Entrainment performance enhancement of HVAC systems with active chilled beams

Guan, Zheming

2017

Guan, Z. (2017). Entrainment performance enhancement of hvac systems with active chilled beams. Doctoral thesis, Nanyang Technological University, Singapore.

<http://hdl.handle.net/10356/72171>

<https://doi.org/10.32657/10356/72171>

Entrainment Performance Enhancement for HVAC Systems with Active Chilled Beams

GUAN ZHEMING

School of Electrical & Electronic Engineering

A thesis submitted to the Nanyang Technological University
in fulfilment of the requirement for the degree Of
Doctorate of Philosophy

2016

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research and has not been submitted for a higher degree to any other university or institution.

.....

Date

.....

author

Acknowledgements

First and foremost, I would like to express my sincere thanks to my supervisor, Prof. Wen Changyun, for his guidance, advise and wisdom. He encourages me to think independently and concur the difficulties come across. To me, he is not only the academic advisor, but also a mentor in life and a sincere friend. His wisdom and attitude to science will guide me through my life.

I also thank Assoc Prof. Cai Wenjian for providing the fully equipped laboratory and Assoc Prof Li Hua for particular guidance on fluid mechanics. I would like to express special thanks to Mr. Li Yi in Australia, for inspiring me about the topic of research. I would like to thank Nanyang Technological University(NTU) for providing high quality of research and learning atmosphere.

I would like to thank my friends and colleagues in NTU. Especially, I would like to thank Zhu Yin Hai, Li Guoqi, Chen Jiawei, Guo Fanghong and Wei Zhe, who are also students to Prof Wen, and inspire me during my research. Importantly, I would like to thank Lin Chen, Zhao Lei, Chen Can, Ji Ke in Process and Instrumental Lab for their help on the use of lab equipments. I would also like to thank the friends who provide selfless help to me: Zhong Jingshan, Jiang Chaoyang, Xu Jinming and Guo Xianggui. I shared precious memories with my other labmates: Wang Mengmeng,

Jing Jin, Qi Yongbin, Wang Hao, Wei Chen, Zhu Qingchang, Yu hang, Zhao Wei, Zhai Deqing, Zheng Wenji. I am grateful that we share a fruitful time studying and working together in NTU. Thank you again for all your help and support, without which none of these can be achieved.

Special thanks must go to my dearest fiance, Xiang Yu, who encourages me and supports me throughout my life as a Ph.D. candidate. I would express my greatest thanks to my dear mother, who always trusts in me and backs me up.

Lastly, I would like to express late thanks to my dad, who passed away half a year before this thesis is done, who would like to see me wearing the gown of doctor degree most, who always loves me and is always loved by all.

R.I.P.

Summary

Heating, ventilation and air conditioning (HVAC) systems with active chilled beams (ACB) terminal units are attracting increasing attention in both theoretical research and practical application, due to their superior benefits compared with variable air volume (VAV) systems. In conventional VAV systems, a large volume of recirculation air is distributed through ductwork by big exhaust and supply fans. The energy consumed by these fans is enormous. As water has a much higher heat capacity compared with air, the ACB system utilizes water instead as the media of heat exchange, saving energy from delivering less volume of air. The volume of water transferring the same cooling is much smaller than air. Therefore, ACB systems are more energy efficient, by taking advantage of using less energy in distributing the chilled water directly to the air-conditioned space. Besides the energy saving benefits, ACB system can save the space above suspended ceilings. As most of the space is occupied by air supplying duct, the bottleneck for buildings with VAV system diminishing the space is to decrease the size of duct. As the ACB system requires less air supplying, it is an inherent character of ACB system to occupy less space above the ceiling, so that the floor-to floor height of a building can be decreased. Our research is dedicated how to enhance the energy efficiency performance of HVAC systems with space-friendly ACB terminal units. To achieve this, we make use of different tools to investigate the

energy efficiency of ACB systems in many aspects, especially by confining the geometric factors of terminal unit. The energy efficiency of the whole system can thus be increased. In the meanwhile, we also suggest reduction of the height of terminal unit, so that the ACB system can show a greater advantage in term of space saving. By these means, the economic value of the ACB system, as well as the user acceptance can be increased.

Accurate measurement of experimental data is the foundation of computations and the subsequent studies. An innovative method of acquiring entrainment ratio is proposed and validated through our experimental studies. With this method, the entrainment performance of a commercial terminal unit is tested by a series of experiments. The performance of the commercial terminal unit is also considered as a benchmark to compare and contrast.

We build a numerical model of the terminal unit in computational fluid dynamics (CFD) software. The software can simulate the air flow pattern inside the terminal unit. By iterative computations, the velocity of every single point inside the terminal unit can be numerically solved. Compared with the experimental data, the model is verified to accord with the experimental data and is thus proved valid.

After the model is established, we make numerous simulations and modification, and finally achieve a model that can largely increase the performance of the terminal units. It is found that by changing the geometry of the mixing chamber and lengthening the nozzle, the modified structure can increase entrainment ratio by 32% under the same working condition and primary air volume flow rate. The height of terminal unit is also diminished to fit more tight space.

The results obtained prove that the entrainment ratio of a terminal unit, as well as

the efficiency of the whole air conditioning system, can be effectively increased by proper modification on its geometry. The findings about the nozzle can also guide the adjustment of other geometric characters of the terminal unit.

Table of contents

Acknowledgements	i
Summary	iii
List of Abbreviations	x
List of Figures	xii
List of Tables	xvi
1 Introduction	1
1.1 Motivation and Objective	1
1.2 Methodology and Approaches	4
1.3 Contributions and outline of the thesis	5
2 Background Knowledge and Literature Review	8
2.1 HVAC system components and ACB Based System Overview	8

2.1.1	System overview	8
2.1.2	ACB terminal unit structure and airflow pattern	9
2.1.3	Entrainment ratio	10
2.1.4	Design constraints	12
2.2	State of the art	13
2.3	The use of CFD in entrainment study	17
3	Experiments on Type I Terminal Unit with a new method of acquiring ER	19
3.1	Introduction	19
3.2	Experiment setup	20
3.3	A new acquisition method of ER	26
3.4	Experiment results	41
3.5	Conclusions	46
4	CFD Model of Type I Terminal Unit	47
4.1	Introduction	47
4.2	Model simplification with loss coefficient	48
4.3	Numerical model of Type I terminal unit	52
4.4	Simulation results	57
4.5	Conclusion	60

5	Type II Structure:Modified for Higher ER	62
5.1	Introduction	62
5.2	The validity of CFD model	63
5.3	Motivation of modifications	64
5.3.1	Nozzle arrangement	66
5.3.2	Height of the mixing chamber	66
5.3.3	Mixing chamber walls	67
5.3.4	Length of nozzles	68
5.4	Optimization of Type II terminal unit	68
5.4.1	Dimensions of mixing chamber wall	72
5.4.2	Distance between nozzles and diameter of nozzles	89
5.4.3	Length of nozzles	94
5.5	Conclusion	98
6	Entrainment Performance Comparison and Ceiling Attachment Test	100
6.1	Introduction	100
6.2	Performance comparison between Type I terminal unit and Type II terminal unit	101
6.3	Performance comparison between Type II terminal unit with another terminal unit available in market	103

6.4	Entrainment performance with different primary air flow rate	105
6.5	Ceiling attachment test	110
6.6	Conclusion	112
7	Conclusion and future work	114
7.1	Conclusion	114
7.2	Recommendation of future work	117
	Author's Publications	118
	Bibliography	119
	Appendix A Simulation Report	127

List of Abbreviations

The following tables describe the acronyms and abbreviations used throughout the thesis.

Acronyms	Meaning
HVAC	heating, ventilation, and air conditioning
ACB	active chill beams
VAV	variable air volume
DOAS	dedicated outdoor air system
COP	coefficient of performance
SD	standard deviation
CFD	computational fluid dynamics
AHU	air handling unit
PIV	Particle Image Velocimetry

Abbreviation	Meaning
ER	entrainment ratio
f	loss coefficient
P	pressure(Pa)
ρ	air density(kg/m^3)
$\dot{Q}$	mass flow rate(kg/s)
S	cross sectional area of the exit of external hood(m^2)
U_n	normal air speed across chilled beams(m/s)
$\dot{V}$	volume flow rate(m^3/h)
v	the air speed measured with air velocity transducer
$\bar{v}$	average air speed at the exit of external hood (m/s)
L	length of nozzle, segments, terminal unit, etc.
D_n	distance between nozzles
W	width
H	height
n	number of blocks at the hood
m	number of nozzles
<i>Subscripts</i>	
1	primary air
2	secondary air
3	mixed air
in	virtual duct inlet
out	virtual duct outlet
a, b, c, d, g, h	coefficient of model
hd	external hood

List of Figures

2.1	The schematic of a typical ACB system, with the emphasis on how the cooling is delivered	9
2.2	The structure of a commercial terminal unit	10
2.3	The schematic diagram of airflow pattern inside terminal unit	11
3.1	The experiment plant containing Type I terminal unit for entrainment study	22
3.2	Coils that is arranged in two layers and the fins inside the terminal unit	23
3.3	The grille covering the coils and fins	23
3.4	The nozzles used in Type I terminal unit	24
3.5	The user interface of pressure transmitter	24
3.6	The control panel of fans	25
3.7	The the vortex precession flowmeter	25
3.8	Validation of proposed measuring method	29

3.9	Data fitting results to validate the primary air volume flow rate model regarding on the primary pressure	44
3.10	Data fitting results to validate the mixed air volume flow rate model regarding on the primary pressure	45
4.1	Geometry of virtual duct with notation of perforated grill, cooling coil and fins	49
4.2	Schematic of the virtual duct, with the emphasis on the pressure drop through it	50
4.3	Loss coefficients under different conditions	52
4.4	Geometric model of Type I terminal unit with hood	53
4.5	The notation of Type I terminal unit to illustrate its detailed dimensions	53
4.6	The generated mesh of Type I terminal unit with hood	55
4.7	The streamlines of Primary and induced room air flow inside Type I terminal unit	58
4.8	The streamlines of primary and induced room air flow inside Type I terminal unit - cross-sectional view	58
4.9	Pressure distribution inside terminal unit	59
4.10	Comparison of ERs obtained with different methods	60
5.1	Geometric model of Type II structure - 3-Dimensional view	65
5.2	Geometric model of Type II structure - cross sectional view	65

5.3	Geometry of the simplified terminal unit model used for optimization analysis	69
5.4	Generated mesh of the simplified terminal unit model used for optimization analysis	70
5.5	Boundary conditions of the simplified terminal unit model used for optimization analysis	71
5.6	The notation of Type II terminal unit to illustrate the detailed dimensions	72
5.7	Correlation matrix of dimensional parameters of Type II terminal unit, ER and $\dot{V}_1$	80
5.8	Sensitive analysis showing the correlation of dimensional parameters of Type II terminal unit, ER and $\dot{V}_1$	81
5.9	Comparison of different $\angle BAN$	83
5.10	Data fitting results to validate the model of $\angle BAN$ and ER	84
5.11	Comparison of different EG	86
5.12	Data fitting results to validate the model of EG and ER	86
5.13	Comparison of different AL	88
5.14	Data fitting results to validate the model of AL and ER	89
5.15	Data fitting results to validate the model of primary air volume flow rate and nozzle diameter	92
5.16	Relationship between D_n and ER	94

5.17	Primary air and induced air flow streamlines near the nozzle	97
6.1	Geometry of a virtual room with ACB terminal unit installed above ceilings	111
6.2	Streamlines inside the virtual room	111
6.3	Vector field inside the virtual room	112

List of Tables

3.1	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $60Pa$	29
3.2	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $70Pa$	30
3.3	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $80Pa$	31
3.4	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $90Pa$	32
3.5	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $100Pa$	33
3.6	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $110Pa$	34
3.7	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $120Pa$	35
3.8	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $130Pa$	36

3.9	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $140Pa$	37
3.10	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $150Pa$	38
3.11	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $160Pa$	39
3.12	Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $170Pa$	40
3.13	Velocities of air(m/s) measured at the exit of the external hoods to compute $\dot{V}_3$ (I)	42
3.14	Velocities of air(m/s) measured at the exit of the external hoods to compute $\dot{V}_3$ (II)	43
3.15	$\dot{V}_1$ and $\dot{V}_3$ under different working conditions	44
4.1	Simulation results of loss coefficient with different air flowing speed .	51
4.2	Dimensions of Type I terminal unit model	54
4.3	Configuration of mesh	56
4.4	Configuration of boundary conditions	57
4.5	ER calculated with empirical model and CFD under different pressures	59
5.1	Range of dimension used by Type II terminal unit	73
5.2	Parameter analysis of dimensions of Type II terminal unit (I)	74

5.3	Parameter analysis of dimensions of Type II terminal unit (II)	75
5.4	Parameter analysis of dimensions of Type II terminal unit (III)	76
5.5	Parameter analysis of dimensions of Type II terminal unit (IV)	77
5.6	Correlation matrix of dimensional parameters of Type II terminal unit	79
5.7	Dimensions set for optimization of $\angle BAN$	81
5.8	Results of simulations about optimization of $\angle BAN$	82
5.9	Dimensions set for optimization of EG	84
5.10	Results of simulations about optimization of EG	85
5.11	Dimensions set for optimization of AL	87
5.12	Results of simulations about optimization of AL	88
5.13	The volume flow rates of primary air with different diameters of nozzles	91
5.14	Results of simulations about the distance between nozzles	93
5.15	Entrainment performance with nozzles of different lengths	95
5.16	Adjusted $\dot{V}_1$ with nozzles of different lengths	96
5.17	Maximum ER with different nozzles	97
6.1	Comparison of ER between Type I terminal unit and Type II terminal unit	102
6.2	Comparison of $\dot{V}_1$ between Type I terminal unit and Type II terminal unit	102

6.3	Performance comparison of two types of terminal units	104
6.4	ER under different volume flow rate of primary air, working condition: $P = 75Pa$	106
6.5	ER under different volume flow rate of primary air, working condition: $P = 100Pa$	107
6.6	ER under different volume flow rate of primary air, working condition: $P = 150Pa$	108
6.7	ER under different volume flow rate of primary air, working condition: $P = 200Pa$	109

Chapter 1

Introduction

1.1 Motivation and Objective

Since Willis Carrier first invented modern air conditioning in 1902, the heating, ventilation and air conditioning (HVAC) systems has largely improved people's life, especially in Singapore, where every day is suffocatingly hot and humid. Even the Singapore's first prime minister and elder statesmen named the air-conditioner as the most important invention of the 20th century. The use of air-conditioning has thoroughly changed the lifestyle of people living in tropic area. Lim Swee Say, Singapore's environment minister, once said, "Air-conditioning plays a crucial role in our economy. Without it, many of our rank-and-file workers would probably be sitting under coconut trees to escape from the heat and humidity, instead of working in high-tech factories." However, the energy consumed by HVAC system is enormous. According to a report released in 2012 by U.S. Energy Information Administration, the electricity cost by heating, cooling and ventilation accounts for 30% of total electricity cost by all buildings[1]. As more and more attentions are paid on energy saving, the HVAC systems with active chilled beams(ACB) terminal units that can fulfil the need of better energy efficiency, are attracting increasing attention in both theoretical research and practical application [2–23]. Their benefits are superior comparing to

variable air volume(VAV) systems [24–28]. In conventional VAV system, large volume of recirculation air is distributed through ductwork by big exhaust and supply fans. The energy consumed by these fans is significant. As water has a much higher heat capacity compared with air, the ACB system utilizes water instead as the media of heat exchange, saving energy from delivering less volume of air. The ACB system is a combination of chilled beams system and a dedicated outdoor air system(DOAS). The chilled beams system can handle most of the temperature-driven sensible loads, and the DOAS system is used for dehumidify the indoor air, supplying fresh air and handle the rest of sensible loads. The combined system can fulfil the requirement of cooling and air quality, as well as the needs of energy conservation. Many researches have shown that, a chilled beam system is more energy-saving compared to VAV system [29–31]. In addition, as the temperature of coolant is higher in chilled beam systems, the coefficient of performance(COP) of the chiller is higher. In the meanwhile, low grade renewable energy can also be used in ACB based HVAC system, which can further reduce the cost of electricity and make ACB system more economic.

Besides its superior performance in energy saving, ACB system may reduce the total construction cost of a building. In most office buildings, the space above the suspension ceiling is occupied by HVAC system ductwork and terminal unit, lighting facilities, cables, etc. If the space above the suspension ceiling can be diminished, the floor-to-floor height can be reduced and so is the cost of construction. In most cases, the bottle neck of reducing this space is the size of the HVAC system ductwork and terminal units. The ductwork used in ACB system is smaller than the VAV system, as the volume rate of primary air needed for ACB system is smaller. The terminal unit of ACB system is usually smaller in height, but there is still room to improve. In fact, the great potential of ACB system has been well recognized in building industry.

The ACB system has been accepted by building service industry, and it is promoted as a "green" HVAC system solution [27, 32].

Despite the fact that ACB system has been used in Europe, North America and Australia for decades and the ACB products are getting mature to be accepted by more and more people, researches about optimization of the terminal unit of ACB systems are very rare, especially in the aspect of the geometry of terminal unit. An ideal ACB system provides cooling mainly through chilled beams and the primary air is mainly for ventilation and dehumidifying. However, the utilization of chilled beams is low in most products due to poor entrainment performance, resulting in the energy wasted in providing more primary air. The energy-saving advantages of ACB system are thus reduced, compared to VAV system. The entrainment performance is the bottle neck to increase the popularity of ACB system. The energy efficiency, especially entrainment performance, of the terminal units used in ACB system still needs to be improved. Therefore, to enhance the entrainment performance of terminal unit as much as possible is our primary objective. Nonetheless, only enhanced entrainment is not enough. In our research, we find it is very easy to enhance the entrainment by reducing the nozzle size, which will decrease the ventilation ability of individual terminal unit, but such modification will result in a suffocating indoor environment or a noteworthy increase in the amount of terminal units used per unit area. This is a contrary to the purpose of our study. Therefore, the comparison of entrainment ability must be within the condition of constant ventilation ability. Besides, as mentioned earlier, the ability of fitting tight space is one of the advantages of ACB system, this advantage must be hold, or even carried forward, through out our study.

In a nutshell, we dedicate to modify the current terminal unit of ACB system to improve its entrainment performance, with the same amount of primary air and indoor

air quality. The modified terminal unit should induce more room air with the same amount of primary air and less energy consumption. In the meanwhile, we also take the height of terminal unit into consideration, to fit more tightened ceiling space.

1.2 Methodology and Approaches

The study and design relating to fluid are usually accomplished in CFD software. Conventionally, the design and related tests are accomplished with prototypes, but the time and cost on manufacturing a prototype are enormous. Besides, it is usually cumbersome to get essential data for design with the prototype. Nonetheless, some physical process, such as air flow pattern and pressure distribution cannot be easily tested, and some data will not be available without proper transmitters. CFD techniques enable to theoretically simulate any physical process with relatively low cost and the cost can be even lower with more powerful computers. After computations, almost all the physical process incurred can be visualized within the software. More importantly, CFD software provides parametric analysis, such as correlation analysis, for a deeper insight for the optimizations.

Nonetheless, CFD technique is not a panacea. The results obtained may vary with different mesh configurations and boundary conditions set for the geometrical model, and they thus determine the validity of simulation results. The mesh and boundary conditions must be set properly. A common practise of verifying the validity of CFD results is to perform simulations on an off-the-shelf product, and conduct experiments on this product as well. By comparing the results from simulations and experiments, the validity of CFD model can be verified, and the results obtained from modified model are thus valid.

