Design, modeling and performance optimization of active air terminal system

Ke, Ji

2019

Ke, J. (2019). Design, modeling and performance optimization of active air terminal system.
Doctoral thesis, Nanyang Technological University, Singapore.

<https://hdl.handle.net/10356/90114>

<https://doi.org/10.32657/10220/48446>

**NANYANG
TECHNOLOGICAL
UNIVERSITY**

SINGAPORE

DESIGN, MODELING AND PERFORMANCE OPTIMIZATION
OF ACTIVE AIR TERMINAL SYSTEM

JI KE

School of Electrical & Electronic Engineering

Nanyang Technological University

2018

DESIGN, MODELING AND PERFORMANCE OPTIMIZATION
OF
ACTIVE AIR TERMINAL SYSTEM

JI KE

School of Electrical & Electronic Engineering

A thesis submitted to the Nanyang Technological University
in partial fulfillment of the requirement for the degree of
Doctor of Philosophy

2018

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

[Date Here]

11-Mar-2019

.....

Date

[Student's Signature Here]

..... Ji Ke

[Student's Name Here]

Ji Ke

.....

Supervisor Declaration Statement

I have reviewed the content and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

[Date Here]

11-Mar-2019

.....

Date

[Supervisor's Signature Here]

.....

[Supervisor's Name Here]

Cai Wenjian

.....

Authorship Attribution Statement

This thesis contains material from 2 papers published in the following peer-reviewed journals where I was the first author.

Chapter 4 is published as Ji Ke, Cai Wenjian, Zhang Xin, Wu Bingjie and Ou Xianhua. ‘Modeling and validation of an active chilled beam terminal unit’. Journal of Building Engineering 22, 161-170 (2019). DOI: 10.1016/j.jobbe.2018.12.009.

The contributions of the co-authors are as follows:

- Prof Cai provided the initial project direction and edited the manuscript drafts.
- I prepared the manuscript drafts. The manuscript was revised by Prof Zhang, Dr Wu and Dr. Ou.
- I co-designed the study with Prof Cai and performed all the laboratory work at the School of Electrical and Electronic Engineering. I also analyzed the data.
- All the model derivation and simulation were conducted by me in the ACMV lab.
- Dr Wu and Dr Ou assisted in the collection of the real-time ACB performance data.

Chapter 5 is published as Ji Ke, Cai Wenjian, Wu Bingjie and Ou Xianhua. ‘Mechanical design and performance evaluation of active thermosiphon beam terminal units’. Building and Environment (2019). DOI:10.1016/j.buildenv.2019.02.033.

The contributions of the co-authors are as follows:

- Prof Cai provided the initial project direction and edited the manuscript drafts.
- I wrote the drafts of the manuscript. The manuscript was revised together with Dr Wu and Dr Ou.
- I performed all the experiments, adjust the indoor conditions, conducted data evaluation and analyze the experimental results.
- Dr Wu assisted in the setting up of ACMV pilot plant.

- Dr Ou suggested the experimental procedures to investigate the performance of the ATB terminal unit.

[Date Here]

03-11-2019

.....
Date

[Student's Signature Here]

.....
[Student's Name Here]
Ji Ke

.....

Acknowledgements

First and foremost, I would like to express my sincere gratitude to my supervisor Professor Cai Wenjian for all the guidance and advice throughout the course of my research work. Without their encouragement and advice, I would not have been able to ensure the smooth completion of the research.

Also, I really deeply appreciate all the help from my friends Dr. Chen Can, Dr. Lin Chen, Dr Wang Xinli, Dr. Wu Bingjie, Dr. Zhai Deqing, Dr. Ou Xianhua, Dr. Chen Haoran, Dr. Shen Suping in Process Instrumentation Laboratory, their extensive knowledge and kind offer supported me in many ways. There were so many good memories within and without the laboratory.

My sincere thanks would be given to School of Electrical and Electronic Engineering NTU and SinBerBEST for providing the financial support for my study.

Last but not least, I thank my parents who have concerned and encouraged me to this day and my friends who have made me full confidence to complete my research.

Table of Contents

Statement of Originality	I
Supervisor Declaration Statement.....	III
Authorship Attribution Statement	V
Acknowledgements	VII
Table of Contents	IX
Summary.....	XIII
List of Figures.....	XV
List of Tables	XVIII
Nomenclatures.....	XIX
Chapter 1 Introduction.....	1
1.1 Background	1
1.2 Overview of active air terminal systems.....	2
1.3 Motivations and objectives	9
1.4 Major contribution	10
1.5 Organization of the thesis	11
Chapter 2 Literature review	13
2.1 Introduction.....	13
2.2 Active air terminal unit	13
2.3 Air flow patterns and thermal comfort.....	16
2.4 System modeling and optimization.....	20
2.5 Terminal unit applications	21
2.6 Summary	24
Chapter 3 Terminal unit design and experimental setup	25

3.1 Introduction.....	25
3.2 The experimental active chilled beam	25
3.3 The experimental active thermosiphon beam	26
3.4 Chiller plant and dedicated outdoor air system.....	29
3.5 Summary	33
Chapter 4 Modeling and validation of an active chilled beam terminal unit.....	35
4.1 Introduction.....	35
4.2 Modeling development of ACB.....	37
4.2.1 Air entrainment model.....	38
4.2.2 Heat transfer model.....	40
4.2.3 Parameter identification.....	43
4.3 Experimental procedure	44
4.4 Model validation	46
4.5 Summary	53
Chapter 5 Mechanical design and performance evaluation of active thermosiphon beam terminal units	55
5.1 Introduction.....	55
5.2 The ATB working principle.....	57
5.3 Experimental study	59
5.3.1 The experimental setup.....	59
5.3.2 The experimental procedures.....	61
5.3.3 Theoretical analysis	62
5.3.4 Assessment criteria	63
5.4 Experimental results.....	65
5.4.1 Primary air plenum pressure	65

5.4.2 Chilled water flow rate	67
5.4.3 Average temperature difference	69
5.4.4 Full duct length	72
5.4.5 Performance comparison with ACB and PDV	74
5.5 Summary	75
Chapter 6 Model-based optimization for ATB system.....	77
6.1 Introduction.....	77
6.2 Model development of ATB system.....	78
6.2.1 Chiller energy model	78
6.2.2 Fan and water pump energy model.....	79
6.2.3 The air flow model.....	80
6.2.4 The cooling coil model	81
6.2.5 The indoor built model	83
6.3 Experimental setup and model validation.....	83
6.3.1 Experimental setup	83
6.3.2 Model validation.....	85
6.4 Global optimization formulation.....	89
6.4.1 Objective function	89
6.4.2 Constrains	90
6.4.3 Optimization strategy of ATB system	92
6.5 Optimization results	96
6.6 Summary	102
Chapter 7 Conclusions and future work.....	103
7.1 Conclusions.....	103
7.2 Future work.....	104

References	107
Author's publications	119

Summary

Air conditioning and mechanical ventilation (ACMV) system, which determines the indoor environment quality and energy efficiency of buildings, attracts increasing attentions throughout the world. In modern society, a series of problems such as the sensation of draught, energy waste arising with the massive usage of air conditioning and sick building syndrome (SBS). Prioritizing green building techniques in ACMV system can improve occupants' fitness level and deliver dramatic energy saving. Among various ACMV schemes, the active air terminals (active chilled beam and active thermosiphon beam) have outstanding performance on energy saving, indoor environment quality improvement and space saving. However, the existing research is still inadequate and some technical difficulties stand as major obstacles for application of the air terminals especially in tropical regions. To fulfil the gaps, this thesis focuses on the performance analysis, terminal unit modeling and operating optimization of the active air terminal based systems. The contributions of this thesis include:

1. A simple yet accurate hybrid model of active chilled beam (ACB) is developed with respect to air buoyancy. The model demonstrates the air entrainment characteristics in the air chamber and the heat transfer process in the cooling coil. Compared with the existing ACB terminal unit model, the proposed model captures the effects of air buoyancy and further reduces the complexity of the cooling coil model. The ACB model includes only two equations with nine unknown model parameters that can be identified through Levenberg-Marquardt method based on experimental measurements. Experimental validation in a mock up room proves that the models can predict the supply air flow rate and heat transfer process in a wide range of operating conditions. The proposed ACB model can be further utilized in optimization and performance evaluation for the ACB system.
2. To eliminate the condensation problem and improve the heat transfer efficiency of the traditional ACB, the mechanical design of the terminal unit is optimized. Combining air entrainment effect and displacement ventilation, the

active thermosiphon beam (ATB) is developed with innovative nozzle arrangement, cooling coil placement and air chamber configuration. The experimental comparisons of ATB and ACB are conducted under various operating conditions to estimate its thermodynamic and hydrodynamic performances. The comparison results indicate that 1) the cooling capacity of ATB is around 40% higher than ACB and passive displacement ventilation (PDV); 2) the ATB has better dehumidification ability with the sensible heat ratio of 0.42; 3) the initial cost of ATB system is the lowest under same cooling load requirement. More importantly, the experimental findings provide a guideline for the operation and optimization of ATB systems.

3. A model-based optimization strategy for the ATB system is developed to reduce the energy consumption and maintain indoor environment quality. The thermal models of the terminal unit and the energy consumption models of different components are established based on the experimental results. Accordingly, the global optimization strategy is formulated to search the optimal operating points of the ATB system with regard to total energy consumption under operating constraints. The experimental results indicate that the optimized operating parameters obtained by the genetic algorithm (GA) can significantly reduce the total energy consumption. The obtained findings indicate that the ATB system is a promising ACMV system in terms of initial cost, thermal comfort and energy saving for a variety of applications.

List of Figures

Figure 1.1 Schematic diagram of ACB terminal unit	3
Figure 1.2 3D mechanical design of ATB terminal unit.....	4
Figure 1.3 The installation of ATB.....	5
Figure 1.4 Performance simulation of the ATB system	6
Figure 1.5 Schematic diagram of ACB system.....	7
Figure 1.6 Water loop of the ATB system.....	8
Figure 2.1 System Layout for Passive Displacement Ventilation	15
Figure 2.2 Typical air distribution of ACB system.....	17
Figure 2.3 The PDV system installed in NTU	24
Figure 3.1 The experimental ACB terminal unit	26
Figure 3.2 The schematic drawing of ATB	27
Figure 3.3 Different models of induction nozzles	27
Figure 3.4 Prototype of the heat exchanger	28
Figure 3.5 Prototypes of full ducts.....	29
Figure 3.6 Front view of the chiller plant	30
Figure 3.7 Back view of the chiller plant.....	31
Figure 3.8 The air handling unit	32
Figure 3.9 The liquid desiccant dehumidification system	32
Figure 4.1 The interaction of the ACB sub-models.....	37
Figure 4.2 Experimental fitting for the primary air volume flow rate	47
Figure 4.3 Model validation for primary air volume flow rate.....	48

Figure 4.4 Model validation for secondary air volume flow rate	49
Figure 4.5 Model validation for secondary air volume flow rate without air buoyancy ..	49
Figure 4.6 Model validation for secondary air volume flow rate under various chilled water inlet temperatures	50
Figure 4.7 Model validation for secondary air volume flow rate under various room temperatures	50
Figure 4.8 Secondary air volume flow rate under various average temperature differences	51
Figure 4.9 Secondary air volume flow rate under various chilled water flow rates	52
Figure 4.10 Model validation for heat transfer rate	53
Figure 5.1 The air flow patterns of the ATB system	57
Figure 5.2 The temperature distribution of the ATB system	58
Figure 5.3 The NTU Eugenia Room	60
Figure 5.4 The influence of primary air plenum pressure on heat transfer rate	66
Figure 5.5 The influence of primary air plenum pressure on SHR	66
Figure 5.6 The influence of primary air plenum pressure on ER	67
Figure 5.7 The influence of chilled water flow rate on heat transfer rate	68
Figure 5.8 The influence of chilled water flow rate on SHR	68
Figure 5.9 The influence of chilled water flow rate on ER	69
Figure 5.10 The influence of chilled water flow rate on supply air temperature	69
Figure 5.11 The influence of average temperature difference on heat transfer rate	70
Figure 5.12 The influence of average temperature difference on SHR	71

Figure 5.13 The influence of average temperature difference on ER.....	71
Figure 5.14 The influence of average temperature difference on supply air temperature	72
Figure 5.15 The influence of fall duct on heat transfer rate.....	73
Figure 5.16 The influence of fall duct on sensible and latent cooling capacity.....	73
Figure 5.17 The influence of fall duct on SHR.....	74
Figure 5.18 The influence of fall duct on ER	74
Figure 6.1 The schematic diagram of the experimental ATB system.....	84
Figure 6.2 Model validation of fan energy consumption.....	85
Figure 6.3 Model validation of pump energy consumption.....	86
Figure 6.4 Model validation of chiller energy consumption.....	86
Figure 6.5 Model validation of the primary air flow rate	87
Figure 6.6 Model validation of the secondary air flow rate.....	87
Figure 6.7 Model validation of the cooling capacity	88
Figure 6.8 Scheme of the optimization strategy	93
Figure 6.9 Flow chart of the optimization strategy	95
Figure 6.10 The indoor heat condition and number of occupants	97
Figure 6.11 The original and optimized primary air flow rate	99
Figure 6.12 The original and optimized chilled water flow rate.....	99
Figure 6.13 The original and optimized energy consumption of the ATB system.....	100
Figure 6.14 Comparison of fan, pump, chiller and total energy consumption	101

List of Tables

Table 4.1 The operating ranges of ACB system	45
Table 4.2 Sensor specification	46
Table 4.3 Summary of identified parameters.....	46
Table 4.4 Summary of the assessment criteria.....	53
Table 5.1 Summary of sensor specification	60
Table 5.2 Summary of system setting points	61
Table 5.3 Performance criteria of the terminal units	75
Table 6.1 Components rated capacities	84
Table 6.2 Prediction accuracy of the models	88
Table 6.3 Classification of state variables	92
Table 6.4 The upper and lower bound of constraints.....	97
Table 6.5 The parameter setting of GA	98
Table 6.6 Summary of the energy consumption between both operation strategies	101

Nomenclatures

A	section area
a	unknown parameters
b	unknown parameters
C	specific heat capacity at constant pressure (J/kg°C)
c	unknown parameters
D	characteristic length (m)
d	unknown parameters
dT	local temperature difference (°C)
dw	local carbon dioxide content difference (g/kg)
E	Energy consumption (W)
e	constant coefficient
f	constant coefficient
g	gravitational acceleration (m/s ²)
H	enthalpy (kJ/kg)
h	heat transfer coefficient (W/ m ² °C)
i	constant coefficient
k	constant coefficient
M	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
n	constant coefficient
P	pressure (Pa)
PLR	part ratio
Pr	Prandtl number
Q	heat transfer rate (W)
R	thermal resistance
Re_D	Reynold number
T	temperature (°C)
$\dot{V}$	volume flow rate (m ³ /h)

W	moisture content (g/kg)
w	carbon dioxide content (g/kg)
x	constant coefficient
ΔT	average temperature difference (°C)

Subscripts

a	air or air side
am	mix air
asoc	secondary air off the cooling coil
c	water or water side
ch	chiller plant
chw	chilled water
chwa	air around the cooling coil
cur	current
f	fan
h	
in	inlet
l	load
lat	latent
min	minimum
max	maximum
out	outlet
p	water pump
pri	primary air
r	real value
rated	rated
req	required
sec	secondary air

sec_0	secondary air without air buoyancy
sen	sensible
sup	Supply air
total	total
z	zone

Greek symbols

ρ	density (kg/m ³)
λ	thermal conductivity
v	flow velocity
μ	fluid absolute viscosity
η	efficiency

Abbreviations

ACMV	Air conditioning and mechanical ventilation
ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineer
ER	Entrainment ratio
GA	Genetic algorithm
HVAC	Heating ventilation and air conditioning
IAQ	Indoor air quality
PDV	Passive displacement ventilation
PID	Proportion integration differentiation
RE	Relative error
RMSRE	Root mean square of relative error
SBS	Sick building syndrome
SHR	Sensible heat ratio

VAV	Variable air volume
VSD	Variable speed drive

Chapter 1 Introduction

1.1 Background

Air conditioning and mechanical ventilation (ACMV) system, which provides desired thermal comfort and satisfied indoor air quality (IAQ), is an essential part of people's daily life. Since ACMV system first designed in 1902 by Alfred Wolff [1], it has been widely used in individual residences and commercial buildings. As urban citizens spend more than 80% of time in indoor environment, the proper operation of ACMV is critical to occupants' performance and productivity. Additionally, people's fitness level is positively correlated with the indoor air quality, especially in tropical countries where ACMV system is running all year around. Currently, indoor air cooling, heating and ventilation applications account for 30% (5.35 Quads) of energy consumption in commercial buildings in the mild regions [2]. In Singapore, where the annual average temperature is 28.4°C [3], the proportion of building energy consumption attributes to cooling and mechanical ventilation applications can reach up to 70% [4]. As a consequence, optimizing the ACMV system is promising for optimizing the energy efficiency and improving the indoor environment quality.

Energy saving, thermal comfort and IAQ are the core objectives in green buildings which determine the orientation of ACMV development. Novel mechanical design, sub-systems, optimization algorithm and machine learning are introduced into ACMV applications. To improve energy efficiency and IAQ, the energy recovery ventilator (ERV) has been developed to utilize the energy contained in the exhaust air to treat the fresh air. The ERV can reduce the energy consumed to pretreat the outdoor ventilation air [5]. Besides, air-water ACMV system and demand-controlled ventilation can separately handle latent/sensible loads and optimize the supply of ventilation air based on the occupants' demands [6]. The technologies of the next generation in ACMV system, which aim at high performance and high efficiency, have gained increasing interests. Among the technology innovation, great efforts have been put on mechanical design and optimal operation of air terminal units. As the ultimate components to treat and deliver air,

terminal units have defining influence on indoor thermal comfort and structure of the ACMV central equipment.

The active air terminal (active chilled beam and active thermosiphon beam) is a potential alternative to the conventional variable air volume (VAV) and fan coil (FCU) system. Active air terminal based systems are typical air-water configuration ACMV systems which have outstanding performance on energy saving, IAQ improvement and space saving etc. The active chilled beam (ACB) terminal units have evolved in Europe for twenty years and become very popular in North America and Asia nowadays. But its limitations of chilled water temperature control and condensation stand as major obstacles for application in tropical regions. The active thermosiphon beam (ATB) system, which combines the advantages of ACB and PDV, can ensure the fresh air supply and meet the heat load in wide operation ranges. Consequently, the ATB has the potential to become the standard equipment for ACMV systems in modern buildings with complex layout and multiple functions. The scope of the thesis focuses on performance evaluation, modeling, and optimization of active air terminal systems.

1.2 Overview of active air terminal systems

The active air terminals technology began with Willis Carrier who invented the first induction system [7]. The induction system named perimeter induction terminal was the ACMV system choice from 1930's to 1970's. The induction units fell into disfavor for some unique negative aspects include: (1) high fan energy consumption issues due to higher pressure primary air requirement; (2) condensation issues during cooling operation; (3) difficult rezoning issues due to building profiles change. Hence, the induction units were gradually replaced by FCU and VAV during the 1970's.

