaluation of a software package for self-tuning of three-term DDC controllers"

   ASHRAE Transactions, Part 1, p. 529-534 (1993).

4. Ho, W.F., Wright, J.A. & Chow, W.K.

   "Simulation study of a simple air-conditioning system for optimum control performance"

   Proceedings of the 1st International Conference on "Energy and the Environment", IMech and Brunel University, Vol. 2, p. 557-564 (1997).

Self-setting software for the programmable digital controller

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Introduction

Direct digital control (DDC) has been used in industrial process control systems for many years. With the development of low-cost microprocessor chip technology, the benefits of direct digital control in the pursuit of accuracy, preciseness, flexibility and cost-effectiveness simultaneously stimulate the application of DDC for commercial applications such as building comfort control. However, the tuning of DDC controllers, which involves the selection of proportional, integral and derivative constants, the controller sampling interval as well as other compensation terms, is complicated and time-consuming for those accustomed to tuning conventional electronic or pneumatic controllers. An automated and reliable optional PID tuning method is obviously a highly desirable tool for control engineers. Such a method could be developed into a software program that could be implemented within the computer-based DDC controllers featuring self-setting of PID values for minimum energy consumption, This can be further developed as a vital aid for commissioning of control loops and thus can dramatically reduce the working time and manpower required for the initial tuning of control systems. In our studies, we have verified the idea by implementing self autocontroller setting software according to the Ziegler–Nichols method on an air duct temperature control system through the use of the ADDA (analog to digital and digital to analog) interface between sensing and activating and the digital computer where digital controller action is implemented.

Equipment set-up and system operation

As shown in Figure 1, the entire system was monitored and controlled by a PC which acted as the programmable DDC controller. The role of the ADDA interface was to convert the analog signal received from the temperature-sensing circuit and power transducer into digital signals and to convert the digital control action signal from the digital computer into analog form to the actuator.

The commanding software was written in BASIC Language which was capable of receiving and evaluating the instantaneous air duct temperature and power consumption of the heater. The error and the corresponding PID control action was then computed and the appropriate control voltage output applied to the actuator which was coupled with the autotransformer in turn to control the supply voltage to the electric heaters, thus maintaining the air duct temperature at the set point.

Ziegler–Nichols open loop method tuning program

This program aimed at automating the entire open-loop step test process and the subsequent numerical computation procedures as devised by Ziegler and Nichols[1] in determining the approximate settings for a PID controller. The flowchart for this program is depicted in Figure 2.

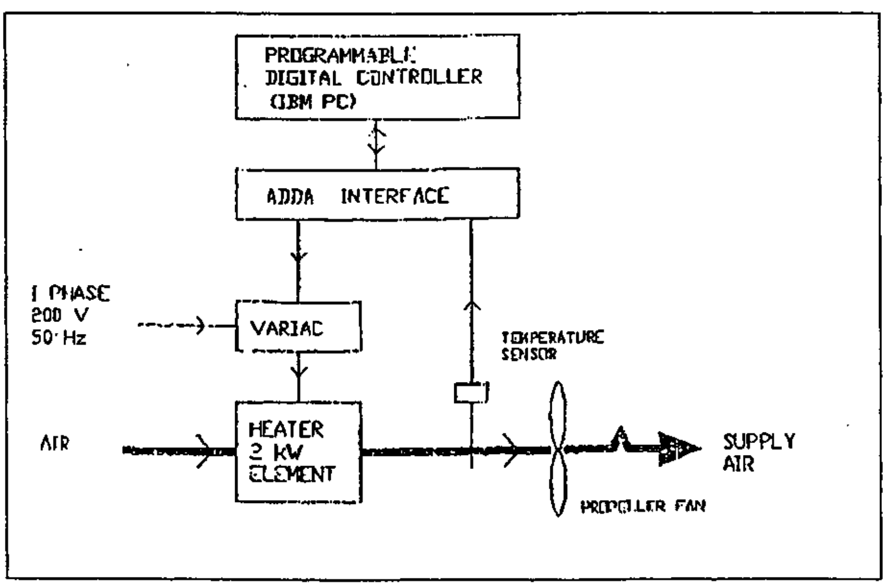

Fig 1 Digital control of air duct temperature

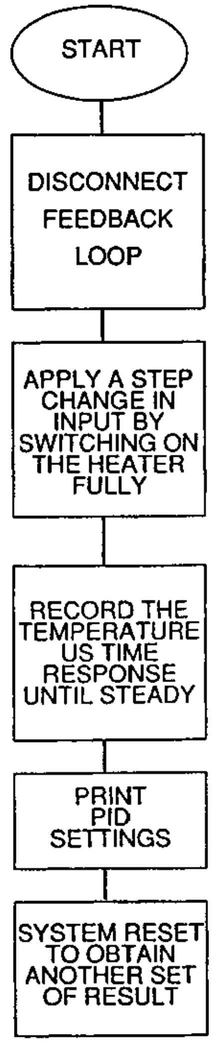

Fig 2 Open loop test method

The open-loop method

This is an empirical method of determining the optimum setting of a controller by performing an open-loop step response test on the actual system. This method has superiority over the model based root-locus or frequency response methods which will be invalid if the transfer function model or frequency response data about the controlled system are not available.

The open loop test method involves opening the control loop and introducing a step change to the input of the controlled process. Data are recorded until a steady state is reached. The three parameters, L—deadtime, Kp—process gain and T—time constant can be obtained from the graphical method as shown in Figure 3 and used to calculate the optimum P, I, D settings using the following formula.

For a PID controller:
proportional gain = $1.2 \times T/(L \times Kp)$
integral time = $2 \times L$
derivative time = $0.5 \times L$

The open loop program

As shown in Figure 2, the open loop program flowchart, the program is composed of the following functional subroutines:

– *Initialization subroutine* to switch on the heater to maximum to stimulate a step change in heating load

– *Data acquisition subroutine* to measure and analyse the temperature response of the air duct system with the temperature sensor with a sampling time of 1 sec.

– *Graphic display subroutine* to display all the temperature readings with respect to time on the graphic screen instantaneously so that the user has an immediate grasp of the testing process.

– *Numerical analysis subroutine* to stop the data acquisition procedure on detection of set-point stabilization. The first and second temperature derivatives with respect to time are then calculated to determine the tangent line of maximum slope. The values of process gain Kp, dead time L and time constant T are determined by geometry and the optimum PID settings are subsequently computed by equation (2):

controller output M(t) =

$$Kp\left[e(t) + Td \times \frac{de}{dt} + \frac{1}{Ti}\int e(t)dt\right] \quad (2)$$

where

Kp = proportional constant
Td = derivative constant
Ti = integral constant
$e(t)$ = error = control variable—set point

and $\frac{de}{dt}$ at $t = kT$ can be approximated

by $\frac{e(k) - e(k-1)}{T}$;

and e(t)dt at $t = kT$ can be approximated by mid-ordinate rule as: T (0.5e(0) = e(1) = ... + $e(K-1)$ = 0.5e(k)) or T ($y(k-1)$ = 0.5e(k)) where $y(k)$ = $y(k-1)$ = $e(k)$ and $y(0) = 0.5\,e(0)$.

Experiment and results

The open loop test was carried out at 23°C with the controller disconnected in the control loop. A step load input of 2 kW was then applied by switching on the 2 kW electric heating element fully. The supply air temperature was recorded and plotted vs time until steady at 37°C as shown in Figure 4. Based on the response curve, the values of PID settings were evaluated according to the Ziegler-Nichols method. These values were: proportional gain = 4, integral action time = 40 sec, derivative action time = 10 sec.

With the control loop closed, these PID values were set to the digital controller and with initial supply air temperature of 27°C, a step load was applied by setting the supply air temperature setpoint to 33°C. The supply air temperature response vs time was plotted in Figure 5.

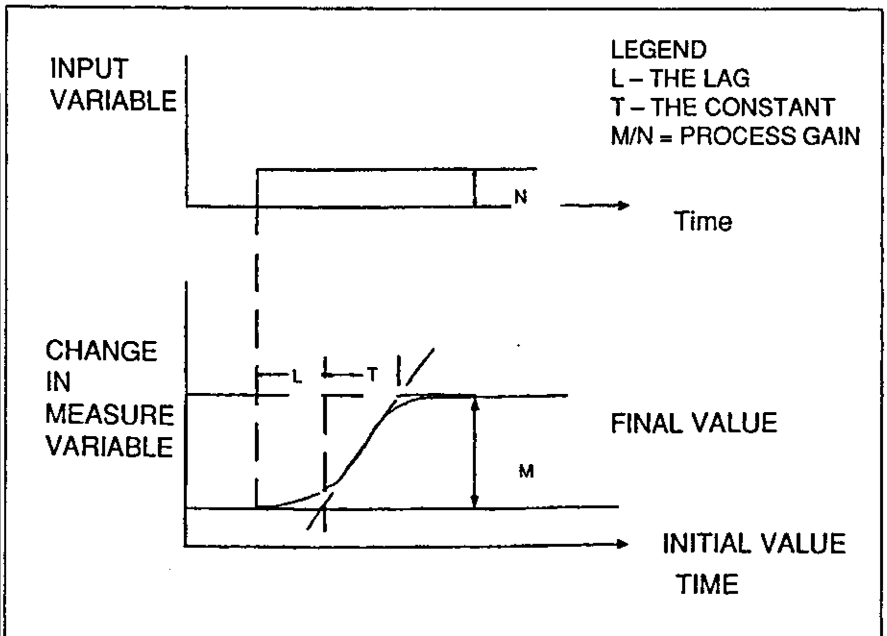

Fig 3 Typical step response curve for open loop test

Analysis of Results

From the response curve shown in Figure 5, we can see that

(i) the control system is stable with only one little overshoot
(ii) the setting time is within 20 min, which is reasonable for a thermal system
(iii) no offset was observed
(iv) from the above, the PID settings are acceptable for the thermal control system.

Discussion

Minimum power consumption strategy

The existing published performance criteria for controller tuning are mostly oriented towards rapid response and minimum overshoot such as the integral error series consisting of minimum integral of square error (ISE), the minimum integral of absolute error (IAE), and the minimum integral of absolute error multiplied by time (ITAE), the Ziegler and Nichols criterion for ¼ decay damping, etc.

The approach taken in this study on the contrary sheds new light on the criterion of minimum power consumption for controller tuning

Software compatibility and development

Regarding the potential of the present software to cater for different control systems where the sensor and actuator interface may vary considerably, simple adjustment to the input and output signal or addition and deletion of channels and extra interfaces will be essential for warranting full compatibility.