Our research is also carried in this way. We first conduct experiments on an off-the-shelf terminal unit. The relevant data on entrainment performance is measured carefully. In fact, a new measurement method is proposed and proven valid to acquire experiment data more accurately. The dimensions of the terminal unit are also measured for modelling in CFD software. The mesh configurations and boundary conditions are carefully selected with reference to the prior works and suggestions from expert. The simulations results obtained are then compared to the experimental results. Once the results are matched well, the model can be proven valid.

We then qualitatively analyze the entrainment performance of the terminal unit and propose hypothesis on enhancing entrainment performance. The geometry of the terminal unit is modified according to the hypothesis, which can then be verified with simulations. Once the entrainment performance is enhanced, we conduct quantitative and systematic optimization. By finding the local optimum of some dimensions, entrainment performance can be further enhanced.

1.3 Contributions and outline of the thesis

This thesis focuses on developing a terminal unit for ACB based HVAC system, with high entrainment ratio and diminished height. The contributions and outline of this thesis are summarized as follow:

1. In Chapter 2, we firstly present ACB based HVAC system. The functions of the constitutive components of ACB terminal unit are explained briefly, followed by the notation of different parts of air that flow inside. Then a very important index measuring the entrainment performance of terminal unit, namely, ER is

introduced. The constraints on enhancing the entrainment performance are also touched. Secondly, we review the current research results, and learn from their advanced achievements. Lastly, we introduce the application of CFD techniques in our study.

2. Chapter 3 is about a series of experiment on entrainment performance of a terminal unit. We build an experiment plant to measure ER of a commercial terminal unit. Attentions are paid on the airflow and induction inside the terminal unit. An innovative method to acquire ER is proposed. We creatively propose to acquire ER by measuring the volume flow rate of mixed air, instead of induced air. Two empirical models of the rate of air flowing inside terminal unit are established. An empirical model of ER is calculated according to the models of air flow rate. ER and primary air volume flow rate under normal working conditions are calculated with these models.
3. In Chapter 4, we build a geometric model of the terminal unit used in our experiment with CFD software. The dimensions are the same as the actual terminal unit. To save the time and cost of computation, loss coefficient is introduced to simplify the model. We also perform simulations to determine the loss coefficient. The configurations of the mesh and boundary conditions are carefully selected. They are also verified by the results of simulations. These results accord well with the experiment results, proving the validity of the CFD model.
4. In Chapter 5, we propose a new terminal unit with superior entrainment performance. Its structure is based on the terminal unit used in the experiment, with a few modifications applied on it. We then optimize the details about the proposed terminal unit. We firstly optimize the dimensions of the mixing cham-

ber wall, including three parameters with strong correlation with ER. Secondly, the distance between nozzles is optimized, under the condition that the supply of primary air holds constant. During the optimization, an empirical model of primary air supplied to nozzle diameter is established. The last optimization is about the length of the nozzles. The robustness of proposed terminal unit is proven during the optimization.

5. In Chapter 7, we firstly compare the entrainment performance of proposed terminal unit with the one used in the experiments, since the enhancement of entrainment performance is one of the major contributions of our study. Afterwards, The proposed terminal unit is compared with another commercial terminal unit. The proposed terminal unit shows superior benefits on the commercial product. Finally, we present simulations that the air discharged from terminal unit is well attached to the ceilings, ensuring the thermal comfort of the occupants in the conditioned room. For the ease of comparing with other products, a reference of entrainment performance under different primary air flow conditions is provided, based on simulations results.
6. Chapter 8 offers our conclusion about our study and suggests topics for future work.

Chapter 2

Background Knowledge and Literature Review

2.1 HVAC system components and ACB Based System Overview

2.1.1 System overview

The ACB based HVAC system and conventional HVAC system share a similar way of generating cooling, but differ in the way of delivering it. In conventional HVAC system, the media for delivering cool is air only, and the ACB system uses both air and water, as shown in Figure 2.1[33]. That is, in ACB based schemes, the chillers provide chilled water both to the air handling unit(AHU) and the terminal units in the conditioned room. A DOAS continuously supplies primary air to the ACB unit which is sufficiently cooled and dehumidified to handle all of the internal moisture-driven latent loads and partially the temperature-driven sensible loads. As one of the major purposes of primary air is for ventilation, it is also called ventilation air. Secondary air is induced up into the ACB terminal units and is cooled by a series of cooling coils

and fins within the beam. It is thus called induced air as well. Sometimes, it is also referred as room air. The primary air and the secondary air are then mixed in the mixing chamber and supplied into the room via linear slots on the edge of the ACB. The mixture of primary air and secondary air is referred as mixed air.

Figure 2.1: The schematic of a typical ACB system, with the emphasis on how the cooling is delivered

2.1.2 ACB terminal unit structure and airflow pattern

Figure 2.2 shows Type I terminal unit schematics in the market. This terminal unit originates from a commercial product of DADANCO[34]. It consists of a primary chamber to supply primary air, a series of nozzles to eject primary air, a mixing chamber where ejected primary air mixes with induced room air, and cooling coils and fins.

The airflow inside the terminal unit is illustrated in Figure 2.3. The sufficiently cooled and dehumidified primary air is delivered to the primary chamber through ductwork

*H:height W:width L:length

Figure 2.2: The structure of a commercial terminal unit

by around $100Pa$ pressure. The actual pressure may vary with working condition, ductwork shape and the distance to the AHU. The primary air is then ejected to the mixing chamber through the nozzles. The high speed ejected air forms a low pressure kernel near the nozzle exits, which induces room air into the mixing chamber. The warm room air, under the effect of suction and convection, will rise from room to terminal unit. The warm room air traverses and exchanges heat with the coils and fins. The cooled room air mixes with the ejected primary air and then is discharged to the air circulation in the room.

2.1.3 Entrainment ratio

The entrainment efficiency of the system, which is commonly called entrainment ratio(denoted by ER), is defined as:

$$ER = \frac{\dot{Q}_2}{\dot{Q}_1} \quad (2.1)$$

Figure 2.3: The schematic diagram of airflow pattern inside terminal unit

where $\dot{Q}_2$ is mass flow rate of induced air, defined as the mass flow of air induced to the mixing chamber through the coils and fins and $\dot{Q}_1$ is mass flow rate of primary air, defined as the mass flow of air ejected to the mixing chamber through nozzles.

It follows that the mixed air mass flow rate is given by

$$\dot{Q}_3 = \dot{Q}_1 + \dot{Q}_2 \quad (2.2)$$

So we have,

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{\dot{Q}_3}{\dot{Q}_1} - 1 \quad (2.3)$$

In ACB system, the pressure inside the primary chamber is normally within $75Pa$ to $150Pa$ gauge pressure, which is neglectable compared to the absolute pressure ($101kPa$), the change of density of air can be neglected. Therefore, the ratio of mass flow rate approximately equals the ratio of volume flow rate, i.e.,

$$\frac{\dot{Q}_3}{\dot{Q}_1} \approx \frac{\dot{V}_3}{\dot{V}_1}, \frac{\dot{Q}_2}{\dot{Q}_1} \approx \frac{\dot{V}_2}{\dot{V}_1} \quad (2.4)$$

Thus, we have

$$ER \approx \frac{\dot{V}_2}{\dot{V}_1} = \frac{\dot{V}_3}{\dot{V}_1} - 1 \quad (2.5)$$

The entrainment ratio is the key parameter to evaluate the entrainment performance of terminal unit, as it is closely related to the cooling ability of ACB system. An ACB system with higher entrainment ratio requires less primary air to handle indoor heat load, and thus has lower running cost. However, the minimum volume flow rate of primary air should not be lower than minimum requirement of ventilation air for hygiene and safety. The lower the entrainment ratio, the larger the volumetric flow rate of primary air to handle indoor heat load, as well as the noise. Therefore, to enhance the entrainment performance of ACB terminal unit, namely, to increase entrainment ratio of the terminal unit, is of great importance.

2.1.4 Design constraints

The design of terminal unit is constrained by its size and primary air flow rate. W and L in Figure 2.2 are fixed to fit suspended ceiling grids commonly used in most offices. Since the ability of fitting tight space is also an advantage of ACB system, it gains additional benefits with smaller H , so that the terminal unit is capable to fit more tight space. As for ventilation, it is obvious that less primary air (smaller Q_1) supplied helps to enlarge ER, but the primary air supplied to the room should meet the requirement of ASHRAE 62.1[35], where the minimum outdoor airflow rate is required. In our design, attentions are paid on keeping the primary air volume flow rate the same with Type I structure for a fair comparison and fulfilling the requirement.

Through our preliminary study, it is found that the ER of a terminal unit depends

on its geometric parameters. So the optimization problem in our study is to maximize the ER by searching for suitable geometric parameters, subject to practical constraints on the terminal unit. However, it is very complicated to find a mathematical expression how the ER depends on the geometric parameters. In fact, there is little theoretical analysis about the airflow and entrainment available. Thus in our study the optimization problem is solved by experimentally finding the maximum ER through comparing its value with different set-ups of geometric parameters meeting the constraints.

2.2 State of the art

In [32], Alamdari and his colleague examine the environmental performance of combined displacement ventilation and chilled ceiling system with both physical and computer modelling techniques. They also discuss the feasibility of low energy cooling systems, such as ground water cooling, thermosiphon cooling, dry coolers, and evaporative cooling. They also address that ground water cooling appears to have considerable potential.

In [30], Stetiu develop a parametric study on the energy and power savings potential of a radiant cooling system. The case study in this paper shows that about 30% of the energy consumption and 27% of the peak power demand can be saved by employing a radiant cooling system. The savings potential is climate-dependent, and is large in retrofitted buildings than in new constructions.

In [24], Roth, et al. discuss the two distinct architectures of chilled beams, its energy saving potentials, and its market factors. The chilled beams are also recommended to

use in laboratories where heat generated by laboratory and process equipment drives cooling loads and supply airflow levels.

In [27], Syed Moazzam Ali has examined a chilled beams system, and results are encouraging and showed HVAC energy consumption is saved up to 30%-40%, compared to a conventional air conditioned building. The simulation results show the chilled beam system together with the cooling strategies has much more advantages over the conventional HVAC system.

In [36], Alexander and O'Rourke address several design considerations in many aspects, including capacity and energy saving, suitability for various spaces, duct design and working static pressures, beam placement and room air distribution, air-side and water-side control, installation and maintenance. These constraints are described in details with full and accurate data.

In [37], Jan Fredriksson and his colleagues investigate the instantaneous velocity profiles of the airflow generated by the chilled beam, as well as the temperature field below the chilled beam with experiments conducted in a mockup of an office room. The results show that the airflow from the chilled beam behaves similarly to a two-dimensional plume but exhibit strong oscillation both sideways and along the chilled beam. They also add that the heat sources in the room may interfere the flow generated by the chilled beam.

In [38], Rumsey and Weale address how chilled beams work, how to apply them in laboratories, and the specific benefits (and limitations) of their use in laboratories. They also present a representative case study where chilled beams have been used successfully.

In [39], Memarzadeh, et al, numerically investigate the performance of ACB in a

general laboratory layout with some equipment intensive areas and the removal effectiveness of such a system. Their use of CFD software is a good example of CFD utilization in ACB related design.

In [40], Cao, et al, describe the measured and predicted air distribution of an active chilled beam in a typical office room. They apply a free convection model for the vertical downward air jet along the wall by superposing a free convection velocity and an isothermal jet velocity. The research fully consider the air flow pattern inside the room and thermal comfort of the occupants.

In [28], Mika Ruponen and his colleague present a new simplified method to measure the entrainment ratio of ACB terminal unit, where they uses a anemometer and a simple purpose built measurement venturi together with the primary airflow rate.

In [41], Cao applies Particle Image Velocimetry(PIV) method to study the turbulent air flow behaviors of the attached plane jet discharged from an ACB. This is a very innovative technology to study the air distribution inside the air conditioned room and the air flow pattern inside the ACB terminal unit.

In [26], Livchak and Lowell discuss what parameters affect active beam performance and introduce new criteria for beam selection. They criticize on the engineers who attempt to reduce system first costs and total number of beams, as this approach compromises the system's energy performance and diminishes advantages of active beam systems over all-air systems.

In [42], Koskela, et al, experimentally investigate the flow aptterns and draft risk in an office environment where cooling and air distribution is implemented with active chilled beams. The experiments are carried in a mock-up room with different heat loads mimicking the four seasons. Their study concentrates on measurement

and modelling results in a single-person office room. Comparisons are made with corresponding results in an open-plan office.

In [43], Chen and his colleagues present a hybrid model of ACB terminal unit, which took both mechanical and thermal issues in consideration. The confined air jet inside the terminal unit and the cooling coil of the chilled beam are divided into two sub-models. The primary air is taken as a confined air jet and the mixing of the primary air and secondary air are simplified with isothermal assumption. This hybrid model is simple and suitable for engineering applications. In [44], Chen and his colleagues also investigate the effect of different circuitry arrangements on heat transfer performance, as well as the effectiveness of circuitry adjustment.

In a nutshell, the feasibility of ACB system has been rigorously demonstrated and analyzed, and its economy and practicality, utilization of low grade cooling source, together with its future development potential are widely accepted by the researchers and engineers. In the research and development of ACB systems, much attention is paid on the thermal comfort of the room, the humidity control of the conditioned room, and the efficiency of the overall ACB system other than terminal units. The research about terminal unit, especially in the geometry of mixing chamber are very rare, so that there is no satisfactory products in the market.

In the aspect of research and development, the results from experiments are reliable and accurate, and that from simulations are convenient to get and efficient. However, it is impractical to obtain numerous data by relying solely on experiments, and the results obtained from simulations only are not convincing. A good way to overcome these difficulties is to conduct experiments on a off-the-shelf product, and also model it in CFD software. By comparing the CFD simulation results with experiment data, the validity of the model can be proved, and the results obtained from the CFD model

are thus convincing.

2.3 The use of CFD in entrainment study

Our study is partially performed in CFD software. CFD is a branch of fluid mechanics that uses numerical analysis and algorithms to solve and analyze problems that involve fluid flows. It is widely used in numerical studies of fluids [45–55]. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions[56]. In industrial design, it always takes long time and high cost to manufacture a prototype. The experiments and measurement performed on the prototype are also time-costing and cumbersome. With the help of CFD techniques, researchers can draw the geometry of desired prototype within the software. By solving the governing equations, simulation results can be viewed in the software. Especially, researchers can visualize the invisibles, such as, the flow pattern of air, pressure distributions and heat exchange process.

The complete process of CFD analysis can be summarized into three main steps: pre-processing, solving, and post-processing. In pre-processing step, we firstly set up the geometry of the terminal unit and define the computational domain with Design Modeler. Secondly, the grid for finite element analysis is generated with ANSYS Mesh. Lastly, we set up the physical and mathematical model, fluid properties and the boundary conditions used for computations. The solving step is finished with computers by iterative computations. The governing equations used in CFD analysis includes Navier-Stokes equations, continuity equation, equation of state and Rayleigh Equation. The results of computation can be viewed in the post-processing step. In post-processing, the results can be visualized with streamlines, vector fields and

contour maps. If the CFD results match the experiment results, the CFD model can be validated. The design and modification of new terminal unit can be applied on the validated CFD model. The CFD techniques allow the design and prototyping of new terminal unit performed in virtual environment. The time and cost can be significantly cut down. The air flow pattern inside the terminal unit can be visualized. With the help of CFD techniques, the study of entrainment performance of terminal unit can be carried out easily.

Chapter 3

Experiments on Type I Terminal Unit with a new method of acquiring ER

3.1 Introduction

For decades, researchers and engineers have dedicated to enhance the entrainment performance of terminal unit, and the products they put on market are various. Different models of products can be purchased from European, North American and Australian markets. The products they provide are mature, with different shapes and functionalities to fit various scenarios. However, the research relating the optimization of terminal unit or the enhancement of entrainment performance is very rare. The terminal units used in the market are not optimized. We believe that through our systematic modification and optimization, we can enhance the entrainment performance of terminal unit in ACB system. Prior to further study, we firstly conduct experiment on existing product. The emphasis is on measuring $\dot{V}_1, \dot{V}_2, \dot{V}_3$ and corresponding ER, and providing basis for future research.

In our research, we study an off-shelf terminal unit manufactured by one of most famous ACB terminal unit providers, DADANCO. For ease of notation, we hereby name the terminal unit Type I terminal unit. Our experiment plant here is built with Type I terminal unit. In our experiment, we measure V_1 , V_3 under different primary pressures, and establish separate model according to the experimental results. The model of ER is then calculated according to (2.5). The data is recorded for analysing the entrainment performance of terminal unit as well as verifying the validity of CFD model in following chapters. In addition, as one of the most important parameters to evaluate a terminal unit, the ER cannot be measured accurately with current methods. We thus propose a new method of acquiring ER by measuring volume flow rate of mixed air. This method is properly calibrated and validated through experiments.

This chapter is organized as follows. In Section 3.2, we present the experiment plant and its constitutive components. In Section 3.3, the proposed ER acquisition method is presented with its calibration and validation. In Section 3.4, the results obtained during the experiments are given in details and the concluding remarks are offered in Section 3.5.

3.2 Experiment setup

The experiment plant locates at the Process Instrumentation Laboratory, Nanyang Technological University, consisting of the following parts, as shown in Fig. 3.1:

1. Type I terminal unit with coils and fins inside. Note that the coils and fins contain a lot of trivial structures, shown in Figure 3.2. They are covered with

perforated grille, shown in Figure 3.3.

2. Two external hoods covering the outlets of the terminal unit.
3. Two centrifugal fans connected in series, supplying primary air to the terminal unit
4. An intelligent vortex precession flowmeter to measure volume flow rate of primary air.
5. A pressure transmitter to measure the pressure inside primary chamber.
6. An air velocity transducer to measure the velocity of discharged mixed air.
7. Ducts that connect all the parts, with length ten times of diameter.

*(1) A terminal unit (upside down) with cooling coils and fins inside. (2) Two centrifugal fans connected in series to supply primary air. (3) Two external hood covering the outlets of terminal unit. (4) An intelligent vortex precession flowmeter to measure volume flow rate of primary air. (5) A pressure transmitter to measure the pressure inside primary chamber. (6) An air velocity transducer to measure the velocity of discharged mixed air. (7) Ducts that connects all the parts.

Figure 3.1: The experiment plant containing Type I terminal unit for entrainment study

Figure 3.2: Coils that is arranged in two layers and the fins inside the terminal unit

Figure 3.3: The grille covering the coils and fins

In this study, as most of our attention is paid on the air flow inside the terminal unit, heat exchange is not taken into consideration. Therefore, our experiments are also carried isothermally. For ease of measurement and installation, we put the terminal unit upside down. The way of installation will not affect the entrainment performance, as gravity has little influence on the results. Figure 3.4 shows the nozzles used in Type I terminal unit.

Figure 3.4: The nozzles used in Type I terminal unit

During the experiments, the pressure inside primary chamber can be read directly from the pressure transmitter and it is tuned by adjusting the speed of the centrifugal fans. The user interface of pressure transmitter is shown in Figure 3.5. The speed of fans can be tuned with the control panel shown in Figure 3.6

Figure 3.5: The user interface of pressure transmitter

Once the speed of fans settles down and the pressure inside terminal unit fluctuates little, we can read the volume flow rate of primary air ($\dot{V}_1$) from the vortex precession flowmeter. Figure 3.7 gives a closer look to the vortex precession flowmeter.

According to (2.3), ERs under different working conditions can be acquired if $\dot{V}_3$ is known. Instead of using current method, we propose a new acquisition method of

Figure 3.6: The control panel of fans

Figure 3.7: The the vortex precession flowmeter

ER. The following section shows how ER is acquired by measuring $\dot{V}_3$.