The ACBs used today share the same core innovation with the induction unit. The terminal units discharge high speed primary air through nozzles to create a vacuum region and induce room air across the cooling coil where the secondary air is conditioned. The ACBs with mature technology improve the mechanical design of nozzles and

terminal unit which increase the induction ratio with 80% lower primary air pressure. In addition, the latest central components (dedicated outdoor air system, LDDS, etc.) improve the stability and accuracy of the ACMV system which insure the dry operation of the cooling coil. With these changes, the ACB is particularly beneficial to be used in office environment in terms of energy efficiency, virtually noiseless and space saving.

To illustrate the working principle of the ACB, a typical schematic diagram of ACB is shown in Figure 1.1.

Figure 1.1 Schematic diagram of ACB terminal unit

The outdoor fresh air is pretreated and pressurized by a DOAS. Then the cooled primary air is charged through a series of nozzles into the mixing chamber. The high speed jet flow of the primary air creates a negative pressure region in the chamber. Hence, the secondary air is entrained through the cooling coil due to the pressure difference. Since the cooling coil is in the air path, the chilled water removes the heat from secondary air. Finally, the secondary air and the primary air mix in the chamber and supply to the occupied zone [8, 9].

Despite the key benefits of the ACB, there are several impediments that have limited their applications in tropical regions. The chilled beams have a relatively low cooling capacity as warmer chilled water is supplied and the air movement is halted at the coil due to the downward movement of cold air. Since the driving force for room air circulation is the entrainment effect, the cooling/heating capacity of ACB is proportional to the pressure in

the primary chamber. For the situations where either ventilation or cooling/heating requirement is large, the system has to be operated to meet the larger demand.

The ATB is an innovative solution to ACMV system which overcomes all the disadvantages of conventional ACBs. Based on the ACB technology, the ATBs have rearranged nozzles, vertically installed heat exchanger and novel air chamber design. The mechanical design and performance simulation of the ATB system are shown in Figure 1.2 and Figure 1.4 respectively.

Figure 1.2 3D mechanical design of ATB terminal unit

The ATB is designed to be suspended from the ceiling or mounted on the wall where the stratification of high temperature return air is formed. In general, the fall duct is installed at the air outlet of ATB which restricts the diffusion of supply air and enhance the ventilation. Moreover, the ATB is recommended to be installed opposite the window for better indoor air circulation as shown in Figure 1.3.

Figure 1.3 The installation of ATB

Comparing with ACB, the utilization of the thermosiphon effect and the additional water drainage system are the core innovation of ATB. Similar to ACB system, the ATB system needs DOAS to continuously supply treated ventilation air to build up the primary chamber pressure. The warm air in the ceiling height will be induced through the heat exchanger due to air entrainment and fluid thermosiphon effects. Due to the gravitational force generated by the high density cooled air, the mix air in the chamber will drop along the air straightener to the floor level and gradually diffuse the occupied zone. During the operation, the condensate water formed on the surface of heat exchangers drains out via gravity.

Figure 1.4 Performance simulation of the ATB system

The active air terminals introduced above adopt the same air handling and distribution structure as illustrated in Figure 1.5, but additional primary air treatment is required in the ACB systems. The fresh air flow rates are controlled by dampers based on feedback from occupants counting or air quality sensor. In the ACB system, the primary air handles the whole latent load and part of the sensible load which constrains the primary air temperature and moisture content. As a consequence, the fresh air is treated by the AHU first. Then, the cooled air needs to be supplied to the Liquid Desiccant Dehumidification System (LDDS) where moisture content of primary air can be reduced as low as 3g/kg and a precise indoor humidity control can be realized. For the ATB system, the primary air is directly treated by conventional AHU which is more feasible and practical in tropical countries with high humidity. In some cases, the additional supply air fan is required in to keep the pressure in terminal unit. Dampers are installed at each branch of ductwork which have two functions: 1) maintain partial operation situations during overtime or weekend usage; 2) control the ventilation rate to the occupied zones. In summary, the primary air system in ACB is relatively complex but with the capability to satisfy 40% of sensible load and entire latent load. In ATB systems, the ventilation air accounts for a fraction of the total cooling capacity, generally around 20%. The control of fresh air supply and cooling capacity are largely decoupled and it enhances the system reliability.

Figure 1.5 Schematic diagram of ACB system

As demonstrated in Figure 1.6, the chilled water loop of the ATB system is with relative simple structure. As condensation is strictly prevented during the operation of ACB system, there should be a preheating system to reheat the overcooled chilled water to 14-18°C before deliver to the ACB. The typically chilled water inlet temperature for ATB is 8-10°C. Hence, the chilled water can be directly supplied to the DOAS and the terminal unit which simplify the water loop structure and reduce the water pressure drop. Dedicated booster pump system regulates the chilled water supply and ensures adequate water pressure. At the entrance of the occupied zones, the motorized valves are installed to regulate chilled water flow rate and maintain indoor environment quality.

Figure 1.6 Water loop of the ATB system

Compared with the conventional ACMV systems, the ACB system has distinct advantages in terms of energy efficiency, IAQ improvement and space saving. However, there are some impediments that limited their applications include high installation cost, condensation prevention, coupled ventilation and cooling capacity. The ATB has novel mechanical configuration and operation principle which overcomes the drawbacks of ACB.

The advantages of the ATB are briefly interpreted as below:

1. Triple effects (thermosiphon, entrainment and Coandă effect) enhance the heat transfer efficiency of terminal unit.
2. Vertically placed coil with water drainage system can eliminate condensation issue and control indoor moisture content.
3. The control of fresh air supply and cooling capacity are largely decoupled. There is still cooling supplied to the space even fresh air is cut off and the circulation is purely governed by buoyancy effect and chilled water supply.
4. Displacement ventilation of ATB is noiseless with better ventilation efficiency. Hence, the ATB system can improve indoor environment and reduce the risk of draught.

1.3 Motivations and objectives

In Singapore, the central-air conditioning systems have been widely used in commercial buildings, hospitals and campus which account for more than 50% of the total power consumption. The optimal design and operation of ACMV system have significant energy conservation and cost reduction potential. The ACB system has just been applied for twenty years and great efforts are put on the system design and unit structure optimization. In real application, model free on-off control is widely used which lead to considerable degeneration in the IAQ and energy efficiency. In addition, the ACB systems are originated and widely utilized in Europe countries where the operating conditions of the ACMV system is different from that in tropical regions. Some general technical issues remain which affect the application of ACB. Considering the ATB is a new designed air terminal unit which is sensitive to the mechanical structure and operation condition. Yet, no research work can be found that investigates the ATB system. Therefore, the active air terminals still have some urgent issues to be resolved:

- The existing ACB model is of great complex and failed to evaluate the effect of air buoyancy on the entrainment effect. The horizontally placed heat exchanger halts the secondary air movement. These factors delay the progression of ACB application and the development of advanced control scheme.
- The ATB is an innovation terminal unit which is sensitive to the operation condition. No performance evaluation or operation characteristic is available in the literature. As a result, the cooling performance and energy efficiency haven't been optimized to suit various working conditions.
- The fresh air supply and cooling capacity of active air terminal systems are severely interacted. The appropriate design and optimization operation of active air terminal system are essential for maintain the indoor environment quality and minimize the energy consumption.

The goals of this thesis are to fully analyze the operating characteristics of active air terminal based systems and develop efficient system optimization strategies. More specifically, the thesis holds whole length tightness between topics and resolves the

aforementioned issues which hinder the improvements and applications of active air terminal systems:

- Develop a hybrid model with brief structure and high precision which quantifies the system coupling and evaluates the air conditioning performance.
- Demonstrate the mechanical design of ATB and estimate the cooling performance as a guideline for practical application.
- Develop a model-based optimization strategy for the ATB system to minimize the energy consumption and maintain indoor thermal comfort.

1.4 Major contribution

The major contributions of this thesis include:

- A simplified hybrid model is developed for the ACB terminal unit based on the conservation of energy and mass. Considering the effect of buoyancy force generated by the temperature gradients, the model demonstrates the air mixing in the air plenum and the heat transfer process in the terminal unit. Experimental validation in the thermal room proved that the model is effective in predicting the supply air flow rate and heat transfer rate with high accuracy. The proposed models can be further examined in the optimization and performance evaluation for ACB systems.
- The performance comparison of ATB and ACB systems under various operating conditions are conducted. The main factors (primary air plenum pressure, average temperature difference, fall duct length, chilled water flow rate) that influence the ATB heat transfer efficiency are tested separately to determine the optimal operation settings. Based on the experimental results, the cooling capacity and energy efficiency of ATB terminal unit can be improved by optimizing the system design. Meanwhile, the findings provide a guideline for the real application of ATB.
- A model-based control strategy is presented to reduce the ATB system energy

consumption and maintain the indoor thermal comfort. The optimal working condition is tracked through genetic algorithm. The simulation results indicate that the optimization scheme can significantly reduce the energy consumption and satisfy the indoor environment quality.

1.5 Organization of the thesis

The thesis is organized as follows:

Chapter 2 reviews some essential knowledge related to the active air terminal systems.

Chapter 3 presents the mechanical design of an ACB and a self-designed ATB terminal unit. The experimental setup is demonstrated as the fundamental of the subsequent experimental research.

Chapter 4 develops a hybrid model for ACB terminal unit. The air entrainment model and heat transfer model are introduced respectively. The validation results demonstrate the effectiveness of the ACB model in predicting the induced air flow rate and heat transfer rate.

Due to the negative influence of air buoyancy on the ACB performance, chapter 5 optimizes the mechanical design of ATB to enhance the unit cooling performance and eliminate the pre-exist drawbacks. A series of tests are done to evaluate the ATB overall performance.

With respect to the experimental results in chapter 5, chapter 6 develops a model-based optimization scheme for the ATB system. The total energy consumption is minimized and the indoor thermal comfort is maintained.

Chapter 7 summarizes the conclusions and presents the foreseeable research orientations.

Chapter 2 Literature review

2.1 Introduction

To achieve the objective as discussed in Chapter 1, a comprehensive review of active air terminal related technology is necessary. As active air terminal technology is still in development phase, some technical difficulties emerge in the practical application. The previous research should be scrutinized to figure out the problem property and provide the potential solutions.

In this chapter, the state of art active air terminal literatures and available sources are summarized as follow. The air terminal unit aerodynamic and thermodynamic optimization designs are introduced in section 2.2. The system air flow patterns and indoor environment quality with active air terminal systems are explained in section 2.3. The active air terminal system modeling and control strategies are investigated in section 2.4. In section 2.5, the terminal unit applications and effectiveness evaluation are examined. The summary is demonstrated in section 2.6.

2.2 Active air terminal unit

The terminal unit is the key component in the ACMV system which determines the system overall cooling capacity and energy efficiency. The air entrainment and heat transfer process within the unit are highly depend on the mechanical design. Hence, the designs of casing, air mix chamber, nozzles and heat exchanger require depth investigation to optimal the unit performance. At present, some studies have been carried out in this area.

Inducing the secondary air across the cooling coil without fan energy requirement (entrainment effect) is the core innovation of active air terminal technology. The effectiveness of the entrainment effect is quantified by the entrainment ratio (the flow rate of secondary air to primary air). Ruponen et al. [10] simplified measurement methods of entrainment ratio for the ACB system. The proposed method used one

velocity transmitter, one venturi and primary air flow rate which showed robust and consistent results. Filipsson et al. [11] presented three acquisition methods to obtain the entrainment ratio. Comparison studies of air velocity, modified capacity and tracer gas methods were conducted under various operation conditions. The experimental validation indicated the modified capacity method is more accuracy.

The air jet flow release from the primary air plenum is coupled with the induction nozzle design. Freitag et al. [12-14] conducted simulations and experiments to investigate the internal and external air flow of ACB. The flow patterns and velocities in the unit were obtained under various air plenum pressures. The results indicated that the entrainment effect could be reinforced through adjusting the nozzle width and bending. In real application, Dadanco [15] provided a series of specially shaped nozzles to strengthen the entrainment efficiency.

To optimal the design of induction nozzles, Guan et al. [16, 17] took advantage of computational fluid dynamic (CFD) technique to optimize nozzle radius and separate distance for the induction process. The simulation results revealed that the nozzle radius had a negative correlation to entrainment ratio while small separate distance could promote air entrainment. Wu et al. [18] conducted CFD simulations to comprehensively exam the effects of nozzle diameter and inlet pressure on the ACB performance. It showed that supply air un-uniformity was severe when the nozzle diameter was large. And the rise of the inlet pressure could aggravate the un-uniformity.

Furthermore, Guan et al. [19] optimized the geometric design of ACB to achieve sufficient entrainment efficiency. The nozzles and negative pressure kernel were relocated at the center of the terminal unit which made the chamber more effectively for air entrainment process. The modified terminal unit structure with 7 mm nozzle could increase the entrainment ratio by 30%.

Active air terminal system is a typical air-water structure ACMV system. The heat exchanger inside unit has some distinctive features compared to conventional cooling coils. Chen et al. [20, 21] systematically studied the cooling coil heat transfer performance with different circuitry arrangements and tube connecting sequences.

Compared the operation characteristics between conventional 1-circuit and multiple-circuits coil design, the refined 2-circuits arrangement achieved significant improvement with respect to heat transfer rate and pressure drop. Dominguez et al. [22] summarized the ongoing research on cooling coil of terminal units and conducted tests for fin-and-tube cooling coil from several configurations of ACB. The results correlated the heat exchanger design and air thermal resistance which offer additional energy saving potential to the terminal unit. An experimental study [23] on the function of 2-pipe ACB revealed that the 2-pipe system can reduce the energy consumption up to 18% less than the conventional 4-pipe one.

In addition, the air outlets of the active air terminal have impacts on the air flow patterns and occupants' comfort level. Bertheussen et al. [24, 25] evaluated the performance of radial swirl jet structure ACB and investigated the influence of internal load distribution. The results showed the CSW chilled beam generated a satisfied thermal environment with higher ventilation effectiveness than the diffuse ceiling system. A practical issue also occurred that additional fresh air was required to safeguard the IAQ as the system is highly correlated with the heat sources.

Figure 2.1 System Layout for Passive Displacement Ventilation

Passive displacement ventilation is an emerging technology which gets increasing attention in Singapore. The buoyance driven displacement ventilation process highly depends on the terminal unit mechanical design and indoor heat source distribution. Experimental studies were made [26-29] to analyze the terminal unit air conditioning efficiency and energy saving potential which pointed out design guidelines and application issues. Betz et al. [30] reviewed several softwares that gave approaches to the simulation of terminal units which was classified as one of the pivotal study requirements by ASHRAE [31].

2.3 Air flow patterns and thermal comfort

The IEQ is positively related to the occupants' health and productivity. The air flow patterns and thermal comfort, which receive increasing concerns of researchers, are key indicators of IEQ. The temperature distribution and air flow patterns in the active air terminal system are more complex for the terminal unit unique working principle. Specific experiment and simulation are required to estimate the terminal unit operation characteristics and provide optimal indoor thermal comfort.

Figure 2.2 Typical air distribution of ACB system

To comprehensive investigated the ACB air flow behaviors, Cao et al. [32-38] conducted a series of researches on the air flow characteristics, supply air velocity decay and turbulence structure along the air flow trajectory. Firstly, the supply air is expelled from terminal unit and flow along the ceiling. The air velocity distribution and airflow structure were investigated using particle image velocimetry (PIV) velocimetry technique [32, 33]. The experimental result revealed that the Coandă effect would attach the air flow to the ceiling and form fully turbulence. This air transfer mode prevented occupants from directly exposure to draught and slowed the air supply velocity. Then the supply air spread over the ceiling and impinge on the ceiling wall corner. The airflow pattern of the supply air flow around the corner was identified [35]. A semi-empirical model and a CFD model were developed to describe the air flow restricted by ceiling and side wall. The experimental findings indicated that the models were effective to describe the maximum speed of air jet at low Reynolds numbers. After the corner, the air continues to drop along the vertical wall to the floor. A free convection model was proposed afterward to calculate the vertical moving air flow along the wall [36]. The wall jet velocity and temperature were recorded at different heights and various horizontal lengths along the

wall. The introduced model and experimental result showed that the velocities got maximum values close to the wall between 25mm to 50mm and decreased quasilinear under the height of 1.7 m. Finally, the cooled air collides the floor corner and spreads around the room. To avoid risk of draught, the velocity of corner airflow was modelled [37]. The returning corner airflow entrained the ambient air and reached maximum velocity at the floor surface. The proposed model could predict the corner region air flow patterns and evaluate the sensation of draught.

The air flow patterns of the ATB and passive displacement ventilation systems have many in common. The primary difference is that the ATB system utilizes the entrainment effect to enhance the ventilation process and improve the heat transfer rate. The air flow behaviors had been investigated [39] with respect to air velocity, thermal load distribution and air temperature. Rees et al. [40] conducted a series of test to investigate the air surface temperature and flow under various conditions. The results revealed that the behaviors of such ventilation systems depend on the operating conditions. Greater internal heat gain and higher ceiling surface temperature provided sufficient momentum to drive the ventilation air flow. Chen et al. [41] presented a methodology for the calculation of passive displacement ventilation system indoor airflow patterns and energy consumption. The turbulence model was developed for indoor airflow computation which showed that the displacement system gave better IAQ with significant energy saving. Further researches were done to analyze the passive air flow characteristics and the energy saving approach of the system [42]. The experiments showed that the displacement airflow pattern was fully satisfied when the internal load equaled to the cooling capacity. And excess heat load might cause thermal discomfort at low levels in the occupied zone.

In addition, the IEQ, which significantly influences the occupants' health and productivity, is desired to be investigated. The main factors that determine the active air terminal system thermal comfort, including heat load distribution and strength, air temperature, air velocity and relative humidity were evaluated. Wu et al. [43] measured the active chilled beam system air velocity and turbulence intensity under isothermal and non-isothermal environments. The air velocity would increase then decrease and detach

from the ceiling from transverse direction. Besides, higher pressure drop could enhance the strength of Coandă effect, while the larger temperature difference between supply air and room air hindered this effect. As a consequence, in the operation of active chilled beam systems, the pressure selection and temperature gradients should be optimal designed to guarantee a satisfied thermal performance. Fredriksson et al. [44] conducted some experiments to build up the temperature field below the chilled beam and visualized the transient velocity patterns of the airflow. The results showed that the air convection generated by heat source might reverse the chilled beam supply air flow pattern and produced strong oscillations through the chilled beam and on the sideways. The oscillations could cause a sensation of draught. Wu et al. [45] evaluated the effect of indoor heat sources configuration and strength on the thermal comfort in a thermal isolated room. Some common thermal comfort indices such as Air Diffusion Performance Index (ADPI), Predicted Mean Vote (PMV), Draft Rate (DR) and Vertical Air Temperature Difference (VATD) were adopted based on the test results. The analysis results showed that symmetrically distributed heat sources could provide better thermal comfort while high indoor thermal load might cause draft risk due to excessive air speed. In addition, Melikov et al. [46-52] comprehensively investigated the air flow patterns and indoor environment quality in the active chilled beam system. Based on the research, some design guidelines, including terminal units install location and considerable heat sources distribution were provided. The conclusion was made that the active chilled beams offered good indoor climate conditions and high level of flexibility with proper system configuration.