The software is now written in discrete files each with different features and functions. Further development to merge the relevant programs is necessary in order to automate the entire self-setting process.

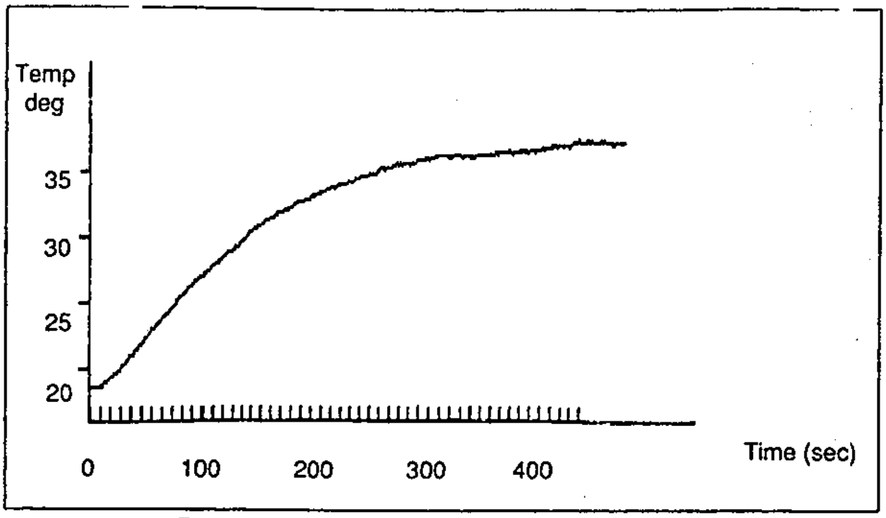

Fig 4 Open loop response of air duct heater system

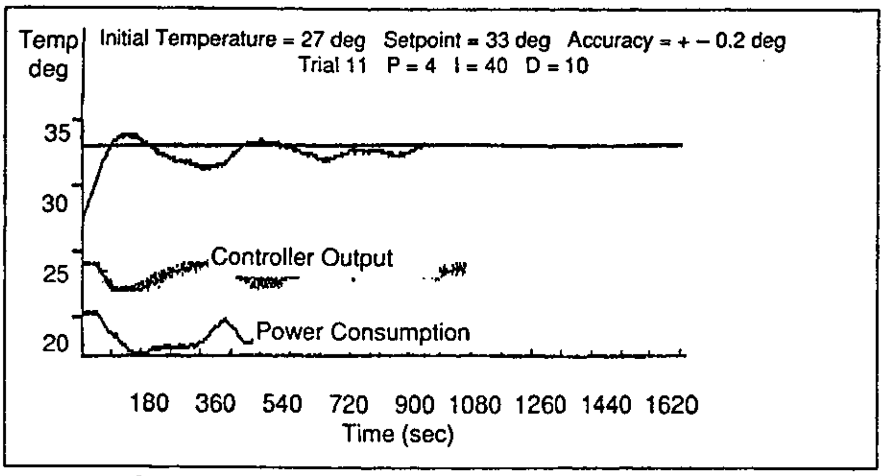

Fig 5 Temperature response of DDC air duct heater system

Application of the self-setting control software package

The package developed in this study will be an indispensable tool for the testing and commissioning engineer or technician during the initial tuning of the newly built or retrofitted control system. It benefits the user by saving the time-consuming repetitive trial and error procedures during the tuning of the DDC controller, at the same time achieving the energy conscious result.

During the running of the system, the self-setting procedure can be used to retune the system on a regular basis to cope with different external conditions and system modification.

The method and approach adopted in the package is notably practical. No complicated transfer function model or frequency analysis of the control system is utilized for system simulation. Instead, the test is intended to be run on the actual control system, thereby precluding the need for understanding the dynamics of the system. Therefore, it is more flexible and simple to establish, understand and handle.

No extra cost for initial equipment purchase is necessary in implementing this software package for a DDC system. It only demands a trained technician to refine the program according to specific applications.

The energy saving advantage will be more pronounced if the concept is introduced to a large operating system where a small increase in performance affects the overall energy bill drastically.

Conclusion

The self-setting software package was tested extensively under the existing control system and found to perform and compute satisfactorily in accordance with the original design intent.

The concept of automation of the complete tuning process has been achieved and accomplished by the DDC computer. The capability to update and retrieve data from the database library and instantaneous display of the various parameters contribute significantly to the success and completion of the study.

Substantial reduction of power requirement during the resetting of the system response against an ambient load change, a disturbance or a change in the set point of the system can be achieved.

It can be concluded also that the commissioning of a programmable digital controller by a self-setting software package using a relatively simple empirical method is possible and acceptable. A self-setting software package using computer searching techniques is being investigated and tested for simple and complicated building services control systems.

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A self-tuning software package for direct digital controllers

This article describes new programs that optimize the P and I settings for simple air duct heating systems

By Wai-fuk Ho and Wai-hung Ng
Student Member ASHRAE

This paper reports on the investigation of the feasibility of adopting self-setting control for a direct digital controller simulated within a PC system. The research work includes the development and application of a series of self-setting computer programs to optimize the P and I settings of a direct digital controller for a simple air duct heating system.

The closed loop response of the DDC system is tested for five different searching algorithms with power consumption as the objective function for the optimization of the P and I setting of the digital controller. The results are compared in terms of searching time and minimum power consumption achieved. The study is confined to the searching of P and I constants because the derivative control affects little on a thermal system due to the large time constant and also the large trial time requirement for three parameter searching. Hence, the D setting is fixed in the study.

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Wai-hung Ng is a graduate trainee in the Building Services Section of the Kowloon Canton Railway Corporation. He received his B.Eng. in building services engineering from Hong Kong Polytechnic.

Successive closed loop testing of the five searching algorithms generated consistent pattern and range of optimum P and I settings but requiring different searching steps and time. It can be concluded that by applying the self-setting software appropriately, a digital controller will be capable of self-commissioning. In addition, substantial energy savings during system initialization can be achieved.

The tuning of the direct digital controllers, which involves the selection of controller settings, the controller sampling time as well as other compensation terms, is often complicated. A need for automation in the tuning of direct digital controllers is obvious.

The prime objective of this research is to investigate the feasibility of self-setting control for DDC systems. The result will be a software program that can be implemented within the computer-based direct digital controllers featuring self-setting of PI values for minimum energy consumption. This can be further developed to fit into any existing energy management or building automation software, forming a vital aid for commissioning of the control loop. It can also dramatically reduce the working time and manpower required for initial tuning of an HVAC&R system.

Equipment set-up/system operation

As depicted in *Figure 1*, the entire system is monitored and controlled by a personal computer that acts as the programmable direct digital controller. The role of the AD/DA interface is to convert the analog signal received from the temperature sensing circuit and power transducer into digital signals. The interface also converts the digital control action signal from the digital computer into analog form to the actuator.

The commanding software is written in BASIC language, which is capable of receiving and evaluating the instantaneous air duct temperature and power consumption of the heater. The error and the corresponding control action is then computed and the appropriate control voltage output applied to the actuator. In turn, the actuator is coupled with the autotransformer to control the supply voltage to the electric heaters, thus maintaining the air duct temperature at the setpoint.

Software programming

Throughout the entire software package, all the programs are written in Basic language because of its simplicity. The self-setting software package developed comprises a total of the five computer programs listed below:

- PID self-setting program—Global Search
- PID self-setting program—One Parameter Search
- PID self-setting program—Random Search
- PID self-setting program—Derivative Search
- PID self-setting program—Hooke and Jeeves Search

The principal features of the self-setting PID tuning program with controller output given in Equation 1 are represented and explained in *Figure 2*.

Controller output:

$$M(t) = Kp \times e(t) + Td \times \frac{de}{dt} + \frac{1}{Ti}\int e(t)\,dt \quad (1)$$

where Kp = Proportional constant
Td = Derivative constant
Ti = Integral constant
e(t) = Error = control variable − setpoint

and $\frac{de}{dt}$ at t = kT can be approximated by:

$$\frac{e(k) - e(k-1)}{T}$$

$\int e(t)dt$ at t = kT can be approximated by mid-ordinate rule as,

$$T(0.5e(0) + e(l) + \ldots + e(k-1) + 0.5e(k))$$
$$= \text{or } T\,(y(k-1) + 0.5e(k))$$

where y(k) = y(k − 1) + e(k) and y(0) = 0.5e(0)

Parameter optimization—searching algorithms

Five computer searching algorithms for PI optimization are employed and compared in this research. They are: global searching algorithm (Box *et al.* 1969); one parameter searching algorithm (Bunday 1984; Box *et al.* 1969); random searching algorithm (Box *et al.* 1969); derivative search algorithm (Bunday 1984; Box *et al.* 1969; Balakerishnan, Neustadt 1964); and Hooke and Jeeves search algorithm (Bunday 1984; Box *et al.* 1969).

The global searching algorithm is the simplest search technique employed, merely by letting each parameter vary in steps over its permitted range in such a way that most combinations of parameters are tried. The value of the objective function is calculated with each combination of parameter values, and the parameters giving the optimum results are selected. As nearly all combinations are tried, this method tends to yield the correct result for a suitable grid step. The drawbacks of this method are that the number of runs and the time required to yield any reasonably accurate result are enormous.

With the one parameter searching algorithm, the searching logic is to perform a series of one parameter searches in which all but one parameter are held constant and each parameter being varied in turn until a "local" minimum is found. Each single parameter search can be done by the bisection method in which each parameter is increased or decreased by the assigned step size until the objective function no longer improves. The step size is then halved and stepped in the opposite direction until the step size is reduced to a specified magnitude. Though involving less trial runs than the global method, this algorithm may locate a local rather than a global optimum.

With the random searching algorithm, the searching principle is to vary the parameters in a random way and calculate the objective function for each combination of trial parameters and select the best objective function. The number of runs needed to locate the optimum depends not only on the starting point chosen but also on chance and, thus, varies a lot.

The popular creeping random search is adopted in this study. Here, the search is initiated from a trial point and random changes are made in all parameter values. If a combination of parameters produces a better objective function value, the random search is restarted about this new point. Subsequent steps are then taken in the direction of improvement until no further improvement is obtained. The average step size varies depending on the success of the preceding trials and will be smaller for increasingly successful trials.