3.3 A new acquisition method of ER

Conventionally, ER is calculated with $\dot{V}_1$ and $\dot{V}_2$ [43]. However, the speed of induced air is hard to measure accurately. First, the speed of induced air is very low and it is very sensitive to external environment. In fact, the speed of induced air so low that even the breath from experiment operator may cause a big error to it. Therefore, the results may easily be influenced by experimental conditions and instruments used. This method is not robust to even a very small disturbance. Secondly, the distribution of air speed is highly uneven. As the area of induced air inlet is relatively large, it is not possible to continuously measure at every point. The averaged speed calculated from measurements at several selected points are not sufficient to represent its average speed. Some researchers use an external hood at the induced air inlet and they believe that air speed can be measured from a velocity meter installed at the other end of the hood[43]. However, the outer hood itself is an interference to the induced air flow. As room air is very sensitive to the environmental condition, this method may also result in large errors.

Instead of installing the polymer hood at low speed induced air inlet, we choose to install a hood made of stainless steel at the mixed air outlets. They are properly sealed so that the discharged mixed air will not leak. The rigid body of the hood made by steel ensures that it is not deformable and will not interfere the mixed air flow. Its inner walls are smooth, so the resistance to the mixed airflow is small. As shown in Figure 3.1, the dimensions of the hood are denoted by L_{hd} , H_{hd} and W_{hd} . L_{hd} is in the direction of air flows and it is long enough to regulate the mixed air and

reduce the turbulence. The cross-sectional area of the hood can be calculated by,

$$S = W_{hd}H_{hd}$$

Each hood is equally divided into n identical blocks, with an area of S/n each. As the inner wall is smooth and the blocks are identical, we assume that at the velocity measured at the center of n th block, v_n can represent the average air flow speed of the block. As two hoods are installed on terminal unit, we have the total volume flow rate of mixed air,

$$\dot{V}_3 = \sum_{i=1}^{2n} v_i \frac{S}{n}$$

Let

$$\bar{v} = \frac{\sum_{i=1}^{2n} v_i}{2n}$$

which is the average speed of the air flow at the exit of the outlet. It is estimated by the average of the speeds at different spots, measured by air velocity transducer. Thus we have,

$$\dot{V}_3 = \bar{v}S$$

It is verified by the simulations that the external hood is long enough to regulate the mixed air and reduce its turbulence. The reading on air velocity transducer fluctuates little when we take the measurement, which proves the turbulence is significantly eliminated. At the outlet of the terminal unit without hood, the velocity of air is not uniformly distributed. The velocity of air is higher near the mixing chamber wall, as the portion of primary air is larger. Once the hood is installed, it will regulate the discharged air, and make the distribution of air more uniform. In our experiment, the velocity transducer measures the velocity of mixed air at 19 different spots equally

spaced near the exit of the hood on each side ($n = 19$). Five measurements with an interval of one second are taken at each spot. The average is considered as the air flow velocity at this spot. The average speed of totally 38 spots multiplied by the cross-sectional area of the hood exit gives the total volume flow rates of mixed air ($\dot{V}_3$) and *ER* can thus be calculated with (2.3).

In order to prove the validity of the new measurement method, we have performed the following tests. The induced air inlet and one side of the outlet are sealed tightly, leaving only one side of outlet open. Consequently, only the primary air will flow out of the terminal unit through the unsealed side outlet. The volume rate of primary air that flows through the calibrated vortex precession flowmeter can be directly read. The velocity of air flowing out of the outlet is measured at 19 above-mentioned spots. The volume flow rate can be calculated by multiplying the average speed with the cross-sectional area of the hood, which should be equal to the reading from flowmeter. The validity can be thus proved. To ensure this method is valid throughout our experiment, the validation processes are carried in different working conditions, from $60Pa$ to $170Pa$, with intervals of $10Pa$.

Table 3.1 to 3.9 give the velocities of air that are measured five times at 19 different spots, under the working conditions from $60Pa$ to $170Pa$, respectively. The comparison between the volume flow rate calculated from the proposed method and the volume flow rate read from flowmeter is presented in Figure 3.8. From the results, it can be observed from the figure that the maximum errors are well kept within $\pm 5\%$. Therefore, this concludes that the proposed method used to measure mixed air is valid.

Figure 3.8: Validation of proposed measuring method

Table 3.1: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $60Pa$

Spot	1st	2bd	3rd	4th	5th	Average	S.D.
1	0.250	0.253	0.258	0.255	0.255	0.254	0.006
2	0.292	0.297	0.295	0.296	0.305	0.297	0.010
3	0.306	0.306	0.314	0.313	0.327	0.313	0.017
4	0.313	0.314	0.324	0.325	0.335	0.322	0.018
5	0.281	0.272	0.274	0.280	0.281	0.278	0.009
6	0.237	0.234	0.231	0.234	0.229	0.233	0.006
7	0.164	0.156	0.151	0.143	0.146	0.152	0.017
8	0.108	0.090	0.095	0.085	0.087	0.093	0.018
9	0.087	0.069	0.078	0.066	0.080	0.076	0.017
10	0.059	0.046	0.057	0.053	0.051	0.053	0.010
11	0.078	0.073	0.074	0.078	0.063	0.073	0.012
12	0.080	0.088	0.082	0.088	0.075	0.083	0.011
13	0.115	0.134	0.119	0.128	0.119	0.123	0.016
14	0.171	0.192	0.177	0.176	0.175	0.178	0.016
15	0.202	0.217	0.207	0.219	0.205	0.210	0.015
16	0.211	0.223	0.213	0.230	0.212	0.218	0.017
17	0.197	0.194	0.198	0.204	0.194	0.197	0.008
18	0.168	0.163	0.154	0.165	0.165	0.163	0.011
19	0.157	0.156	0.142	0.153	0.148	0.151	0.012

Table 3.2: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $70Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.325	0.318	0.317	0.331	0.331	0.325	0.016
2	0.339	0.316	0.336	0.325	0.348	0.333	0.025
3	0.371	0.371	0.371	0.369	0.377	0.372	0.006
4	0.384	0.380	0.374	0.378	0.380	0.379	0.007
5	0.332	0.334	0.318	0.335	0.336	0.331	0.015
6	0.270	0.270	0.248	0.265	0.266	0.264	0.018
7	0.189	0.178	0.161	0.174	0.180	0.176	0.020
8	0.125	0.120	0.105	0.125	0.122	0.119	0.017
9	0.086	0.085	0.066	0.087	0.086	0.082	0.018
10	0.078	0.082	0.057	0.083	0.084	0.077	0.023
11	0.071	0.090	0.059	0.075	0.085	0.076	0.024
12	0.093	0.123	0.086	0.098	0.112	0.102	0.030
13	0.131	0.154	0.125	0.137	0.146	0.139	0.023
14	0.209	0.234	0.210	0.223	0.222	0.220	0.021
15	0.220	0.247	0.223	0.231	0.227	0.230	0.021
16	0.234	0.260	0.239	0.241	0.230	0.241	0.023
17	0.211	0.235	0.215	0.219	0.224	0.221	0.019
18	0.172	0.183	0.173	0.171	0.164	0.173	0.014
19	0.176	0.180	0.180	0.172	0.188	0.179	0.012

Table 3.3: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $80Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.428	0.404	0.400	0.396	0.403	0.406	0.025
2	0.380	0.363	0.353	0.360	0.369	0.365	0.020
3	0.453	0.440	0.439	0.437	0.448	0.443	0.014
4	0.437	0.429	0.420	0.427	0.442	0.431	0.017
5	0.359	0.368	0.357	0.363	0.377	0.365	0.016
6	0.298	0.311	0.308	0.305	0.317	0.308	0.014
7	0.184	0.198	0.192	0.192	0.203	0.194	0.014
8	0.003	0.022	0.010	0.016	0.020	0.014	0.016
9	0.072	0.086	0.074	0.083	0.084	0.080	0.013
10	0.082	0.092	0.068	0.081	0.076	0.080	0.018
11	0.072	0.083	0.063	0.071	0.067	0.071	0.015
12	0.105	0.109	0.081	0.096	0.095	0.097	0.022
13	0.172	0.181	0.145	0.157	0.158	0.163	0.028
14	0.254	0.264	0.240	0.248	0.252	0.252	0.018
15	0.281	0.297	0.266	0.278	0.287	0.282	0.023
16	0.280	0.287	0.266	0.279	0.287	0.280	0.017
17	0.259	0.264	0.253	0.259	0.263	0.260	0.009
18	0.213	0.218	0.205	0.214	0.212	0.212	0.009
19	0.207	0.218	0.206	0.198	0.202	0.206	0.015

Table 3.4: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $90Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.427	0.426	0.430	0.406	0.416	0.421	0.020
2	0.416	0.430	0.400	0.411	0.405	0.412	0.023
3	0.488	0.501	0.478	0.481	0.484	0.486	0.018
4	0.513	0.531	0.479	0.515	0.517	0.511	0.038
5	0.421	0.445	0.406	0.422	0.436	0.426	0.030
6	0.322	0.341	0.329	0.340	0.348	0.336	0.021
7	0.168	0.183	0.184	0.180	0.193	0.182	0.018
8	0.112	0.120	0.138	0.113	0.128	0.122	0.022
9	0.078	0.093	0.113	0.080	0.084	0.090	0.029
10	0.063	0.083	0.104	0.064	0.070	0.077	0.034
11	0.064	0.081	0.125	0.069	0.073	0.082	0.049
12	0.099	0.112	0.114	0.093	0.120	0.108	0.022
13	0.078	0.101	0.141	0.090	0.126	0.107	0.052
14	0.223	0.247	0.242	0.239	0.276	0.245	0.039
15	0.288	0.319	0.298	0.307	0.339	0.310	0.039
16	0.315	0.349	0.316	0.331	0.364	0.335	0.043
17	0.297	0.346	0.304	0.315	0.334	0.319	0.041
18	0.239	0.272	0.245	0.244	0.266	0.253	0.029
19	0.231	0.249	0.225	0.226	0.234	0.233	0.019

Table 3.5: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $100Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.492	0.500	0.475	0.467	0.478	0.482	0.027
2	0.438	0.453	0.420	0.430	0.422	0.433	0.027
3	0.528	0.552	0.521	0.533	0.519	0.531	0.026
4	0.539	0.565	0.529	0.562	0.531	0.545	0.034
5	0.481	0.504	0.484	0.512	0.474	0.491	0.032
6	0.360	0.382	0.361	0.400	0.351	0.371	0.040
7	0.230	0.245	0.223	0.273	0.224	0.239	0.042
8	0.148	0.160	0.141	0.161	0.143	0.151	0.019
9	0.084	0.098	0.071	0.098	0.080	0.086	0.023
10	0.083	0.096	0.084	0.088	0.069	0.084	0.020
11	0.088	0.094	0.096	0.074	0.076	0.086	0.020
12	0.125	0.134	0.134	0.107	0.122	0.124	0.022
13	0.198	0.206	0.202	0.176	0.194	0.195	0.023
14	0.303	0.306	0.317	0.280	0.297	0.301	0.027
15	0.376	0.382	0.390	0.344	0.368	0.372	0.035
16	0.357	0.360	0.364	0.343	0.356	0.356	0.016
17	0.362	0.368	0.368	0.360	0.372	0.366	0.010
18	0.268	0.279	0.264	0.273	0.275	0.272	0.012
19	0.225	0.225	0.206	0.232	0.240	0.226	0.025

Table 3.6: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $110Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.556	0.574	0.542	0.552	0.542	0.553	0.026
2	0.480	0.497	0.487	0.483	0.471	0.484	0.019
3	0.603	0.622	0.603	0.609	0.587	0.605	0.025
4	0.607	0.620	0.608	0.620	0.577	0.606	0.035
5	0.500	0.509	0.497	0.520	0.469	0.499	0.038
6	0.359	0.354	0.335	0.369	0.314	0.346	0.044
7	0.248	0.240	0.214	0.241	0.201	0.229	0.040
8	0.175	0.162	0.132	0.145	0.132	0.149	0.038
9	0.117	0.121	0.075	0.074	0.103	0.098	0.045
10	0.104	0.122	0.055	0.044	0.095	0.084	0.066
11	0.106	0.116	0.044	0.056	0.089	0.082	0.062
12	0.141	0.136	0.072	0.094	0.124	0.113	0.059
13	0.211	0.213	0.145	0.160	0.191	0.184	0.061
14	0.352	0.346	0.291	0.295	0.328	0.322	0.057
15	0.422	0.416	0.366	0.350	0.418	0.394	0.068
16	0.402	0.394	0.353	0.359	0.403	0.382	0.049
17	0.384	0.378	0.362	0.357	0.382	0.373	0.025
18	0.295	0.288	0.285	0.288	0.293	0.290	0.008
19	0.254	0.245	0.250	0.257	0.244	0.250	0.011

Table 3.7: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $120Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.566	0.551	0.535	0.573	0.524	0.550	0.041
2	0.600	0.563	0.549	0.606	0.533	0.570	0.064
3	0.681	0.610	0.629	0.681	0.605	0.641	0.075
4	0.709	0.621	0.641	0.686	0.602	0.652	0.089
5	0.637	0.547	0.572	0.581	0.519	0.571	0.088
6	0.463	0.439	0.395	0.414	0.350	0.412	0.086
7	0.323	0.303	0.241	0.267	0.216	0.270	0.088
8	0.200	0.192	0.140	0.151	0.101	0.157	0.081
9	0.112	0.138	0.102	0.121	0.062	0.107	0.057
10	0.108	0.139	0.119	0.104	0.069	0.108	0.051
11	0.068	0.101	0.109	0.083	0.050	0.082	0.048
12	0.091	0.104	0.120	0.110	0.070	0.099	0.039
13	0.192	0.223	0.237	0.206	0.177	0.207	0.048
14	0.333	0.359	0.377	0.361	0.324	0.351	0.044
15	0.398	0.412	0.407	0.416	0.371	0.401	0.036
16	0.408	0.426	0.403	0.427	0.374	0.408	0.043
17	0.411	0.412	0.395	0.419	0.392	0.406	0.023
18	0.321	0.315	0.279	0.330	0.303	0.310	0.039
19	0.294	0.282	0.263	0.315	0.284	0.288	0.038

Table 3.8: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $130Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.623	0.631	0.613	0.633	0.607	0.621	0.023
2	0.572	0.588	0.555	0.580	0.573	0.574	0.024
3	0.751	0.760	0.734	0.736	0.738	0.744	0.022
4	0.714	0.727	0.710	0.696	0.730	0.715	0.027
5	0.614	0.628	0.627	0.597	0.626	0.618	0.026
6	0.461	0.495	0.495	0.457	0.488	0.479	0.037
7	0.252	0.304	0.290	0.250	0.295	0.278	0.051
8	0.146	0.207	0.201	0.149	0.201	0.181	0.061
9	0.059	0.130	0.120	0.084	0.128	0.104	0.063
10	0.043	0.108	0.111	0.057	0.116	0.087	0.069
11	0.080	0.113	0.134	0.083	0.106	0.103	0.045
12	0.087	0.125	0.121	0.099	0.099	0.106	0.032
13	0.233	0.263	0.251	0.242	0.224	0.243	0.030
14	0.384	0.408	0.374	0.379	0.351	0.379	0.041
15	0.441	0.442	0.411	0.443	0.421	0.432	0.029
16	0.468	0.460	0.457	0.460	0.462	0.461	0.008
17	0.463	0.467	0.454	0.451	0.467	0.460	0.015
18	0.330	0.314	0.320	0.316	0.331	0.322	0.016
19	0.307	0.321	0.313	0.275	0.296	0.302	0.036

Table 3.9: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $140Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.701	0.644	0.683	0.694	0.712	0.687	0.052
2	0.664	0.619	0.636	0.669	0.656	0.649	0.042
3	0.768	0.749	0.768	0.781	0.763	0.766	0.023
4	0.764	0.775	0.786	0.766	0.746	0.767	0.030
5	0.659	0.679	0.684	0.672	0.638	0.666	0.037
6	0.462	0.480	0.498	0.478	0.433	0.470	0.049
7	0.260	0.247	0.265	0.256	0.190	0.244	0.061
8	0.178	0.125	0.147	0.138	0.090	0.136	0.064
9	0.142	0.073	0.119	0.117	0.068	0.104	0.064
10	0.148	0.091	0.138	0.145	0.084	0.121	0.062
11	0.166	0.128	0.142	0.176	0.130	0.148	0.043
12	0.161	0.111	0.113	0.151	0.105	0.128	0.052
13	0.285	0.231	0.225	0.271	0.213	0.245	0.062
14	0.449	0.392	0.378	0.433	0.376	0.406	0.067
15	0.511	0.456	0.457	0.497	0.456	0.475	0.053
16	0.513	0.461	0.468	0.472	0.458	0.474	0.045
17	0.473	0.488	0.502	0.495	0.491	0.490	0.022
18	0.335	0.366	0.389	0.375	0.392	0.371	0.046
19	0.263	0.286	0.324	0.344	0.279	0.299	0.067

Table 3.10: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $150Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.532	0.592	0.577	0.580	0.583	0.573	0.047
2	0.656	0.716	0.699	0.706	0.688	0.693	0.046
3	0.770	0.820	0.776	0.791	0.805	0.792	0.041
4	0.805	0.799	0.813	0.823	0.782	0.804	0.031
5	0.732	0.714	0.751	0.749	0.696	0.728	0.047
6	0.544	0.591	0.574	0.575	0.491	0.555	0.079
7	0.277	0.251	0.316	0.304	0.236	0.277	0.068
8	0.172	0.131	0.173	0.193	0.115	0.157	0.065
9	0.136	0.082	0.098	0.159	0.074	0.110	0.073
10	0.171	0.115	0.109	0.176	0.150	0.144	0.062
11	0.134	0.080	0.078	0.141	0.124	0.111	0.060
12	0.134	0.097	0.194	0.144	0.151	0.144	0.070
13	0.205	0.256	0.212	0.243	0.247	0.233	0.045
14	0.398	0.462	0.456	0.431	0.478	0.445	0.062
15	0.479	0.510	0.494	0.499	0.553	0.507	0.056
16	0.527	0.532	0.535	0.527	0.574	0.539	0.040
17	0.479	0.468	0.481	0.463	0.512	0.481	0.038
18	0.422	0.388	0.383	0.383	0.398	0.395	0.033
19	0.384	0.363	0.337	0.358	0.346	0.358	0.036

Table 3.11: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $160Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.783	0.793	0.733	0.813	0.756	0.776	0.063
2	0.716	0.726	0.665	0.735	0.690	0.706	0.057
3	0.813	0.858	0.849	0.860	0.823	0.841	0.043
4	0.834	0.865	0.867	0.848	0.836	0.850	0.031
5	0.731	0.754	0.772	0.766	0.736	0.752	0.036
6	0.556	0.576	0.609	0.599	0.578	0.584	0.042
7	0.315	0.344	0.373	0.354	0.321	0.341	0.048
8	0.193	0.200	0.223	0.199	0.185	0.200	0.028
9	0.101	0.100	0.121	0.092	0.122	0.107	0.027
10	0.139	0.147	0.161	0.131	0.156	0.147	0.024
11	0.128	0.126	0.127	0.106	0.133	0.124	0.021
12	0.128	0.126	0.122	0.122	0.152	0.130	0.025
13	0.265	0.265	0.262	0.265	0.280	0.267	0.014
14	0.435	0.452	0.437	0.452	0.455	0.446	0.019
15	0.500	0.518	0.524	0.508	0.527	0.515	0.023
16	0.547	0.561	0.561	0.546	0.588	0.561	0.034
17	0.520	0.527	0.510	0.500	0.544	0.520	0.034
18	0.426	0.412	0.419	0.413	0.442	0.422	0.025
19	0.328	0.305	0.333	0.305	0.340	0.322	0.033

Table 3.12: Velocities of air flow(m/s) measured for measuring method validation when pressure in primary chamber is $170Pa$

Spot	1st	2nd	3rd	4th	5th	Average	S.D.
1	0.789	0.758	0.773	0.794	0.792	0.781	0.031
2	0.752	0.723	0.747	0.735	0.724	0.736	0.026
3	0.927	0.881	0.903	0.877	0.876	0.893	0.044
4	0.967	0.935	0.929	0.915	0.898	0.929	0.051
5	0.877	0.834	0.820	0.813	0.795	0.828	0.062
6	0.758	0.729	0.708	0.727	0.694	0.723	0.048
7	0.344	0.351	0.336	0.355	0.353	0.348	0.016
8	0.163	0.194	0.177	0.197	0.210	0.188	0.037
9	0.088	0.092	0.097	0.105	0.128	0.102	0.032
10	0.108	0.131	0.154	0.117	0.174	0.137	0.054
11	0.154	0.151	0.158	0.122	0.188	0.155	0.047
12	0.129	0.168	0.153	0.136	0.211	0.159	0.065
13	0.229	0.254	0.234	0.202	0.251	0.234	0.042
14	0.500	0.507	0.479	0.419	0.466	0.474	0.070
15	0.580	0.586	0.566	0.507	0.554	0.559	0.063
16	0.611	0.600	0.595	0.531	0.567	0.581	0.064
17	0.553	0.566	0.589	0.522	0.582	0.562	0.053
18	0.419	0.412	0.437	0.388	0.400	0.411	0.037
19	0.379	0.396	0.383	0.359	0.356	0.375	0.034

3.4 Experiment results

The flow rate of primary air supplied to the primary chamber may vary in different working conditions. Its actual value may be tuned to fulfil the requirement of fresh air. The distance from AHU to terminal unit may also affect the primary pressure at the terminal unit, as pressure attenuates in the ducts. Therefore, our experiments are performed in a series of working conditions with different pressures.