The indoor environment quality in the room with passive displacement ventilation was also studied and compared with the ACB system [27, 53, 54]. The environmental variables were measured from different positions near the subjects and thermal comfort sensations were compared. The passive displacement ventilation system had advantages in terms of uniformity gradients of temperature and air velocity distribution. Meanwhile, the displacement ventilation system was not sufficient to satisfy the thermal comfort with high heat load and could raise the pollutant concentration into the breath zone.

2.4 System modeling and optimization

Modeling and optimization of ACMV systems have long been investigated in research. Many advanced modeling methods and air conditioning system simulation tools are available in the literature [55-59]. Active air terminal is both air diffusion device and air conditioning component, the range of modeling and optimization for active air terminal system is quite different from other environments. Up to now, a few researches have been involved in this region.

To predict the entrainment process, Filipsson et al. [11] measured the quantity of the supply air for ACB system in three methods. Based on the experimental measurements, the air loop model was developed considering the primary air flow rate, chilled water temperature and heat source radiation. The proposed model was adopted to improve the self-regulating characteristics of ACB terminal units. In addition, the water side heat transfer process was also described [22]. Fernando et al. derived a generic numerical model of plate fin-and-tube cooling coil for ACB terminal unit. Steady state experimental data was used along with the coil model to determine the correlation for the air-side heat transfer. Model validation showed that the coil model was with reasonable accuracy and could be used for water circuit design. Filipsson et al. [60] proposed an ACB thermal model based on NTU analysis. The model captured the influence of air buoyance forces and minimized the extensive measurements. The thermal model predicted the cooling coil heat transfer rate with high accuracy in various operating conditions. Furthermore, the air-loop and water-loop operation characteristics of active chilled beam system were summarized. Chen et al. [61] developed an ACB model which coordinated the experimental results and first principles in hybrid manner. The model combined the unit configuration and thermodynamic of the entrainment process and cooling coil in the terminal unit. Showing robustness and high accuracy, the model could be applied to wide control and optimization applications.

The passive ventilation system is characterized by thermal stratification which is quite different from the overhead air conditioning system. Mateus et al. [62] presented an approach to model the thermal stratification in the displacement ventilation system using

three air temperature nodes. The simplified model could predict the indoor temperature gradient with significantly improved accuracy. Limit system inputs to height, size of heat sources, the model was easy to implement. Carrilho et al. [63] developed models for vertical temperature variations and heat transfer prediction in the passive displacement ventilation system. The model gave insights into mechanisms and system parameters that determine the airflow pattern and vertical temperature profile. The modelling of chilled ceilings and passive chilled beam were also available in the references [64-68]. The modeling approaches and performance could be utilized to comprehensively understand the features of active air terminal system.

To fully develop the energy saving and indoor thermal comfort potential of active air terminals, application of advanced control and optimization methods are required. Chen et al. [69] was the first one developed the fuzzy controller for ACB systems. The strong nonlinearities of the system were relaxed by T-S fuzzy method. Simulations were conducted to test the LQR methodology performance based on the mock up room and verified terminal unit. The fuzzy controller could achieve good closed-loop performance and adjust the room temperature under various operating conditions.

In real application of ACB systems, the control strategies have been greatly simplified. Trox technic [70] developed a flow limiter to adjust primary air flow rate while the room temperature was maintained through on-off control of water supply. Dadanco [71] kept primary air flow constant volume while varied the primary air temperature and humidity for cooling applications. FlaktWoods [72] installed the Pi Function accessory to modify the flow rate of fresh air which in turn affect the cooling capacity while the chilled water were kept constant at the predetermined setting points.

2.5 Terminal unit applications

Active air terminal system is not a panacea. The system design subjects to many requirements: 1) the fresh air requirement, 2) the ceiling space, 3) the cooling capacity, 4) the climate influence, 5) the application scenarios and etc. Some studies have been done

to provide the design guidelines and evaluate the practical effectiveness of active air terminal system.

Loudermilk et al. [73, 74] presented the design guidelines for the ACB system, considering thermal comfort, sizing and locating of terminal units based on the ANSI/ASHRAE standard. The case study was done to evaluate the indoor air velocity and moisture content. The results indicated that active chilled beam systems could significantly improve the IEQ in terms of the noise, draft conditions, and temperature inconsistency. In addition, Alexander et al. [75] introduced the active chilled beam systems and gave some design considerations in various conditions. The main concerns of application, including duct design and air supply static pressure, air distribution and beam placement, installation and air/water side control, were presented respectively. The energy saving potential and suitability for different spaces were also discussed. Rumsey et al. [76] expounded the application of ACB upon successful installations. Accordingly, some commissioning, operations, and maintenance issues were given. The initial costs of chilled beam system and conventional system were also compared, chilled beam system costed more on equipment level while reduced ducting and piping costs. To further minimize the operation cost of active chilled beam system, Livchark et al. [77] put forward that the design objective of the system was to minimize primary airflow and maximize use of water coil for cooling and heating. The mathematical description proved that the cooling energy produced by per volume primary air significantly influenced the active chilled beam energy efficiency.

The buoyance driven passive displacement ventilation system was used in high thermal load condition for many years. Nowadays, the system has gain increasing interests to provide comfort ventilation in low thermal load conditions. Nielsen et al. [26] presented the passive displacement ventilation system practical design procedure. Some design calculations were provided to optimal the selection of room stratification height and concentration distribution which would influence indoor temperature and velocity distribution. Emmerich et al. [78] evaluated the potential benefits and limitations of displacement ventilation system based on simulation study of energy impacts in an office building. The experimental findings indicated that stable thermal stratification depended

on the comprehensive design of internal loads, room configurations and temperatures. Naydenov et al. [28] illustrated experiments in mock up rooms with passive displacement ventilation which comprised thermal condition measurements and occupants' response collection. The results showed that the displacement ventilation required detailed design and consideration to satisfy the thermal comfort.

In the real application, certain spaces are appropriate for active air terminal use while others are not suitable for the technology [79]. Accordingly, the usage of active air terminals is largely confined to commercial buildings, offices and school. Rumsey et al. [80] investigated how to apply active chilled beam in laboratories. Based on initial cost and energy consumption calculation, the active chilled beam was proved to lower both construction costs and operation costs with refined system design. Barnett et al. [81] illustrated the energy efficiency of active chilled beam usage in cooling and heating laboratory. Energy simulations were conducted through an hourly analysis program which showed active chilled beam could save about 50% energy with roughly same first cost. Devlin et al. [82] used full scale prototype tests and simulations to verify the selection of ACB in hospitals. The results showed active chilled beam was an appropriate solution for the hospital as the system could promote a uniform temperature distribution and reduce the airborne cross-infection risk.

In addition, the passive displacement ventilation is widely utilized to regulate the indoor environment. Some literatures are available which investigate the effectiveness of the system. Shan et al. [83] conducted a field experiment to evaluate human subjects' thermal comfort in the passive displacement ventilation system. The PDV system provided satisfactory IEQ in terms of draft sensation and temperature profile. Li et al. [84] investigated the application of displacement ventilation in hospital environments. The experimental findings indicated the displacement ventilation performed better than mixing ventilation in certain conditions, especially in office, classrooms, theaters and non-critical rooms in the hospital. In practice, the passive displacement ventilation system is widely used in Nanyang Technological University as a green design approach in Figure 2.3 The PDV system installed in NTU. The tutorial rooms and sports hall use displacement ventilation system which can save 30% energy consumption.

Figure 2.3 The PDV system installed in NTU

2.6 Summary

In this chapter, the current state of the art research progress in active air terminal systems is introduced. The existing studies focus on the mechanical design of the terminal unit, indoor air flow pattern and thermal comfort, system modeling and application analysis. Based on the literatures, the researches into control and optimization of active air terminal system are still inadequate. Since active air terminal systems have just evolved for more than twenty years, researchers devote great effort on the terminal unit optimization and system design. In real applications, the control schemes have been simplified. Model-free control and optimization strategies in the active air terminal systems inevitably lead to considerable degeneration in the energy efficiency as well as the indoor environment quality. Thus, there is still a long way for the researchers to go to compensate the study and improve the system overall efficiency.

Chapter 3 Terminal unit design and experimental setup

3.1 Introduction

In practice, there are multiple designs of ACB terminal units to apply for different application environments. The distinct designs, such as heat exchanger location, nozzle dimension and etc., lead to crucial difference in air conditioning performance. In this thesis, a typical 2-way discharge ACB is investigated. Besides, as the ATB is an innovation ACMV solution, the terminal unit mechanical structure and working principle need to be clarified.

The air flow patterns and operation characteristics of ACB and ATB systems are quite different. The ACBs are usually installed at the central part of the ceiling to produce uniform air supply. The ATBs are recommended to mount on the wall to enhance the displacement ventilation process. In order to verify the active air terminals performance under various working conditions, two different experimental platforms are set up.

In this chapter, a two-way discharge ACB and an independent developed ATB are introduced. Besides, the thermal room and the ACMV system are specified.

3.2 The experimental active chilled beam

The ACBs are manufactured as shown in Figure 3.1. The terminal unit has a dimension of $0.6\text{ m} \times 1.2\text{ m} \times 0.3\text{ m}$. For the air side, the diameter is the primary air inlet is 150mm. Twenty-nine induction nozzles are installed evenly on both sides of the ACB primary air outlets. In the experiments, leak proof rubber nozzles with 7mm inner diameter are adopted to strength the entrainment effect. In the water loop, the plain fin and copper tube cooling coil consists of total 16 tubes. Besides, the coil's fin thickness and distance are 0.5 mm and 4.35 mm respectively.

Figure 3.1 The experimental ACB terminal unit

3.3 The experimental active thermosiphon beam

The schematic drawing of the ATB is demonstrated in Figure 3.2. The terminal unit consists of a housing, a drainage system, a mixing chamber, a row of customized nozzles and a heat exchanger. The shapes of housing and primary air chamber are optimized through CFD simulation to produce adequate primary air flow with minimum chamber pressure. The housing is constructed with 3 mm galvanized steel sheets and accommodates components of the terminal unit. In addition, the internal thermal insulator is attached on the inner surface of the housing to prevent condensation outside the housing and heat loss. A drain pan is installed below the heat exchanger to collect the condensate water drops from the cooling coil. The external thermal insulator is attached on the outer surface of the tray to prevent condensation and water leakage outside the terminal unit.

Figure 3.2 The schematic drawing of ATB

Thirty rubber nozzles are distributed evenly on the primary air outlet plate. Different sizes of nozzles, shown as in Figure 3.3, have been designed to accommodate difference ventilation/cooling load ratios. The nozzle is made of fire-resistant materials and specially designed to reduce the noise level. The 7mm diameter nozzles are used to balance the cooling capacity and fresh air requirement in the experiments.

Figure 3.3 Different models of induction nozzles

The cooling coil adopted in the ATB is demonstrated in Figure 3.4. A self-designed 2 circuits arrangement finned tube heat exchanger is manufactured. The heat exchanger is constructed with aluminum fins and copper pipes. Compared with the conventional cooling coil configuration, the 2 circuits arrangement coil can increase the heat transfer efficiency. Based on the simulation and actual test, the fin thickness and interval are

chosen as 0.5mm and 4.16 mm to enhance the heat transfer process. Total twenty copper tubes, which have an external diameter of 12.7mm, are distributed evenly with 2-rows staggered layout. The performance of the nozzles and the heat exchanger have been tested to validate the effectiveness.

Figure 3.4 Prototype of the heat exchanger

The prototype of various kind of full duct is shown in Figure 3.5. Various constructional conditions need to be taken into consideration when estimating the cooling capacity of terminal units. The utilization of fall duct can improve the ATB overall performance. The fall duct is made of fireproof and heat insulating phenolic foam board. To reduce air resistance, the fall duct is covered with smooth aluminum foil. The depth and width of the fall duct are 200mm and 1200mm respectively while the height can be adjusted according to the space condition. Meanwhile, the air outlet on the fall duct is of dimension 1000mm×150mm.

Figure 3.5 Prototypes of full ducts

3.4 Chiller plant and dedicated outdoor air system

A pilot plant is setup to investigate the active air terminal unit performance, including entrainment ratio, cooling capacity, water loop pressure drops and so on. The ACB and ATB system, which consists of air loop and water loop, are air-water configuration ACMV system. The air loop is designed to consistently provide pretreated fresh air to the active terminal units and maintain air plenum pressure. The water loop cools down the circulation water and supplies chilled water to the active air terminals.

Two photographs of the chiller plant setup are shown in Figure 3.6 and Figure 3.7. The system water loop has the following major components: condenser, compressor, evaporator, water tank, separator, receiver, water pump, electric expansion valve, flowmeter and control cabinet. Two sets of chiller plant are constructed to supply chilled water to terminal units and dedicated outdoor air system (DOAS) separately. The specification of the components are selected based on the internal and external load calculation.

Figure 3.6 Front view of the chiller plant

A 350W water circulating pump with a capacity of 0-6 m³/h is installed in the main pipe to maintain the water loop pressure between the evaporator and the terminal units. A water tank is placed before the pump to store chilled water and minimize water temperature fluctuation. The motorized valves are installed to regulate the chilled water flow rate.

Figure 3.7 Back view of the chiller plant

The Bitzer 4CES-6Y-40S semi-hermetic compressor is utilized to compress the refrigerant. Its rated power input is 6.0kW with the rpm 1450. The permissible cooling capacity control is from 3.04kW to 21.7kW. The Eden G3 matrix air-cool condenser with the rated capacity of 12.3kW helps ready the refrigerant for the cooling process. The chilled water temperature is maintained by the electric expansion valve which modulates the refrigerant flow rate into the evaporator.

Figure 3.8 The air handling unit

In the air loop, an AHU and a liquid desiccant dehumidification system (LDDS) operate in sequence to treat the fresh air as presented in Figure 3.8 and Figure 3.9. The AHU uses EC fan to supply air which provides better energy efficiency and speed control. The fan rated power consumption is 200W and air volume flow rate is 2000 m³/h. The AHU rated cooling capacity is 23.4kW with designed chilled water inlet temperature 6°C. In the ACB systems, the whole latent load is handled by the primary air. Consequently, the LDDS is needed to further absorb the moisture in the supply air. After treated by the DOAS, the moisture content of the primary air can be reduced to 3g/kg and meet the indoor latent load requirement.

Figure 3.9 The liquid desiccant dehumidification system

3.5 Summary

This chapter presents the experimental setup of chiller plant and DOAS. Besides, the mechanical design of the ACB and ATB terminal units are described. The pilot plant was setup based on the load calculation and the HVAC system layout. As the proposed air conditioning system has a high degree of freedom in adjusting the operation parameters. The aerodynamic and thermodynamic performance of the terminal units can be comprehensively investigated. In the following chapters, the research into active air terminal system will be presented.

Chapter 4 Modeling and validation of an active chilled beam terminal unit

4.1 Introduction

As discussed in Chapter 2, the ACB system has just evolved for more than two decades. Great effort has been paid on ACB terminal unit design and optimal operation while the research on system control and optimization is still inadequate. In real application, the control strategy has been simplified as far as possible which seriously hinders the system energy efficiency and indoor thermal comfort. Hence, develop an accurate ACB model, which is sufficient to control and optimization applications, becomes the primary goal.

The configuration of the ACB system includes two loops: in the air loop, the primary air and the secondary air mix in the terminal unit and supplied to the occupied zone; in the water loop, the chilled water in the heat exchanger cooled down the entrained air. Up to now, some existing research on ACMV system could be incorporated into the modeling of ACB terminal unit. Ruangtrakoon et al. [85] analyzed the effect of nozzle geometry on the entrainment ratio with various pressures and temperatures. Enjalbert et al. [86] developed an entrainment effect model based on Reichardt's hypothesis with respect to the conservation of momentum and mass. Ariafar et al. [87, 88] demonstrated a series of investigation into air flow out the primary nozzles. The turbulence jet models require detailed information including nozzle diameters, distribution and boundary conditions to predict the output air flow rate. Hence these models are more suitable for the ACB terminal unit design rather than system control application. Filipsson et al. [11] showed three different methods to measure the induction ratio and investigated the experiment parameters that may influence the induction ratio. The experimental findings proved that the air buoyancy also influenced the strength of entrainment effect. Then, the following model captures the entrainment effect and air buoyancy to predict the flow rate of induced air.

Considering the modeling of cooling coil, many researches have investigated the process of heat transfer between the cooling coil and the induced air. Wang [89] and Ou [90]

developed cooling coil models and analyzed the heat and mass transfer characteristics inside the LDDS based on the hybrid modeling approach to monitor the system performance. Lee [91] adopted the multi node approach and developed a simplified explicit model which can estimate the heat transfer rate of the chilled water cooling coil under both dry and wet condition. Li et al. [92] and Afram [93] integrated first principles, real time experimental results and system constructions in order to develop a cooling coil model in the AHU based on hybrid method which could detect system operation fault and efficiency. Constrained by the unique working principle of ACB, the induced air off coil temperature is difficult to measure. Hence, the modeling approach should reduce the assistant information and maintain the model accuracy. Chen et al. [61] firstly developed a hybrid model for ACB terminal unit which caught the thermodynamic and mechanical aspects of the heat exchanger and air jet. Although the air jet model was simple, the proposed model failed to evaluate the influence of air buoyancy force and the complexity of the heat exchanger model undermined its practicability. Hence, the cooling coil model requires further simplification for the monitoring, control and optimization of ACB system.

In this chapter, a hybrid model of ACB terminal unit is developed to predict the induced air flow rate and heat transfer process in the terminal unit. The model is derived based on physical and thermodynamic principles using hybrid modeling approach. The models are derived from heat transfer mechanism and the energy balance principle, while the parameters are identified by experimental data. The ACB model is combined with two sub models, namely the air entrainment model and heat transfer model. The air entrainment model captures the entrainment effect and the air buoyancy force. The simplified cooling coil model describes the heat transfer process with no more than three lumped parameters. The air side and chilled water side information has been encapsulated to expand the model application range. The unknown characteristics parameters are identified using Levenberg Marquardt method with respect to the experimental results. The model validation is conducted to verify the model effectiveness.

This chapter is structured as follows: the model development and experimental procedures are presented in Section 4.2 and 4.3; the validation results are illustrated in Section 4.4; and the summary is given in Section 4.5.

4.2 Modeling development of ACB

The interactions of the sub-models of the ACB system is demonstrated in Figure 4.1 The interaction of the ACB sub-models.

Figure 4.1 The interaction of the ACB sub-models

The following assumptions are adopted to simplify the mathematical derivation of the model:

1. Condensation is avoided during the test.
2. The air is homogeneously mixed in the air chamber.
3. The chilled water temperature field distributes evenly in the heat exchanger.
4. The heat storage is neglected in the tube of cooling coil.

5. The joints of the cooling coil and tubes of the temperature sensor are adiabatic.

4.2.1 Air entrainment model

In ACMV systems, the air flow rate is generally described by flow resistance and pressure in the duct. The plenum pressure is more accessible compared with the air side flow rate. The total flow resistance in the ACB terminal unit is affected by the dimension of the air chamber, the shape of the nozzles and so on. During the test, the flow resistance is considered as a constant and the chamber pressure is the unique variable in the entrainment process. As a consequence, it is suitable to reflect the entrainment effect via the air plenum pressure.

$$\dot{V} = \sqrt{\frac{P}{R}} \quad (4.1)$$

where $\dot{V}$ is the air volume flow rate, P is the pressure in the primary air plenum, R is the flow resistance of the total air passage.