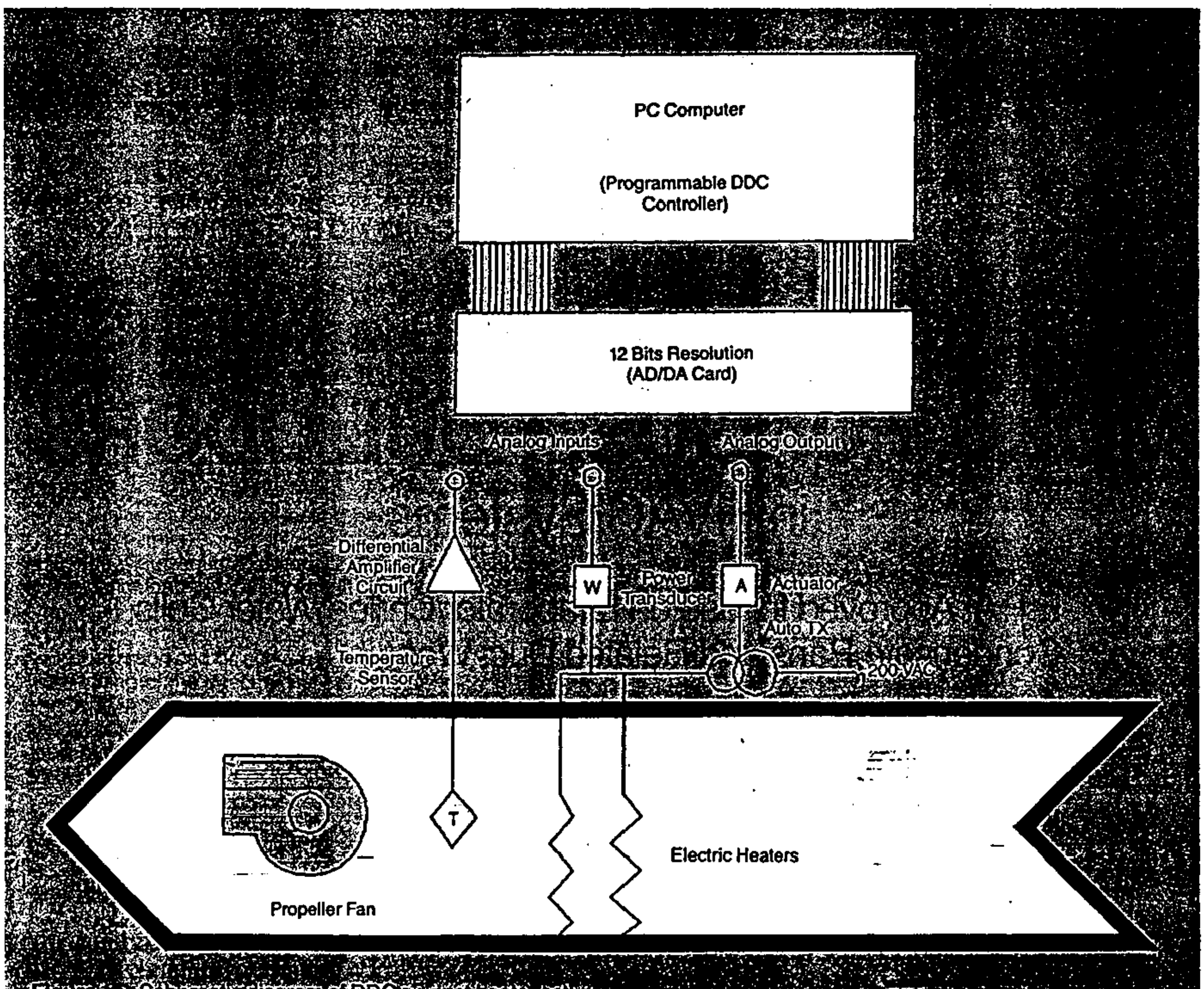


Figure 1. Schematic diagram of DDC air duct heater setup

With the derivative search algorithm, the objective function is calculated for some initial trial values of parameters and small changes in step size are made, in turn, to parameter values. The objective function is evaluated twice more, and partial derivatives of objective function with respect to each parameter can be calculated. The direction of the steepest descent path is calculated from these derivatives and a step is made along that direction by an amount proportional to the slope itself. In case of no improvement at the new point, the gradient is calculated again and a further step is made.

With the Hooke and Jeeves search algorithm, the search comprises two repeating basic operations of simple exploration about a pattern point to determine the best direction in which to move and a pattern move from a base point using knowledge gained in the preceding trial. This method is more direct compared with the derivative method, which involves derivatives evaluation. The merit is that the new set of trial parameter values is extrapolated from current and past values.

During the last three decades, much research in the area of parameters optimization has been done and many papers have been published. Many search routines have been developed, but no universal best method has been determined. It is often necessary to match the method to the problem or try many methods with the same problem to ensure that the correct answer is found.

The search algorithms employed in this study are typical of direct search and gradient methods. They are chosen because they give the correct answer with as little analytical and/or computational calculation as possible. They are fairly simple to understand and can be implemented in a digital computer.

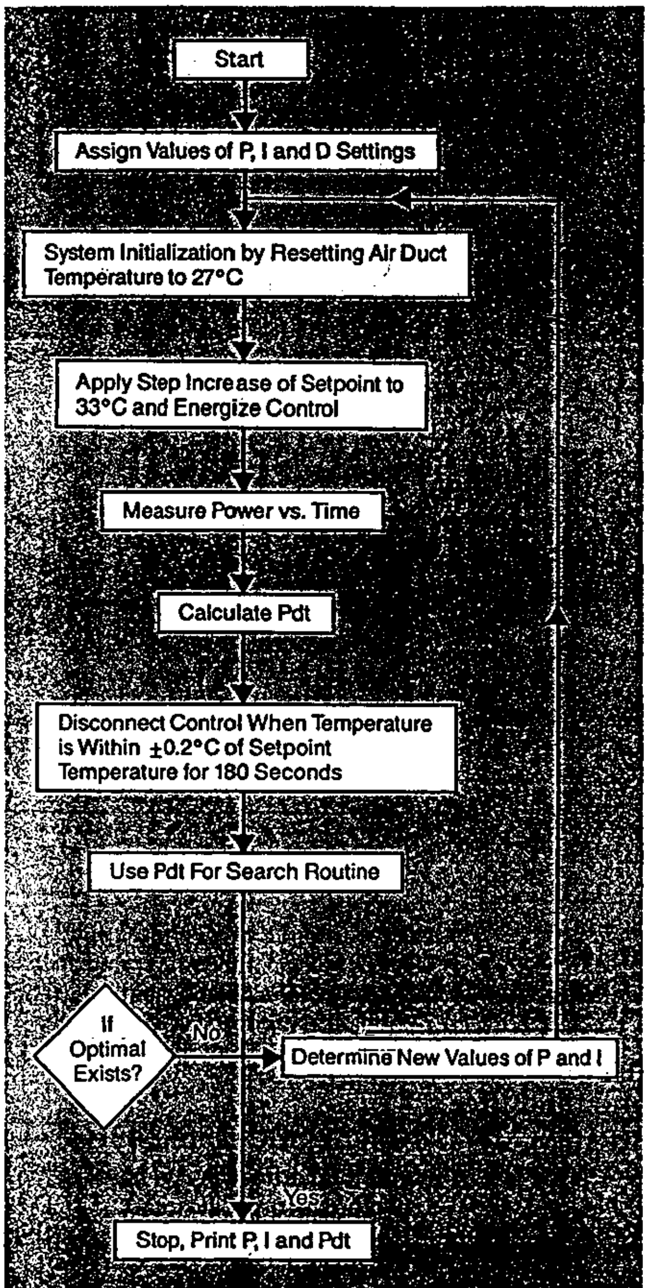


Figure 2. Flowchart for self-setting PID tuning program

PID self-setting control results

The results and the response curves of the various searching methods are summarized in *Table 1* and *Figure 3*. As reflected from the results of the searching algorithms shown in the table, the number of trials, time required and trajectory of searching adopted by different methods differ substantially.

For the global and derivative searching algorithms, the searchings were stopped after 1,501 seconds because no convergence was observed. The energy for that time period was calculated for each trial and a comparison was then made based on the energy calculated for optimization.

The other searching algorithms located the optimum PI settings for energy conservation within 18 trials, where each trial was stopped when an accuracy of ±0.2°C around the setpoint temperature lasting for 180 seconds was obtained. Comparison of the results in *Table 1* suggests the Hooke and Jeeves algorithm is the fastest, most effective method, owing to the least number of trials and time required to find the solution.

However, it should be noted that, even with the same method of searching, the actual trial runs will depend on the exact step size chosen, variations of step size, and initial trials. The probability factor in the case of the random search and Hooke and Jeeves search might not be universally applicable.

Discussion

The existing published performance criteria for controller tuning (Kaya, Scheib 1988; Radke 1988; Cheung 1988) are mostly oriented toward rapid response and minimum overshoot such as the integral error series. This consists of minimum integral of square error (ISE), the minimum integral of absolute error (IAE), and the minimum integral of absolute error multiplied by time (ITAE); and the Ziegler and Nichols criterion for 1/4 decay damping, etc.

On the contrary, the approach taken in this study sheds new light on the criterion of minimum power consumption for controller tuning. This study intends to develop a self-tuning package for the commissioning and tuning of temperature control systems in large commercial buildings. Since hundreds of such control systems are often considered, the need for a tuning package for minimum objective energy is obvious and appropriate. Hence, the criterion of minimum power consumption is adapted in this tuning package.

The following explicit relationships between power consumption and temperature response are established during the study:

- Power consumption increases with overshoot and sustained oscillation around the setpoint during the tuning process;
- Power consumption increases dramatically with sustained offset during the steady state; and

• Power consumption decreases with sluggish response during the tuning process.