Considering that the system may stay in transition states after the speed of fans is changed, the reading from vortex precession flowmeter for $\dot{V}_1$ is taken after 5 minutes. The velocity of discharged mixed air is also measured after 5 minutes, to ensure the airflow has reached steady states.

The velocity of discharged mixed air measured at the exit of the external hoods is shown in Table 3.13 and 3.14. Note that v_1 to v_{19} are measured at the hood further from the fans, and v_{20} to v_{38} are measured at the hood nearer to the fans. Table 3.15 records $\dot{V}_1$ read from flowmeter and $\dot{V}_3$ calculated from Table 3.13 and 3.14.

Through data fitting based on Table 3.15, a suitable function describing the relationship between $\dot{V}_1$ and pressure is found to be

$$\dot{V}_1 = aP^b \quad (3.1)$$

where $a = 10.4978$, $b = 0.5131$, and the relationship between $\dot{V}_3$ and pressure can be described as

$$\dot{V}_3 = cP^d \quad (3.2)$$

Table 3.13: Velocities of air(m/s) measured at the exit of the external hoods to compute $\dot{V}_3$ (I)

Spot	60Pa	70Pa	75Pa	80Pa	90Pa	100Pa
v_1	0.426	0.445	0.574	0.49	0.494	0.639
v_2	0.436	0.472	0.548	0.512	0.572	0.656
v_3	0.492	0.541	0.551	0.624	0.519	0.672
v_4	0.579	0.63	0.615	0.672	0.608	0.721
v_5	0.665	0.713	0.695	0.763	0.791	0.81
v_6	0.502	0.54	0.521	0.598	0.615	0.646
v_7	0.533	0.602	0.598	0.659	0.649	0.736
v_8	0.603	0.656	0.606	0.705	0.706	0.848
v_9	0.569	0.634	0.5936	0.668	0.712	0.724
v_{10}	0.4568	0.576	0.508	0.636	0.647	0.734
v_{11}	0.448	0.483	0.436	0.499	0.561	0.595
v_{12}	0.408	0.443	0.412	0.487	0.498	0.626
v_{13}	0.425	0.474	0.437	0.465	0.535	0.556
v_{14}	0.308	0.354	0.277	0.389	0.408	0.459
v_{15}	0.292	0.317	0.262	0.313	0.428	0.406
v_{16}	0.326	0.327	0.307	0.363	0.445	0.432
v_{17}	0.333	0.334	0.292	0.347	0.373	0.364
v_{18}	0.31	0.345	0.293	0.329	0.404	0.414
v_{19}	0.285	0.337	0.382	0.338	0.452	0.489
v_{20}	0.883	0.917	0.837	1.025	1.085	1.066
v_{21}	0.808	0.838	0.756	0.891	0.974	1.054
v_{22}	0.819	0.902	0.928	0.974	1.055	1.153
v_{23}	0.786	0.846	0.989	0.908	0.968	1.128
v_{24}	0.733	0.754	0.754	0.871	0.885	0.878
v_{25}	0.518	0.608	0.595	0.66	0.684	0.704
v_{26}	0.429	0.522	0.499	0.549	0.563	0.635
v_{27}	0.373	0.432	0.264	0.477	0.434	0.274
v_{28}	0.23	0.268	0.181	0.276	0.263	0.198
v_{29}	0.216	0.172	0.196	0.208	0.223	0.187
v_{30}	0.241	0.206	0.249	0.232	0.219	0.253
v_{31}	0.258	0.249	0.283	0.275	0.294	0.298
v_{32}	0.295	0.286	0.349	0.338	0.376	0.355
v_{33}	0.354	0.354	0.433	0.374	0.402	0.489
v_{34}	0.357	0.401	0.459	0.401	0.416	0.526
v_{35}	0.371	0.386	0.422	0.427	0.389	0.459
v_{36}	0.307	0.346	0.322	0.342	0.312	0.349
v_{37}	0.268	0.296	0.315	0.295	0.305	0.332
v_{38}	0.268	0.312	0.281	0.314	0.317	0.326

Table 3.14: Velocities of air(m/s) measured at the exit of the external hoods to compute $\dot{V}_3$ (II)

Spot	110Pa	120Pa	125Pa	130Pa	140Pa	150Pa
v_1	0.539	0.552	0.641	0.582	0.62	0.87
v_2	0.574	0.615	0.562	0.73	0.707	0.732
v_3	0.663	0.669	0.677	0.772	0.872	0.71
v_4	0.79	0.85	0.794	0.904	0.94	0.926
v_5	0.858	0.889	0.868	0.874	0.995	1.012
v_6	0.649	0.695	0.604	0.782	0.762	0.83
v_7	0.69	0.8	0.709	0.869	0.85	0.82
v_8	0.798	0.795	0.835	0.843	0.926	0.906
v_9	0.764	0.847	0.808	0.881	0.909	0.87
v_{10}	0.776	0.765	0.765	0.844	0.896	0.914
v_{11}	0.658	0.643	0.613	0.643	0.712	0.784
v_{12}	0.566	0.611	0.629	0.69	0.662	0.764
v_{13}	0.589	0.64	0.643	0.648	0.699	0.743
v_{14}	0.525	0.529	0.582	0.488	0.573	0.649
v_{15}	0.425	0.523	0.482	0.413	0.455	0.566
v_{16}	0.413	0.545	0.547	0.447	0.482	0.606
v_{17}	0.403	0.449	0.315	0.334	0.502	0.553
v_{18}	0.396	0.499	0.489	0.438	0.463	0.596
v_{19}	0.388	0.534	0.547	0.414	0.483	0.641
v_{20}	1.109	1.193	1.284	1.144	1.184	1.415
v_{21}	0.935	1.063	1.084	1.137	1.245	1.246
v_{22}	1.104	1.1823	1.208	1.255	1.343	1.311
v_{23}	1.032	1.104	1.141	1.145	1.279	1.245
v_{24}	1.006	0.952	0.855	1.126	1.182	1.065
v_{25}	0.776	0.694	0.675	0.749	1.006	0.703
v_{26}	0.611	0.463	0.622	0.617	0.731	0.617
v_{27}	0.482	0.447	0.424	0.613	0.64	0.334
v_{28}	0.313	0.277	0.278	0.332	0.344	0.261
v_{29}	0.242	0.218	0.219	0.244	0.262	0.249
v_{30}	0.224	0.262	0.272	0.284	0.281	0.264
v_{31}	0.294	0.339	0.374	0.315	0.306	0.359
v_{32}	0.392	0.378	0.371	0.414	0.41	0.454
v_{33}	0.449	0.468	0.428	0.496	0.547	0.522
v_{34}	0.444	0.516	0.419	0.561	0.602	0.525
v_{35}	0.467	0.458	0.411	0.549	0.611	0.483
v_{36}	0.372	0.386	0.355	0.402	0.486	0.437
v_{37}	0.337	0.347	0.289	0.364	0.417	0.357
v_{38}	0.343	0.291	0.283	0.366	0.393	0.33

Table 3.15: $\dot{V}_1$ and $\dot{V}_3$ under different working conditions

Pressure(Pa)	$\dot{V}_1(m^3/h)$	$\dot{V}_3(m^3/h)$
60	85.3	307.6
70	93.0	333.2
75	93.9	327.7
80	101.1	358.2
90	107.3	374.3
100	112.0	403.6
110	117.9	407.3
120	123.4	427.2
125	121.0	420.2
130	128.9	449.4
140	132.4	487.1
150	137.0	485.1

where $c = 38.8053$ and $d = 0.5033$.

The experimental data of $\dot{V}_1$ and $\dot{V}_3$ and their corresponding description functions are shown in Figure 3.9 and 3.10, respectively, which validate (3.1) and (3.2).

Figure 3.9: Data fitting results to validate the primary air volume flow rate model regarding on the primary pressure

Figure 3.10: Data fitting results to validate the mixed air volume flow rate model regarding on the primary pressure

According to (2.5) - (3.2), an empirical model that relates ER to pressure is obtained as:

$$ER = 3.6965P^{-0.0098} - 1 \quad (3.3)$$

It can be seen from (3.1) and (3.2) that both $\dot{V}_1$ and $\dot{V}_3$ are approximately proportional to the square root of primary pressure, making the ER of Type I terminal unit robust to the pressure fluctuation. Note that, when the terminal unit is working under $P = 100Pa$, the volume flow rate of primary air is $111.5m^3/h$ according to (3.1). At the same time, the ER is 2.5334 according to (3.3).

3.5 Conclusions

In this chapter, we have presented an experiment on terminal unit entrainment performance. An experiment plant is built to measure ER of a commercial terminal unit. The experiment plant shows the actual air flow and induction occurring in real life scenario. At the same time, to ensure the reliability of the experiment data, we have proposed a mixed air flow rate measurement method. Instead of measuring the low speed and sensitive room air, our proposed method makes an innovative progress to measure the mixed air, which is high speed and less susceptible. Its error is well kept within 5% and it is more accurate than the conventional method. With the proposed method, the mixed air flow rate can be measured more accurately, and thus the calculated ER can reflect the entrainment performance of terminal unit precisely. Two empirical models of primary air volume flow rate and mixed air volume flow rate to pressure are then established. It is worth to mention that the volume flow rate of primary air is $111.5m^3/h$ under normal working condition. An empirical model of ER to primary pressure is established based on the models of primary and mixed air volume flow rate. According to this model, we have calculated $ER=2.5334$, under normal working conditions, $P = 100Pa$.

Chapter 4

CFD Model of Type I Terminal Unit

4.1 Introduction

In Chapter 3, we have studied an off-shelf product. We aim to achieve an optimized terminal unit with high ER, by modifying Type I terminal unit. Nonetheless, it usually takes a lot of time to manufacture a new structure and the cost is significantly high to perform experiments on a large scale. With the development of computing techniques, the cost of computing becomes lower and massive numerical computation is realizable, making CFD techniques more and more popular. By solving the governing equations of the fluid, the CFD software can simulate the physical processes in complicated systems, such as fluid flow and heat transfer. The solution may contain the distribution of basic parameters, such as velocity, pressure and temperature, as well as their variance with time in a continuous domain. The flow of the fluid can be thus simulated within the software.

There are many commercial CFD software available, such as FLUENT, CFX, STAR-CD, PHOENICS. In our numerical study, we use ANSYS®Workbench™2.0 Frame-

work Version 14.5 and its built-in components CFX tools for entrainment simulation. In this chapter, we mainly simulate the fluid flow inside the terminal unit used in Chapter 3. By comparing the experiment data and simulation results, we can improve and validate the physical and mathematical model. Furthermore, with a valid model, we will analyse the factors that may affect ER.

This chapter is organized as follows. In Section 4.2, we introduce the loss coefficient used in CFD model and simplify the geometric model with loss coefficient. In Section 4.3, we present the mesh configuration and boundary conditions of the CFD model. Section 4.4 will present the simulation results and compare it with the experiment data set. Concluding remarks of this chapter are given Section 4.5.

4.2 Model simplification with loss coefficient

In our analysis, the model is supposed to be built as close as to the actual experiment set, including the external hood for measuring the mixed air. Note that the coils and fins with a perforated grille covering the coils and fins are installed at the opening where secondary air flows through. They will form resistance to the secondary air and its pressure will thus drop. As the coils and fins are made of very trivial structures and the perforated grille contains hundreds of holes, it is very time-consuming to model and compute the pressure drop across these trivial structures in every simulation. Therefore, we introduce loss coefficient to calculate pressure drop in an easier way.

According to [57], the pressure drop is calculated by

$$\Delta P_{loss} = \frac{1}{2} f \rho U_n^2 \quad (4.1)$$

where ρ is the air density, U_n is normal air speed and f is the specified loss coefficient. The loss coefficient is an inherent property of the coils and fins, and it is obtained from another simulation, as shown below.

The pressure drops as air flows through the perforated grille, coils and fins, which is simulated in a virtual duct. The geometry of the duct is shown in Figure 4.1.

Figure 4.1: Geometry of virtual duct with notation of perforated grill, cooling coil and fins

The size of the duct $304mm \times 304mm$, with its length of $300mm$. Inside the duct, the perforated grille, the cooling coils and fins are figured according to the experiment plant. Air flows from $-z$ to $+z$ (as notated in Figure 4.1). In each simulation, the speed of air flowing in is kept constant at $\widehat{U}_n$. Then the air goes through the grille, the coils and fins, respectively. The pressure at the inlet $\widehat{P}_{in}$ and the outlet $\widehat{P}_{out}$ can be easily obtained from simulation. Then the loss coefficient can be calculated as,

$$f = \frac{2(\widehat{P}_{in} - \widehat{P}_{out})}{\rho \widehat{U}_n^2} \quad (4.2)$$

Figure 4.2: Schematic of the virtual duct, with the emphasis on the pressure drop through it

According to the experiment results obtained in Chapter 3, the ER of Type I terminal unit is 2.5 and we wish to increase it further by 20% to 60%. When the primary air is supplied at $111.5 \text{ m}^3/\text{h}$, the minimum $\dot{V}_2$ should be approximately $200 \text{ m}^3/\text{h}$, and the maximum ER is around $440 \text{ m}^3/\text{h}$, which defines the range of $\widehat{U}_n$. $\widehat{U}_n$ can be calculated by dividing the volume flow rate of induced air by the effective area of the induced air inlet, which is 0.33 m^2 for the terminal unit used. 9 different possible values of $\widehat{U}_n$ are selected to study the resulting loss coefficients. The range of these values are selected large enough to cover our study, and the conclusion derived from the simulations is thus valid through our study. The average pressure at the inlet and outlet, $\widehat{P}_{in}$ and

$\widehat{P}_{out}$, are computed through simulations. With these data available, we can calculate loss coefficient with (4.2). The values of $\widehat{U}_n$ and the corresponding $\dot{V}_2$, $\widehat{P}_{in}$ and $\widehat{P}_{out}$ from simulations, together with calculated loss coefficient are shown in Table 4.1.

Table 4.1: Simulation results of loss coefficient with different air flowing speed

$\widehat{U}_n(m/s)$	Corresponding $\dot{V}_2(m^3/h)$	$\widehat{P}_{in}(Pa)$	$\widehat{P}_{out}(Pa)$	Loss Coefficient
0.2028	240.92	0.651337	0.0247072	25.7369
0.2278	270.62	0.79563	0.0353969	24.7469
0.2528	300.32	0.956313	0.0479865	24.0087
0.2778	330.02	1.13308	0.0625842	23.4315
0.3028	359.72	1.32405	0.0786335	22.9448
0.3278	389.42	1.52766	0.0955905	22.5127
0.3528	419.12	1.74455	0.113919	22.1299
0.3778	448.82	1.97435	0.133604	21.7847
0.4028	478.52	2.21683	0.154408	21.4724

The loss coefficients with different induced air flowing rates are shown in Figure 4.3, together with their average, 10% above the average and 10% below the average, shown with dotted line in the middle, at the top and at the bottom, respectively. The average of the nine points is 23.20. It can be seen from the figure that the loss coefficient decreases with the volume flow rate of induced air. That is, as the increase of ER, the loss coefficient will be smaller. However, the deviation of loss coefficient is not significant. Under normal condition when primary air flows in at $111.5m^3/h$ and $ER=2.53$, on one hand, the induced air flows in at $282.1m^3/h$ as in Type I terminal unit. The loss coefficient, in this case, will be smaller than 24.74. On the other hand, when induced air flows in at $478.52m^3/h$, the loss coefficient is still larger than 21, and ER, in this case, is 4.29, which is hardly achievable during current stage of study. In both extreme cases discussed above, the loss coefficient is well kept within $\pm 10\%$ of its average value. That is to say loss coefficient is not sensitive to the normal air speed in the range of interest in our study. As the loss coefficient becomes smaller as ER increases, we use 23 for the subsequent study.

Figure 4.3: Loss coefficients under different conditions

In this way, the calculation of the pressure drop across the coils and fins are simplified with loss coefficient, without which the calculation will be extremely time-wasting. In fact, each simulation we performed to find proper value for loss coefficient may cost half a day to give the converged result with high-performance computer. Not to mention more complicated, larger in size model of terminal unit. By introducing the loss coefficient, the pressure drop across the grille, coils and fins is only calculated once, instead of in every simulation. The time cost in calculation is significantly saved.

4.3 Numerical model of Type I terminal unit

The geometric model sketched for numerical calculation is shown in Figure 4.4 and 4.5. To keep consistency with the experiment plant, the external hoods are also included in the CFD model. The detailed dimensions of Type I terminal unit from

our measurement are listed in Table 4.2. As the terminal unit is axially symmetrical, only left side is noted.

Figure 4.4: Geometric model of Type I terminal unit with hood

Figure 4.5: The notation of Type I terminal unit to illustrate its detailed dimensions

The induced air inlet is simplified with loss coefficient method mentioned in Section 4.2. The computation mesh, consisting of around 66k nodes and 300k elements, is shown in Figure 4.6. The mesh configuration is listed in Table 4.3. The basic

Table 4.2: Dimensions of Type I terminal unit model

Lines and Angles	Measurement
AB	$60mm$
BC	$172.44mm$
CD	$43.32mm$
DE	$201.5mm$
AF	$80mm$
FG	$30mm$
GH	$61.69mm$
HI	$20mm$
FM	$80mm$
MN	$600mm$
AP	$520mm$
CJ	$174.6mm$
CK	$114.79mm$
KL	$276.49mm$
$\angle BAP$	155°
$\angle ABC$	145.06°
$\angle BCD$	90°
$\angle CDE$	150°
$\angle FGH$	136.7°
$\angle AFG$	150°

governing equations, namely the Navier-Stokes equations and the energy equations are solved numerically with the finite volume techniques. The turbulence of the flow is modelled with the standard k- ϵ model.

Figure 4.6: The generated mesh of Type I terminal unit with hood

As different boundary conditions may give different simulation results and lead to biased conclusions, the boundary conditions are one of the key factors to determine if a CFD model is valid. Through extensive studies, we find that the set of boundary conditions shown in Table 4.4 gives the best fit of the experiment results.