To avoid sophisticated air entrainment theories, empirical relationship is utilized to describe how the air flow rates vary with the plenum pressure.

$$\dot{V}_{pri} = aP^b \quad (4.2)$$

$$\dot{V}_{sec_0} = cP^d \quad (4.3)$$

where $\dot{V}_{pri}$ is the primary air volume flow rate, $\dot{V}_{sec_0}$ is the second air volume flow rate without air buoyance, a, b, c and d are the unknown constant coefficients.

The induced secondary air with low ventilation velocity is sensitive to operation environment. The air through the coiling coil and the air in the conditioned zone are of different temperatures which would drive the air flow due to air buoyancy. To describe the air movement, the Boussinesq approximation is adopted.

The equation of air motion based on acceleration due to gravity:

$$g' = g \frac{\rho_{chwa} - \rho_z}{\rho_{chwa}} \quad (4.4)$$

where g' is the air effective gravity, g is the acceleration of gravity, ρ_z is the zone air density, ρ_{chwa} is the air density around the ACB cooling coil.

Refer to the ideal gas law $P\dot{V} = nRT$, under ideal condition $\dot{V} = R \frac{nT}{P}$, $\rho = \frac{pM}{RT}$, the effective gravity is described by:

$$g' = g \frac{T_z - T_{chwa}}{T_z} = g \frac{\Delta T}{T_z} \quad (4.5)$$

where T_z is the room temperature, T_{chwa} is the average chilled water supply temperature, ΔT is the average temperature difference between the room chilled water.

Adopt empirical relationship, the constants including the dimension of the room, size of the coil inlet grille, the resistance of air flow and etc. are lumped into the constant of proportionality. Combined with Eq. (4.3) the room temperature and average temperature difference are the additional manipulated parameters. The secondary air flow rate can be described by:

$$\dot{V}_{sec} = cP^d + k\left(\frac{\Delta T}{T_z}\right)^n \quad (4.6)$$

where $\dot{V}_{sec}$ is the second air volume flow rate under air buoyance, k and n are unknown constant coefficients.

The entrainment effect of the ACB terminal unit is described by entrainment ratio. The relation between the primary air flow rate, the second air flow rate, the supply air flow rate and the ER can be obtained as follows:

$$ER = \frac{\dot{V}_{sec}}{\dot{V}_{pri}} = \frac{\dot{V}_{sup}}{\dot{V}_{pri}} - 1 \quad (4.7)$$

$$V_{\text{sup}} = V_{\text{pri}} + V_{\text{sec}} = aP^b + cP^d + k\left(\frac{\Delta T}{T_z}\right)^n \quad (4.8)$$

$$ER = \frac{c}{a} P^{(d-b)} + \frac{k}{a} \frac{\Delta T^n}{P^b T_z^n} \quad (4.9)$$

where ER is the entrainment ratio, V_{sup} is the supply air flow rate.

As mentioned in the assumptions, the air is homogeneously mixed in the air chamber. The supply air temperature and second air off coil temperature can be calculated as:

$$T_{\text{sup}} = \frac{T_{\text{pri}} V_{\text{pri}} + T_{\text{sec}} V_{\text{sec}}}{V_{\text{pri}} + V_{\text{sec}}} \quad (4.10)$$

$$T_{\text{sec}} = \frac{(T_{\text{sup}} - T_{\text{pri}}) V_{\text{pri}} + T_{\text{sup}} V_{\text{sec}}}{V_{\text{sec}}} = \frac{(T_{\text{sup}} - T_{\text{pri}}) + T_{\text{sup}} * ER}{ER} \quad (4.11)$$

where T_{sec} , T_{pri} and T_{sup} are induced air off coil temperature, primary air temperature, supply air temperature.

4.2.2 Heat transfer model

The condensation is strictly avoided during the heat transfer process in the ACB. Hence, only sensible heat transfer exists between the cooling coil and the induced secondary air. The amount of heat transfer from hot secondary air to chilled water due to temperature difference can be expressed as:

$$Q = \frac{T_{a.in} - T_{c.in}}{R_h} \quad (4.12)$$

where Q , $T_{a.in}$, $T_{c.in}$ and R_h are the heat transfer rate, the secondary air inlet temperature, the chilled water inlet temperature and the overall thermal resistance respectively.

The overall resistance is composed of three parts: the thermal resistance of the air convection, the thermal resistance of the cooling coil conduction and the thermal resistance of the chilled water convection. However, the wall of the cooling coil is made of copper which has good thermal conduct. Therefore, the thermal resistance of the cooling coil can be neglected. Then the overall thermal resistance can be expressed as:

$$R_h = R_a + R_c \quad (4.13)$$

where R_a and R_c are the thermal resistance of secondary air convection and chilled water convection.

The chilled water and primary air are driven mechanically by pump and fan. So, the heat transfer between the air and the chilled water is forced convection heat transfer. The heat transfer coefficient influenced by the cooling coil diameter and the fluid thermal conductivity which can be calculated by the Reynold number Re_D and Prandtl number Pr [94]:

$$\frac{hD}{\lambda} = C Re_D^e Pr^f = C \left(\frac{D\rho v}{\mu} \right)^e \left(\frac{C_p \mu}{\lambda} \right)^f \quad (4.14)$$

where C, e and f are constant coefficients, D is the characteristic length, h is the heat transfer coefficient, λ is the thermal conductivity, ρ is the fluid density, v is the fluid flow velocity, μ is the fluid absolute viscosity, C_p is the fluid specific heat.

The assumption is made that the temperature and fluid in the cooling coil are evenly distributed. Accordingly, the chilled water density and velocity remain unchanged when steady state is obtained. Then the heat transfer coefficient can be rewritten as:

$$h = \frac{\lambda}{D} C \left(\frac{D\rho V}{\mu A} \right)^e \left(\frac{C_p \mu}{\lambda} \right)^f = \frac{\lambda}{D} C \left(\frac{4\dot{m}}{\mu \pi D} \right)^e \left(\frac{C_p \mu}{\lambda} \right)^f = x \dot{m}^e \quad (4.15)$$

where $x = \frac{\lambda}{D} C \left(\frac{4}{\mu \pi D} \right)^e \left(\frac{C_p \mu}{\lambda} \right)^f$, A is the fluid section area, V is the fluid volume flow rate, $\dot{m}$ is the fluid mass flow rate.

The total heat resistance can be rewritten as:

$$R_h = \frac{1}{h_a A_a} + \frac{1}{h_c A_c} \quad (4.16)$$

For $h = x\dot{m}^e$

$$R_h = \frac{1}{h_a A_a} + \frac{1}{h_c A_c} = \frac{x_a A_a \dot{m}_a^e + x_c A_c \dot{m}_c^e}{x_a A_a \dot{m}_a^e x_c A_c \dot{m}_c^e} \quad (4.17)$$

where $h_a, h_c, A_a, A_c, \dot{m}_a$ and $\dot{m}_c$ are the heat transfer coefficients, the mass flow rates, the heat transfer areas of induced air and chilled water respectively. x_a and x_c are constant parameters need to be identified.

Combining Eqs. (4.12) and (4.17), the heat transfer rate in the cooling coil can be expressed as:

$$Q = \frac{x_a A_a \dot{m}_a^e x_c A_c \dot{m}_c^e}{x_a A_a \dot{m}_a^e + x_c A_c \dot{m}_c^e} (T_{a.in} - T_{chw.in}) = \frac{x_c A_c \dot{m}_c^e}{1 + \frac{x_c A_c \dot{m}_c^e}{x_a A_a \dot{m}_a^e}} (T_{a.in} - T_{chw.in}) \quad (4.18)$$

Based on the conservation of energy, the heat incremental of secondary air is equal to the heat transfer rate in the cooling coil as the heat specific of cooling coil is neglected. Then we obtained:

$$\begin{aligned} Q &= -C_a \dot{m}_a (T_{a.in} - T_{a.out}) = \frac{x_c A_c \dot{m}_c^e}{1 + \frac{x_c A_c \dot{m}_c^e}{x_a A_a \dot{m}_a^e}} (T_{a.in} - T_{chw.in}) \\ &= \frac{b_1 \dot{m}_c^e}{1 + b_2 \left(\frac{\dot{m}_c}{\dot{m}_a}\right)^e} (T_{a.in} - T_{chw.in}) \end{aligned} \quad (4.19)$$

where $b_1 = x_c A_c$, $b_2 = x_c A_c / x_a A_a$, C_a is the heat specific capacity of air, $\dot{M}_a$ is the mass flow rate of secondary air. $T_{a.in}$ and $T_{a.out}$ are secondary air on and off coil temperature.

Then, the heat transfer rate of the cooling coil is rewritten as:

$$Q = \frac{b_1(\rho_a \dot{m}_c)^e}{\rho_a^e + b_2 \left[\frac{\dot{m}_c}{cP^d + k \left(\frac{\Delta T}{T_z} \right)^n} \right]^e} (T_{a.in} - T_{chw.in}) \quad (4.20)$$

where ρ_a is the air density.

The proposed ACB model is of brief structure and characterized by fewer parameters which can be identified through experimental results using nonlinear least square method.

4.2.3 Parameter identification

To estimate the model parameters, the nonlinear least square method is adopted.

$$\begin{aligned} f(u) &= \sum_{i=1}^N r_i^2(u) = \sum_{i=1}^N (aP_i^b + cP_i^d + k \left(\frac{\Delta T_i}{T_{z,i}} \right)^n - V_i)^2 \\ f(c) &= \sum_{j=1}^N r_j^2(c) = \sum_{j=1}^N \left(\frac{b_1 \dot{m}_{c,j}^e}{1 + b_2 \left(\frac{\dot{m}_{c,j}}{\dot{m}_{a,j}} \right)^e} (T_{a.in,j} - T_{chw.in,j}) - Q_j \right)^2 \end{aligned} \quad (4.21)$$

where f is the sum of the squares of the residuals between identified results and experimental results; r_i is the residuals between identified results and experimental results; $u = [k \ n]^T$ and $c = [b_1 \ b_2 \ e]^T$ are the parameter vectors to be identified; V_i is the experimental supply air flow rate; Q_j is the experimental heat transfer rate.

The Levenberg-Marquardt method is adopted to search for the optimal solution for the unidentified parameters. The descent direction is obtained as follows:

$$(J^{(k)})^T(c)J^{(k)}(c) + \lambda^{(k)}I)P^{(k)}(c) = -J^{(k)T}(c)r^{(k)}(c) \quad (4.22)$$

where $r(c)=[r_1(c) \ r_2(c) \ \dots \ r_N(c)]^T$, $\lambda^{(k)} \geq 0$ is a scalar, I is the identity matrix, $P^{(k)}$ is in a descent direction.

The Jacobian matrix is defined as:

$$J = \begin{bmatrix} \frac{\partial r_1}{\partial b_1} & \frac{\partial r_1}{\partial b_2} & \frac{\partial r_1}{\partial e} \\ \frac{\partial r_2}{\partial u_1} & \frac{\partial r_2}{\partial b_2} & \frac{\partial r_2}{\partial e} \\ \vdots & \vdots & \vdots \\ \frac{\partial r_N}{\partial u_1} & \frac{\partial r_N}{\partial b_2} & \frac{\partial r_N}{\partial e} \end{bmatrix} \quad (4.23)$$

In each iteration, suitable value should be given to $\lambda^{(k)}$ which can be expressed as:

$$\lambda^{(k+1)} = \begin{cases} \lambda^{(k)} / \nu & \text{if } f^{(k+1)} < f^{(k)} \\ \lambda^{(k)} \nu & \text{if } f^{(k+1)} > f^{(k)} \end{cases} \quad (4.24)$$

And

$$c^{(k+1)} = c^{(k)} + P^{(k)} \quad (4.25)$$

The iteration ends if $|c^{(k+1)} - c^{(k)}| < \mu$, where μ is the tolerance set as 1×10^{-5} .

4.3 Experimental procedure

To develop an accuracy and feasible model for ACB terminal unit, experiment are conducted under a wide operation range. Both the air loop and water loop parameters are varied during the experiments. Fans and pumps are equipped with VSD to modulate the primary air and chilled water volume flow rate. In addition, the heating panels and chillers can provide various thermal load and cooling capacity to meet different cooling demands. The steady state data sets are recorded in the data acquisition system. 3 cases of experiments are conducted to comprehensively evaluate the ACB terminal unit

performance. In case 1, the influence of chamber pressure on the air flow rates are investigated. During experiments, the temperatures of the chilled water, primary air and thermal room are set to be the same to eliminate the disturbance of air buoyancy. The pressure in the primary air chamber is adjusted from 30Pa to 270 Pa and the corresponding primary air flow rate and secondary air flow rate are recorded. For case 2, the room temperature (22-28°C), chilled water flow rate (0.02-0.2L/s) and chilled water temperature (13-18°C) are regulated separately to estimate the effect of buoyancy on the secondary air flow rate. For each data set, the operating conditions and the corresponding primary air flow rate and secondary air flow rate are recorded. In case 3, the operation parameters of the cooling coil are recorded under typical working conditions for the identification of the heat transfer model. The primary air chamber pressure, room temperature, chilled water flow rate and chilled water temperature are regulated separately. In addition, the chilled water inlet temperature is kept above the dew point. The operation parameters and the chilled water outlet temperature are recorded to calculate the heat transfer rate.

The operating ranges of the system are shown in Table 4.1.

Table 4.1 The operating ranges of ACB system

Parameters	Range	Unit
Room temperature	22.0-28.0	°C
Plenum pressure	30-270	Pa
Chilled water inlet temperature	13-18	°C
Chilled water flow rate	0.02-0.2	L/s

In addition, the experimental results are obtained in steady states. The air loop steady state is confirmed when the reading variation from the room temperature is within 0.1 °C and the pressure transmitter is within 5Pa for 5 minutes. The thermal equilibrium of the air side and water side heat transfer processes are confirmed when the temperature variations are within 0.1 °C for 15 minutes.

The specifications of all the sensors installed in the duct and water pipes are depicted in Table 4.2.

Table 4.2 Sensor specification

Sensors	Product Model	Accuracy	Measuring Range
Air temperature	EE21	± 0.2 °C	-40~60°C
Air humidity	EE21	$\pm 2\%$ RH	0~100% RH
Differential pressure	Dwyer MS-111	2%	0~250Pa
Air flowmeter	8710 DP-CALC	$\pm 3\%$	42~4250 m ³ /h
Water temperature	Siemens QAE21	± 0.3 °C	-30~130°C
Water pressure difference	EJA110A	$\pm 0.065\%$ of Span	1~100kPa
Water flow rate	LWGY-A	1%	100~600L/h

4.4 Model validation

The effectiveness of the proposed model is evaluated through relative error and root mean square of relative error, which is expressed as follow:

$$RE = \frac{|V_c - V_r|}{V_r} \times 100\% \quad (4.26)$$

$$RMSRE = \sqrt{\frac{\sum_{i=1}^N (RE_i)^2}{N}} \quad (4.27)$$

where V_c is the calculated value, V_r is the actual measured value, N is the number of fitted points.

Table 4.3 Summary of identified parameters

Model	Identified model parameters
Air entrainment	$a=10.1770$, $b=0.5173$, $c=22.4322$, $d=0.5474$, $k=-54.4590$, $n=1.1099$
Heat transfer	$b_1=1383.5998$, $b_2=2.4672$, $e=0.8125$

For the validation of the air entrainment model, totally 31 data sets of primary air volume flow rate and corresponding plenum pressure are collected. Among the data sets, 11 data sets are randomly selected for model identification while the rest 20 data sets are used to compare with the predicted value. The primary air fitting and validation results are given in Figure 4.2 and Figure 4.3.

Figure 4.2 Experimental fitting for the primary air volume flow rate

Figure 4.3 Model validation for primary air volume flow rate

As the entrainment process is more susceptible to interference, repeat tests are conducted to minimize the experimental error. Meanwhile, to estimate the influence of air buoyancy on entrainment effect, the air loop and water loop parameters are adjusted within a large range. Totally 165 data sets are collected in respect to secondary air flow rate, chamber pressure difference, room temperature and chilled water inlet temperature. 45 data sets are randomly selected to identify the model coefficients and the rest 120 data sets are used for model validation. The model validation is illustrated in Figure 4.4.

Figure 4.4 Model validation for secondary air volume flow rate

Based on the identified coefficients, the air entrainment model is validated under various primary air plenum pressures and average temperature differences (the chilled water inlet temperature and room temperature are adjusted separately) as shown in Figure 4.5 to Figure 4.7.

Figure 4.5 Model validation for secondary air volume flow rate without air buoyancy

Figure 4.6 Model validation for secondary air volume flow rate under various chilled water inlet temperatures

Figure 4.7 Model validation for secondary air volume flow rate under various room temperatures

The influence of the chilled water on the entrainment effect is investigated. The test results indicate that the average temperature differences have adverse impact on the entrainment effect. As shown in Figure 4.8, the entrained secondary air is negatively correlated with the average temperature difference. The secondary air flow rate is

reduced by 7% when the average temperature difference approaches 12°C. Meanwhile, the water side flow rate has insignificant effects on the induced flow rate of secondary air as described in Figure 4.9. Compared the differences in air flow between the 3 cases, the maximum difference is within 5L/s (approximately 1% of the measured flow rata). Both tests are conducted under the same condition except horizontal axis parameters.

Figure 4.8 Secondary air volume flow rate under various average temperature differences

Figure 4.9 Secondary air volume flow rate under various chilled water flow rates

For the heat transfer model, totally 58 data sets in wide operation range are recorded. After the steady state obtained, the corresponding chilled water inlet temperature, chilled water flow rate, primary air plenum pressure and induced air temperature are collected. Randomly select 26 data sets to calibrate the model and the rest 32 for model validation. The model validation is illustrated in Figure 4.10.

Figure 4.10 Model validation for heat transfer rate

Table 4.4 Summary of the assessment criteria

Error index	V_{pri}	V_{sec}	Q
RE	2.62%	0.93%	4.85%
RMSE	2.41%	2.38%	5.50%

The comparison results of air entrainment model and heat transfer model are summarized in Table 4.4. For the air entrainment model, the average RE is 1.17%. While for the heat transfer model, the average RE is 4.85%. Based on the curve fitting and model validation results, the proposed ACB model is precise and sufficient for control and optimization applications.

4.5 Summary

In this chapter, a simple yet accurate ACB model is developed under hybrid manners. The simple and precise air entrainment model captures air buoyancy and entrainment effect with six identified parameters. The heat transfer model is derived into only three

lumped parameters by analyzing first principles and experimental results. On the basis of experimental findings, the conclusions are summarized below:

1. The primary air plenum pressure and the average temperature difference are the main variables that determine the flow rates of primary air and secondary air. While the chilled water flow rate has insignificant effects on the induced air flow rate.
2. The air buoyancy generated by temperature difference impedes the entrainment process which reduces the flow rate of secondary air by 7% in the cooling mode. Both the room temperature and chilled water inlet temperature are correlated with the entrainment ratio.
3. The ACB sub-models have good agreements with the experimental results under different operation conditions (average RE<5%). The proposed ACB model simplifies the calculations and processes for real engineering applications.