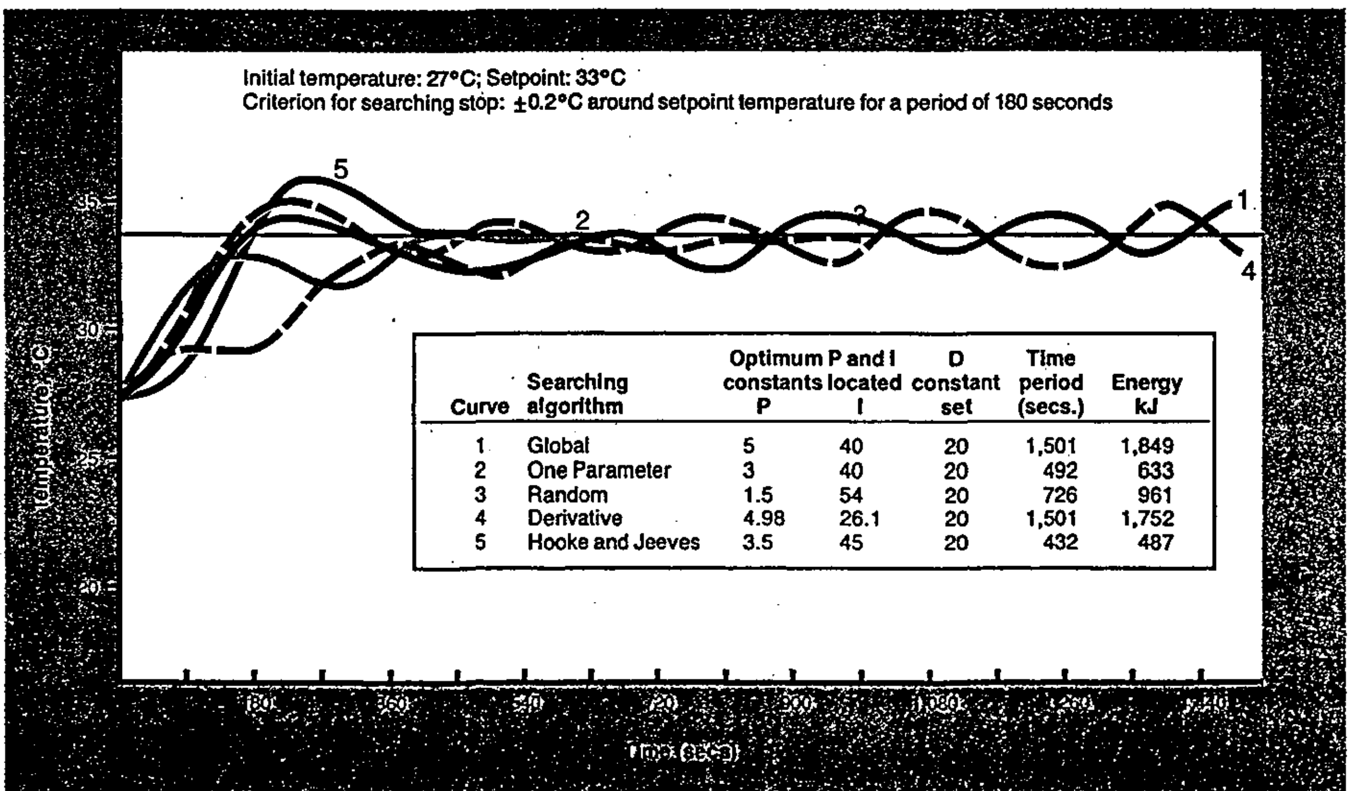
Nevertheless, practical application of the above relationships in pursuing minimum power consumption may be invalid under some exceptional circumstances.

To cite a counter example, consider an optimal start control system where the plants are deliberately started at a certain time interval before the building is occupied to allow the system to reach and settle at the design operating conditions during the occupancy period. In this case, tuning the system to sluggish response criterion in the pursuit of energy conservation may not be feasible and may contradict with the original objective of achieving comfort. Thus, thorough study of the requirements and performance of the system to obtain a compromise between time and energy should be carried out before the implementation of this strategy in a control system.

Being the most preliminary method, the global search method should be selected as the first step to parameter

Table 1. Searching Algorithms Results

Searching Algorithm	Optimum P and I Constants Located P	I	Power Consumption For the Optimum Run kJ	Time Period For the Optimum Run (Secs.)	Number of Trial	Accumulated Power Consumption for Complete Search kJ	Accumulated Time Required For Complete Search (Secs.)	Remarks
Global	5	40	1,849	1,501	18	34,983	27,018	Not converge, stopped at 1,501 secs. for energy comparison
One Parameter	3	40	633	492	6	28,523	20,405	
Random	1.5	54	961	726	18	31,459	22,770	
Derivative	4.98	26.1	1,752	1,501	19	35,100	28,253	Not converge, stopped at 1,501 secs. for energy comparison
Hooke and Jeeves	3.5	45	487	432	11	14,299	12,976	



Curve	Searching algorithm	Optimum P and I constants located P	I	D constant set	Time period (secs.)	Energy kJ
1	Global	5	40	20	1,501	1,849
2	One Parameter	3	40	20	492	633
3	Random	1.5	54	20	726	961
4	Derivative	4.98	26.1	20	1,501	1,752
5	Hooke and Jeeves	3.5	45	20	432	487

Figure 3. Optimum PID control response results for different searching algorithms.

DDC software package

searching. This method attempts to evaluate the objective function extensively throughout the entire parameters universe. Disregarding the time factor, it will be a good start to locate the region of the possible minimum for subsequent testing by other precise search methods.

The other four searching methods are characterized by their own set of searching algorithms and, hence, inherent advantages. It would be best to develop a unique algorithm that can aggregate the advantages of the different searching methods. This new algorithm may possess the following features:

- Location of a global optimum instead of a local by suitable selected initial trial and variable step size in successive trials. (Although convergence to local minimum did not occur in this study, it may occur in certain control systems under optimum search.)
- Intelligent enough to evaluate the best direction to propagate to achieve the optimum objective function.
- Capable of stopping the searching process at the right time.

The experimental results showed that the Hooke and Jeeves method is the most desirable searching algorithm because of the least number of trials and time required for the searching. The searching time for this method is about 216 minutes, which is well acceptable for thermal systems. If this method is adapted for optimum PI searching, a setting of 300 minutes for the searching time limit will ensure that the searching will be completed.

Regarding the potential of the present software to cater to different control systems where the sensor and actuator interface may vary considerably, simple adjustment to the input and output signals or adding/deleting channels and extra interfaces will be essential for warranting full compatibility. The software is now written in discrete files, each with different features and functions. Further development to merge the relevant programs together is necessary to automate the entire self-setting process.

The self-setting software package developed in this study will be an indispensable tool for the testing and commissioning engineer or technician during the initial tuning of the newly built or retrofitted control system. It benefits the user by saving the time-consuming, repetitive trial-and-error procedures during the tuning of the direct digital controller and, at the same time, achieving the energy conscious results. During the running of the system, the self-setting procedure can be used to retune the system on a regular basis to cope with different external conditions and system modifications.

The method and approach adopted in this self-setting software package are notably practical. No complicated transfer function model or frequency analysis of the control system is utilized for system simulation. Instead, the test is intended to be run on the actual control system, discarding the need for understanding the dynamics of the system. Therefore, it is more flexible and simple to establish, understand and handle.

No extra cost for initial equipment purchase is necessary in implementing this software package for a DDC system. It only needs a trained technician to refine the program according to specific applications.

The energy saving advantage will be more pronounced if the concept is introduced to a large operating system where a small increase in performance affects the overall energy bill drastically.

Conclusion

The different programs in this self-setting software package were tested extensively under the existing control system and found to perform and compute satisfactorily according to the original design intent.

DDC software package

The concept of automating the complete tuning process has been achieved and accomplished by the DDC computer. The capability to update and retrieve data from the database library and the instantaneous display of the various parameters contributed significantly to the success and completion of the study.

Substantial reduction of power requirements during the resetting of the system response against an ambient load change, a disturbance or a change in the setpoint of the system can be achieved by this self-setting software package.

Recommendations

For further development of the existing software package, it is recommended that the open loop program based on Ziegler and Nichols empirical open loop step response test (Ziegler, Nichols 1945; Blickley 1988; Ho, Ng 1989) be run first to determine the approximate settings. Then run the global search to locate the probable minimum region and avoid the risk of local minimum. The final step is to select the best searching method from the four remaining algorithms or use a compromise method for parameter searching.

The test results could be more solid and consistent if the following modifications and improvements are made to the existing system:

- The ambient environment in the testing system should be kept constant throughout the tuning process by placing the test rig in a properly controlled, air-conditioned area.
- The external surface of the air duct should be insulated so that heat exchange of the air duct with the ambient during the test can be kept to a minimum.
- The test should be run with a larger size propeller fan or electric heater to speed up the heating process and hence the system response. The current system takes more than 20 minutes to reach and settle around the setpoint. Therefore, it requires enormous time for repetitive tuning during the searching process.
- The effect of varying sampling time and also dead time on the power consumption can be another subject of study besides the PID settings.
- The variation of the three parameters (P, I and D) together with respect to the minimum power consumption will also be an interesting subject of research despite the tremendous combination trials required.

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DEVELOPMENT AND EVALUATION OF A SOFTWARE PACKAGE FOR SELF-TUNING OF THREE-TERM DDC CONTROLLERS

W.F. Ho

ABSTRACT

A computer software package for self-tuning of digital PID controllers was developed using a computer searching technique for optimization (Bunday 1984). The software program developed was verified against a simulated room model (Borrensen 1981) and the performance was satisfactory in terms of manpower saving in the initial tuning and energy saving during control.

INTRODUCTION

Conventional controllers require controller settings to be tuned manually during system startup. Retuning is necessary if there are system modifications or changes in process conditions. Manpower is required to retune these controllers, and errors in the manual tuning of these controllers can cause unsatisfactory control and lead to a waste of energy. The use of programmable direct digital control (DDC) makes possible self-tuning of digital controllers using computer optimization search technique. DDC is computer software based, hence initial tuning and retuning can be automated. In this study, self-tuning of a digital controller using a computer search technique was investigated. Previous work on self-tuning for PI (proportional plus integral) control using computer search techniques has been carried out by Ho and Ng (1989, 1990). The current investigation extended the work to self-tuning of a PID (i.e., proportional, integral, and derivative) DDC controller, which was then verified against the air temperature control of a simulated room model. Although it has been stated (Ho and Ng 1990) that the derivative term had little effect on thermal control systems, such systems were used to develop and verify the PID, self-tuning software. Despite the tremendous searching time required (131 minutes in the present study of a simulated room temperature control system), the derivative term can increase the system effective damping, thereby improving the stability of thermal control systems. The main objective of the study is the development of a PID, self-tuning package that can be applied to other control systems.

DEVELOPMENT OF A SELF-TUNING SOFTWARE PACKAGE FOR OPTIMUM P, I, and D SETTINGS

Work by Ho and Ng (1990) suggested that the Hooke and Jeeves search was an effective search technique for optimum digital PI controller settings; therefore, it was employed in this study for searching optimum PID values. The same air duct temperature control system (Ho and Ng 1990) was used. The equipment setup and system operation is described in the Appendix. The flowchart of the complete software package for optimum PID search is given in Figure 1 and consists of the following subroutines.