In CFD, the boundary conditions are given to the governing equations and they will be iteratively solved. For each simulation, the solutions are iterated until the RMS residual is smaller than 10^{-5} . A simulation report showing all the detailed mesh and boundary conditions, as well as the simulation results, is attached in Appendix A.

Table 4.3: Configuration of mesh

Defaults	
Physics Preference	CFD
Solver Preference	CFX
Relevance	0
Sizing	
Use Advanced Size Function	On:Curvature
Relevance Center	Fine
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angel Center	Medium
Curvature Normal Angel	Default(45.0°)
Min Size	Default($1.9488e - 004m$)
Max Face Size	Default($1.9488e - 002m$)
Max Size	Default($3.8975e - 002m$)
Growth Rate	Default(1.850)
Minimum Edge Length	$5.e - 004m$
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.77
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Advanced	
Shape Checking	CFD
Element Mindside Nodes	Dropped
Straight Sided Elements	
Number of Retries	0
Extra Retries for Assembly	Yes
Rigid Body Behavior	Dimensionally Reduced
Mesh Morphing	Disabled
Defeaturing	
Pinch Tolerance	Default($1.7539e - 004m$)
Generate Pinch On Refresh	No
Automatic Mesh Based Defeaturing	On
Defeaturing Tolerance	Default($9.7438e - 005m$)

Table 4.4: Configuration of boundary conditions

Boundary Name	Type	Boundary Details
Primary Air Inlet	Inlet	Mass And Momentum: Total Pressure, 100[Pa] Flow Direction: Normal to Boundary Condition Turbulence: Zero Gradient
Room Air Inlet	Opening	Mass And Momentum: Entrainment, 0[Pa] Pressure Option: Static Pressure Loss Coefficient: 23 Turbulence: Zero Gradient
Mixed Air Outlet	Opening	Mass And Momentum: Opening Pres. and Dirn, 0[Pa] Flow Direction: Normal to Boundary Condition Turbulence: Zero Gradient

The rest boundaries are no slip wall with default settings

4.4 Simulation results

Figure 4.7 and 4.8 show the air flow pattern inside the terminal unit. It is clear that the air is supplied at a relatively high speed. Then the air circulates inside the Primary chamber, forming a uniform distribution. The primary air is ejected through nozzles at high speed, which is also the force that induces the room air. Figure 4.8 highlights the air induced. It mixes with primary air near the nozzles. Note that in the center (marked with rectangle), the streamlines are sparse, while the streamlines are intensive at the margins (marked with ellipses).

Figure 4.9 shows the pressure distribution inside the terminal unit. The pressure inside the primary chamber is uniformly around $100Pa$. The lowest pressure inside the mixing chamber is $-12.37Pa$, which is formed by the ejection of primary air.

The ERs of Type I structure, calculated respectively with the empirical model and CFD, are presented in Table 4.5. The pressure ranges from $60Pa$ to $200Pa$, and its corresponding ER reminds stable around 2.5. The error between ER from CFD and that from the empirical model is very small.

Figure 4.7: The streamlines of Primary and induced room air flow inside Type I terminal unit

Figure 4.8: The streamlines of primary and induced room air flow inside Type I terminal unit - cross-sectional view

Figure 4.9: Pressure distribution inside terminal unit

Table 4.5: ER calculated with empirical model and CFD under different pressures

Pressure(Pa)	ER from CFD	ER from Model	error
60	2.555	2.551	0.15%
70	2.546	2.546	0.01%
75	2.548	2.543	0.18%
80	2.54	2.541	-0.04%
90	2.542	2.537	0.20%
100	2.557	2.533	0.93%
110	2.546	2.530	0.63%
120	2.547	2.527	0.79%
125	2.554	2.526	1.12%
130	2.549	2.524	0.98%
140	2.572	2.522	1.99%
150	2.574	2.519	2.17%
160	2.574	2.517	2.26%
170	2.554	2.515	1.55%
180	2.556	2.513	1.71%
190	2.556	2.511	1.78%
200	2.557	2.509	1.89%

The ER from CFD and that from the empirical model are shown in the Figure 4.10 for comparison. It shows that the CFD model agrees well with the experimental data, with a little deviation when $P > 150Pa$. The figure shows that the deviation is well kept within $\pm 5\%$ and thus can be neglected. The mesh and boundaries conditions used in our research are thus valid. Therefore, the CFD model for Type I terminal unit is valid, and the same mesh generation and boundary conditions can be used for further research.

Figure 4.10: Comparison of ERs obtained with different methods

4.5 Conclusion

In this chapter, a geometric model for CFD analysis is established. The model is based on the Type I terminal unit used in Chapter 3. The geometric dimensions are the same as the actual experiment plant. To speed up the computations, we simplified

the geometric model by introducing loss coefficient. The value of loss coefficient is determined accordingly. The mesh and boundary conditions of the geometric model are duly evaluated to ensure the validity of CFD model. The streamlines and the pressure inside the terminal unit are also examined to match with the experiment set. Comparing the CFD result and the measurement taken in Chapter 3, we can say that the CFD model accords with the experiment plant well, which proves the validity of our CFD model.

Chapter 5

Type II Structure: Modified for Higher ER

5.1 Introduction

In our research, we have studied most of the commercial terminal units, including some widely-used mature products. The ER of most products are around 2. Few of them are above 3. Some may have superior ER up to 4 to 5, but with the cost of less primary air. We also note that most of the products arrange nozzles close to the margin of the primary chamber, instead of close to the center, and primary air is ejected vertically or almost vertically[40]. Our Type I terminal unit is one of them. The nozzles of Type I terminal unit are arranged far from the center and the angle of ejection is about 30° from the vertical direction. In such way of arrangement, the length of the vertically ejected jetflow is limited by the height of the terminal unit. The induction process has not fully finished when the air is discharged. The mixed air still has the kinetic energy to induce the room air. However, as the ejected primary air is discharged to the conditioned room, there is no more induction. In the meanwhile, it is not feasible to increase the height of terminal unit, as the ceiling space is limited. Therefore, the height of terminal unit and the way of installation of

nozzle make the primary air discharged to the conditioned room too early before it fully induces the room air.

Based on this concern, we creatively move the nozzles to the center of the mixing chamber, with the nozzle installed horizontally. In this way, the length of the jetflow is no longer limited by the height of terminal unit. It fully uses the space at the center of the terminal unit, leading to a longer jetflow of induction. More room air is induced and thus higher ER can be achieved. In the meanwhile, the side walls are shaped into circular arcs, which can change the direction of air flow without losing much kinetic energy of mixed air. The thermal comfort of the conditioned room is also taken into consideration. At the end of the side walls, where they contact to the conditioned room ceilings, a small angle is kept with the ceilings, so the discharged air may attach to the ceiling under Coanda effect.

This chapter is organized as follows. In Section 5.3, we highlight the modifications made to Type II terminal unit and explain the motivation of these modifications. Section 5.4 will present our optimization on Type II terminal unit in several aspects including the detailed dimensions of mixing chamber wall, the distance between nozzles and the length of nozzles. Concluding remarks of this chapter are given Section 5.5.

5.2 The validity of CFD model

In Chapter 4, we have built a CFD model of Type I terminal unit, which is proved valid with the experiment results obtained in Chapter 3. The entrainment performance of Type I terminal unit is enhanced by the modifications in its geometry, which is

also carried with simulations. Note that only geometry of Type I terminal unit is changed, while the key factors to determine the validity of a model, such as the mesh configurations, the boundary conditions, the turbulence model, etc. are still kept the same. Since the geometry is not fundamentally changed but only a little, the air flow pattern will also change little. Therefore, the validity of the CFD model is not affected.

5.3 Motivation of modifications

To enhance the ER, based on the obtained Type I model, we make several changes on the mixing chamber and the nozzles. Through extensive investigations, we propose a modified terminal unit. For the ease of notation, we hereby name it Type II terminal unit. Its length and width are kept the same as those of Type I terminal unit. Figure 5.1 shows a 3-dimensional view of Type II terminal unit for a comprehensive understanding of it. Note that the coordinate axis at the bottom right may be referred for ease of explanation. Figure 5.2 shows its cross sectional view in Z-direction, highlighting the mixing chamber wall and nozzle arrangement. Note that the mixing chamber and the nozzles shown in this figure are not optimized, and their optimization will be shown in following sections.

The changes of the mixing chamber are focused on the following four aspects for reasons given in the subsequent paragraphs, while the rest of the terminal unit is kept the same for the fairness of comparison.

1. The nozzles are moved closer to the center axis of mixing chamber and the direction of ejection is changed to horizontal. The bottom of primary chamber

Figure 5.1: Geometric model of Type II structure - 3-Dimensional view

Figure 5.2: Geometric model of Type II structure - cross sectional view

changes accordingly.

2. The height of mixing chamber is reduced by $56mm$.
3. The wall of mixing chamber uses more smooth curves.
4. The nozzles are longer than the ones used in Type I terminal unit.

5.3.1 Nozzle arrangement

In the perspective of energy conservation, the induction of primary air on induced air is an exchange of kinetic energy between primary air and room air. In Type I structure, limited by the height of the mixing chamber, the path where ejected air travels in the mixing chamber is short. When discharged from terminal unit, the mixed air is still at a relatively high speed. It has not fully induced the room air. Most of its kinetic energy is wasted by early discharging to the room. Due to this inherent drawback of Type I terminal unit, it cannot achieve a high ER, especially when the pressure goes up, as more kinetic energy is wasted. Therefore, by moving the nozzles close to the center of the mixing chamber and ejecting horizontally, the path where air travels from the nozzle exit to the terminal unit outlet is lengthened. The ejected primary air and the induced room air mix better. In this case, the mixed air will not be discharged at a high speed. Instead, it will induce more room air up and be discharged to the conditioned room.

5.3.2 Height of the mixing chamber

Compared with conventional VAV system, the ACB system is shorter in height and has the advantage of occupying less space above the suspending ceiling. Type II

structure does better at this point. In Type I structure, the primary air is ejected almost vertically. The mixing chamber has to be high enough to ensure the air mixing. However, the primary air is ejected horizontally in Type II structure, which allows the height of the mixing chamber to be further diminished, as long as it is not too narrow for air to flow. In our case, The total height of the terminal unit is reduced by $56mm$, which is 19.38% decrements. Moreover, as we only diminish the mixing chamber, the height of the overall terminal unit can be further diminished by using lower primary chamber. However, to highlight the effectiveness of our modification on the mixing chamber wall, we give up the diminishment of the primary chamber. It will not affect the ER much if the height of primary chamber is further diminished in Y-direction. This modification enables Type II structure to fit more tight space without sacrificing its entrainment ability.

5.3.3 Mixing chamber walls

The mixing chamber walls are designed to enhance the room air entrainment and change the direction of airflow. The wall of mixing chamber is modelled with smooth curve instead of straight and cornered lines, to eliminate unnecessary kinetic energy loss while changing the direction of jet flow. Specifically, the smooth wall is made up of two circular arcs connected together, making an S-shape. The circular arcs can achieve higher entrainment performance with relatively lower cost, as the circular arcs are easier to fabricate than other curvature shapes. The walls near terminal unit outlets are also designed to align with the ceiling. The angle it forms with the ceiling is around 20° . The discharged mixed air can well attach to the ceiling under Coanda effect.

5.3.4 Length of nozzles

It is obvious that the nozzles used in Type I terminal unit are very short. It is also common in most of the commercial products. As their primary air is vertically ejected, the height of mixing chamber has limited the use of longer nozzles. It is proved and explained in our following study that the length of the nozzle has great influence on the entrainment performance of terminal unit. Therefore, by using the longer nozzles, our proposed Type II terminal unit can improve the entrainment performance of ACB system.

5.4 Optimization of Type II terminal unit

Although we have changed the arrangement of the nozzles and reshaped the mixing chamber wall, the design of Type II terminal unit is not finalized. The modifications made are mostly qualitative. The detailed dimensions are selected empirically based on our extensive research. We believe through systematic optimization, the ER can be further increased. In this section, we mainly discuss the optimization of the modified terminal unit.

Prior to further study, the model used for analysis is further simplified. In CFD analysis, it usually costs hours or even days to perform a detailed simulation, which is unfavorable to massive computations. As much more simulations are needed during the optimization process. The time cost becomes a major concern. In order to shorten the computation time, the geometric model used in optimization is further simplified based on its symmetry. As terminal unit contains a series of identical nozzles, it can be divided into many repeating units shown in Figure 5.3. Therefore, we isolate this

unit and take this single unit as a representative of the entrainment performance of the whole terminal unit set. We omit the influence made by the margins of terminal unit as its effects are limited. Besides, the primary chamber is further simplified to an air inlet duct, as it does not affect ER much and is not our main focus of study. Despite of the simplifications and assumptions we made, it is still proven by simulations that the entrainment performance of this simplified single unit is similar to the Type II terminal unit. By simplifying this single unit, the simulation runs much faster and we can quickly analyze factors that affect entrainment performance.

Figure 5.3: Geometry of the simplified terminal unit model used for optimization analysis

The mesh configurations are the same to the ones we use in Chapter 4. The mesh generated is shown in Figure 5.4. The boundary conditions are also the same except that the wall noted with red arrows are set to be "Symmetry" and the face noted with black arrows are configured as the supply of primary air, as shown in Figure 5.5.

By setting the corresponding walls "Symmetry", the resulting model can be taken as a terminal unit with infinite length.

Figure 5.4: Generated mesh of the simplified terminal unit model used for optimization analysis

The optimization of Type II terminal unit contains three parts: the optimization of the dimensions of mixing chamber, including several lengths of lines and degrees of arcs, are presented in Section 5.4.1, the optimization about the nozzle placement and the diameter of nozzles are presented in Section 5.4.2, and the length of nozzles that is suitable for Type II terminal unit is presented in Section 5.4.3.

Figure 5.5: Boundary conditions of the simplified terminal unit model used for optimization analysis

5.4.1 Dimensions of mixing chamber wall

The dimensions of the Type II terminal unit, such as the angles of the arcs, the length of certain walls, are mostly empirical. Their dimensions are proposed based on our experience of extensive study of commercial terminal units. It is also a fact that some of the dimension may have a significant influence on the ER, while others have little. It is not worthy to optimize every dimension of the terminal unit. Therefore, we firstly perform parameter correlation analysis on the dimensions that we determined empirically. The ones with more significant influence on entrainment performance have been chosen to be optimized. We selected the values of dimensions randomly in a certain range, and the sensitive analysis is performed with ANSYS Workbench built-in tools. The dimensions we analysed and their notation are shown in Figure 5.6 and Table 5.1.

Figure 5.6: The notation of Type II terminal unit to illustrate the detailed dimensions

Table 5.1: Range of dimension used by Type II terminal unit

Point	Type	Value Range
AB	Chamber Wall, Straight line	$0mm$ to $24.2mm$
$\angle BAN$	Angle with Ceiling	0° to 50°
$\angle CKD$	Angle of Arc	52.65° to 64.35°
LK	Auxiliary Line	$27.9mm$ to $34.1mm$
AL	Auxiliary Line	$140mm$ to $180mm$
EF	Distance of Nozzles Deviating from Center	$30mm$ to $50mm$
EG	Distance of Nozzles to the Chamber Wall	$22mm$ to $35mm$

Parameter analysis

We use the Parameter Correlation toolbox to analyse the correlation between each parameter and ER. The build-in toolbox is able to compute a correlation matrix, which is used to investigate the dependence between multiple dimensions with ER at the same time. Firstly, Random values within the range mentioned in Table 5.1 are selected to configure the geometric model. They are considered as the inputs to the geometric model. Secondly, simulations are run to obtain corresponding ER and primary air flow rate. Note that for ease of comparison, the primary air flow from a single unit is adjusted by multiplying the number of nozzles, which is equivalent to the primary volume flow rate of the whole terminal unit set. It is better to keep adjusted $\dot{V}_1$ around $111.5m^3/h$, which is the normal primary air volume flow rate under normal working condition according to (3.1). The simulation is executed 90 times with different parameters, which is sufficient for our analysis. The value set for all the parameters and the corresponding simulation results are shown in Table 5.2 to Table 5.5. Lastly, we can obtain a correlation matrix with the toolbox based on the simulation results. The correlation matrix shows how well each parameter affects ER.

Table 5.2: Parameter analysis of dimensions of Type II terminal unit (I)

Name	$\angle BAN(^{\circ})$	$AB(mm)$	$AL(mm)$	$EG(mm)$	$LK(mm)$	$\angle CKD(^{\circ})$	$EF(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
1	36.06	2.39	157.79	31.13	31.03	63.30	31.07	114.24	2.89
2	44.47	11.28	153.88	29.91	30.73	60.29	46.09	110.79	2.77
3	40.14	4.23	171.63	25.48	32.60	53.13	35.33	114.87	2.54
4	43.41	7.98	160.38	27.09	32.00	61.95	48.28	112.67	2.51
5	22.73	5.52	145.17	32.68	29.23	54.63	45.03	115.48	2.76
6	24.45	18.02	167.61	28.38	28.49	62.59	37.58	109.14	3.14
7	32.20	14.01	177.15	23.65	28.89	56.64	43.89	113.58	2.60
8	47.13	22.32	148.48	28.89	29.92	55.62	32.06	114.77	2.84
9	28.43	14.77	140.52	22.64	33.29	58.56	39.17	111.62	2.67
10	31.03	19.54	174.10	34.46	33.62	58.23	41.77	114.52	2.95
11	36.28	14.43	169.71	29.21	33.50	64.12	31.05	114.82	2.74
12	25.05	3.65	179.24	24.36	31.46	54.64	40.65	112.66	2.96
13	22.61	6.05	149.75	33.81	29.29	60.37	34.71	113.66	3.29
14	47.00	19.89	173.41	28.43	28.17	56.99	33.52	114.11	2.56
15	42.80	19.09	143.11	26.38	32.90	53.55	36.09	113.04	2.73
16	47.38	2.40	161.87	32.17	32.17	59.06	45.87	111.26	2.67
17	32.63	10.51	154.14	33.51	30.38	55.19	43.97	114.97	3.02
18	38.54	15.28	157.10	25.27	29.10	63.13	47.42	110.42	2.55
19	31.26	8.63	147.81	22.53	30.98	61.18	38.88	109.79	2.49
20	28.74	22.41	167.19	30.09	32.55	57.83	49.03	112.07	3.15
21	35.40	22.76	143.44	27.47	31.22	55.43	40.89	112.80	3.04
22	21.48	1.36	162.65	30.79	32.46	63.00	38.29	109.87	3.25
23	43.02	10.81	166.35	23.56	33.30	59.28	48.12	110.35	2.29
24	46.39	4.23	159.48	33.78	31.94	53.22	36.60	110.92	2.88
25	27.86	12.75	177.13	29.17	28.36	54.22	46.49	111.61	2.94

Table 5.3: Parameter analysis of dimensions of Type II terminal unit (II)