The proposed ACB model can be further applied to real time performance evaluations and optimizations. The parameters, which affect the entrainment ratio, are analyzed and they can improve the ACB performances including avoiding condensation and increasing the operation efficiency. However, there are more researches to be noted, such as modifying the ACB structures to take advantage of air buoyancy and optimizing the system performance using the proposed ACB model.

Chapter 5 Mechanical design and performance evaluation of active thermosiphon beam terminal units

5.1 Introduction

In the last chapter, a model of ACB terminal unit is developed with respect to air buoyancy. The experimental results indicate that the air buoyancy dramatically hinders the operation efficiency of ACB. Hence, it is necessary to improve the mechanical design of the terminal units to achieve better operation efficiency. Combining air entrainment effect and displacement ventilation, the ATB is developed as an innovation solution to ACMV system. As there is no dedicated study on the ATB until now, some references on the ACB and passive displacement ventilation systems are incorporated into the research of ATB.

The PDV is an emerging terminal unit which gains incremental interests in America and Asian countries [95, 96]. The design and performance of PDV system have been investigated to further improve the system energy efficiency and IAQ. Chen et al. [97] provided design guidelines and calculation standard for the PDV system. The results indicated that the PDV system could provide high quality indoor environment and save energy under high cooling load conditions if designed properly. Novoselac et al. [27, 53, 98] studied the combined PDV-cooled ceiling system and compared the performance to the VAV system. The combined system had advantages in terms of IAQ and thermal distribution but required certain temperature gradient to remove air contaminant. Shen et al. [83] compared the performances of PDV system and mixing ventilation with respect to indoor thermal comfort, short term performance and SBS in two tutorial rooms. The results revealed that PDV had less overall draft sensation and satisfactory short term performance with proper system control. In reality, many theoretical researches had optimized the overall performance of the PDV system. Hunt et al. [99] analyzed the ventilation driven by buoyance force. A theoretical model was built which can be applied to predict natural ventilation in the building. Fredriksson and Nelson [100, 101] investigated the effect of thermal load configuration and false ceiling on the efficiency of displacement ventilation. Koskela et al. [102, 103] characterized the ACB operating

performance under summer, winter and midseason. Li [104] and Xu [105] investigated the SHR and total cooling capacity of direct expansion ACMV system under various working conditions.

Based on the above research, the mechanical design and working principles of ATB are demonstrated in this chapter. Meanwhile, the ATB system, which captures the advantages of PDV and ACB systems, is sensitive to the operation condition and internal load distribution. Experiments are conducted under various working condition to investigate the hydrodynamic and thermodynamic characteristics of the terminal unit. Based on the experimental results, the ATB system has outstanding performances in regulating the indoor environment with respect to energy saving, IAQ improvement and thermal comfort. In addition, the ATB overcome all the drawbacks of the ACB system including condensation, high initial cost and poor chilled water temperature control.

In this chapter, the mechanical design and working principle of ATB are introduced in Section 5.2; the experimental setup and theoretical analysis are proposed in Section 5.3; the experimental results are illustrated in Section 5.4; a brief summary is given in Section 5.5.

5.2 The ATB working principle

Figure 5.1 The air flow patterns of the ATB system

The air flow patterns of the ATB system are demonstrated in Figure 5.1. Comparing with ACB, the utilization of the thermosiphon effect (passive heat exchange based on natural convection across air temperature gradients) and the additional water drainage system are the core innovation of ATB. The ATB terminal units are installed in the layer of warm air formed above the occupied zone. And the terminal units can operate in two models: active mode and passive mode. For the active mode, the DOAS is needed to continuously deliver fresh air to build up primary chamber pressure. The primary air is then discharged through the nozzles with high velocity which generates the negative pressure kernel behind the cooling coil. The warm air in the ceiling height will be induced through the heat exchanger due to air entrainment and fluid thermosiphon effects. The primary air and induced air combine in the chamber. The cooled air moves to the floor level along the fall duct and delivers slowly across the room. During the operation, the condensate water formed on the surface of the cooling coil drops into the tray and drained out via gravity.

When the primary air is cut off, the ATB can operate in passive mode. A fully-stratified displacement air distribution system is demonstrated in Figure 5.2. Due to the gravitational force caused by the high density cooled air, the primary air in the chamber will drop along the air straightener to the floor level and then spread over the occupied zone. Once the high density supply air encounters the heat source, it absorbs the internal heat and rises towards the ceiling. As a consequence, the low density warm air forms above the ceiling and induced to the cooling coil. The internal load is removed by this ventilation process.

Figure 5.2 The temperature distribution of the ATB system

The typical chilled water inlet temperature of ATB is 8°C, the condensate water forms on the heat exchanger will drop to the drain pan and discharge through the water drainage system. This process can remove the indoor moisture content and maintain the relative humidity in the occupied zone. Consequently, the supply air volume flow rate from the DOAS is reduced to meet the basic fresh air requirement without considering the latent load.

5.3 Experimental study

5.3.1 The experimental setup

An ATB system is setup at NTU Eugenia Room (a meeting room at N2.1-B2-17). The Eugenia room can accommodate up to 20 people with the diameter of 7m×5m×2.8m (+1m above the ceiling). The ATB terminal unit (external diameter 450mm*1200mm) used in the test adopts plain fin-and-tube cooling coil with a dimension of 1062mm×325mm×78mm. Total thirty nozzles with 7mm inner diameter are evenly distributed inside the unit. To optimize the ventilation process, the ATB is mounted on the wall above the ceiling and 20 pieces of false ceiling are replaced by return air grille. In addition, one chiller plant is equipped with the capability to adjust the water pressure and control the water temperature. A booster fan with VSD is installed in the fresh air duct to keep sufficient static pressure of primary air system. The indoor temperature and moisture content are monitored by a thermostat. The primary air plenum pressure is obtained by a pressure difference sensor. The pitot tube is adopted to record the primary air flow rate. The water side temperatures are evaluated by PT-100 platinum resistance temperature transmitters. And a turbine flowmeter is installed to measure the chilled water flow rate. The data acquisition system is set up to collect experimental results with a sampling rate of 1s. All the water loop sensors and pipes are covered with insulation foam to minimize the uncertainty of measurements.

Figure 5.3 The NTU Eugenia Room

The sensors installed are depicted in Table 5.1.

Table 5.1 Summary of sensor specification

Sensors	Product Model	Accuracy	Measuring Range
Air temperature	EE21	± 0.2 °C	-40~60°C
Air humidity	EE21	$\pm 2\%$ RH	0~100%RH
Air velocity	TSI 8475	$\pm 3\%$	0~2.5m/s
Air flow rate	KIMO-C310	$\pm 5\%$	0-600m ³ /h
Water temperature	Siemens QAE21	± 0.3 °C	-30~130°C
Water flow rate	LWGY-A	0.5%	1~600L/h
Differential pressure	Dwyer MS-111	2%	0~250Pa

5.3.2 The experimental procedures

This study is carried out to analyze the operation characteristics of the ATB terminal unit. To achieve a comprehensive conclusion, the tests are carried out under various configurations and operating conditions. The influences of primary air plenum pressure, average temperature difference (temperature difference between chilled water and the occupied zone), chilled water flow rate and fall duct length on the ATB performance are evaluated individually.

Three cases of experiments are conducted to investigate the ATB cooling performance. For the first case, the VSD booster fan is modulated to estimate the effect of primary air plenum pressure on the ATB cooling performance. In the second case, the chilled water volume flow rate and average temperature difference are adjusted to investigate the coil cooling heat transfer characteristics. In the last case, the length of air straighter is regulated to estimate the influence of air straighter on the cooling capacity of the unit.

The experimental results are recorded in steady state. Hence, the ATB system is in thermal equilibrium with constant air flow movement and temperature distribution. In addition, repeat tests are done to eliminate the uncertainty error caused by the experiment condition and improve the results accuracy. The operation ranges of the parameters are listed in Table 5.2.

Table 5.2 Summary of system setting points

Parameter	Set value	Unit
Zone temperature	24	°C
Zone humidity	55	%RH
Plenum pressure	20-130	Pa
Chilled water flow rate	100-550	L/h
Fall duct length	1.0-2.5	m
Chilled water inlet temperature	8.0-12.0	°C

5.3.3 Theoretical analysis

The heat transfer rate between the secondary air and chilled water under steady state can be calculated by the water side information.

$$Q_{total} = C_{chw} \rho_{chw} \dot{V}_{chw} (T_{chw.out} - T_{chw.in}) \quad (5.1)$$

where Q_{total} is the cooling coil heat transfer rate, C_{chw} is the water specific heat capacity, ρ_{chw} is the chilled water density, $\dot{V}_{chw}$ is the chilled water flow rate, $T_{chw.out}$ and $T_{chw.in}$ are the chilled water outlet and inlet temperature.

With a known function introduced previously, the flow rate of primary air can be derived from primary chamber gauge pressure.

$$\dot{V}_{pri} = f(P_a) \quad (5.2)$$

As the nominal chilled water inlet temperature for the ATB system is 8°C, the condensation would occur during the ATB operation. Hence, the ATB can meet both sensible and latent load requirement. The secondary air volume flow rate can be calculated via the law of conservation of energy:

$$H_{pri} \rho_a \dot{V}_{pri} + H_{sec} \rho_a \dot{V}_{sec} - Q_{total} = H_{am} \rho_a (\dot{V}_{pri} + \dot{V}_{sec}) \quad (5.3)$$

$$H = g(T \text{ } RH) \quad (5.4)$$

$$\dot{V}_{sec} = \frac{(H_{am} - H_{pri}) \rho_a \dot{V}_{pri} + Q_{total}}{(H_{sec} - H_{am}) \rho_a} \quad (5.5)$$

where ρ_a is the air density, $\dot{V}_{pri}$ is the primary air flow rate, H_{pri} is the primary air enthalpy, $\dot{V}_{sec}$ is the second air flow rate, H_{sec} is the second air enthalpy and H_{am} is the mix air enthalpy.

Combine with Eq. (5.5), the secondary air off-coil temperature and relative humidity can be calculated by inverting the air mixing process.

$$T_{pri} \dot{V}_{pri} + T_{asoc} \dot{V}_{sec} = T_{am} (\dot{V}_{pri} + \dot{V}_{sec})$$

$$T_{asoc} = \frac{T_{am} \dot{V}_{pri} - T_{pri} \dot{V}_{pri} + T_{am} \dot{V}_{sec}}{\dot{V}_{sec}} \quad (5.6)$$

$$W_{pri} \rho_a \dot{V}_{pri} + W_{asoc} \rho_a \dot{V}_{sec} = W_{am} \rho_a (\dot{V}_{pri} + \dot{V}_{sec})$$

$$W_{asoc} = \frac{W_{am} \rho_a \dot{V}_{pri} - W_{pri} \rho_a \dot{V}_{pri} + W_{am} \rho_a \dot{V}_{sec}}{\rho_a \dot{V}_{sec}} \quad (5.7)$$

where T_{pri} is the primary air temperature, T_{asoc} the secondary air off-coil temperature, T_{am} is the mix air temperature. W_{pri} is primary air moisture content, W_{am} is mix air moisture content and W_{asoc} is secondary air off-coil moisture content.

Then, the sensible and latent cooling capacity of the heat exchanger can be calculated separately

$$Q_{sen} = \dot{V}_{sec} \rho_a C_a (T_{as} - T_{asoc}) \quad (5.8)$$

$$Q_{lat} = Q_{total} - Q_{sen} \quad (5.9)$$

where Q_{sen} is the sensible cooling capacity and Q_{lat} is the latent cooling capacity.

The overall cooling capacity of ATB terminal unit is the sum of cooling from primary air and heat exchanger.

$$Q_{sum} = Q + Q_{pri} = C_{chw} \rho_{chw} \dot{V}_{chw} (T_{chw.out} - T_{chw.in}) + (H_{sec} - H_{pri}) \rho_a \dot{V}_{pri} \quad (5.10)$$

where Q_{sum} is the total cooling capacity of ATB terminal unit.

5.3.4 Assessment criteria

The entrainment ratio, sensible heat ratio and heat transfer efficiency are used as the performance indexes to evaluate heat transfer performance of the ATB.

Inducing the secondary air across the cooling coil without fan energy requirement (entrainment effect) is one of the core innovations of ATB technology. Entrainment ratio is used to evaluate the entrainment effect.

$$ER = \frac{\dot{V}_{sec}}{\dot{V}_{pri}} \quad (5.11)$$

where $\dot{V}_{sec}$ is the secondary air volume flow rate, ER is the entrainment ratio and $\dot{V}_{pri}$ is the primary air volume flow rate.

The ATB takes advantage of chilled water to handle the indoor latent load and sensible load. In the ATB system, the primary volume flow rate is minimized to satisfy the fresh air requirement without considering the indoor latent load. The sensible heat ratio is applied to estimate the terminal unit latent cooling capacity.

$$SHR = \frac{Q_{sen}}{Q_{total}} \quad (5.12)$$

where SHR is the sensible heat ratio

Under the same operation condition, the heat transfer effectiveness of ATB is also influenced by the unit configuration, nozzle design and etc. As a consequence, the heat transfer effectiveness is used to evaluate the terminal unit mechanical design and cooling performance. It is defined as the ratio of the water side heat transfer rate to the primary air volume flow rate.

$$\eta_h = \frac{Q_{total}}{\dot{V}_{pri}} \quad (5.13)$$

where η_h is the heat transfer effectiveness.

5.4 Experimental results

The influences of primary air plenum pressure, average temperature difference, fall duct length and chilled water flow rate on the performance of ATB are investigated with respect to total cooling capacity, ER, SHR, and heat transfer effectiveness.

5.4.1 Primary air plenum pressure

Figure 5.4 - Figure 5.6 illustrate the variations of heat transfer rate, SHR and secondary air volume flow rate under different plenum pressures. The primary air plenum pressure has a significant influence on the ATB cooling performance which determines the entrainment ratio and affects the flow rate of secondary air. Based on the experimental results, the secondary air volume flow rate and heat transfer rate grow exponentially with the increase of primary air plenum pressure. As the increment of secondary air flow rate facilitates the heat exchange between the induced air and the cooling coil. From Figure 5.5 and 5.6, the SHR and ER decrease with the plenum pressure which reverses the tendency of secondary air flow rate. For the ATB system, the fluid thermosiphon effect induces the majority of secondary air when the plenum pressure is low. Meanwhile, the thermosiphon process is susceptible to the indoor conditions and the fluctuation of the SHR is larger. The variations of the thermosiphon process in turn influence the sensible and latent cooling capacity of the ATB. When the plenum pressure increases, the entrainment effect becomes stronger and more secondary air is induced. However, the entrainment process has a coupling effect on the thermosiphon process which hinders the increment of secondary air and drives a drop in entrainment ratio.

Figure 5.4 The influence of primary air plenum pressure on heat transfer rate

Figure 5.5 The influence of primary air plenum pressure on SHR

Figure 5.6 The influence of primary air plenum pressure on ER

5.4.2 Chilled water flow rate

Figure 5.7 and Figure 5.8 illustrate how the heat transfer rate of cooling coil varies with the chilled water flow rate. The average temperature difference is set at 12°C, the primary air plenum pressure is maintained at 120Pa and fall duct length is 2.5m. The increment of chilled water flow rate leads to a larger heat transfer rate. The heat transfer rate increases with the chilled water flow rate and finally reaches 2400W. Larger heat transfer rate results in lower secondary air off coil temperature and more moisture content is removed from the induced air. Hence, the incremental rate of sensible cooling capacity declines faster than latent cooling capacity and the SHR decrease with the increase of chilled water flow rate. Figure 5.9 and Figure 5.10 demonstrate the influence of chilled water flow rate on the ER and the supply air temperature respectively. Larger chilled water flow rate leads to higher secondary air flow rate and lower supply air temperature. However, the growth rate of heat transfer rate and ER decrease rapidly when the chilled water flow rate exceed 400L/h/. In addition, the volume flow rate of chilled water is associated with the pressure drop of water pipe, the increment of the chilled water flow rate consumes higher pump energy. The chilled water flow rate is recommended to be less than 400L/h.

Figure 5.7 The influence of chilled water flow rate on heat transfer rate

Figure 5.8 The influence of chilled water flow rate on SHR

Figure 5.9 The influence of chilled water flow rate on ER

Figure 5.10 The influence of chilled water flow rate on supply air temperature

5.4.3 Average temperature difference

Figure 5.11 reveals the variations of heat transfer rate under different average temperature differences. In the test, the primary air plenum pressure is constant at 120 Pa, the chilled water flow rate is set at 400L/h and fall duct is modified to 2.5m. The fluid

thermosiphon effects mainly depend on the air buoyancy across temperature gradients. As shown in Figure 5.13, the average temperature difference between chilled water and displacement ventilation air can reinforce the ventilation process which in turn enhances the cooling capacity. The experimental results indicate that the changes of heat exchange rate and heat transfer effectiveness follows the variation of the average temperature difference. As shown in Figures 5.12 and 5.14, the secondary air off coil temperature drops with the increase of average temperature difference and stabilizes at 15°C. Meanwhile, the latent cooling capacity of ATB increases and finally reaches 1300W. Hence, there is a U-type curve of the SHR and the minimum value locates between 12 - 12.5°C. According to Figure 5.14, the supply air temperature drops below 15°C when the average temperature difference exceeds 13°C. As the supply air temperature significantly influences the indoor thermal comfort, the supply air temperature is recommended to be higher than 15°C. Otherwise, the occupants may have the sensation of draught at floor level.

Figure 5.11 The influence of average temperature difference on heat transfer rate

Figure 5.12 The influence of average temperature difference on SHR

Figure 5.13 The influence of average temperature difference on ER

Figure 5.14 The influence of average temperature difference on supply air temperature

5.4.4 Full duct length

Figure 5.15-Figure 5.17 demonstrate the influence of the fall duct on the ATB cooling performance. To emphasize the influence of fall duct, the chilled water flow rate is set at 500L/h while the primary air plenum pressure is 120Pa and the average temperature difference is 12.5°C. The ATB operates under the same condition except for the length of air straightener and the corresponding data sets are recorded in one hour. The heat transfer rate of the ATB with 2.5m fall duct is almost 450W larger on average than that with 1.0m fall duct. In the ATB system, the off-coil air is cooled and quickly drops through the fall duct. As illustrated in Figure 5.18, the fall duct can restrict the air mass to descend to the floor level without spreading at half height. Hence, the air straightener can promote the ventilation air flow rate which improves the heat transfer rate between the chilled water and secondary air. However, higher ventilation rate leads to higher secondary air off coil temperature and the SHR of the 2.5m fall duct is slightly higher than that of the 1m fall duct as shown in Figure 5.17. In addition, the length of fall duct determines the ceiling height and the cooling space. Based on the air flow pattern within the ATB system, the warm air above the terminal unit will remain untreated which can reduce the internal load and minimize energy consumption. Consequently, the

interactions between cooling capacity and energy efficiency need to be considered when choosing the length of fall duct.