PID Control Subroutine

The PID control was implemented by the following:

$$c(t) = K_p \cdot e(t) + \frac{1}{T_i}\int e(t)\,dt + T_d \cdot \frac{de(t)}{dt}, \tag{1}$$

where

c = controller output (V),
K_p = proportional constant (V/°C),
T_i = integral constant (s·°C/V),
T_d = derivative constant (s·V/°C),
$e(t)$ = error (setpoint - control variable) (°C).

The $de(t)/dt$ can be approximated by $[e(k) - e(k-1)]/T$, where T is the sampling time in seconds, and $\int e(t)dt$ at $t = kT$ can be approximated by the mid-ordinate rule as $T\,[0.5e(0) + e(1) + ... + e(k-1) + 0.5e(k)]$ or $T\,[y(k-1)+0.5e(k)]$, whereby $y(k) = y(k-1) + e(k)$, and $y(0) = 0.5e(0)$.

Ziegler and Nichols Open-Loop Tuning Program

This program developed by Ho and Ng (1989) automated the entire Ziegler and Nichols open-loop test (Ziegler 1945) for a step increase of air duct setpoint temperature from 27°C to 33°C and the subsequent numerical computation for PID values. The results were used as the

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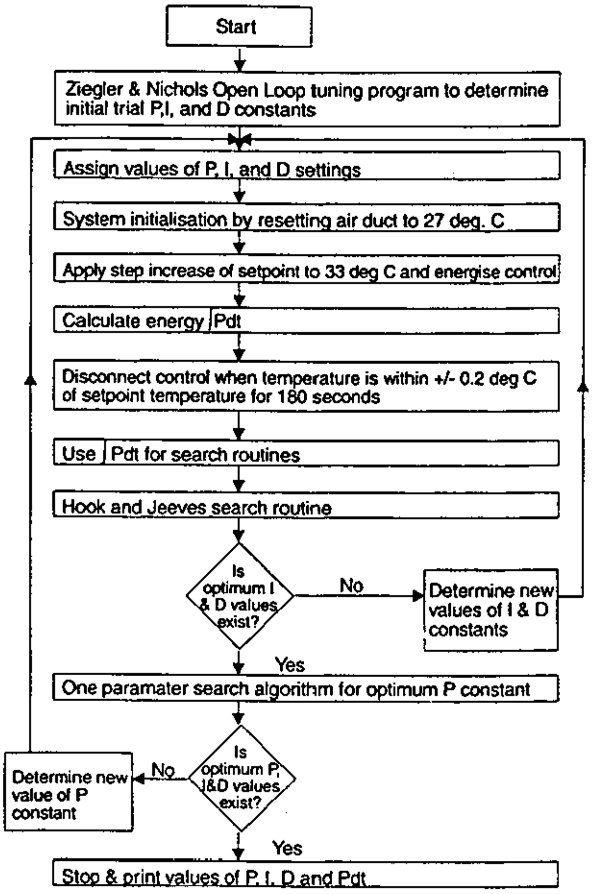

Figure 1 *Flowchart of self-tuning PID tuning program for air duct temperature control system.*

initial predicated PID values for the three-parameter optimum search routine.

Search Algorithm Subroutines

The search algorithm was the working software for the optimum PID values search. The objective criterion of PID optimization was to obtain the combination of PID values that consumed minimum power energy for a step load change of setpoint. Setting the DDC controller at these optimum values would save the "most" energy during control. The working logic of the search algorithm was first to search for the "best" I and D values at fixed P value, using the search algorithm. The next stage was to change the P value and repeat the first stage for the "best" I and D values. This stage, using one parameter search algorithm for optimum P value, would stop when the optimum set of PID values were located. The two search subroutines were described as below:

(i) Hooke and Jeeves Search algorithm for optimum I and D constants at a constant P value: The purpose of this search was to find out the optimum I and D values at a fixed P value. The search (Bunday 1984) was composed of two repeating operations of simple explorations about a pattern point to determine the best direction in which to move and a pattern move from a base point using knowledge gained in the preceding trial. The objective of the search was for minimum energy consumption, and details of each search are given below.

Input: Step increase of air duct setpoint temperature from 27°C to 33°C.
Objective function of the search: Energy consumption.
Stability criterion for stopping search: Air duct temperature within ± 0.2°C of setpoint (i.e., 33°C) for a period of 180 seconds.

Direct measurement of power energy consumption used for raising the air duct temperature was achieved through a power transducer; hence energy consumption was used as the objective function in the search to save computation time rather than using conventional objective functions, such as ISE (integral square-error), IAE (integral absolute-error), or ITAE (integral-of-time-multiplied absolute-error). The search was restarted at 27°C after waiting for the system to cool down. The system was artificially cooled by a fan (blowing ambient air at 17.5 ± 2°C) to avoid lengthy waiting times. The waiting time for each trial was less than one minute. The stability criterion used to stop the search was arbitrarily chosen. Since only small fluctuations around the air duct temperature setpoint were observed, the stability criterion chosen was justified.

(ii) One-parameter search algorithm for optimum P value search giving optimum set of PID values: The purpose of this search was to find out the proportional value at which the optimum combination of PID values was obtained. The search (Bunday 1984) was done by the bi-section method in which the parameter was increased or decreased by the assigned step size until the objective function no longer improved. The step size was then halved and stepped in the opposite direction until the step size was reduced to a specified magnitude.

The two searches (i) and (ii) were run in alternate steps until the optimum set of P, I and D values was found.

VERIFICATION OF THE SEARCH SOFTWARE PACKAGE BY TESTING AGAINST A ROOM TEMPERATURE CONTROL MODEL

The search software program was verified by testing against a simulated digital PID) room temperature control system. The PID temperature control loop of the model room consisted of four components, PID DDC controller, control valve, cooling coil, and room. Using the design conditions of the hypothetical air-conditioning system and room mentioned in previous studies (Ho 1987; Borresen 1981), the dynamic models of these components were developed and described by Ho (1987) and Borresen (1981). Since the time constant (13.3 minutes) of the room temperature responding to the conditioned supply air temperature was more than four times larger than that of the air side components, only the steady state relationships of the air components were used here. The mathematical relationships of components of the room temperature control system are summarized as follows:

Room

$$\Delta T_r / \Delta T_{is} = G_1(s) = 0.87/(13.3s + 1), \quad (2)$$

where

Δ = change,
T_r = room temperature (°C),
T_{is} = air inlet supply temperature (°C),
s = complex variable.

Cooling Coil

$$\Delta T_{is}/\Delta F_w = K_1 = -35°C \cdot s/kg, \quad (3)$$

where F_w = chilled-water mass flow (kg/s).

PID Controller

$$\Delta c/\Delta e = G_{2(s)} = [K_p + 1/T_i \cdot s + T_d \cdot s]. \quad (4)$$

Control Valve

$$\Delta F_w / \Delta c = K_2 = -0.472\ kg/s \cdot V. \quad (5)$$

Hence the overall open-loop transfer function of the system is:

$$\Delta T_r/\Delta e = [K_p + 1/T_i \cdot s + T_d \cdot s] \cdot (-0.472) \cdot (-35) \cdot (0.87)/(13.3s + 1). \quad (6)$$

The block diagram of the closed-loop control of the room temperature is shown in Figure 2.

With the estimated "optimum" set of PID values obtained from the search above, a step increase of 1°C of the room temperature setpoint T_{rs} from 22°C to 23°C was applied. The resulting response of the room temperature against time was observed until stability occurred. The ITSE (integral-of-time-multiplied square-error criterion) objective function was then computed.

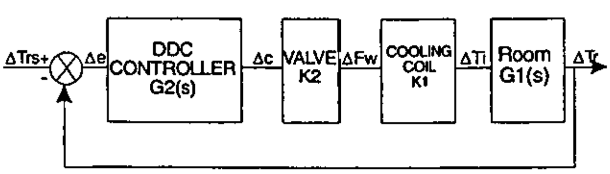

Figure 2 *Block diagram of the room air temperature control system.*

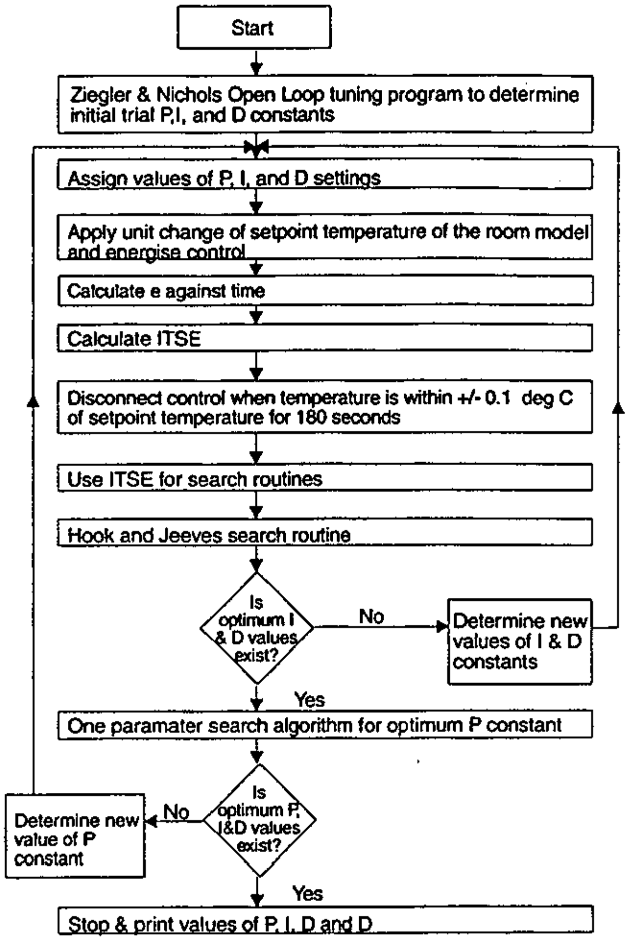

Figure 3 *Flowchart of self-tuning PID tuning program for model room temperature control system.*

The above was repeated with another set of PID values, as dictated by the PID search routine, until the optimum set of P, I, and D) values was located.

The flowchart of the program is shown in Figure 3. The details of the test were as follows:

Input: Unit step increase in room setpoint temperature from 22°C to 23°C.
Objective function of the search: ITSE.
Stability criterion for stopping each trial of search: Room temperature within ± 0.1°C of setpoint temperature (23°C) for a period of 180 seconds.