Name	$\angle BAN(^{\circ})$	$AB(mm)$	$AL(mm)$	$EG(mm)$	$LK(mm)$	$\angle CKD(^{\circ})$	$EF(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
26	25.35	16.00	147.24	33.49	30.04	60.30	45.61	114.10	3.19
27	33.23	5.55	149.11	22.11	28.95	56.83	33.94	114.82	2.46
28	38.17	7.75	153.50	25.50	30.88	61.81	42.64	114.44	2.51
29	47.69	18.05	169.82	31.73	29.57	63.56	34.46	113.91	2.54
30	30.11	20.94	172.30	26.48	33.68	57.94	30.04	112.06	3.00
31	45.67	8.22	166.01	30.96	28.01	58.39	47.72	109.63	2.67
32	41.70	3.95	141.87	29.71	30.81	58.92	35.40	114.80	2.87
33	24.79	18.45	154.75	22.47	32.19	53.12	45.19	115.15	2.89
34	20.79	0.54	161.44	26.21	29.73	60.73	39.34	110.79	2.99
35	48.92	6.70	169.01	28.03	33.75	54.21	42.17	113.81	2.51
36	39.04	22.12	148.33	24.35	29.96	62.45	40.16	111.76	2.53
37	36.32	10.11	178.22	24.85	31.51	61.51	33.19	111.90	2.40
38	29.89	20.76	173.01	33.86	29.06	55.98	36.09	113.09	3.08
39	34.08	15.43	146.50	33.05	32.50	57.15	30.49	115.26	3.36
40	27.32	12.95	156.19	31.20	33.29	64.04	48.19	109.94	2.99
41	30.89	15.15	145.88	24.30	31.78	53.39	36.19	113.20	2.91
42	21.35	9.00	166.32	24.85	33.52	57.78	43.37	115.26	2.98
43	41.08	18.88	163.93	26.53	29.74	55.94	30.72	111.69	2.72
44	40.16	6.45	172.26	22.16	28.07	60.02	44.85	114.14	2.14
45	27.73	14.21	157.60	27.24	30.15	61.92	40.07	112.08	2.87
46	34.58	10.16	148.71	34.92	28.66	54.05	49.13	111.82	3.06
47	45.95	4.65	143.25	28.61	32.75	63.08	39.09	109.67	2.73
48	49.78	23.47	169.37	30.20	33.44	59.36	46.43	110.39	2.50
49	38.06	0.01	178.19	32.54	31.35	57.13	33.59	113.49	2.63
50	23.24	20.75	154.43	31.72	30.61	64.26	34.31	115.22	3.32

Table 5.4: Parameter analysis of dimensions of Type II terminal unit (III)

Name	$\angle BAN(^{\circ})$	$AB(mm)$	$AL(mm)$	$EG(mm)$	$LK(mm)$	$\angle CKD(^{\circ})$	$EF(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
51	43.21	16.33	149.50	34.69	31.46	60.71	38.95	110.70	3.00
52	46.42	0.94	170.93	31.74	28.67	57.87	36.50	110.71	2.64
53	34.94	11.43	178.84	26.27	29.88	54.96	48.81	113.48	2.65
54	31.80	18.08	152.98	25.29	28.15	55.77	31.89	112.53	2.86
55	39.15	5.82	156.33	22.25	33.69	59.48	33.48	114.94	2.31
56	25.67	13.82	146.72	27.40	29.49	61.87	47.97	110.79	3.12
57	21.00	4.81	165.59	29.11	32.14	63.39	41.65	113.94	3.13
58	36.04	8.14	140.24	30.85	32.58	53.21	45.10	114.76	3.25
59	27.40	20.45	175.08	32.84	33.24	56.88	35.37	114.03	3.19
60	49.24	24.02	163.43	24.10	30.72	62.50	43.10	114.80	2.13
61	22.74	22.88	162.88	30.35	28.52	58.45	32.47	115.34	3.28
62	24.55	8.58	173.68	31.76	30.79	61.41	47.74	113.72	2.98
63	43.36	17.79	177.65	23.85	31.68	62.81	41.93	113.08	2.19
64	32.02	5.59	147.94	25.41	31.27	57.28	48.29	111.67	2.75
65	33.08	4.01	155.63	28.02	28.34	59.73	37.89	109.73	2.96
66	47.56	12.78	167.17	34.55	30.24	52.83	44.90	114.89	2.63
67	39.80	10.58	140.72	33.08	33.19	63.54	35.70	114.28	3.00
68	35.55	2.23	169.37	26.24	32.66	54.58	31.32	110.57	2.83
69	44.87	15.63	150.96	22.54	29.57	59.19	38.94	109.93	2.31
70	26.45	20.41	157.20	29.03	33.94	55.64	43.32	114.09	3.27
71	45.00	2.59	159.16	26.14	28.58	55.69	32.14	112.85	2.57
72	20.69	11.25	148.91	29.12	30.65	60.20	30.05	113.38	3.34
73	48.31	13.56	147.95	25.42	33.93	61.61	38.87	110.32	2.46
74	26.54	17.27	166.28	32.73	33.20	59.22	34.54	114.34	3.26
75	41.88	9.19	174.34	33.98	29.72	63.78	42.86	113.05	2.51

Table 5.5: Parameter analysis of dimensions of Type II terminal unit (IV)

Name	$\angle BAN(^{\circ})$	$AB(mm)$	$AL(mm)$	$EG(mm)$	$LK(mm)$	$\angle CKD(^{\circ})$	$EF(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
76	31.15	1.66	153.05	32.20	32.01	54.60	45.57	114.22	3.13
77	37.51	14.60	168.16	27.40	32.82	53.20	40.37	111.72	2.87
78	38.29	19.44	176.72	24.40	30.07	57.85	37.89	109.37	2.47
79	25.12	5.94	161.79	22.70	31.23	62.02	49.06	110.39	2.46
80	33.58	23.97	141.50	30.74	28.36	56.18	46.23	111.13	3.33
81	25.82	11.16	148.75	26.74	29.25	63.75	46.05	113.61	2.89
82	40.46	22.83	143.72	23.36	32.39	57.18	42.14	113.46	2.46
83	49.72	18.14	166.75	34.53	30.25	62.83	44.75	112.68	2.52
84	44.19	1.44	153.57	24.82	28.63	53.06	41.81	112.21	2.52
85	43.33	8.42	170.81	27.80	32.96	59.44	38.40	109.39	2.56
86	21.98	5.18	173.16	28.86	31.70	55.03	49.61	111.97	3.08
87	30.21	3.47	147.70	29.87	31.09	61.15	30.69	110.88	3.15
88	35.35	16.70	177.27	22.52	30.42	60.58	32.01	112.21	2.32
89	33.53	14.21	156.16	32.45	33.70	58.39	36.89	110.27	3.03
90	26.68	21.73	163.95	31.98	28.29	54.38	34.44	114.74	3.23

It can be seen that the maximum ER obtained in the set of simulations is 3.36, and the smallest is 2.13. It is obvious that one or more parameters have great influence on the ER. It is worthwhile to optimize the dimensions of the mixing chamber wall. However, only from the simulation results, we can not figure out which parameter can affect the ER most. Therefore, the following correlation analysis is performed. The correlation matrix is shown in Table 5.6.

Table 5.6: Correlation matrix of dimensional parameters of Type II terminal unit

	$\angle BAN$	AB	AL	EG	LK	$\angle CKD$	EF	V_1	ER
$\angle BAN$	1.0000	0.0248	0.0478	-0.0400	0.0276	-0.0299	-0.0359	-0.1469	-0.6609
AB	0.0248	1.0000	-0.0025	0.0021	-0.0123	0.0089	-0.0473	0.1169	0.0936
AL	0.0478	-0.0025	1.0000	-0.0037	-0.0123	-0.0184	-0.0162	-0.0279	-0.2437
EG	-0.0400	0.0021	-0.0037	1.0000	-0.0131	0.0062	0.0098	0.1536	0.5744
LK	0.0276	-0.0123	-0.0123	-0.0131	1.0000	0.0203	-0.0224	0.0284	0.0266
$\angle CKD$	-0.0299	0.0089	-0.0184	0.0062	0.0203	1.0000	-0.0230	-0.2013	-0.1221
EF	-0.0359	-0.0473	-0.0162	0.0098	-0.0224	-0.0230	1.0000	-0.1832	-0.0815
V_1	-0.1469	0.1169	-0.0279	0.1536	0.0284	-0.2013	-0.1832	1.0000	0.1630
ER	-0.6609	0.0936	-0.2437	0.5744	0.0266	-0.1221	-0.0815	0.1630	1.0000

A graphic presentation of correlation matrix and sensitive histogram based on the correlations matrix is shown in Figure 5.7 and 5.8. It can be observed that some of the parameters affect the ER remarkably while some are not. Specifically, $\angle BAN$, EG and AL have significant influence on ER. $\angle BAN$ and AL are inversely correlated to ER and EG has a positive influence on ER. Their influence on $\dot{V}_1$ can be neglected. In the following simulations, the only thing that we need to do about $\dot{V}_1$ is to constantly check its value: as long as it does not deviate from $111.5m^3/h$, the result we obtained can be considered as valid. The rest of our optimization on dimensions is to investigate their detailed influence on ER. To avert the noise of other factors, we will remain the rest parameters fixed, and analyse the corresponding factors separately.

Figure 5.7: Correlation matrix of dimensional parameters of Type II terminal unit, ER and $\dot{V}_1$

Optimization on $\angle BAN$

Since we have figured that the following three parameters, $\angle BAN$, EG and AL , influence the entrainment performance of Type II terminal unit most, we will investigate

Figure 5.8: Sensitive analysis showing the correlation of dimensional parameters of Type II terminal unit, ER and $\dot{V}_1$

them in details. we wish to find out their relationship to ER with regression method. The angle of terminal unit with ceiling has the strongest correlation with ER among all the parameters of our interest. The correlation of $\angle BAN$ with ER is -0.6609 , whose absolute value is the largest in all the parameters. The value of $\angle BAN$ is set to range from 18.45° to 34.60° . The other parameters are fixed at the values shown in Table 5.7.

Table 5.7: Dimensions set for optimization of $\angle BAN$

Name	Value
$\angle CKD$	58.48°
$\angle BAN$	18.45° to 34.60°
AB	$14.33mm$
AL	$156.26mm$
EF	$39.22mm$
EG	$31.16mm$
LK	$31.16mm$

The results of the simulation are listed in Table. 5.8. Note that the volume flow rate

obtained from simulation is the primary air ejected from one nozzle. The primary air volume flow rate of overall terminal unit is the number obtained from simulation multiplied by the number of nozzles. Although the average value of $\dot{V}_1$ is slightly smaller than $111.5\text{m}^3/\text{h}$, the error is small enough to say the simulation is valid. We can see a downward trend of ER as $\angle BAN$ gets bigger. The maximum ER is about 3.4 and the minimum ER is about 3.0, with $\angle BAN$ ranges from 18.45° to 34.60° . Figure 5.9 shows the comparison of geometries of terminal unit when $\angle BAN = 18.45^\circ$

Table 5.8: Results of simulations about optimization of $\angle BAN$

$\angle BAN(^{\circ})$	Adjusted $\dot{V}_1(\text{m}^3/\text{h})$	ER
18.45	110.54	3.1044
19.60	109.73	3.1379
20.76	110.23	3.2192
21.91	110.09	3.2244
23.06	110.83	3.2344
24.22	111.15	3.2624
25.37	110.86	3.2323
26.52	111.31	3.2252
27.68	110.99	3.2308
28.83	110.30	3.2108
29.98	110.19	3.1633
31.14	110.19	3.1286
32.29	110.91	3.1005
33.44	111.70	3.0359
34.60	110.80	2.9772

and $\angle BAN = 34.60^\circ$, respectively. It can be seen that as $\angle BAN$ gets bigger, the space of mixing chamber is bigger.

A quadratic regression is performed and the regression model of $\angle BAN$ and ER is

$$ER_{\angle BAN} = -0.0034\theta_{\angle BAN}^2 + 0.1689\theta_{\angle BAN} + 1.1450 \quad (5.1)$$

where $\theta_{\angle BAN}$ ranges from 18.45° to 34.60° . Figure 5.10 shows the simulation results

Figure 5.9: Comparison of different $\angle BAN$

together with the regression model. It shows on the figure that the model fits the simulation results well and the maximum ER of 3.24 can be achieved when $\angle BAN = 24.83^\circ$ according to the model.

Optimization on EG

The distance of the nozzle to the mixing chamber wall is another design point to be optimized, with correlation of 0.5744 with ER. This distance makes difference on the property of the jet: if the distance is smaller, the primary air ejected more close to the mixing chamber wall and it is a plane wall jet. While the distance gets larger, the primary air is ejected far from the mixing chamber wall and it is more close to a free jet. The value of EG is set to range from 24.928mm to 37.392mm . The rest of the parameters are shown in Table 5.9.

Figure 5.10: Data fitting results to validate the model of $\angle BAN$ and ER .Table 5.9: Dimensions set for optimization of EG

Name	Value
$\angle CKD$	58.48°
$\angle BAN$	28.83°
AB	$14.33mm$
AL	$156.26mm$
EF	$39.22mm$
EG	$31.16mm \pm 20\%$
LK	$31.16mm$

The results of the simulation are presented in Table 5.10. It can be seen from the table that in the prescribed range, $\dot{V}_1$ is well kept around $111.5m^3/h$. The ER increases from 2.79 to a maximum of 3.27 and then decrease to 3.0, as EG increases from 24.928 to 37.392.

Table 5.10: Results of simulations about optimization of EG

$EG(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
24.928	110.94	2.79041
26.1744	109.90	2.94905
27.4208	109.94	3.03333
28.6672	112.05	3.11032
29.9136	109.92	3.23095
31.16	110.06	3.27192
32.4064	110.96	3.2261
33.6528	109.41	3.22293
34.8992	110.62	3.1457
36.1456	109.58	3.08704
37.392	111.75	2.99936

The terminal units with the minimum EG , the maximum EG , and the one giving biggest ER are superposed into Figure 5.11. If the nozzle is placed too close to the chamber wall, the part of room air that mixes with the primary air at the space between the primary air and the chamber wall is little. If the nozzles are placed too far from the chamber wall, they are placed too low and the board NRST(as in Figure 5.6) will hinder the flow of mixed air.

A quadratic regression is also performed and the regression model of EG and ER is

$$ER_{EG} = -0.009L_{EG}^2 + 0.5743L_{EG} - 5.9568 \quad (5.2)$$

where L_{EG} ranges from $24.928mm$ to $37.392mm$. From the model, we can have the maximum ER of 3.2 when $L_{EG} = 31.9mm$.

Figure 5.11: Comparison of different EG Figure 5.12: Data fitting results to validate the model of EG and ER

Optimization on AL

Although AL has the smallest correlation with ER in the three parameters we optimized, it is still significant to the entrainment performance of terminal unit. AL is an auxiliary line so that $AL \perp DL$. L is thus the horizontal position where the arc CD joint with segment DE . It can also be taken as where the curvature starts in the direction of primary air flows. The value of AL is set to range from $125.008mm$ to $187.512mm$. The rest of the parameters are shown in Table 5.11.

Table 5.11: Dimensions set for optimization of AL

Name	Value
$\angle CKD$	58.48°
$\angle BAN$	28.83°
AB	$14.33mm$
AL	$156.26mm \pm 20\%$
EF	$39.22mm$
EG	$31.16mm$
LK	$31.16mm$

The results of the simulation are presented in Table 5.12. It can be seen from the table that in the prescribed range, $\dot{V}_1$ is well kept around $111.5m^3/h$. The ER increases from 2.66 to a maximum of 3.33 and then decrease to 2.83, as AL increases from 125.008 to 187.512.

The terminal units with the minimum AL , the maximum AL , and the one giving biggest ER are superposed into Figure 5.13. It can be seen that as AL gets bigger, the mixing chamber get higher, with the width of the top of mixing chamber smaller. Neither too high nor too low can give high ER. The best AL is around $143.75mm$

The quadratic regression performed on AL and ER shows that

$$ER_{AL} = -0.0004323L_{AL}^2 + 0.1327L_{AL} - 6.9659 \quad (5.3)$$

Table 5.12: Results of simulations about optimization of AL

$AL(mm)$	Adjusted $\dot{V}_1(m^3/h)$	ER
125.008	110.83	2.66243
131.2584	110.04	3.09545
137.5088	112.05	3.24472
143.7592	110.11	3.32532
150.0096	111.16	3.2498
156.26	110.09	3.15947
162.5104	112.06	3.01534
168.7608	109.81	3.06321
175.0112	109.47	2.99504
181.2616	110.91	2.84761
187.512	110.54	2.83109

Figure 5.13: Comparison of different AL

where L_{AL} ranges from $125.008mm$ to $187.512mm$. From this model we can calculate the maximum ER of 3.2 when $L_{AL} = 153.59mm$.

Figure 5.14: Data fitting results to validate the model of AL and ER

5.4.2 Distance between nozzles and diameter of nozzles

In Section 5.4.1, we have discussed the dimensions of the mixing chamber wall. Besides the chamber wall, it is commonly accepted that diameter of nozzles and the distance between nozzles have great influence on entrainment performance. On one hand, works have been done to prove smaller nozzle can significantly increase ER. However, it is not practical to use very small nozzles in all the situations because smaller diameter of nozzles brings about less supply of primary air. As a need for ventilation, the reduction of primary air may give rise to a suffocating room. It also reduces the ability to handle the latent load of the conditioned room, and increases the risk of "office rain". On the other hand, the distance between nozzles has a positive correlation with ER, under the conditions that the nozzles are the same.

This is also impractical, as it has the same problem with the diameter of nozzles. Under the condition that the length of terminal unit is held constant, the increase of the distance between nozzles reduces the amount of nozzle used, which also reduces the primary air supply. Increasing the length of terminal unit will increase the first cost of ACB system, making it financially infeasible. In the research of diameter of nozzles as well as other parameters of terminal units, it should be avoided to increase the entrainment performance by reducing the flow rate of primary air. We hold the opinion that these two factors should not be treated separately, as the change of either one may result in the change of primary air. In our research, we consider these two factors as a whole. That is to say, we consider the distance between nozzles as a variable and the diameter of nozzles is a dependent variable with distance while keeping the primary air constant. To achieve this, we firstly model the relationship between nozzle diameter and primary air flow rate of a single nozzle. The amount of nozzles used is determined by dividing the length of terminal unit by the distance between nozzles, and the diameter of nozzles is calculated with the amount of nozzles and corresponding primary air flow rate.

Primary Air Volume Flow Rate as a Function of Nozzle Diameter

Under the working condition of $100Pa$, we have performed simulations on the single unit. The diameter of nozzle varies from $3mm$ to $10mm$. The detailed corresponding primary air volume flow rate is listed in shown in Table 5.13. and their relationship is shown in Figure 5.15

According to the results shown in Table 5.13, the primary air flow of a single nozzle is regressed as

Table 5.13: The volume flow rates of primary air with different diameters of nozzles

Diameter of Nozzles(mm)	Volume Flow Rate of Primary Air(m^3/h)
3	0.2143
3.5	0.3066
4	0.4016
4.5	0.5086
5	0.6261
5.5	0.7615
6	0.815
6.5	0.9473
7	1.183
7.5	1.3384
8	1.5213
8.5	1.7213
9	1.989
9.5	2.2011
10	2.4864

Note that the nozzle used in Type II has the shape of truncated cone, with air flowing from the base with larger diameter (bottom) to the one with small diameter (top). The term "Diameter" here refers the small one, while the diameter of bottom is kept to be $19mm$.

$$\dot{V}_{10} = gD_N^h \quad (5.4)$$

where $g = 0.0251$, $h = 1.9797$ are constants and D_N is the diameter of nozzles and $\dot{V}_{10}$ is referred as primary air volume flow rate of a single nozzle. The volume flow rate of primary air with different diameters of nozzles and (5.4) is shown in Figure 5.15. It can be seen that (5.4) fits the simulations results well.

Figure 5.15: Data fitting results to validate the model of primary air volume flow rate and nozzle diameter

The Influence of Distance between Nozzles on ER with Constant Primary Air Flow Rate

The volume flow rate of the primary air under normal working condition, $111.5\text{m}^3/\text{h}$, is also kept in our study of distance between nozzles. Once again, as the distance between nozzles increases, the amount of nozzle used is reduced. Therefore the diameter of nozzles must be increased to keep the primary air flow rate constant. We

assume the amount of nozzle installed in each terminal unit is $2m$, and the preferred volume flow rate of a single nozzle is $\frac{111.5}{2m}$. The diameter of nozzle needed can be calculated with (5.4). The simulation results are shown in Table 5.14.