Figure 5.15 The influence of fall duct on heat transfer rate

Figure 5.16 The influence of fall duct on sensible and latent cooling capacity

Figure 5.17 The influence of fall duct on SHR

Figure 5.18 The influence of fall duct on ER

5.4.5 Performance comparison with ACB and PDV

The performance comparisons of ATB, ACB and PDV are analyzed to investigate the system operation characteristics. The comparison tests of the ATB, ACB and PDV are conducted under the same room conditions including the indoor temperature (24°C) and

indoor relative humidity (60%). For the ATB and PDV system, the chilled water flow rate and chilled water inlet temperature are set at 360L/h and 8°C respectively. As condensation is strictly avoided during the operation of ACB, the chilled water inlet temperature of the system is set at 14°C and the flow rate is 360L/h. In addition, the dimensions of the heat exchangers of the three terminal units are the same. During the test, there is no obstacle that block the air circulation.

Table 5.3 Performance criteria of the terminal units

Terminal unit	Cooling capacity (W)	ER	SHR	η_h
ATB	2150	3.66	0.42	27.3
ACB	1400	2.63	0.73	7.78
PDV	1520	NA	0.52	NA

As shown in Table 5.3, the ATB has distinct advantages in terms of cooling capacity and heat transfer effectiveness. In addition, the ATB and PDV have better dehumidification ability as the most of indoor moisture content is removed by condensation process on the surface of the cooling coil. For the ACB system, the entire latent load is handled by the primary air which impairs the energy efficiency as excess fresh air is pretreated and supplied to the occupied zone. In addition, the number of ATB terminal units is the lowest under same heat load requirement which significantly reduces the initial cost.

5.5 Summary

In this work, a primary study of ATB was conducted under various operation conditions to investigate the influences of relevant parameters on the performance of the terminal unit. On the basis of experimental findings, the conclusions are summarized below:

- The plenum pressure is associated with the flow rate of secondary air and heat transfer rate of the cooling coil. Higher primary air plenum pressure gives higher cooling capacity and lower entrainment ratio. In the real application, the plenum pressure can be modulated based on indoor heat load and the number of occupants.

- The ATB heat transfer rate increases with chilled water volume flow rate. However, larger chilled water flow rate leads to slower growth rate of heat transfer rate and higher energy consumption. The chilled water flow rate is recommended to be less than 400L/h.
- The temperature difference between the chilled water and the secondary air can improve the displacement ventilation and the heat transfer efficiency. Meanwhile, large average temperature difference may lead to the sensation of draught. The results from the experiments indicate that the average temperature difference should be maintained around 12°C to provide considerable heat transfer capacity and thermal comfort
- The fall duct enhances the displacement ventilation by forming the temperature gradients and straightening the air mass towards the floor level. It is reasonable to set the fall duct length around 3m to balance the heat transfer rate and cooling space.
- The ATB can handle both sensible load and latent load. In the experiments, the sensible heat ratio (SHR) can reach below 50% which proved that the ATB is adequate to meet extreme load conditions (the test is conducted in the tropical region and the Eugenia Room is occupied in general).

Based on the experimental results and the theoretical analysis, the ATB can provide sufficient sensible and latent cooling capacity with high energy efficiency. The experimental evaluation provides a guideline for the applications of ATB systems. The performance and energy efficiency of the ATB system can be further improved by adjusting system design and optimizing the operation parameters. Consequently, the active thermosiphon beam has the potential to become the standard equipment for ACMV systems in modern buildings with complex layout and multiple functions. As the proposed ATB is still in the preliminary stage, there is more research work can to be done. For example, the air flow patterns in the occupied zone can be investigated and the model of ATB terminal unit can be developed for real-time control and optimization.

Chapter 6 Model-based optimization for ATB system

6.1 Introduction

The performance of ATB is evaluated in the last chapter which shows that the terminal unit can provide adequate cooling capacity and satisfied thermal comfort. Moreover, the performance and energy efficiency of the ATB system can be further improved by optimizing system design and operation parameters. It is of great essence to develop a model-based optimization strategy for the ATB system to maintain the indoor air quality and reduce the energy consumption.

In order to precisely predict the system performance and formulate the optimization strategy, it is necessary to review some existing research works on the optimization of ACMV system. Henrique et al. [106] proposed an optimization scheme to search for the optimal setting points of the ACMV system. Three models were developed to simulate the indoor thermal comfort and energy usage of the ACMV system while GA was used to maximize the occupant's comfort level and minimize the electricity usage. Congradac et al. [107] used GA to optimize the indoor CO₂ concentration control with regard to power saving. The GA was proved to be a robust and efficient stochastic optimization method to search the optimal setting points among the appropriate scope of solutions. Huang et al. [108] proposed an adaptive learning algorithm based on GA for the automatic tuning of the PID controller in the ACMV system to achieve optimal performance. The simulation results showed that the GA was valid for tuning of PID parameters, yielding minimum overshoot and setting time. Seo et al. [109] adopted multi-island GA to optimize the design and minimize the energy conservation of the ACMV system. The experimental results revealed that the optimization method was capable of reducing the primary energy demand in the apartment house. Ge et al. [68] proposed a model-based control strategy for a liquid desiccant-chilled ceiling system with the objective of optimizing the indoor thermal comfort and reducing the power consumption. The GA could resolve the multi-objective optimization problem in terms of maintaining the indoor environment and minimizing the energy consumption. The GA is widely used as an approach to

complicated problem and global optimization for air conditioning and mechanical ventilation system.

This chapter is structured as follows: the model development of each component in the ATB system is proposed in Section 6.2; the global optimization formulation and optimization strategy of ATB system are presented in Section 6.3; the model validation results are illustrated in Section 6.4; in Section 6.5 the optimization results are illustrated; Section 6.6 draws the summary.

6.2 Model development of ATB system

Predicting the system performance and energy consumption with high accuracy is the basic of real-time optimization procedures. The mathematics and physical models can accurately forecast the air conditioning system cooling and ventilation performance, but the additional complexities of the models outweigh the advantage. Therefore, the models of each component in the ATB system are developed under hybrid manners which can attain higher model accuracy while reducing the model complexity. In addition, the energy consumption models of the chiller, fans and water pumps are deduced based on conservation equations of energy and mass. The hybrid models are expressed as follows.

6.2.1 Chiller energy model

The chiller plant is a complicated system which mainly consists of compressor, evaporator and condenser. The energy consumption of chiller plant is associated with the rotation speed of compressor, the heat transfer rate between evaporator and condenser, the flow rate of the refrigerant and etc. [110]. From Chang's study, the energy consumption of the chiller plant could be described by part load ratio which is defined as the ratio of current cooling capacity and the rated refrigerating capacity of the chiller [111]. Then, the part ratio of chiller plant can be calculated as follows:

$$PLR_{ch} = \frac{Q_{ch.cur}}{Q_{ch.rated}} \quad (6.1)$$

where PLR_{ch} is the part ratio of the chiller plant, $Q_{ch.cur}$ is the current cooling capacity of the chiller and $Q_{ch.rated}$ is the rated cooling capacity of the chiller.

The cooling requirements from the air handling unit and the indoor air terminal units determine the current load of the chiller plant. Therefore, the real time cooling load is calculated based on the water side information:

$$Q_{ch.cur} = C_{chw} \dot{m}_{chw} (T_{chw.in} - T_{chw.out}) \quad (6.2)$$

where C_{chw} is the specific heat of water, $\dot{m}_{chw}$ is the chilled water mass flow rate, $T_{chw.in}$ is the chilled water inlet temperature and $T_{chw.out}$ is the chilled water outlet temperature.

In the steady state operating conditions, the power consumption of chiller can be expressed as:

$$E_{ch} = a_{ch,3} PLR_{ch}^3 + a_{ch,2} PLR_{ch}^2 + a_{ch,1} PLR_{ch} + a_{ch,0} \quad (6.3)$$

where E_c is the chiller power consumption, $a_{ch,0}$, $a_{ch,1}$, $a_{ch,2}$ and $a_{ch,3}$ are unknown constant coefficients.

6.2.2 Fan and water pump energy model

A booster fan is utilized to maintain the air flow to the ATB. The real time energy consumption of fan can be calculated as a cubic function of the ratio of the fluid volume flow rate to the rated volume flow rate [112].

$$PLR_f = \frac{\dot{m}_a}{\dot{m}_{a.rated}} \quad (6.4)$$

$$E_f = E_{f,rated} (a_{f,3} PLR_f^3 + a_{f,2} PLR_f^2 + a_{f,1} PLR_f + a_{f,0}) \quad (6.5)$$

where PLR_f is the part ratio of booster fan, $\dot{m}_a$ is the current mass flow rate of primary air, $\dot{m}_{a,rated}$ is the rated mass flow rate of primary air. E_f is the real time fan power consumption, $E_{f,rated}$ is the fan power consumption at rated air flow rate, $a_{f,1}$, $a_{f,2}$ and $a_{f,3}$ are unknown constant coefficients.

Similarly, the power consumption of water pump is expressed as follows:

$$PLR_p = \frac{\dot{m}_w}{\dot{m}_{w,rated}} \quad (6.6)$$

$$E_p = E_{p,rated} (a_{p,3} PLR_p^3 + a_{p,2} PLR_p^2 + a_{p,1} PLR_p + a_{p,0}) \quad (6.7)$$

where PLR_p is part ratio of water pump, $\dot{m}_w$ is current mass flow rate of chilled water, $\dot{m}_{w,rated}$ is the rated mass flow rate of chilled water. E_p is the real time pump power consumption, $E_{p,rated}$ is the pump power consumption at rated water flow rate, $a_{p,0}$, $a_{p,1}$, $a_{p,2}$ and $a_{p,3}$ are unknown constant coefficients.

6.2.3 The air flow model

The primary air in ATB serves as the fresh air supply which also enhances the room air passing through cooling coils. The ATB terminal units capture air entrainment effect and thermosiphon effect to enhance the air ventilation in the occupied zone. The air entrainment process is described by the entrainment ratio.

$$ER = \frac{\dot{V}_{sec}}{\dot{V}_{pri}} \quad (6.8)$$

where $\dot{V}_{\text{sec}}$ is the secondary air flow rate and $\dot{V}_{\text{pri}}$ is the primary air flow rate.

Adopt empirical relationship, the parameters including configuration of the terminal unit, size of the full duct, resistance of air flow and etc. are lumped into the constant of proportionality. The primary air plenum pressure, room temperature and average temperature difference are the manipulated parameters. The flow rates of primary air flow rate and secondary air can be described as follows:

$$V_{\text{pri}} = a_{\text{pri}} P^{b_{\text{pri}}} \quad (6.9)$$

$$V_{\text{sec}} = a_{\text{sec}} P^{b_{\text{sec}}} + k_{\text{sec}} \cdot \frac{\dot{m}_{\text{chw}}^{n_1}}{P^{n_2}} \cdot \left(\frac{\Delta T}{T_z} \right)^{n_3} \quad (6.10)$$

where P is the primary air plenum pressure, a_{sec} , b_{sec} , k_{sec} , n_1 , n_2 and n_3 are unknown constant coefficients, ΔT is the average temperature difference, T_z is the room temperature.

In the tests, the chilled water inlet temperature and room temperature are maintained at the set points. Hence the secondary air flow rate can be simplified as:

$$V_{\text{sec}} = a_{\text{sec}} P^{b_{\text{sec}}} + k_{\text{sec}} \cdot \frac{\dot{m}_{\text{chw}}^{n_1}}{P^{n_2}} \quad (6.11)$$

6.2.4 The cooling coil model

The cooling coil of the ATB is operating under wet condition, thus both sensible load and latent load can be eliminated by the terminal unit. Given the air side and water side measurements, the cooling coil model can predict the heat transfer rate and the secondary air off coil temperature under various working conditions. The hybrid model of cooling coil is given as follows:

$$Q = \frac{b_1 \dot{m}_{chw}^e}{1 + b_2 \left(\frac{\dot{m}_{chw}}{\dot{m}_{sec}} \right)^e} (T_{sec.in} - T_{chw.in}) \quad (6.12)$$

where b_1 , b_2 and e are unknown constant coefficients, $\dot{m}_{sec}$ is the secondary air mass flow rate, $T_{sec.in}$ is the secondary air inlet temperature, $T_{chw.in}$ is the chilled water inlet temperature.

Based on the conservation of energy, the heat incremental of secondary air is equal to heat transfer rate in the cooling coil as the heat specific of the cooling coil is neglected. Then we obtained:

$$Q = C_{chw} \dot{m}_{chw} (T_{chw.out} - T_{chw.in}) \quad (6.13)$$

$$Q = -\dot{m}_{sec} (H_{sec.out} - H_{sec.in}) \quad (6.14)$$

where $H_{sec.out}$ is secondary air off coil enthalpy and $H_{sec.in}$ is the secondary air on coil enthalpy.

With reference to ASHRAE data [113], the saturation air enthalpy can be expressed as the cubic function of air temperature within the range of 0°C to 50°C.

$$h_a = 9.3839 + 1.71137T_a + 0.0222T_a^2 + 0.00063T_a^3 \quad (6.15)$$

where h_a is the saturation air enthalpy and T_a is the saturation air temperature.

Based on Eq. (6.11) the secondary air off coil temperature can be calculated. The sensible cooling capacity is expressed as follows:

$$Q_{sen} = \dot{m}_{sec} c_a (T_{sec.in} - T_{sec.out}) \quad (6.16)$$

where Q_{sen} is the sensible cooling capacity, c_a is the specific heat of air, $T_{sec.in}$ is the secondary air on coil temperature, $T_{sec.out}$ is the secondary air off coil temperature.

6.2.5 The indoor built model

As analyzed in Chapter 5, the sensible heat ratio of ATB approximates 0.45. Therefore, the flow rate of primary air is regulated to maintain the indoor CO₂ level while leaving the indoor moisture content reach the steady state. In the occupied room, the room temperature and CO₂ level can be expressed with respect to the heat balance and mass balance equations:

$$M_a C_a \frac{dT_r}{dt} = -\dot{m}_a c_a (T_r - T_{pri}) - Q_{sen} + Q_{l, sen} \quad (6.17)$$

$$M_a \frac{dw_r}{dt} = -\dot{m}_a (w_r - w_{pri}) + D_l \quad (6.18)$$

where M_a is the indoor air mass, t is time, T_{pri} is the primary air temperature, $Q_{l, sen}$ is the sensible load, w_r is the indoor carbon dioxide content, w_{pri} is the primary air carbon dioxide content, D_l is the CO₂ production rate.

6.3 Experimental setup and model validation

6.3.1 Experimental setup

The configuration of the ATB system is illustrated in Figure 6.1. The pilot plant consists of an air-cool chiller, an air handling unit, a water pump, a booster fan and 2 ATB terminal units. As analyzed in Chapter 5, the SHR of the ATB system can reach 50% which indicates that the terminal units have efficient dehumidification ability. Hence, the room temperature and CO₂ concentration are regulated according to the pre-set value while the moisture content is left to reach the steady value. The room temperature is controlled by modulating the chilled water volume flow rate while the CO₂ level is modulated through adjusting the primary air volume flow rate.

Figure 6.1 The schematic diagram of the experimental ATB system

The unknown constant parameters of the components' energy consumption models and terminal unit model are identified based on the system nominal parameters and operating data. The nominal parameters are listed in Table 6.1.

Table 6.1 Components rated capacities

Component	Rated power	Nominal output
Chiller	6.00kW	21.7kW
Fan	200W	262 m ³ /h
Pump	350W	0.6L/s

6.3.2 Model validation

The effectiveness of the energy consumption models and ATB models are evaluated through relative error and root mean square of relative error. To comprehensively validate the accuracy of the proposed model, the testing data of each components of the ATB system is recorded over a wide operating range. The predicted energy consumptions of each component are compared with the measured values respectively as illustrated from Figure 6.2 to Figure 6.4. From the comparison results, the proposed models of energy consumptions are well corresponding to the measured values with the margin of RE less than 5%.

Figure 6.2 Model validation of fan energy consumption

Figure 6.3 Model validation of pump energy consumption

Figure 6.4 Model validation of chiller energy consumption

To estimate the effectiveness of the proposed models, the air loop and water loop parameters are adjusted within a large scale. Totally 60 data sets are collected in respect to secondary air volume flow rate, primary air volume flow rate, room temperature, chilled water inlet/outlet temperature and chilled water flow rate. The model validation

results are illustrated from Figure 6.5 to Figure 6.7. The validation results of the energy models and ATB models are summarized in Table 6.2.

Figure 6.5 Model validation of the primary air flow rate

Figure 6.6 Model validation of the secondary air flow rate

Figure 6.7 Model validation of the cooling capacity

Table 6.2 Prediction accuracy of the models

Model	RE	RMSE
Fan energy consumption	2.12%	2.64%
Pump energy consumption	1.37%	1.69%
Chiller energy consumption	1.81%	2.11%
Primary air model	1.93%	2.68%
Secondary air model	3.04%	3.36%
Cooling coil model	2.46%	2.87%

Based on the curve fitting and model validation results, the energy consumption models and ATB terminal unit models have good consistency with experimental results. The proposed models are precise and sufficient for control and optimization applications.

6.4 Global optimization formulation

The global optimization strategy is formulated to find the optimal set points of the ATB system. The total energy consumption of the system is minimized while the indoor environment quality is maintained based on the proposed strategy. The following assumptions are adopted to simplify mathematical calculation of optimization:

- The primary air and secondary are homogeneously mixed in the terminal unit.
- The chilled water temperature field distributes evenly in the heat exchanger.
- The heat storage in the cooling coil tube is neglected.
- The joints of the cooling coil and sensor are adiabatic.
- The air mass in the room is constant.
- The primary air supply is constant during the test,
- The chilled water inlet temperature is constant during the optimization process.

Then the optimization strategy is developed in terms of energy consumption and the system constraints.

6.4.1 Objective function

By analyzing the energy models in Eqs. (6.3), (6.5) and (6.7), the energy consumption of the ATB system is composed of three parts: chiller, pump and fan. The objective function is to minimize the total energy consumption as shown below:

$$E_{total} = E_f + E_{ch} + E_p \quad (6.19)$$

where E_{total} is the total energy consumption of the ATB system.

6.4.2 Constrains

In the practical ATB systems, certain constrains must be satisfied to meet the air conditioning and ventilation requirements. The constraints concentrate on the operation range of different components and the coupling of the system parameters.

- Chiller cooling capacity

The chiller plant regulates the cooling capacity through a frequency converter. The operation frequency of the chiller has a lower bound to prevent overheat of the motor. Hence, the constraints of the chiller plant are set as below:

$$Q_{ch.min} \leq Q_{ch} \leq Q_{ch.max} \quad (6.20)$$

where $Q_{ch.min}$ is the lower bound of chiller cooling capacity and $Q_{ch.max}$ is the upper bound of chiller cooling capacity.

- The primary air flow rate

The primary air flow rate is adjusted to satisfy the ventilation requirement and maintain positive pressure in the room [114]. Meanwhile, the primary air flow rate is restricted by the physical limitation of the fan.

$$\dot{m}_{pri.min} < \dot{m}_{pri} < \dot{m}_{pri.max} \quad (6.21)$$

where $\dot{m}_{pri.min}$ is the lower bound of the primary air mass flow rate and $\dot{m}_{pri.max}$ is the upper bound of the primary air mass flow rate.

- Chilled water flow rate

The chilled water flow rate is limited by the operation frequency of the motor:

$$\dot{m}_{chw.min} < \dot{m}_{chw} < \dot{m}_{chw.max} \quad (6.22)$$

where $\dot{m}_{chw.min}$ and $\dot{m}_{chw.max}$ are the limits of the chilled water mass flow rate.