Direct measurement of cooling energy was not available for the room temperature control model. Therefore, ITSE was used as the objective function because of its simplicity to save complicated calculation. The ITSE criterion stipulated that in the unit step response test, a large initial error was weighed lightly, while errors occurring late in the transient response were more heavily penalized. Hence the use of ITSE was appropriate for the simulated room temperature control system. PID control was applied to the simulated system, and no offset in the room temperature was expected.

However, a small tolerance of ±0.1°C around the new setpoint room temperature, which lasted for a period of 180 seconds, was used as a settling criterion. This was imposed to allow for any occurrence of disturbance in practical application.

RESULTS

Self-Tuning Control for Optimum P, I, and D Settings of Air Duct Temperature Control System and Room Temperature Control System

The results of the searches are given below.

Air Duct Temperature Control System

(i) Open-loop test results:
Energy consumption for the open loop test: 220 kJ
Time required for the open loop test: 112 seconds
Optimum values of P,I and D: $K_p = 5$ V/°C
$T_j = 40$ s·°C/V
$T_d = 22.5$ V·s/°C

The resulting air duct temperature response for the step increase of air duct setpoint temperature from 27°C to 33°C using PID values found is shown in Figure 4. From the response curve, the energy consumption in the system control was 3552 kJ, and a period of 1172 seconds was required for the system to become steady.

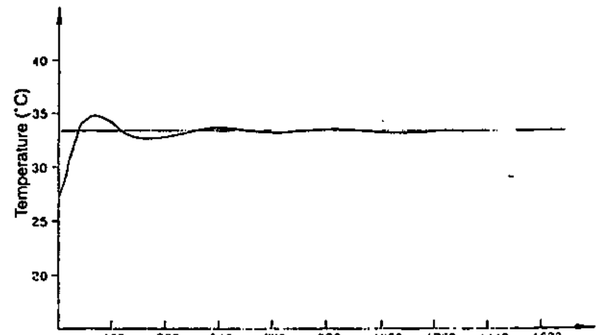

Figure 4 *Temperature response of the air duct system using PID values obtained by the open loop test. Initial temperature = 27°C. Setpoint = 33°C. Trial I, P = 5 V/°C, T_i = 40 s·°C/V, T_d = 22.5 s·V/°C. Total energy = 3552 KJ. Accuracy = ±0.2°C.*

(ii) Optimum search algorithm results:
No. of trials: 44
Total time spent in the searches (including waiting time for restart): 705 minutes
Total energy consumption in the searches: 127206 kJ
Optimum values of P,I and D: $K_p = 1.625$ V/°C
$T_i = 37.5$ s·°C/V
$T_d = 12.5$ V·s/°C
Energy consumption in system control at the optimum trial: 857 kJ
Time spent for the optimum trial to become steady: 241 seconds

The resulting air duct temperature response for the step increase of air duct setpoint temperature from 27°C to 33°C at the optimum PID values found is shown in Figure 5.

Room Air Temperature Control Model

No. of trials: 9
Total time spent for the searches (excluding the waiting time): 86 minutes
Estimated total waiting time for restarting searches: 45 minutes
Optimum values of P, I, and D: $K_p = 2$ V
$T_i = 20$ s·°C/V
T_d 12.5 V·s/°C
Time spent for the optimum trial to become steady: 20 minutes

The resulting room temperature response for the step increase of room setpoint temperature from 22°C to 23°C at the optimum PID values found is shown in Figure 6.

DISCUSSION

Stability of Control Systems

From the response curves shown in Figures 5 and 6, for both the air duct and simulated room temperature control systems set with optimum PID values, it was found that

— the control systems are stable,
— no offset from the setpoint is observed, and
— the settling times are acceptable.

System Response

From the response curves of both systems, it was observed that the system response of the air duct temperature control system is faster than that of the room temperature control system as expected due to the thermal mass of the air inside the air duct is much smaller than that inside the room.

Search Algorithm

In the search, the P value is fixed for searching of optimum I and D values in the I-D plane. The process is then repeated at different values of P, according to the one-parameter search, until the optimum set of P, I, and D values is located. The P value is chosen to be the fixed variable in the three-parameter PID optimum search because the P value

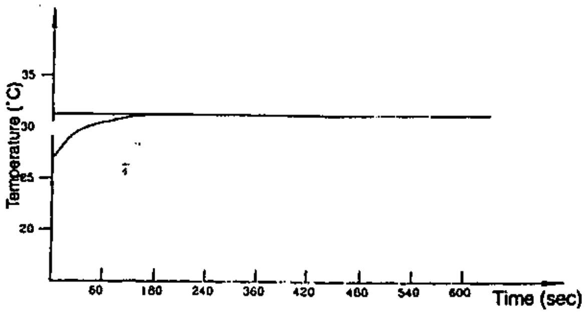

Figure 5 *Temperature response of the DDC air duct system using optimum PID values found by Hooke and Jeeves search. Initial temperature = 27°C. Setpoint = 33°C. Trial 4, P = 1.625 V/°C, Ti = 37.5 s·°C/V. T_d = 22.5 s·V/°C. Total energy = 857.06 KJ.*

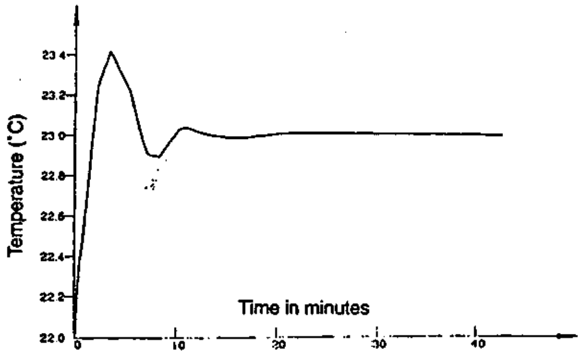

Figure 6 *Temperature response of the DDC room temperature control system. Initial temperature = 22 °C. Setpoint room temperature = 23 °C. Trial 9, P = 2 V/°C, T_i = 20 s·°C/V, T_d = 12.5 s·V/°C.*

is the dominant constant in PID control. This can be seen from the PID control equation (Equation 1).

In the optimum searching, local minima in the I-D plane for each of P values tested are avoided by using the optimum I and D values determined from the open loop test as the starting values for search in the I-D plane. The Ziegler and Nichols open loop test method for optimum setting of controllers is an experimental method that was tested under different combinations of controller settings. Hence, it is expected that local minima in the I-D plane can be avoided by using the I and D values obtained from the open loop test as the initial trial set.

Time and Energy Consideration in the Search

The search can be compared with a conventional open-loop test for optimum setting of controllers. From the experimental results on the air duct temperature control system, with the same setpoint, step setpoint increase, and initial conditions, the total time and energy consumption required in the complete search are approximately 378 and 533 times larger than that required by the open loop test (705 minutes to 112 seconds and 117206 kJ to 220 kJ). For the same step load increase under the same initial conditions, the energy consumption and the "settling" time at the optimum PID values obtained from the search are both less than one quarter of that at the PID values found from the open-loop test (857 kJ and 241 seconds compared with 3552 kJ and 1172 seconds). Hence, if the large consumption of energy and time for initial tuning by the search method can be tolerated, then once the control system is tuned up, there will be a considerable saving of energy and time in implementing the control to the setpoint under load disturbances. This is true for different operation conditions except that there are system modifications.

Room Model Consideration

The room model used for the verification of the three-parameter optimum search program was discussed by Borresen (1981) and found particularly suitable for room air temperature systems. Hence it is used in this study.

Disturbance Consideration

The search software package is developed from load disturbance. Transient disturbances are allowed in the stability criterion of the dead band around the setpoint temperature. Hence constant or transient disturbances of load will not significantly affect the performance of the software.

Assumptions

For both systems under study, the time delays approximate a few seconds and are therefore ignored in the study. However, it is recommended that further study should be made on the effect of time delay.

Advantages

The advantages of the software package can be summarized as follows:

- Self-tuning for DDC controllers can be applied to control systems using PID DDC controllers. This results in saving of manpower during initial commissioning of PID digital controllers.
- The self-tuning software, including derivative control, improves the stability of control systems.
- No complicated analysis is required for setting PID digital controllers.
- Increased energy savings can be achieved in im-

plementing control under load disturbances rather than using the controller values found by the open loop test controller.
— Retuning as a result of system modifications is easy.
— No extra cost for initial equipment purchase is necessary in implementing this software package for DDC systems. It demands only a trained technician to refine the program according to specific applications. The energy savings will be increased if the concept is introduced to a large operating system, where a small increase in performance drastically affects the overall energy bill.

Although the room model (Borresen 1981) is considered acceptable for verifying the software package, it is recommended to confirm the robustness of the software by testing it against different real situations in the future.

CONCLUSION

The self-tuning software package for the three-term digital DDC controller based on the Hooke and Jeeves method was developed and tested successfully for the air duct temperature control system. The software was verified by applying it on the room temperature control model system, where it was found to perform and compute satisfactorily in accordance to the original design intent.

ACKNOWLEDGMENTS

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NOMENCLATURE

V = volt (V)
T_r = room temperature (°C)
T_{rs} = room temperature setpoint (°C)
T_{is} = air inlet supply temperature (°C)
e = error signal (°C)
c = controller output signal (V)
F_w = chilled water mass flow (kg/s)
K_p = proportional constant (V/°C)
T_j = integral constant (s·°C/V)
T_d = derivative constant (s·V/°C)
T = sampling time (s)
s = complex variable

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APPENDIX

Equipment Setup and System Operation

The experimental setup is shown in Figure A-1. The entire system is monitored and controlled by a personal computer. The AD/DA interface converts the analog signals received from the temperature sensing circuit and power transducer into digital signals. It also converts the digital control action signal from the digital computer into analog form to the actuator.