Table 5.14: Results of simulations about the distance between nozzles

Amount of nozzles on each side	Distance between Nozzles(mm)	Nozzle Diameter(mm)	Adjusted $\dot{V}_1(m^3/h)$	ER
24	47.58	9.67	111.87	2.9540
28	40.79	9.04	112.07	3.0376
32	35.69	8.45	109.23	3.1147
36	31.72	7.96	108.14	3.1685
40	28.55	7.55	107.15	3.2014
44	25.95	7.19	106.76	3.2482
48	23.79	6.88	106.99	3.1732
52	21.96	6.61	106.93	3.1501
56	20.39	6.37	106.17	3.1668

In Table 5.14, amount of nozzles on each side, distance between nozzles and nozzle diameter are the parameters we set for simulations and ER and primary air volume flow rate are obtained from simulations. The primary air volume flow rates are well kept constant during the simulations. The amount of nozzle on each side ranges from 24 to 58. with the ER from 2.95 to 3.25. Figure 5.16 shows the trend of ER with distance between nozzles. In the range of our interest, the ER is high when the distance between nozzles is around 26mm. When the distance between nozzles is between 20mm to 40mm, the ER can be well kept above 3.

Quadratic regression shows that the relationship between D_n and ER is

$$ER_{D_n} = -0.000531D_n^2 + 0.0274D_n + 2.8351 \quad (5.5)$$

According to the model, the maximum ER is 3.18, when $D_n = 25.8$. The result derived from the regression model matches with our simulations well.

Figure 5.16: Relationship between D_n and ER

5.4.3 Length of nozzles

We have studied the nozzle arrangement in last section and we will study the nozzle itself in this section. As the nozzles play one of the most important roles in room air induction, it is important to research how the nozzles affect ER. The nozzles can be optimized in many aspects, such as the radius of nozzles, the shape of their opening, the distance between nozzles and their arrangement, the angle of ejection and the distance to the chamber wall. In this section, we investigate the nozzles in Type II structure by varying their length L_n as demonstrated in Figure 5.6, aiming to find out the optimum L_N that gives the maximum ER. The length of the nozzles is set to be different values listed in Table 5.15. Nozzles with different lengths are tested separately at working conditions of $75Pa$, $100Pa$, $150Pa$, and $200Pa$. The adjusted primary air volume flow rates are recorded in Table 5.16. Note that according to the experiment model (3.1), the volume flow rates of primary air under the condition of $P = 75Pa, 150Pa, 200Pa$ are $96.20m^3/h, 137.29m^3/h, 159.13m^3/h$, respectively.

From the table, we can see the simulation results accord with the experiment results well, which also proves the consistency and validity of our study.

Table 5.15: Entrainment performance with nozzles of different lengths

Nozzle Length(mm)	ER at $P = 100Pa$	ER at $P = 75Pa$	ER at $P = 150Pa$	ER at $P = 200Pa$
6	2.4141	2.4860	2.4524	2.3661
10	2.5173	2.5909	2.5307	2.4751
20	2.8063	2.6134	2.5194	2.5250
30	2.7815	2.8086	2.8342	2.8144
40	2.9732	3.0430	3.2267	3.0300
50	3.1532	3.4079	3.3709	3.3709
52	3.3501	3.3757	3.3347	3.3269
55	3.2990	3.2375	3.2092	3.2214
58	3.2019	3.2384	3.1819	3.1638
60	3.1596	3.2096	3.1767	3.1720
65	3.1112	3.1291	3.0873	3.0744
70	2.9476	2.9736	2.9326	2.9184
75	2.7969	2.8098	2.7739	2.7620
80	2.6403	2.6707	2.6363	2.6235
85	2.5161	2.5376	2.5026	2.4903
90	2.3887	2.4204	2.3837	2.3717
95	2.3039	2.3272	2.2854	2.2755
100	2.2079	2.2310	2.1923	2.1846
110	1.9824	2.0155	1.9862	1.9787

The results in Table 5.15 show that in normal working condition, ER increases with nozzle length when $L_N < 50mm$ and decreases when $L_N > 60mm$. When $50mm < L_N < 60mm$, the ER can reach up to 3.35. Under other working conditions, ER also increases with nozzles to a maximum and then decreases. Table 5.17 shows corresponding L_N under different working condition that gives maximum ER.

The results in Table 5.16 show that despite of nozzles with various length, the primary volume flow rate fluctuates little. There is a trend that $\dot{V}_1$ increases with nozzle length, but the incremental can be neglected. In other working conditions, $\dot{V}_1$ also matches

Table 5.16: Adjusted $\dot{V}_1$ with nozzles of different lengths

Nozzle Length(mm)	$\dot{V}_1(m^3/h)$ at $P = 100Pa$	$\dot{V}_1(m^3/h)$ at $P = 75Pa$	$\dot{V}_1(m^3/h)$ at $P = 150Pa$	$\dot{V}_1(m^3/h)$ at $P = 200Pa$
6	105.77	91.49	129.70	149.94
10	109.26	94.52	133.97	154.87
20	111.59	96.57	136.88	158.16
30	113.02	97.82	138.69	160.20
40	114.85	99.52	140.63	162.44
50	110.47	95.66	135.26	156.21
52	112.53	97.47	137.85	159.20
55	111.04	96.20	136.03	157.07
58	115.19	99.75	141.02	162.86
60	110.35	95.53	135.13	156.07
65	111.79	96.74	136.83	158.01
70	112.99	97.80	138.29	159.71
75	112.13	97.06	137.23	158.50
80	113.13	97.96	138.53	160.02
85	113.45	98.24	138.92	160.48
90	113.75	98.50	139.30	160.93
95	113.46	98.23	138.93	160.49
100	110.74	95.90	135.66	156.70
110	109.25	94.58	133.76	154.55

with the Type I model. Therefore, the simulations are valid.

Table 5.17: Maximum ER with different nozzles

Working Condition	Optimal Nozzle Length	Maximum ER
100Pa	52mm	3.35
75Pa	50mm	3.40
150Pa	50mm	3.37
200Pa	50mm	3.37

The theory of increasing ER with longer nozzles is demonstrated as follow. For the ease of description, the room air is divided into two parts: A and B, as shown in Figure 5.17. A is the part of the air that is induced up and mixes with the ejected

Figure 5.17: Primary air and induced air flow streamlines near the nozzle

air under the jet flow, and B is the part of air that is induced up and goes between the nozzles. Different from A, B mixes with the ejected air at the space between jet flow and the chamber wall. From the streamlines we can see that, most of A mixes with primary air near the nozzle exit and lengthening the nozzle does not affect the induction of A. However, B part of air needs a longer nozzle so that the space between nozzles is big enough for the supply of air.

It is also worth to mention that the air directions near the room air inlet are both going upwards, as shown in Figure 4.8 and 5.17. Thus, the air flow patterns near room air inlet are similar in Type I and Type II terminal units, which also proves that the loss coefficient does not change much in both models.

Note that the optimization of nozzle length is carried in different working conditions. Pressure supplied to primary chamber varies from $75Pa$ to $200Pa$. The terminal unit shows consistent performance in all the scenarios. The ER is held on 3.3 to 3.4, and the optimal L_N is around $50mm$. Therefore, we can deduce that the Type II terminal unit is robust to the working conditions

5.5 Conclusion

In this chapter, we present the modification we made on Type II terminal unit, and create the Type II terminal unit accordingly. The modifications are reflected in four aspects. Firstly, the installation of nozzles is moved from the edge to the center. Secondly, the height of the mixing chamber has been diminished and the total height of terminal unit can also be diminished. Thirdly, the wall of mixing chamber has been changed to a curvature S-shape to eliminate unnecessary kinetic energy loss. Last but not least, the length of nozzles are changed to fit the proposed terminal unit. It is obvious that the mixing chamber of Type II terminal unit is significantly different from Type I terminal unit, and these modifications have brought a great leap in the entrainment performance of terminal unit.

As the preliminary design of terminal unit is based on empirical research of commercial products, we can not say the proposed terminal unit is the optimal solution of ACB

terminal unit. Therefore, we have undergone a series of optimization studies. Firstly, the dimensions of the mixing chamber are optimized. The dimensions of the mixing chamber contain a lot of line segments and arcs whose actual values are not optimized. We perform a parameter analysis to determine which is worthwhile to optimize. We then select three dimensions with strong correlation with ER to study. Secondly, we study the effect of distance between nozzles on ER. The study is carried under the condition that the primary air volume flow rate is held constant, which is also an innovative approach in this field. An empirical model of primary air volume flow rate and nozzle diameter under certain conditions is created, and the influence of diameters of nozzles on ER is studied accordingly. Lastly, the length of nozzles is studied through a series of simulations. The study is carried in different working conditions, which also proves that the proposed terminal unit is robust in different situations. The results also show that the nozzle that can optimize the entrainment performance of Type II terminal unit is fairly different than the ones used in Type I terminal unit.

The modification and optimization ensure the superior performance of Type II terminal unit and it is an important portion of our study.

Chapter 6

Entrainment Performance Comparison and Ceiling Attachment Test

6.1 Introduction

Through intensive experimental study presented in Chapter 3, validation of simulation model presented in Chapter 4, and model modification and optimization presented in Chapter 5, we finally achieved a systematically optimized terminal unit, Type II terminal unit. It has many advantages comparing to other products in the market, which facilitate the popularity of ACB system. Besides, as enhanced entrainment performance can increase the efficiency of the overall system, it also significantly benefits the entire system in terms of energy saving.

In this chapter, we will illustrate and verify the superior advantages of Type II terminal unit through comparing and contrasting, which is carried in three parts. Firstly, we will summarize the performance of Type II terminal unit obtained so far and compare it with Type I terminal unit. Secondly, we will choose another off-the-shelf

terminal unit and compare it with Type II terminal unit. Note that the comparison is also carried under the condition of the same volume flow rate of primary air. Lastly, as all the experimental and simulation studies are conducted under this condition, it is impractical to compare it with other products with different flow rate of primary air. Therefore, we deliberately modify the diameters of the nozzles to adjust the volume flow rate of primary air, so that it is possible to compare with any other product in the market. Besides, to ensure the Type II terminal unit better adapt to the market, we do not only focus on ER, but also the thermal comfort of the occupants in the conditioned room. A simulation is conducted with an ACB terminal unit installed in a virtual room, to test how the discharged air attaches to the room.

This chapter is organized as follow. Section 6.2 summarizes the performance of Type II terminal unit and compares it with Type I terminal unit. Section 6.3 compares the entrainment performance of Type II terminal unit with another commercial product. Section 6.4 provide a reference of Type II terminal unit entrainment performance with different primary air flow rate. Section 6.5 shows that the discharged mixed air is well attached to the ceilings connected to the terminal unit, and conclusion is given in Section 6.6

6.2 Performance comparison between Type I terminal unit and Type II terminal unit

From the empirical model of entrainment performance of Type I terminal unit, we can calculate that the ER of Type I terminal unit under normal working condition is 2.5334. The corresponding primary air volume flow rate is $111.5m^3/h$. The en-

trainment performance of Type II terminal unit so far is 3.35 with the same working conditions and primary air volume flow rate, according to the simulations presented in Chapter 5. The increase of ER, in this case, is equivalent to 32.23%. For other working conditions, the increase of ER is also significant. Table. 6.1 compares the ER under different working conditions, and Table. 6.2 compares the corresponding volume flow rate of primary air.

Table 6.1: Comparison of ER between Type I terminal unit and Type II terminal unit

Working Conditions	ER of Type I	ER of Type II	% incremental
$P = 75Pa$	2.5433	3.40	33.68%
$P = 100Pa$	2.5334	3.35	32.23%
$P = 150Pa$	2.5193	3.37	33.77%
$P = 200Pa$	2.5095	3.37	34.29%

Table 6.2: Comparison of $\dot{V}_1$ between Type I terminal unit and Type II terminal unit

Working Conditions	$\dot{V}_1(m^3/h)$ of Type I	$\dot{V}_1(m^3/h)$ of Type II
$P = 75Pa$	96.20	95.66
$P = 100Pa$	111.5	112.53
$P = 150Pa$	137.29	135.26
$P = 200Pa$	159.13	156.21

It can be seen from the tables that the increase of ER is all above 30%, with similar primary air supply. Type II terminal unit shows great enhancement in entrainment performance. Besides the entrainment performance, Type II terminal unit takes less space in height. As a design constraint, the width and the length of the terminal unit are kept constant, but the height is better to be smaller. The height of Type I terminal unit is 289.39mm, while the height of Type II terminal unit is 233mm. Type II terminal unit is 56mm shorter in height. In fact, the height of the terminal unit can be further diminished, without affecting its performance. In summary, Type II terminal unit is far better than Type I terminal unit, either in entrainment performance or in

the space taken. It is obvious that the modifications and optimizations make Type II terminal unit show great advantages over Type I terminal unit.

6.3 Performance comparison between Type II terminal unit with another terminal unit available in market

Type II terminal unit has been proven better than Type I terminal unit, but it may not be true that Type II terminal unit is one of the best terminal units in the market. To make the results obtained more convincing, another commercial product is introduced for comparison. We choose one of the best product, made by Trox technix, DID-622-US series with Type M nozzles, to compare the entrainment performance under different working conditions.

Note that, throughout our study, we keep the volume flow rate of primary air constant. In the meanwhile, the length of terminal unit is also kept the same as the original design for the fairness of comparison. The purpose is to limit the total cost of terminal units used in the conditioned room. Strictly speaking, the installation cost is not only proportional to the number of terminal units used in the room, but also the length of the terminal unit. Obviously, the larger the size of the terminal unit, the higher it costs. What we want to keep constant is actually the volume flow rate of primary air per unit length. As the terminal unit used in our study is 4 feet in length, the primary air volume rate used for comparison in this section is divided by 4.

Besides the primary air, the comparison should be made with the same working con-

ditions, i.e. primary pressure. However, we do not have the data for the entrainment performance of DID-622-US working in the conditions used in our study and with comparable primary air flow rate. For a fair comparison, we set the pressures close to the ones listed in the user guide of DID-622-US. For the primary air of Type II terminal unit, we choose to change the nozzle diameter to adjust primary air flow rate as in previous chapters. The performance data of DID-626-US is retrieved from [58]. The nozzles we used in Type II still have the shape of truncated cone, with air flowing from the base with larger diameter (bottom) to the one with small diameter (top). The diameter of bottom is kept $19mm$, and the diameter of top is $5mm$

Table 6.3: Performance comparison of two types of terminal units

Product Name	Primary Airflow ^b (m^3/h per foot)	Pressure (Pa)	ER
DID-622-US ^a	8.50	42.35	4.8
	13.59	112.09	
	18.69	211.73	
Type II	8.26	45	5.29
	13.46	120	5.34
	18.26	220	5.30

^a As the system of units is different in the US, data retrieved from [14] has been converted to the units used in this paper, for the ease of comparison.

^b Data is acquired as the primary volume flow rate in m^3/h divided by the length of terminal unit in feet.

As shown in Table 6.3, the DID-622-US gives a constant ER in different working conditions. Its ER of 4.8 is fairly good compared with other commercial products. However, under the same working conditions, our simulations results show ER of Type II terminal unit is about 0.5 in average higher than DID-622-US. Note that comparing with our previous result, the increase in ER of Type II terminal unit is caused by utilizing small nozzle. Note that small nozzle may reduce the volume flow rate of primary air, and thus increase ER.

By comparing DID-622-US and Type II terminal units, it can be concluded that Type II structure is about 10% better than DID-622-US, with similar primary air rate and pressure. In a nutshell, the results illustrate that our Type II terminal unit has better entrainment performance compared with DID-622-US with Type M nozzles, which also proves that our geometric modification on the terminal unit has a great contribution in the energy efficiency performance of ACB system.

6.4 Entrainment performance with different primary air flow rate

In previous sections, we compare the entrainment performance of Type II terminal unit with Type I terminal unit and the DID-622-US. It is a relatively easy work to compare with Type I terminal unit, as all the experiments and simulations are performed under the same volume flow rate of primary air. There is no need to convert the results. However, the comparison with DID-633-US is not straight forward. As the volume flow rate of primary air per unit length designed for each product is different, direct comparison is meaningless. Thus we have to modify the nozzle used to achieve the same primary air volume flow rate per unit length. The comparison is thus valid. However, it is still impractical to compare with the rest of the product in the market. Once we have another product on hand, it is still not straightforward to compare their performance. Therefore, we perform another series of simulations with different size of nozzles installed. It provides a guideline for comparing with the rest terminal units in the market. The results are shown in Table 6.4 to Table 6.7. The simulation of ER under different volume flow rate of primary air is also conducted with different working conditions. Once again, the entrainment performance of terminal unit with

different flow rates cannot be compared by simply comparing their corresponding ER. For example, although the ER has reached to 18.6488 when the nozzle with the diameter of $1.5mm$ is installed, we cannot say its entrainment performance is better than the one installed with the diameter of $11mm$, whose ER is 2.0908.

Table 6.4: ER under different volume flow rate of primary air, working condition: $P = 75Pa$

Nozzle Diameter(mm)	$\dot{V}_1(m^3/h)$	ER
1.5	3.67	18.8181
2	6.93	13.7763
2.5	10.90	11.1060
3	15.11	9.4469
3.5	21.30	7.8875
4	28.08	6.7579
4.5	35.51	5.9062
5	44.04	5.2478
5.5	52.29	4.7462
6	61.89	4.3162
6.5	73.43	3.9480
7	86.17	3.6028
7.5	98.71	3.3327
8	111.60	3.0954
8.5	125.77	2.8895
9	141.19	2.6914
9.5	156.70	2.5065
10	173.35	2.3458
10.5	185.16	2.2473
11	204.03	2.0952

If the diameter of nozzles is too small, the ACB terminal unit behaves more like a passive chilled beam. The coils and fins almost provide all the cooling, but lack the ability to ventilate and handle the latent load. Although the efficiency of the system is achieved, but with the risk of "office rain" and high installation cost. On the other hand, if the diameter of a nozzle is too large, the ventilation and dehumidification ability is assured, but the ER is too low. Such system is more close to a VAV system.

Table 6.5: ER under different volume flow rate of primary air, working condition:
 $P = 100Pa$

Nozzle Diameter(mm)	$\dot{V}_1(m^3/h)$	ER
1.5	4.25	18.6488
2	8.02	13.8190
2.5	12.61	11.0268
3	17.47	9.4350
3.5	24.63	7.8133
4	32.46	6.6903
4.5	41.03	5.8541
5	50.88	5.1987
5.5	60.40	4.7248
6	71.47	4.3254
6.5	84.79	3.9580
7	99.49	3.6008
7.5	113.97	3.3317
8	128.85	3.1027
8.5	145.22	2.8892
9	163.05	2.6904
9.5	181.02	2.4998
10	200.29	2.3398
10.5	213.93	2.2420
11	235.79	2.0908

Table 6.6: ER under different volume flow rate of primary air, working condition:
 $P = 150Pa$

Nozzle Diameter(mm)	$\dot{V}_1(m^3/h)$	ER
1.5	5.22	18.6455
2	9.85	13.9502
2.5	15.49	10.9110
3	21.44	9.3796
3.5	30.20	7.7189
4	39.80	6.5973
4.5	50.29	5.7856
5	62.33	5.1738
5.5	73.97	4.7482
6	87.51	4.3342
6.5	103.80	3.9675
7	121.81	3.5973
7.5	139.55	3.3380
8	157.80	3.0995
8.5	177.89	2.8873
9	199.79	2.6911
9.5	221.89	2.4920
10	245.55	2.3320
10.5	262.41	2.2417
11	289.14	2.0856

Table 6.7: ER under different volume flow rate of primary air, working condition:
 $P = 200Pa$

Nozzle Diameter(mm)	$\dot{V}_1(m^3/h)$	ER
1.5	6.04	18.4768
2	11.38	14.0768
2.5	17.90	10.8762
3	24.78	9.4171
3.5	34.90	7.6561
4	45.98	6.5460
4.5	58.07	5.7760
5	71.95	5.1990
5.5	85.38	4.7702
6	101.02	4.3532
6.5	119.83	3.9703
7	140.63	3.5992
7.5	161.14	3.3385
8	182.25	3.0998
8.5	205.49	2.8876
9	230.82	2.6863
9.5	256.44	2.4917
10	283.83	2.3265
10.5	303.21	2.2214
11	334.18	2.0817

The cooling is more dependent on the primary air, making the efficiency of the system lower.