Meanwhile, there are interactives between the system parameters which should be disposed in the optimization procedures. To maintain the indoor thermal comfort, the room temperature and CO₂ level are regulated to satisfy the requirements.

$$T_r = T_{req} \quad (6.23)$$

$$w_r \leq w_{req} \quad (6.24)$$

where T_{req} is the room temperature set point and w_{req} is the indoor carbon dioxide content requirement.

Moreover, the indoor sensible load and carbon dioxide load can be expressed as

$$Q_{req} = \dot{m}_{sec} C_a (T_{req} - T_{sec.out}) + \dot{m}_{pri} C_a (T_{req} - T_{pri}) \quad (6.25)$$

$$D_{req} \leq \dot{m}_{pri} (w_{req} - w_{pri}) \quad (6.26)$$

where Q_{req} is the indoor sensible load and D_{req} is the indoor carbon dioxide load.

Consequently, the optimization formulation of the ATB system can be summarized as follow:

$$\begin{aligned} \min \quad & E_{total} = E_f + E_{ch} + E_p \\ \text{subject to:} \quad & T_r = T_{req} \\ & w_r \leq w_{req} \\ & Q_{ch.min} \leq Q_{ch} \leq Q_{ch.max} \\ & \dot{m}_{pri.min} < \dot{m}_{pri} < \dot{m}_{pri.max} \\ & \dot{m}_{chw.min} < \dot{m}_{chw} < \dot{m}_{chw.max} \end{aligned} \quad (6.27)$$

To reduce the dimension of optimization variables and simplify the computation complexity, the variables are classified into three categories:

- Uncontrollable variables (T_{pri} , w_{pri} , $T_{chw.in}$, T_{req} , w_{req} , Q_{req} , D_{req}): the primary air supply temperature and chilled water inlet temperature (T_{pri} and $T_{chw.in}$) are determined by the building management system. The primary air CO₂ content

(w_{pri}) is determined by the atmosphere condition. The indoor air requirements (T_{req} and w_{req}) are determined by the occupants. The internal loads (Q_{req}, D_{req}) are determined by the outdoor conditions and number of occupants. All the uncontrollable variables are kept constant within each optimization process.

- Controllable variables ($\dot{m}_{pri}, \dot{m}_{chw}$): the fresh air and chilled water are controlled by the VSD fan and water pump respectively. The proposed two variables determine the cooling capacity of the terminal units and the indoor CO₂ removal efficiency. The optimization scheme is developed to search the optimal set points of the controllable variables to minimize the energy consumption with acceptable thermal comfort.
- Dependent variables ($Q_{ch}, T_{chw.out}, \dot{V}_{sec}, Q_{sen}, T_{sec.out}, H_{sec.out}$): in the air conditioning and mechanical ventilation process, the variations of dependent variables are determined by the uncontrollable variables and the independent variables.

The classification of the variables is summarized in Table 6.3.

Table 6.3 Classification of state variables

Variable categories	Variables
Uncontrollable variables	$T_{pri}, w_{pri}, T_{chw.in}, T_{req}, w_{req}, Q_{req}, D_{req}$
Controllable variables	$\dot{m}_{pri}, \dot{m}_{chw}$
Dependent variables	$Q_{ch}, T_{chw.out}, \dot{V}_{sec}, Q_{sen}, T_{sec.out}, H_{sec.out}$

6.4.3 Optimization strategy of ATB system

An optimization strategy is developed to find the optimal operating parameters for the ATB system. The proposed algorithm is expected to maintain the IAQ and minimize the energy consumption. The optimization schematic for the ATB system is illustrated in Figure 6.8.

The model updater consists of terminal unit models, indoor built models and energy consumption models. The models predict the heat transfer rate and ventilation rate of the ATB system under various setting points. The corresponding indoor thermal environment and system energy consumption are obtained based on the prediction results. The optimization module analyzes the energy efficiency with respect to the system constraints and uncontrollable variables. Then, the chilled water flow rate and primary air flow rate are optimized by the strategy to maintain the IAQ and reduce the energy consumption.

Figure 6.8 Scheme of the optimization strategy

The genetic algorithm (GA) is adopted to search for the optimal operating parameters for the ATB system. GA is a stochastic optimization method based on the natural selection in the evolution process. The algorithm, which has been extensively used in combinatorial optimization, machine learning and signal processing, has good global optimization ability without additional requirement on the continuity of objective function and

derivation calculation. The process of GA starts with a potential population (candidate solutions). The offspring is generated through selection, crossover, and mutation while the fittest individuals will be chosen on the biases of elimination of inferior. After multiple genetic iterations, the most adaptive population is selected as the optimal solution of optimization problems. The optimization processes for the ATB system is illustrated in Figure 6.9.

Figure 6.9 Flow chart of the optimization strategy

Step 1: Identify the parameters of the ATB terminal unit model and the energy consumption models and determine the uncontrollable and controllable variables based on the outdoor condition and IAQ requirement.

Step 2: Evaluate the operational limits of the components in the ACMV system and analyze the interactions between the parameters. Load the constraints to the optimization module.

Step 3: Initialize the parameters for genetic algorithm and set the initial population. Then the initial population will be coded into binary strings for optimization.

Step 4: Calculate the fitness value of each individual in the initialized population based on the objective function with respect to the energy consumption.

Step 5: Generate the offspring by performing selection, crossover and mutation for the chromosome. Selection is to choose the individuals with the higher fitness values through the method of roulette wheel as the next generation. Crossover is to exchange parts of the binary strings in the chromosomes to generate new individuals. Mutation is to randomly select individuals to mutate a digit of certain chromosomes.

Step 6: Repeat the iteration steps 4-5 until the maximum generation is achieved or the fluctuation of fitness value is within the termination criterion. The maximum fitness value is recorded. Then the chromosomes with maximum fitness value are decoded and the optimal setting points are identified.

6.5 Optimization results

To verify the energy-saving performance of the optimization scheme, experiments are demonstrated to compare the energy consumption of the ATB system under the original control logic and the optimized strategy. The test is conducted throughout a whole day from 8:30 am to 19:30 pm while the number of occupants and the indoor cooling load are recorded as shown in Figure 6.10. According to the ASHRAE standard, the setting points of temperature and CO₂ concentration are 24.5°C and 800ppm respectively. In addition, the upper and lower bound of the components in the ATB system are listed in Table 6.4.

Figure 6.10 The indoor heat condition and number of occupants

Table 6.4 The upper and lower bound of constraints

Constraints	Lower bound	Upper bound	Unit
$\dot{V}_{pri}$	37.8	262	m ³ /h
$\dot{m}_{chw}$	0.1	0.6	L/s
E_{ch}	2	6	kW

The primary air flow rate and chilled water flow rate of the ATB system are optimized by the GA for each instance (1 hour) to maintain the indoor air quality and reduce the total energy consumption. The main parameters of the generic algorithm are summarized in Table 6.5.

Table 6.5 The parameter setting of GA

Parameters for GA	Value
Population size	60
Max generation	50
Selection	Roulette wheel selection
Probability of crossover	0.8
Probability of mutation	0.01

The original and optimized setting points of the ATB system are shown in Figure 6.11 and Figure 6.12. The optimized primary air volume flow rates are less than that of the original ones which reduce the energy consumption of the booster fan. Meanwhile, the optimized chilled water flow rates are regulated to handle the additional indoor cooling load. In tropical regions, the outdoor fresh air is of high temperature and humidity which consumes a large amount of energy to be treated before supply to the occupied zone. The optimized operation of the ATB system can reduce the energy consumption of the chiller and the booster fan which improve the overall energy efficiency.

Figure 6.11 The original and optimized primary air flow rate

Figure 6.12 The original and optimized chilled water flow rate

Figure 6.13 illustrates the energy consumptions of the system with original strategy and the optimized strategy. The comparison results show that the proposed model-based optimization strategy can significantly reduce the total energy consumption in the experimental range. The power consumption of each component in the ATB system is shown in Figure 6.14. On the basis of the calculation, the energy efficiency of the chiller is improved which accounts for most of the energy consumption. After optimization, the fresh air flow rate is minimized while additional chilled water is required to cover the indoor cooling load. As a result, the optimized water pump consumes more power than the original strategy.

Figure 6.13 The original and optimized energy consumption of the ATB system

Figure 6.14 Comparison of fan, pump, chiller and total energy consumption

The comparison results of energy consumption within the test period are summarized in Table 6.6. The proposed optimization strategy provides a total energy saving of 9.3% which proves that the strategy is sufficient for real time optimization applications.

Table 6.6 Summary of the energy consumption between both operation strategies

Components	Energy consumption (kWh)		Energy saving (%)
	Original strategy	Genetic algorithm	
Fan	1.73	1.41	18.5
Pump	1.32	1.56	-18.2
Chiller	39.69	35.80	9.8
Total	42.74	38.77	9.3

6.6 Summary

In this chapter, a model-based optimization approach is proposed to optimize the performance of the ATB system. The energy consumption models of each component and thermal models of terminal unit are developed with hybrid manner. Based on the experimental validation, the proposed models show good agreement with the measurement results. The global optimization strategy is formulated to find the optimal set points of the ATB system with respect to the total energy consumption. The experimental results indicate that the optimized operating parameters obtained by the GA can significantly reduce the total energy consumption by 9.3% and maintain indoor thermal comfort when compared with the original strategy. Moreover, the simulation and experimental results show that the ATB system can provide satisfied indoor thermal comfort with high energy efficiency and reduced initial cost.

Chapter 7 Conclusions and future work

7.1 Conclusions

If the active air terminal units are properly designed and operated, the ACMV system can have significant improvements on the indoor environment quality and energy consumption. Hence, it is desired to optimize the mechanical design of the active air terminals and investigate the system operation characteristics to fulfill the energy conservation potential. The main contributions of the thesis are as follows.

Considering the influence of air buoyancy, a simple yet accurate ACB model was developed by adopting hybrid manner. For the air side, the influences of the primary air and the chilled water on the entrainment effect had been investigated. Meanwhile, by analyzing first principles and experimental results, the cooling coil model was derived which contained only three lumped parameters. The proposed model could accurately predict the air side volume flow rate and water side heat transfer rate in a wide operating range. In the model validation, the final accuracy was within $\pm 5\%$. It was found that the air buoyancy impeded the air entrainment process and reduced the secondary air flow rate up to 10% in the cooling mode.

Based on the performance analysis of the ACB, the ATB was developed with an innovative mechanical design. Combined with basic theoretical analysis, the experimental comparisons of ATB and ACB were conducted under various operation conditions to estimate the terminal unit thermodynamic and hydrodynamic performances. The main operation parameters that influence the ATB heat transfer rate were tested separately to determine the optimal operation settings. In the air side, increasing the primary air plenum pressure and extending the length of air straighter could effectively improve the heat transfer rate. As higher plenum pressure would reduce the entrainment ratio and lead to extra booster fan energy consumption, the pressure was recommended to be less than 120Pa to balance the cooling capacity and the heat transfer efficiency. In the water loop, the chilled water supply could enhance the heat transfer rate, but simultaneously it might lead to overcooled supply air. The average temperature difference between chilled water and occupied zone should be higher than 13°C to avoid

the sensation of draught. The performance of ATB could be substantially improved with appropriate system settings.

Based on the ACB and ATB tests, the main components of the ACMV system were modeled to predict the system ventilation rate, cooling capacity and energy consumption. Accordingly, a model-based optimization approach was developed to minimize the system energy consumption and maintain the indoor thermal comfort. Genetic algorithm was used to search for the optimal set points of the chilled water flow rate and the fresh air flow rate. The simulation results showed that the optimization strategy could provide the desired indoor temperature and CO₂ concentration and achieve significant energy saving.

7.2 Future work

Based on the conclusions, the proposed active air terminal units have outstanding performance in terms of indoor environment quality, energy efficiency and cost saving. In addition, the model-based control strategy is sufficient to minimize the system energy consumption and maintain the desired indoor environment quality. Despite all the achievements in the thesis, there still remains a lot of foreseeable works to fulfill the development of active air terminal based system.

- The researches on ACB system indoor air flow patterns, operation characteristics and modeling methods are confined to the system cooling mode. And the air buoyancy is proved to hinder the inducing of secondary air. In the heating mode, the secondary air is of higher temperature and the air buoyancy acts vertically which significantly change the indoor air flow patterns and temperature gradients. It is necessary to figure out ACB system operation characteristic and heating performance in frigid regions.
- The cooling performance and air flow patterns of ATB system are highly depend on the displacement ventilation. Driven by the density difference between cold air and warm air, the ventilation process is sensitive to the configuration and strength

of indoor heat sources. Therefore, it could be beneficial to evaluate the effect of thermal load distribution and strength on the indoor thermal comfort and terminal unit performance. The experimental results could give insights on the design and operation of the ATB systems.

- The development of the ATB model is the foundation of precision control and optimization for the ACMV system. In the ATB system, there is tight coupling between the displacement ventilation and the air entrainment process. Besides, the thermosiphon effect is influenced by the internal load which increase the difficulties to develop an ideal model to predict the supply air flow rate. In addition, the chilled water inlet temperature is below the dew point and condensation will occur on the surface of the cooling coil. Consequently, the terminal unit can handle both sensible load and latent load which increases the system complexity. Hence, it is highly desired to develop a model of the terminal unit so that the system control and optimization could be more effective and accurate.
- In general, the ATB system fresh air flow rate is regulated to maintain the indoor CO₂ level while the chilled water flow rate and temperature is adjusted to control the indoor temperature. However, no humidity control strategy has been developed so far. In the ATB system, the indoor moisture is removed by condensation process on the surface of the cooling coil. The dehumidification efficiency is linked with the secondary air flow rate, chilled water flow rate, secondary air temperature and chilled water temperature. Thus, the ATB dehumidifying capacity should be evaluated experimentally to promote the practical application.
- In principle, the operation characteristics of ATB is similar to PTB except the additional primary air supply. The system test methods, modeling approaches and optimization algorithms of the ATB systems can be transferred to PTB systems with few adjustments. As the system configuration of the passive displacement ventilation system is simple and reliable which reduce the complexity of system control and cost of maintenance. It is reasonable to extend the current research to

the PTB system to achieve desired indoor environment quality and energy efficiency.

References

- [1] S. D. Swenson, *HVAC: heating, ventilating, and air conditioning*. American Technical Publishers, 2004.
- [2] W. Goetzler, R. Shandross, J. Young, O. Petritchenko, D. Ringo, and S. McClive, "Energy Savings Potential and RD&D Opportunities for Commercial Building HVAC Systems," Navigant Consulting, Burlington, MA (United States)2017.
- [3] (2016). *ANNUAL CLIMATE ASSESSMENT 2016 SINGAPORE*. Available: <http://www.weather.gov.sg/wp-content/uploads/2017/07/Annual-Climate-Assessment-Report-2016.pdf>
- [4] B. Z. A.P. Boranian, J.N. Sarvaiya, N.Y. Jadhav, P. Pawar, Z. Zhang. (2014). *Building energy efficiency R&D roadmap*. Available: <http://www.nrf.gov.sg>
- [5] N. K. Huynh, H. Li, Y. C. Soh, and W. Cai, "Numerical analysis and model-based control of energy recovery ventilator in HVAC system," in *Sustainable Technologies (WCST), 2016 World Congress on*, 2016, pp. 76-77: IEEE.
- [6] H. E. Feustel and C. Stetiu, "Hydronic radiant cooling—preliminary assessment," *Energy and buildings*, vol. 22, no. 3, pp. 193-205, 1995.
- [7] M. Ackermann, *Cool comfort: America's romance with air-conditioning*. Smithsonian Institution, 2013.
- [8] K. Roth, J. Dieckmann, R. Zogg, and J. Brodrick, "Chilled beam cooling," *Ashrae journal*, vol. 49, no. 9, p. 84, 2007.
- [9] G. F. Hundy, A. R. Trott, and T. C. Welch, in *Refrigeration and Air-Conditioning (Fourth Edition)*Oxford: Butterworth-Heinemann, 2008, p. vii.
- [10] M. Ruponen and J. A. Tinker, "A novel method to measure the air entrainment ratio of an active chilled beam," *International Journal of Ventilation*, vol. 7, no. 4, pp. 299-308, 2009.
- [11] P. Filipsson, A. Trüschel, J. Gräslund, and J.-O. Dalenbäck, "Induction ratio of active chilled beams– Measurement methods and influencing parameters," *Energy and Buildings*, vol. 129, pp. 445-451, 2016.

- [12] H. Freitag and D. Müller, "Modeling the internal airflow in active chilled beams," in *Ventilation 2012: 10th International Conference on Industrial Ventilation*, 2012.
- [13] H. Freitag, M. Schmidt, D. Müller, H. Koskela, and P. Mustakallio, "Particle image velocimetry measurements of the internal air flow in active chilled beams," in *Roomvent 2014: 13th SCANVAC International Conference on Air Distribution in Rooms*, 2014, pp. 566-573.
- [14] H. Koskela, P. Saarinen, H. Freitag, M. Schmidt, D. Müller, and P. Mustakallio, "LES simulation of the active chilled beam flow pattern," in *Roomvent 2014: 13th SCANVAC International Conference on Air Distribution in Rooms*, 2014, pp. 582-589.
- [15] Dadanco. *Active chilled beams*. Available: <http://dadanco.com/>
- [16] Z. Guan and C. Wen, "Numerical investigation of geometry parameters for designing efficient terminal units in active chilled beam," in *Industrial Electronics and Applications (ICIEA), 2014 IEEE 9th Conference on*, 2014, pp. 1114-1118: IEEE.
- [17] Z. Guan and C. Wen, "Further optimization on Type II terminal unit in active chilled beam system," in *Industrial Electronics (ISIE), 2017 IEEE 26th International Symposium on*, 2017, pp. 1513-1520: IEEE.
- [18] B. Wu, W. Cai, Q. Wang, C. Chen, C. Lin, and H. Chen, "The Air Distribution Around Nozzles Based On Active Chilled Beam System," 2016.
- [19] Z. Guan and C. Wen, "Geometric optimization on active chilled beam terminal unit to achieve high entrainment efficiency," *Applied Thermal Engineering*, vol. 98, pp. 816-826, 2016.
- [20] C. Chen, W. Cai, Y. Wang, and C. Lin, "Performance comparison of heat exchangers with different circuitry arrangements for active chilled beam applications," *Energy and Buildings*, vol. 79, pp. 164-172, 2014.
- [21] C. Chen, W. Cai, Y. Wang, and C. Lin, "Further study on the heat exchanger circuitry arrangement for an active chilled beam terminal unit," *Energy and Buildings*, vol. 103, pp. 352-364, 2015.