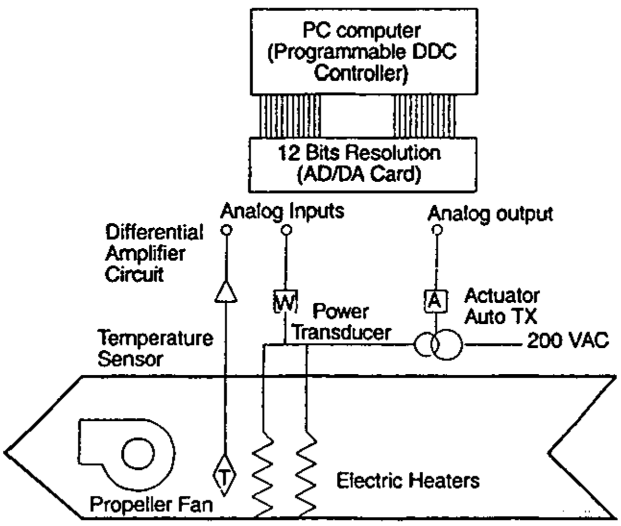

Figure A1 Schematic diagram of DDC air duct heater setup.

The commanding software is written in BASIC language, which is capable of receiving and evaluating the instantaneous air duct temperature and power consumption of the heater. The error and the corresponding control action are then computed. The control voltage output is received by the actuator, which is coupled with the autotransformer to control the supply voltage to the heaters, thus maintaining the air duct temperature at the setpoint.

SIMULATION STUDY OF A SIMPLE AIR-CONDITIONING SYSTEM FOR OPTIMUM CONTROL PERFORMANCE

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ABSTRACT

Performance of a simple typical air-conditioning control system was simulated using the system simulation program HVACSIM+[1]. The study shows that 'mean square error' is a good and effective objective function in designing a simple air-conditioning control system. The mean square error criterion satisfies the requirements of system response and control of the air-conditioning control system. Using mean square error as the objective function, the performances of five cooling coil supply temperature control systems of increasing coil sizes were studied. The best and the worst designs were determined and tested against a typical summer weather day data of Hong Kong using HVACSIM+. The performance in terms of control accuracy, stability and energy efficiency of the best design is better than the worst design. This indicates that system design using mean square error as the system selection criterion is possible and promising.

Keywords:- Air-conditioning system design, performance, optimization, control, objective function

INTRODUCTION

Optimization studies of HVAC systems would enable engineers to get the best design for the system under certain criteria on the initial cost, space and operating conditions so as to provide thermal comfortable environment, efficient use of energy, ease of maintenance and longer the operation life of the individual components. Traditionally, in the design process of HVAC systems, a set of schematic diagrams for the system is prepared under the space constraint. Sizing and matching of the individual components are determined by the design conditions such as the loading of the systems. Engineers would design the system based on their experience, and perhaps using standard charts, tables, etc., provided in the CIBSE guides [2], ASHRAE standards or handbooks [3]. The actual performance of individual components cannot be estimated at the design stage, and optimization studies (if exists !) cannot be performed in a systematically fashion. That might be the reason why so many HVAC systems are poorly designed and oversizing. With the development of system simulation on describing the performance of the individual components by mathematical models, it is possible to estimate how the entire system would work upon changing the operation conditions of the components. Using the technique of system simulation, the effects of changing different components can also be studied. This would enable the optimization studies of the HVAC systems to be performed at the design stage. In the optimization studies, objective functions of the system are defined. This would give a numerical value on how the proposed design is deviated from the optimum. Typical candidates for the objective functions are the net energy consumption, initial cost, operating cost and the payback period. Optimal search algorithm is then used to search the system variables and equipment ranges to get the minimum values of the objective functions.

As discussed and argued by Fisk [4], the mean square error when used as a criterion for temperature control systems can be computed easily. It is also related to both the mean value of the Predicted Percentage Dissatisfied (PPD) in assessing thermal comfort suggested by Fanger and Humphreys [5]; and to the probability of occupants' acceptance to change of their thermal environment. It was also argued that the development of criteria suitable for design with optimal control theory would at the same time has implications for predicting the energy saving that can be associated with conventional thermal controls. It was concluded [4] that the mean square error, besides easy and simple to be computed, appeared to provide a very useful penalty function approximation for thermal control work. Following Fisk's argument [4], mean square error has been used in a simulation study of the performance of a simple air-conditioning system. The simulation study aims to optimize the coil sizing of a simple air-conditioning system for optimum control performance. The resulting best and worst designs are then verified against a typical Hong Kong weather day.

EXAMPLE OPTIMIZATION STUDY OF A SIMPLE AIR-CONDITIONING SYSTEM

A simple HVAC system of cooling coil temperature control was used to illustrate the concept of optimization of HVAC system using the simulation program HVACSIM+. HVACSIM+ was developed at the National Bureau of Standards (NBS), USA (now the National Institute of Standards and Technology). It can be used for detail simulation of entire building systems: the HVAC systems, the equipment control system, the building shell, the physical plant, and the dynamic interactions among these subsystems. In the example, different cooling coil designs were used in the cooling coil temperature control system and their performances were evaluated and compared in term of an objective function. For each cooling coil design, the controller was set with the optimum setting obtained from the open loop test [6]. With the same initial variable values used, the performance of the resulting 'best' and 'worst' cooling coil designs obtained from the example study were tested by simulation against a typical summer day (11, July) of the Example Weather Year 1989 [7] in Hong Kong. The cooling coil temperature control system studied in this example is commonly used in the central air-conditioning systems for commercial buildings in Hong Kong. The simple HVAC system is a cooling coil temperature control system as illustrated in Figure 1. The cooling coil outlet air temperature is monitored by a temperature sensor that sends the temperature signal to the proportional and integral (P+I) controller. The P+I controller, on receiving the temperature signal, compares the signal with the setpoint temperature and results a control signal being sent to the control valve. The control valve regulates the chilled water flow to offset the temperature difference between the setpoint and the outlet air temperature. As the control valve is a normally open type, a reversing relay is used to reverse the control signal generated by the direct acting controller.

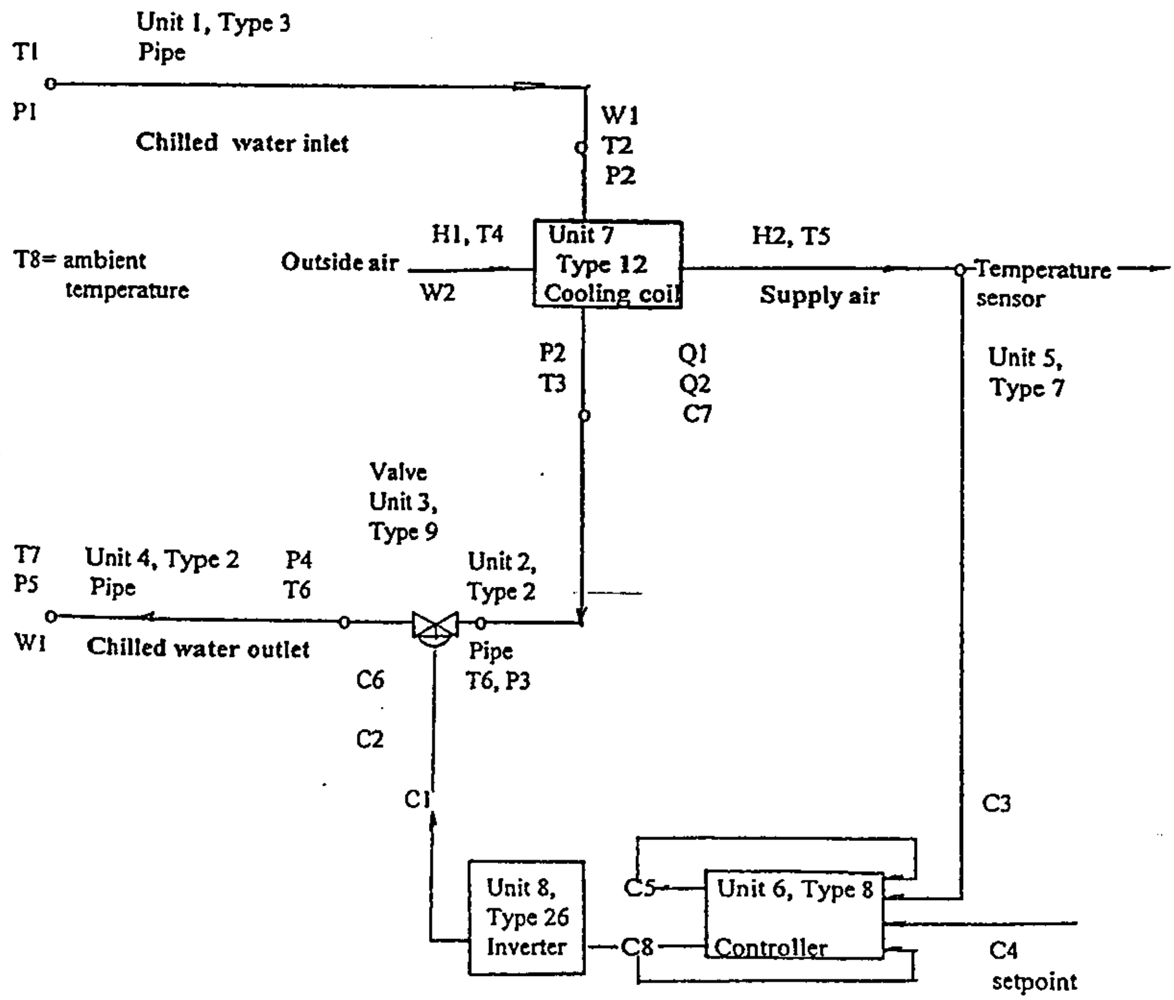

Figure 1: Schematic for cooling coil temperature control system with components and types shown

Table 1: Simulation configuration of the system

Unit	Type	Input Variable	Output Variable
No.1	Type 3, Inlet Pipe	P1-Inlet water pressure P2-Outlet water pressure T1-Inlet water temperature T8-Ambient air temperature T2-Outlet water temperature	T2-Outlet water temperature W1-Water mass flow rate
No.2	Type 2, Outlet Pipe	W1-Water mass flow rate P3-Outlet pressure T3-Inlet temperature T8-Ambient temperature T6-Outlet temperature	T6-Outlet temperature P2-Inlet pressure
No.3	Type 9, Linear Valve with Pneumatic Actuator	P4-Pressure at outlet W1-Water mass flow rate C1-Input control signal C2-Actuator relative position	C2-Actuator relative position P3-Pressure at inlet C6-Valve stem relative position
No.4	Type 2, Outlet Pipe	W1-Water mass flow rate P5-Outlet pressure T6-Inlet temperature T8-Ambient temperature T7-Outlet temperature	T7-Outlet temperature P4-Inlet pressure
No.5	Type 7,Temperature sensor	T5-Temperature input C3-Temperature sensor signal	C3-Temperature sensor output
No.6	Type 8, P+I Controller	C3-Controlled variable C4-Setpoint temperature C5-Integral of control signal C8-Output control signal	C5-Integral of control signal C8-Output control signa
No.7	Type 12, Cooling Coil	W1-Water mass flow rate T2-Inlet water temperature W2-Dry air mass flow rate T4-Inlet dry bulb air temperature H1-Inlet air humidity ratio T3-Outlet water temperature T5-Outlet air dry bulb temperature H2-Outlet air humidty ratio	T3-Outlet water temperature T5-Outlet dry bulb temperature H2-Oulet air humidity ratio Q1-Total cooling load Q2-Sensible cooling load C7-Wet fraction of coil surface area
No.8	Type 26, Inverter	C8-Input control signal	C1-Output control signal

In the simulation, eight components are identified for the system. They are represented by unit 1 to unit 8 as indicated in Figure 1 and consequently programmed (with the specific types) in the simulation.The parameters of the system components used are similar to those used in the simple examples described in the User Manual [1]. The initial values of state and boundary variables of the system components are typical of the cooling coil control systems used in Hong Kong. The configuration of the system components are summarized in Table 1. The simulation was tested against the time dependent boundary conditions of ambient temperature and inlet air humidity ratio that had initial values of 28°C and 0.019 respectively for five different cooling coil designs listed in Table 2. The optimum controller setting for each coil size was determined by the open loop test [6]. The coils chosen are with increasing distinctive sizes. Such distinctive sizes are designed to differentiate the performance of alternative designs.

Table 2: The five cooling coil designs

Cooling Coil Design	1	2	3	4	5
Primary (tube exterior) surface area, Ap (m^2)	6.571	6.491	6.41	6.33	6.249
Secondary surface area, As (m^2)	43.647	58.196	72.745	87.294	101.844
Internal surface area, Ai (m^2)	6.199	6.199	6.199	6.199	6.199
Ratio of minimum air flow area to face area	0.562	0.555	0.548	0.542	0.535
Number of fins per unit lenght (per cm)	2.3622	3.1496	3.937	4.7244	5.5118
Optimum controller setting:					
Proportional gain	2.574	1.4625	0.858	0.828	0.6773
Integral gain	0.39	0.277	0.13	0.12545	0.1026

The performances of the coil designs in term of outlet air temperature against time were recorded for step changes of inlet air at time 1500 seconds after the simulation started. The step change is from 28°C dry bulb and 0.019 humidity ratio to 30°C dry bulb and 0.022 humidity ratio respectively. To allow stability in the initial conditions, the comparison of performances between different coils started at time 1500 seconds. Considering Fisk's argument, the mean square error had been used as the objective function in evaluating the performances of the five coil designs in addition to that using controllability and stability analysis.

THE RESULTS

The responses of the cooling coil outlet temperature against time for the five cooling coil designs under step changes of ambient air temperature and humidity ratio at time 1500 seconds (taken as the reference time 0 second) are shown in Figure 2. The mean square errors for the five cooling coil designs from the reference time 0 second for a period of 1500 seconds are given in Table 3. From table 3, design 1 is considered the best design and design 4 the worst design. With the same initial values as used in the coil performance study, these two designs were further verified against a typical summer day (11, July 1989) in Hong Kong. During the typical day, the time dependent boundary conditions of ambient temperature and inlet air humidity ratio vary from 27.65°C to 30.65°C, and 0.01845 to 0.02175 respectively.The response of the cooling coil outlet air temperature against time of the day, 11 July 1989 for cooling coil designs 1 and 4 are shown in Figure 3.

Table 3: The mean square errors of the five cooling coil designs

Cooling Coil Design	1	2	3	4	5
Mean square error for time error time period 1500 seconds from the reference time 0 second °(C^2)	0.08710	0.10017	0.10933	0.13620	0.13430

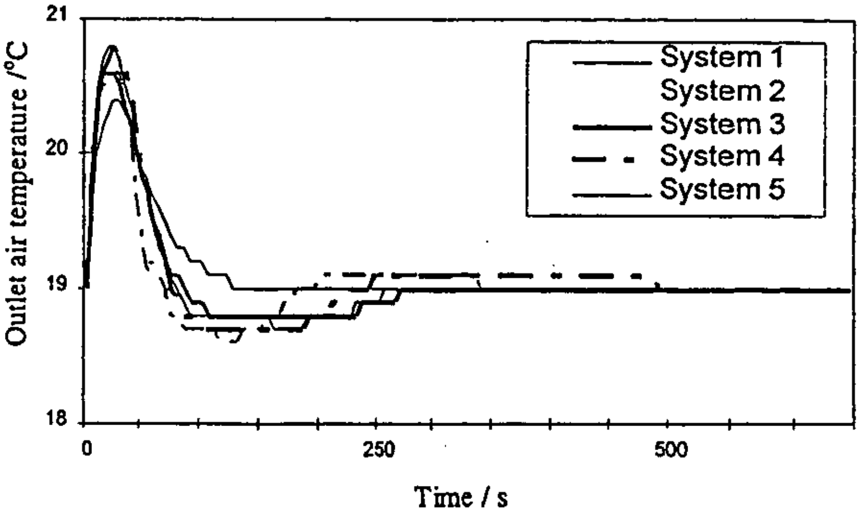

Figure 2 : Cooling outlet temperature control

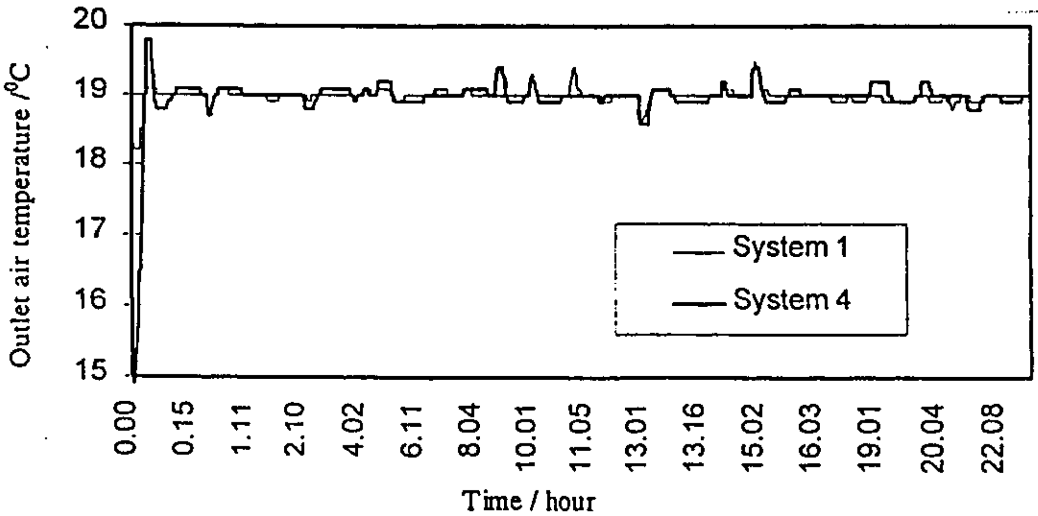

Figure 3 : Cooling coil outlet temperature control for 11 July 1989, Hong Kong

ANALYSIS

The optimization of cooling coil design by step change test

In the simulation test, the cooling coil control is under a step change at time zero and settles before the second step change at time 1500 seconds. The time at 1500 seconds can be considered as the reference time 0 second because the system is stable at that time. Analysis is thus made after the step change at the reference time 0 (that is, 1500 seconds) for a period of 1500 seconds. From the response curves of the five designs shown in Figure 2, the followings are observed:

i. The cooling coil outlet temperature for all five designs, when under the step change in the boundary variables, settles and can be controlled at the setpoint temperature of 19°C with only little hunting.

ii. The value of mean square error is in the order:
Design 1 < Design 2 < Design 3 < Design5 < Design 4

iii. The maximum overshoot is in the order: Design 1 < Designs 2 and 3 < Designs 4 and 5
Except design 1, all other designs have undershoots.

iv. The settling time to the setpoint temperature of 19° is in the order:
Design 1 < Design 5 < Design 3 < Design 4 < Design 2
If a reasonable tolerance of +/- 0.1°C is allowed in the final steady state value of the supply air temperature, the settling time will be in the order:
Design 1 < Design 2 < Design 4 < Design 5 < Design 3

v. The response rate is in the order: Design 3 > Designs 2 and 5 > Designs 1 and 4.

Considering the above, design 1 has the less overshoot when under the step change. The response is good and the system settles at the setpoint temperature at a short time period without much overshoot. Design 2 and design 3 have more or less the same overshoot magnitudes when under the step change. However, design 2 is better in response and settles within +0.1°C faster. Design 4 and design 5 have about the same amount of overshoots and theirs overshoots are larger than the other designs when under the same step change. The settling time for design 4, when a tolerance of 0.1°C is allowed, is a bit shorter than design 5. Their responses are more or less the same. However design 5 has less undershoot period than design 4. In conclusion, the goodness of designs, from control point of view, is in the order: Design 1 > Design 2 > Design 3 > Design 5 >= Design 4 . This order is in good agreement with the criterion of using the mean square error (the mean square error of design 4 is marginally larger than that of design 5). That is to say mean square error is a good and effective objective function or indicator for evaluating the performance of the temperature control system.

The verification of resulting coil designs against a typical summer day in Hong Kong

From the response curves of Figure 3, the followings are observed:

I. The cooling coil outlet air temperature is within 14.9 to 19.8 °C and can be controlled a 19°C without much hunting.

ii. The cooling coil outlet air temperature response is more stable in design 1 than in design 4 Design 4 has more hunting and longer hunting period than design 1. Also design 4 has large overshoot magnitudes. Design 1 has shorter settling time.

Both cooling coil designs can fulfill the requirements of the coil outlet air temperature control. Yє design 1 is more energy efficient and cost effective than the design 4 (less hunting, smalle overshoot and shorter settling time). Moreover, the system response is also found more stable.

CONCLUSIONS

For the simple cooling coil control system, simulation study using HVACSIM+ indicates that ther exist good and bad designs. The performances of different designs are evaluated against th performance index 'mean square error.' The performances of different designs are also evaluate and compared by control considerations of stability and response. It is found that the performanc evaluation using mean square error is generally in good agreement with that using contrc considerations. The mean square error is a good performance index for the simple cooling coi control system. The resulting coil designs are also verified against a typical summer weather da data of Hong Kong. Hence, the mean square error should be considered for use in evaluating th performance of an entire HVAC system.

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