- [22] F. Dominguez-Muñoz, "Analysis of the thermal performance of heat exchangers for active chilled beam units," 2018.
- [23] A. Afshari, R. G. Norouzi, G. Hultmark, and C. Niels, "Two-pipe chilled beam system for both cooling and heating of office buildings," in *11th REHVA World Congress and 8th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings*, 2013, pp. 16-19.
- [24] B. Bertheussen, P. Mustakallio, R. Kosonen, and A. K. Melikov, "Comparison of the Performance of Chilled Beam with Swirl Jet and Diffuse Ceiling Air Supply: Impact of Heat Load Distribution," in *Clima 2013 Indoor Air Quality, Ventilation & Energy Conservation in Buildings*, 2013.
- [25] B. Bertheussen, P. Mustakallio, A. K. Melikov, and R. Kosonen, "Performance of Chilled Beam with Radial Swirl Jet and Diffuse Ceiling Air Supply in Heating Mode," in *Clima 2013 Indoor Air Quality, Ventilation & Energy Conservation in Buildings*, 2013.
- [26] P. V. Nielsen, "Displacement ventilation," *Aalborg University*, 1993.
- [27] D. Loveday, K. Parsons, A. Taki, and S. Hodder, "Displacement ventilation environments with chilled ceilings: thermal comfort design within the context of the BS EN ISO7730 versus adaptive debate," *Energy and Buildings*, vol. 34, no. 6, pp. 573-579, 2002.
- [28] K. Naydenov, G. Pitchurov, G. Langkilde, and A. K. Melikov, "Performance of displacement ventilation in practice," in *Proceedings of Roomvent*, 2002, pp. 483-486.
- [29] S. Riffat, X. Zhao, and P. Doherty, "Review of research into and application of chilled ceilings and displacement ventilation systems in Europe," *International Journal of Energy Research*, vol. 28, no. 3, pp. 257-286, 2004.
- [30] F. Betz, J. McNeill, B. Talbert, H. Thimmanna, and N. Repka, "Issues arising from the use of chilled beams in energy models," *proceedings of SimBuild*, vol. 5, no. 1, pp. 655-667, 2012.
- [31] M. Vaughn, "ASHRAE Research Report," 2014.
- [32] G. Cao, M. Sivukari, J. Kurnitski, M. Ruponen, and O. Seppänen, "Particle Image Velocimetry (PIV) application in the measurement of indoor air distribution by an

- active chilled beam," *Building and Environment*, vol. 45, no. 9, pp. 1932-1940, 2010.
- [33] G. Cao, M. Sivukari, J. Kurnitski, and M. Ruponen, "PIV measurement of the attached plane jet velocity field at a high turbulence intensity level in a room," *International Journal of Heat and Fluid Flow*, vol. 31, no. 5, pp. 897-908, 2010.
 - [34] G. Cao, M. Ruponen, R. Paavilainen, and J. Kurnitski, "Modelling and simulation of the near-wall velocity of a turbulent ceiling attached plane jet after its impingement with the corner," *Building and Environment*, vol. 46, no. 2, pp. 489-500, 2011.
 - [35] G. Cao, M. Ruponen, and J. Kurnitski, "Experimental investigation of the velocity distribution of the attached plane jet after impingement with the corner in a high room," *Energy and Buildings*, vol. 42, no. 6, pp. 935-944, 2010.
 - [36] G. Cao, J. Kurnitski, P. Mustakallio, and O. Seppänen, "Active chilled beam wall jet prediction by the free convection model," *International Journal of Ventilation*, vol. 7, no. 2, pp. 169-178, 2008.
 - [37] G. Cao, J. Kurnitski, M. Ruponen, P. Mustakallio, and O. Seppänen, "Plane-air-jet corner zone modelling in a room ventilated by an active chilled beam," *International Journal of Ventilation*, vol. 7, no. 4, pp. 287-297, 2009.
 - [38] G. Cao, "Modelling the attached plane jet in a room," 2009.
 - [39] F. Alamdari, "Displacement ventilation and cooled ceilings," in *Proceedings of ROOMVENT*, 1998, vol. 98, no. 1, pp. 197-204.
 - [40] S. J. Rees and P. Haves, "An experimental study of air flow and temperature distribution in a room with displacement ventilation and a chilled ceiling," *Building and Environment*, vol. 59, pp. 358-368, 2013.
 - [41] Q. Chen and J. Van Der Kooi, "A methodology for indoor airflow computations and energy analysis for a displacement ventilation system," *Energy and Buildings*, vol. 14, no. 4, pp. 259-271, 1990.
 - [42] B. J. Wachenfeldt, M. Mysen, and P. G. Schild, "Air flow rates and energy saving potential in schools with demand-controlled displacement ventilation," *Energy and buildings*, vol. 39, no. 10, pp. 1073-1079, 2007.

- [43] B. Wu, W. Cai, H. Chen, and K. Ji, "Experimental investigation on airflow pattern for active chilled beam system," *Energy and Buildings*, vol. 166, pp. 438-449, 2018.
- [44] J. Fredriksson, M. Sandberg, and B. Moshfegh, "Experimental investigation of the velocity field and airflow pattern generated by cooling ceiling beams," *Building and environment*, vol. 36, no. 7, pp. 891-899, 2001.
- [45] B. Wu, W. Cai, and K. Ji, "Heat source effects on thermal comfort for active chilled beam systems," *Building and Environment*, vol. 141, pp. 91-102, 2018.
- [46] V. Zbořil, A. Melikov, B. Yordanova, L. Bozhkov, and R. Kosonen, "Airflow distribution in rooms with active chilled beams," in *Proceedings of the 10th International Conference on Air Distribution in Rooms–Roomvent 2007*, 2007, pp. 1-7.
- [47] R. Kosonen, A. Melikov, L. Bozhkov, and B. Yordanova, "Impact of heat load distribution and strength on airflow pattern in rooms with exposed chilled beams," in *Proceedings of ROOMVENT*, 2007.
- [48] R. Kosonen, M. Virta, and A. Melikov, "The impact of thermal loads on indoor air flow," in *Proceedings of CLIMA*, 2007.
- [49] A. Melikov, B. Yordanova, L. Bozhkov, V. Zboril, and R. Kosonen, "Impact of the airflow interaction on occupants' thermal comfort in rooms with active chilled beams," in *The 6th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings*, 2007: Citeseer.
- [50] A. Melikov, B. Yordanova, L. Bozhkov, V. Zboril, and R. Kosonen, "Human response to thermal environment in rooms with chilled beams," in *Proceedings of CLIMA*, 2007: Citeseer.
- [51] J. True, V. Zboril, R. Kosonen, and A. Melikov, "Consideration for minimising draught discomfort in Rooms with Active Chilled Beams," in *Proceedings of Clima, Wellbeing Indoors*, 2007.
- [52] P. Mustakallio, Z. Bolashikov, K. Kostov, A. Melikov, and R. Kosonen, "Thermal Conditions in a Simulated Office Environment with Convective and Radiant Cooling Systems," in *Proceedings of Clima*, 2013.

- [53] A. Novoselac and J. Srebric, "A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems," *Energy and buildings*, vol. 34, no. 5, pp. 497-509, 2002.
- [54] M. Behne, "Indoor air quality in rooms with cooled ceilings.: Mixing ventilation or rather displacement ventilation?," *Energy and Buildings*, vol. 30, no. 2, pp. 155-166, 1999.
- [55] M. Trčka and J. L. Hensen, "Overview of HVAC system simulation," *Automation in Construction*, vol. 19, no. 2, pp. 93-99, 2010.
- [56] D. B. Crawley, J. W. Hand, M. Kummert, and B. T. Griffith, "Contrasting the capabilities of building energy performance simulation programs," *Building and environment*, vol. 43, no. 4, pp. 661-673, 2008.
- [57] P. Jacobs and H. Henderson, "State-of-the-art review of whole building, building envelope, and HVAC component and system simulation and design tools," *Architectural Energy Corporation*, 2002.
- [58] T. Hong, S. Chou, and T. Bong, "Building simulation: an overview of developments and information sources," *Building and environment*, vol. 35, no. 4, pp. 347-361, 2000.
- [59] J. L. Hensen, "Application of modelling and simulation to HVAC systems," in *Proc. 30th Int. Conf. MOSIS*, 1996, vol. 96.
- [60] P. Filipsson, A. Trüschel, J. Gräslund, and J.-O. Dalenbäck, "A thermal model of an active chilled beam," *Energy and Buildings*, vol. 149, pp. 83-90, 2017.
- [61] C. Chen, W. Cai, K. Giridharan, and Y. Wang, "A hybrid dynamic modeling of active chilled beam terminal unit," *Applied Energy*, vol. 128, pp. 133-143, 2014.
- [62] N. M. Mateus and G. C. da Graça, "A validated three-node model for displacement ventilation," *Building and Environment*, vol. 84, pp. 50-59, 2015.
- [63] G. C. da Graça and P. Linden, "A simple model for heat transfer in displacement-ventilation," in *Proceedings of ROOMVENT*, 2004.
- [64] J. Kim, "Characterizing the in-situ performance of passive chilled beams," *ASHRAE Transactions*, vol. 121, p. 1N, 2015.

- [65] J. Kim, A. Tzempelikos, and J. E. Braun, "Review of modelling approaches for passive ceiling cooling systems," *Journal of Building Performance Simulation*, vol. 8, no. 3, pp. 145-172, 2015.
- [66] S. J. Rees, "Modelling of displacement ventilation and chilled ceiling systems using nodal models," © Simon John Rees, 1998.
- [67] S. J. Rees and P. Haves, "A nodal model for displacement ventilation and chilled ceiling systems in office spaces," *Building and Environment*, vol. 36, no. 6, pp. 753-762, 2001.
- [68] G. Ge, F. Xiao, and X. Xu, "Model-based optimal control of a dedicated outdoor air-chilled ceiling system using liquid desiccant and membrane-based total heat recovery," *Applied energy*, vol. 88, no. 11, pp. 4180-4190, 2011.
- [69] C. Chen, W. Cai, Y. Wang, and Z. Liu, "Design of a fuzzy controller for the active chilled beam system," in *Industrial Electronics and Applications (ICIEA), 2014 IEEE 9th Conference on*, 2014, pp. 723-728: IEEE.
- [70] T. technix, "Chilled beam design guide," ed: TB012309.
- [71] DADANCO, "Considerations for Active Chilled Beam Designs," ed: Design Bulletin, 2009.
- [72] F. Woods, "Chilled beam iQ Star WEGA II," ed: Product manual, 2015.
- [73] P. Ken Loudermilk, "Chilled Beams Thermal Comfort FOR," 2009.
- [74] K. J. Loudermilk and D. S. Alexander, "Efficient space humidity control with active chilled beam systems," *ASHRAE Journal*, vol. 54, no. 1, pp. 28-35, 2012.
- [75] D. Alexander and M. O'Rourke, "Design considerations for active chilled beams," *ASHRAE Journal*, vol. 50, no. 9, pp. 50-58, 2008.
- [76] P. RUMSEY and R. Engineers, "Chilled Beams," *Heating/Piping/Air Conditioning Engineering*, vol. 82, no. 1, pp. 46-49, 2010.
- [77] A. Livchak and C. Lowell, "Don't turn active air terminals into expensive diffusers," *ASHRAE Journal*, vol. 54, no. 4, pp. 52-60, 2012.
- [78] S. J. Emmerich and T. P. McDowell, *Initial evaluation of displacement ventilation and dedicated outdoor air systems for US commercial buildings*. US Department of Commerce, Technology Administration, National Institute of Standards and Technology, 2005.

- [79] A. K. Darwich, "Holistic HVAC design," *ASHRAE Journal*, vol. 55, no. 4, pp. 40-45, 2013.
- [80] P. Rumsey and J. Weale, "Chilled beams in labs: eliminating reheat & saving energy on a budget," *Ashrae Journal*, vol. 49, no. 1, p. 18, 2007.
- [81] B. M. Barnet, "Chilled beams for labs: using dual energy recovery," *ASHRAE Journal*, vol. 50, no. 12, pp. 28-36, 2008.
- [82] N. Devlin, "Validation of an active chilled beam design for a healthcare facility," *People*, vol. 190, p. 133, 2011.
- [83] X. Shan, J. Zhou, V. W.-C. Chang, and E.-H. Yang, "Comparing mixing and displacement ventilation in tutorial rooms: Students' thermal comfort, sick building syndromes, and short-term performance," *Building and Environment*, vol. 102, pp. 128-137, 2016.
- [84] Y. Li, P. V. Nielsen, and M. Sandberg, "Displacement ventilation in hospital environments," *ASHRAE J*, vol. 53, pp. 86-88, 2011.
- [85] N. Ruangtrakoon, T. Thongtip, S. Aphornratana, and T. Sriveerakul, "CFD simulation on the effect of primary nozzle geometries for a steam ejector in refrigeration cycle," *International Journal of Thermal Sciences*, vol. 63, pp. 133-145, 2013.
- [86] N. Enjalbert, D. Galley, and L. Pierrot, "An entrainment model for the turbulent jet in a coflow," *Comptes Rendus Mecanique*, vol. 337, no. 9-10, pp. 639-644, 2009.
- [87] K. Ariafar, D. Buttsworth, N. Sharifi, and R. Malpress, "Ejector primary nozzle steam condensation: area ratio effects and mixing layer development," *Applied Thermal Engineering*, vol. 71, no. 1, pp. 519-527, 2014.
- [88] K. Ariafar, D. Buttsworth, G. Al-Doori, and N. Sharifi, "Mixing layer effects on the entrainment ratio in steam ejectors through ideal gas computational simulations," *Energy*, vol. 95, pp. 380-392, 2016.
- [89] X. Wang, W. Cai, J. Lu, Y. Sun, and X. Ding, "A hybrid dehumidifier model for real-time performance monitoring, control and optimization in liquid desiccant dehumidification system," *Applied energy*, vol. 111, pp. 449-455, 2013.

- [90] X. Ou, W. Cai, X. He, D. Zhai, and X. Wang, "Dynamic modeling and validation of a liquid desiccant cooling and dehumidification system," *Energy and Buildings*, vol. 163, pp. 44-57, 2018.
- [91] C. Lee, "A simplified explicit model for determining the performance of a chilled water cooling coil," *International Journal of Refrigeration*, vol. 43, pp. 167-175, 2014.
- [92] L. Song and G. Wang, "Using a hybrid method to construct a computational efficient cooling coil model for an automated single-duct variable air volume system fault detection and diagnosis," *Energy and Buildings*, vol. 92, pp. 363-373, 2015.
- [93] A. Afram and F. Janabi-Sharifi, "Black-box modeling of residential HVAC system and comparison of gray-box and black-box modeling methods," *Energy and Buildings*, vol. 94, pp. 121-149, 2015.
- [94] G.-Y. Jin, W.-J. Cai, Y.-W. Wang, and Y. Yao, "A simple dynamic model of cooling coil unit," *Energy Conversion and Management*, vol. 47, no. 15-16, pp. 2659-2672, 2006.
- [95] X. Yuan, Q. Chen, and L. R. Glicksman, "A critical review of displacement ventilation," *ASHRAE transactions*, vol. 104, p. 78, 1998.
- [96] A. Svensson, "Nordic experience of displacement ventilation system," *ASHRAE Transaction.*, vol. 95, no. 2, pp. 1013-1017, 1989.
- [97] X. Yuan, Q. Chen, and L. R. Glicksman, "Performance evaluation and design guidelines for displacement ventilation," *ASHRAE Transactions*, vol. 105, p. 298, 1999.
- [98] W. Shan and D. Rim, "Thermal and ventilation performance of combined passive chilled beam and displacement ventilation systems," *Energy and Buildings*, vol. 158, pp. 466-475, 2018.
- [99] G. Hunt and P. Linden, "Steady-state flows in an enclosure ventilated by buoyancy forces assisted by wind," *Journal of Fluid Mechanics*, vol. 426, pp. 355-386, 2001.

- [100] J. Fredriksson and M. Sandberg, "The effect of false ceiling on the cooling capacity of passive chilled beams," *Building and Environment*, vol. 44, no. 7, pp. 1426-1430, 2009.
- [101] I. C. Nelson, C. H. Culp, J. Rimmer, and B. Tully, "The effect of thermal load configuration on the performance of passive chilled beams," *Building and Environment*, vol. 96, pp. 188-197, 2016.
- [102] H. Koskela, H. Hägglom, R. Kosonen, and M. Ruonen, "Air distribution in office environment with asymmetric workstation layout using chilled beams," *Building and Environment*, vol. 45, no. 9, pp. 1923-1931, 2010.
- [103] H. Koskela, H. Hägglom, R. Kosonen, and M. Ruonen, "Flow pattern and thermal comfort in office environment with active chilled beams," *HVAC&R Research*, vol. 18, no. 4, pp. 723-736, 2012.
- [104] Z. Li and S. Deng, "An experimental study on the inherent operational characteristics of a direct expansion (DX) air conditioning (A/C) unit," *Building and Environment*, vol. 42, no. 1, pp. 1-10, 2007.
- [105] X. Xu, L. Xia, M. Chan, and S. Deng, "Inherent correlation between the total output cooling capacity and equipment sensible heat ratio of a direct expansion air conditioning system under variable-speed operation (XXG SMD SHR DX AC unit)," *Applied thermal engineering*, vol. 30, no. 13, pp. 1601-1607, 2010.
- [106] H. Pombeiro, M. J. Machado, and C. Silva, "Dynamic programming and genetic algorithms to control an HVAC system: Maximizing thermal comfort and minimizing cost with PV production and storage," *Sustainable Cities and Society*, vol. 34, pp. 228-238, 2017.
- [107] V. Congradac and F. Kulic, "HVAC system optimization with CO₂ concentration control using genetic algorithms," *Energy and Buildings*, vol. 41, no. 5, pp. 571-577, 2009.
- [108] W. Huang and H. Lam, "Using genetic algorithms to optimize controller parameters for HVAC systems," *Energy and Buildings*, vol. 26, no. 3, pp. 277-282, 1997.

- [109] J. Seo, R. Ooka, J. T. Kim, and Y. Nam, "Optimization of the HVAC system design to minimize primary energy demand," *Energy and Buildings*, vol. 76, pp. 102-108, 2014.
- [110] F. Yu and K. Chan, "Modelling of the coefficient of performance of an air-cooled screw chiller with variable speed condenser fans," *Building and environment*, vol. 41, no. 4, pp. 407-417, 2006.
- [111] Y.-C. Chang, J.-K. Lin, and M.-H. Chuang, "Optimal chiller loading by genetic algorithm for reducing energy consumption," *Energy and Buildings*, vol. 37, no. 2, pp. 147-155, 2005.
- [112] L. Lu, W. Cai, Y. S. Chai, and L. Xie, "Global optimization for overall HVAC systems—Part I problem formulation and analysis," *Energy conversion and management*, vol. 46, no. 7-8, pp. 999-1014, 2005.
- [113] ASHRAE., *ASHRAE handbook of fundamentals*. Mar Lin Book Company, 1985.
- [114] ASHRAE., *Thermal environmental conditions for human occupancy* (no. 2004). American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2004.

Author's publications

1. Ke Ji, Wenjian Cai, Bingjie Wu, Xianhua Ou, Mechanical design and performance evaluation of active thermosiphon beam terminal units, *Building and Environment*, 153 (2019), 241-252.
2. Ke Ji, Wenjian Cai, Bingjie Wu, Xin Zhang, Modelling and validation of an active chilled beam terminal unit, *Journal of Building Engineering*, 22 (2019), 161-170.
3. Ke Ji, Wenjian Cai, Fuzzy model based predictive control for active chilled beam systems, in 12th IEEE Conference on Industrial Electronics and Applications (ICIEA), 2017, pp.807-812.
4. Ke Ji, Wenjian Cai, Bingjie Wu, Performance analysis of heat transfer rate and negative air ion application for passive thermosiphon beam system, in 13th IEEE Conference on Industrial Electronics and Applications (ICIEA), 2018, pp.1609-1614.