6.5 Ceiling attachment test

To ensure the thermal comfort of the occupants in the conditioned room, the mixed air discharged from a well designed terminal unit should closely attach to the ceilings jointly connected to it. When we modify the mixing chamber wall, we have taken the thermal comfort of occupants into consideration. The attachment of mixed air to the ceilings is proved in this section.

Ideally, the mixed air flows close to the ceiling. In the meanwhile, it exchanges heat with the room and falls to the ground at the room walls, forming a full cycle of indoor air flow. In Figure 5.17, it can be noted that the outgoing mixed air has attached to the end of mixing chamber wall notated in circle. To ensure the novelty and validity of our design, we performed another simulation to test the ceiling attachment of Type II terminal unit.

Type II terminal unit is attached to a $3m \times 3m$ virtual room, with the height of the ceiling $2.9m$. There is an opening door on one of the walls, and it is considered as the room air outlet. The geometry of the virtual room can be viewed in Figure 6.1. As our goal is to test the ceiling attachment only, complicated obstacles and heat sources are omitted.

Figure 6.2 shows the streamlines of the airflow. It is proved that the mixed air flows along the ceiling and fall to the room gradually. This test has proved the proposed Type II terminal unit is able to supply cooling with good thermal comfort.

Figure 6.1: Geometry of a virtual room with ACB terminal unit installed above ceilings

Figure 6.2: Streamlines inside the virtual room

Figure 6.3 shows the vector field inside the virtual room. It can be seen from the figure that the velocity near the outlet of terminal unit is the highest, and the discharged air keeps a relatively high speed when flowing beneath the ceilings. As it gradually falls to the ground, its speed reduces. The air velocity in the middle of the room is disorganized, representing a low-speed distribution. Such velocity vector field can fully ensure the thermal comfort of occupants inside the room.

Figure 6.3: Vector field inside the virtual room

6.6 Conclusion

In this chapter, we demonstrate the superior performance of the modified terminal unit with a series of comparisons. Firstly, the entrainment performance and the height of Type II terminal unit are respectively compared with Type I terminal unit. The results show that The Type II terminal unit is 33% better than the Type I terminal unit, by taking 19% less space. These results are obtained without sacrificing its ven-

tilation ability. The energy performance of the overall system can thus be increased. The comparison with another commercial product also shows the advantage of Type II terminal unit. By comparing with an off-the-shelf product in the US, Type II terminal unit also surpasses the DID-622-US, no matter how much primary air is supplied. The results show that Type II terminal unit is better than one of the representative product in the market. As it is impractical to compare with every product in the market, we provide a guideline for comparison with a series of nozzles. Note that the dimensions, such as, $\angle BAN$, EG , AL , as well as the distance between nozzles, are optimized with $\dot{V}_1 = 111.5 m^3/h$. The ER provided in the guideline may be further increased with proper optimization, with the method provided in this study.

A good terminal unit should not only provide good entrainment performance but also ensure the thermal comfort of the occupants. According to the simulations, the air discharged to the room closely attach to the ceiling, and will not blow the occupants below the terminal unit. Based on these facts, Type II terminal unit is a well-designed product.

Chapter 7

Conclusion and future work

7.1 Conclusion

ACB system has been used for more than 20 years, but there are still many problems in the tropic markets to be resolved. Nonetheless, its energy-saving potential indicates that it will have a broad development prospect in tropical regions. Our research about ACB terminal unit has great significance on its popularity. During our study, we have achieved the followings:

1. We have built an experiment plant and completed a series of experiments. During the experiments we do not simply repeat what has been done before, but innovatively proposed a new acquiring method of ER by measuring volume flow rate of mixed air. The advantages of measuring mixed air instead are obvious. Firstly, the average speed of mixed air is much larger than induced air. It is easy to measure and robust to external disturbance. Secondly, the volume flow rate of mixed air is larger, which means the relative error is smaller. The proposed method ensures the accuracy of measurement and subsequent analysis.
2. In the end of the experiment, we have measured the volume flow rates of primary air and mixed air under different working conditions. In the meanwhile, we built

empirical models of the flow rates of the two kinds of air with pressure. A model of ER with pressure is derived accordingly. Through the experiments, we have gained sufficient understanding of terminal unit and it provides the basis for the subsequent analysis.

3. After the experiments, we build a CFD model on the basis of the experiment. The dimensions of the model are configured close to the Type I terminal unit as much as possible. However, to save the cost of computation, The model is duly simplified. In the simulations, the trivial and complex structure of the cooling coils and fins significantly increases the complexity of computations. Therefore, we introduce the loss coefficient to simplify the process of pressure drop when air passes through the coils and fins. With the help of loss coefficient, we only have to perform the complex simulation once, and hundreds of simulations are speeded up. After carefully configuring the mesh and boundary conditions and iteratively solving procedure, we can obtain the simulation results. The simulation results are found to be consistent with experiment results. The result of ER and primary air flow rate also match well with experiments. The consistency of the simulation results proves the validity of the CFD model.
4. The validity of CFD model will not be void by the modifications on the geometry. Several modifications are applied on Type I terminal unit. They mainly come in four aspects. Firstly, the nozzles are installed in the center. Secondly, the height of the terminal unit is diminished. Thirdly, the wall of mixing chamber is changed to an S-shape curvature and last but not least, the nozzles used in Type II terminal unit is lengthened.
5. Besides the qualitative analysis, we also performed quantitative optimization. Three main dimensions with stronger correlation with entrainment are opti-

mized. The distance between nozzles is then optimized under the condition of constant primary air flow. To achieve this, a model of volume primary air with nozzle diameter is derived from regression. The length of the nozzles that fits the Type II terminal unit is optimized. The optimization of nozzle length is completed in different working conditions, proving the robustness of Type II terminal unit in the meanwhile.

6. The ER of Type II terminal unit is compared both with Type I terminal unit and DID-622-US. The comparison is completed under the condition of constant primary air flow. The entrainment performance of Type II terminal unit is around 30% better than Type I terminal unit. The height of Type II terminal unit is also smaller and can be further lowered. ER of Type II terminal unit is also 10% higher than DID-622-US, which is a mature product in the US. A guideline of ER with different primary air volume flow rates is provided. The comparison between Type II terminal unit and other products in the market can be easily done.
7. For the ease of comparing to any other products in the market, we perform a series of simulations as a reference. When we get another terminal unit and wish to see if its entrainment performance is better than Type II terminal unit, we can look up its primary air flow rate and find the corresponding diameter of nozzles and ER. However, note that the optimum of some geometric factors, such as the dimension of terminal unit, distance between nozzles and nozzle length, are obtained under normal conditions and $\dot{V}_1 = 111.5m^3/h$. When changed with other sizes of nozzles, the terminal unit may not achieve optimum entrainment performance.
8. Besides the superior entrainment performance, Type II terminal unit shows

good ceiling attachment. Air discharged to the room is closely attached to the ceilings of the conditioned room, and will not affect the thermal comfort of the occupants.

7.2 Recommendation of future work

There are several aspects with prospects for development. Firstly, the pressure inside primary chamber is commonly considered as static and uniform. However, from Figure 4.9, it can be seen that the pressure distribution is not as uniform as expected, so that the pressure to each nozzle is different. This may influence the overall performance of terminal unit. Secondly, we also mentioned that the primary chamber can be further lowered. However, as the primary chamber is not our focus, we do not clarify the smallest height it can achieve. Lastly, limited by time and cost, we are unable to build an actual prototype of Type II terminal unit. The advantages of Type II terminal unit are only verified in simulations. From the simulations of Type II terminal unit, the ER of Type II terminal unit is around 32% better. Because of the estimation and simplification made in the simulations, the actual performance of Type II terminal unit may be slightly different. An actual prototype may provide a more strong evidence for the superior performance and a more accurate measurement of entrainment performance, which will thus be a great contribution to the popularity of ACB system.

Author's publications

Journal Papers

1. **Guan Z**, Wen C. Geometric optimization on active chilled beam terminal unit to achieve high entrainment efficiency[J]. Applied Thermal Engineering, 2016, 98: 816-826.

Conference Papers

1. **Guan Z**, Wen C. Numerical investigation of geometry parameters for designing efficient terminal units in active chilled beam[C]//2014 9th IEEE Conference on Industrial Electronics and Applications. IEEE, 2014: 1114-1118.
2. **Guan Z**, Wen C. Further Optimization on Type II Terminal Unit in Active Chilled Beam System [C]//2017 IEEE 26th International Symposium on Industrial Electronics (ISIE). (Revising).

Patents

1. A novel terminal unit used in ACB based HVAC system (Under Preparation)

Bibliography

- [1] U.S. Energy Information Administration. Annual energy review, 2011. <https://www.eia.gov/totalenergy/data/annual/archive/038411.pdf>. Accessed Dec 14, 2016.
- [2] Bingjie Wu, Wenjian Cai, Qinguo Wang, Can Chen, Chen Lin, and Haoran Chen. The air distribution around nozzles based on active chilled beam system. 2016.
- [3] Paul Mathis, Henning Freitag, Daniela Hegemann, Martin Schmidt, and Dirk Müller. New concepts for the ventilation of shopping centers: Reducing air change rate, applying active chilled beams and elevating cold water supply temperature. *Indoor and Built Environment*, page 1420326X16677571, 2017.
- [4] Changhoon Kim, Jinho Lee, Daehui Lee, and Sunho Choi. Optimized operation method for an active chilled beam with vav system. *Science and Technology for the Built Environment*, 22(4):372–378, 2016.
- [5] Can Chen, Wenjian Cai, Youyi Wang, and Chen Lin. Further study on the heat exchanger circuitry arrangement for an active chilled beam terminal unit. *Energy and Buildings*, 103:352–364, 2015.
- [6] Kyu-Nam Rhee, Mi-Su Shin, and Sun-Ho Choi. Thermal uniformity in an open

- plan room with an active chilled beam system and conventional air distribution systems. *Energy and Buildings*, 93:236–248, 2015.
- [7] Jérôme Le Dréau, Per Heiselberg, and Rasmus Lund Jensen. A full-scale experimental set-up for assessing the energy performance of radiant wall and active chilled beam for cooling buildings. In *Building Simulation*, volume 8, pages 39–50. Springer, 2015.
- [8] Alessandro Maccarini, Alireza Afshari, Niels Christian Bergsøe, Göran Hultmark, Magnus Jacobsson, and Anders Vorre. Innovative two-pipe active chilled beam system for simultaneous heating and cooling of office buildings. In *13th International Conference on Indoor Air Quality and Climate*, pages 7–12, 2014.
- [9] Kyu-Nam Rhee and Sun-Ho Choi. Experimental study on air diffusion performance and local discomfort in the space with an active chilled beam system. *Journal of the architectural institute of Korea planning & design*, 30(7):203–210, 2014.
- [10] Rouzbeh Gordnrouzi, Göran Hultmark, Alireza Afshari, and Niels Christian Bergsøe. Study of a two-pipe chilled beam system for both cooling and heating of office buildings. 2013.
- [11] Can Chen, Wenjian Cai, Youyi Wang, and Zhitao Liu. Design of a fuzzy controller for the active chilled beam system. In *2014 9th IEEE Conference on Industrial Electronics and Applications*, pages 723–728. IEEE, 2014.
- [12] Can Chen, Wenjian Cai, Youyi Wang, Chen Lin, and Lei Wang. Operating characteristics and efficiencies of an active chilled beam terminal unit under variable air volume mode. *Applied Thermal Engineering*, 85:71–79, 2015.

-
- [13] Chul-Ho Kim, Ja-Kang Yang, Seung-Eon Lee, Ki-Hyung Yu, and Kang-Soo Kim. Review of new technologies' energy conservation rate in high-performance buildings. *KIEAE Journal*, 16(1):57–65, 2016.
- [14] Ian C Nelson, Charles H Culp, Julian Rimmer, and Brad Tully. The effect of thermal load configuration on the performance of passive chilled beams. *Building and Environment*, 96:188–197, 2016.
- [15] Alessandro Maccarini, Göran Hultmark, Alireza Afshari, Niels C Bergsøe, and Anders Vorre. Transferring heat among building zones through a room-temperature water loopinfluence of climate and occupancy level. In *Building Simulation*, pages 1–14. Springer, 2016.
- [16] YY Liang, DD Wang, JP Chen, YG Shen, and J Du. Temperature control for a vehicle climate chamber using chilled water system. *Applied Thermal Engineering*, 106:117–124, 2016.
- [17] Peter Filipsson, Anders Trüschel, Jonas Gräslund, and Jan-Olof Dalenbäck. Induction ratio of active chilled beams- measurement methods and influencing parameters. *Energy and Buildings*, 129:445–451, 2016.
- [18] Jerome Le Dreau, Per Heiselberg, and Rasmus Lund Jensen. Experimental investigation of the influence of the air jet trajectory on convective heat transfer in buildings equipped with air-based and radiant cooling systems. *Journal of Building Performance Simulation*, 8(5):312–325, 2015.
- [19] G Cammarata and G Petrone. A numerical investigation on active chilled beams for indoor air conditioning. In *Proceedings of the COMSOL Conference, Ed: B. Nilsson*, 2008.

- [20] Alessandro Maccarini, Göran Hultmark, Anders Vorre, Alireza Afshari, and Niels C Bergsøe. Modeling of active beam units with modelica. In *Building Simulation*, volume 8, pages 543–550. Springer, 2015.
- [21] Panu Mustakallio, Zhecho Bolashikov, Kalin Kostov, Arsen Melikov, and Risto Kosonen. Thermal environment in simulated offices with convective and radiant cooling systems under cooling (summer) mode of operation. *Building and Environment*, 100:82–91, 2016.
- [22] Esmail M Saber, Kwok Wai Tham, and Hansjürg Leibundgut. A review of high temperature cooling systems in tropical buildings. *Building and Environment*, 96:237–249, 2016.
- [23] J Le Dréau and P Heiselberg. Sensitivity analysis of the thermal performance of radiant and convective terminals for cooling buildings. *Energy and Buildings*, 82:482–491, 2014.
- [24] Kurt Roth, John Dieckmann, Robert Zogg, and James Brodrick. Chilled beam cooling. *ASHRAE JOURNAL*, 49(9):84, 2007.
- [25] John Vastyan, Common Ground, and Pa Manheim. Chilled-beam basics. *Heating Plumbing Air Conditioning*, 83(7), 2011.
- [26] Andrey Livchak and Chris Lowell. Dont turn active beams into expensive diffusers. *ASHRAE Journal*, pages 52–60, 2012.
- [27] Syed Moazzam Ali and BALU NAIK Banoth. Low energy consumption hvac systems for green buildings using chilled beam technology. *International Journal of Advanced Research in Engineering & Technology (IJARET)*, 4(3):316–324, 2013.

-
- [28] Mika Ruponen and John A Tinker. A novel method to measure the air entrainment ratio of an active chilled beam. *International Journal of Ventilation*, 7(4):299–308, 2009.
- [29] Corina Stetiu. Radiant cooling in us office buildings: towards eliminating the perception of climate-imposed barriers. Technical report, Lawrence Berkeley National Lab., CA (United States), 1998.
- [30] Corina Stetiu. Energy and peak power savings potential of radiant cooling systems in us commercial buildings. *Energy and buildings*, 30(2):127–138, 1999.
- [31] Stanlet A Mumma. Dedicated outdoor air in parallel with chilled ceiling system. *Engineered Systems*, 18(11):56, 2001.
- [32] F Alamdari, DJG Butler, PF Grigg, and MR Shaw. Chilled ceilings and displacement ventilation. *Renewable Energy*, 15(1):300–305, 1998.
- [33] barcolair. Active chilled beam system. <http://www.barcolair.cn/eng/product.aspx?menuId=80>. Accessed Jan 4, 2014.
- [34] DADANCO. Acb40(2-way) acb50 (1-way) active chilled beams (metric). http://www.master.ca/documents/regroupements/ACB40_50_Metric_US.pdf. Accessed Jan 4, 2014.
- [35] ASHRAE. The standards for ventilation and indoor air quality. <https://www.ashrae.org/resources--publications/bookstore/standards-62-1--62-2>. Accessed July 3rd, 2015.
- [36] Darren Alexander PEng and Mike O’Rourke. Design considerations for active chilled beams. *ASHRAE Journal*, 50(9):50, 2008.

- [37] Jan Fredriksson, Mats Sandberg, and Bahram Moshfegh. Experimental investigation of the velocity field and airflow pattern generated by cooling ceiling beams. *Building and environment*, 36(7):891–899, 2001.
- [38] Peter Rumsey and John Weale. Chilled beams in labs: eliminating reheat & saving energy on a budget. *ASHRAE JOURNAL*, 49(1):18–+, 2007.
- [39] Farhad Memarzadeh, Jane Jiang, and Andy Manning. Numerical investigation on ventilation strategy for laboratories: An approach to control thermal comfort and air quality using active chilled beams. In *ASME 2007 Energy Sustainability Conference*, pages 409–416. American Society of Mechanical Engineers, 2007.
- [40] Guangyu Cao, Jarek Kurnitski, Panu Mustakallio, and Olli Seppänen. Active chilled beam wall jet prediction by the free convection model. *International Journal of Ventilation*, 7(2):169–178, 2008.
- [41] Guangyu Cao, Markku Sivukari, Jarek Kurnitski, Mika Ruponen, and Olli Seppänen. Particle image velocimetry (piv) application in the measurement of indoor air distribution by an active chilled beam. *Building and Environment*, 45(9):1932–1940, 2010.
- [42] Hannu Koskela, Henna Häggblom, Risto Kosonen, and Mika Ruponen. Flow pattern and thermal comfort in office environment with active chilled beams. *HVAC&R Research*, 18(4):723–736, 2012.
- [43] Can Chen, Wenjian Cai, Karunagaran Giridharan, and Youyi Wang. A hybrid dynamic modeling of active chilled beam terminal unit. *Applied Energy*, 128:133–143, 2014.
- [44] Can Chen, Wenjian Cai, Youyi Wang, and Chen Lin. Performance comparison

- of heat exchangers with different circuitry arrangements for active chilled beam applications. *Energy and Buildings*, 79:164–172, 2014.
- [45] Panu Mustakallio, Risto Kosonen, and Anna Korinkova. Full-scale test and cfd-simulation of radiant panel integrated with exposed chilled beam in heating mode. In *Building Simulation*, volume 10, pages 75–85. Springer, 2017.
- [46] Lei XU and Hui-xin LUO. The technology of numerical simulation based on ansys icem cfd and cfx software [j]. *Mechanical Engineer*, 12:049, 2008.
- [47] Szabolcs Varga, Armando C Oliveira, and Bogdan Diaconu. Numerical assessment of steam ejector efficiencies using cfd. *International Journal of Refrigeration*, 32(6):1203–1211, 2009.
- [48] Muhammad Mahmood Aslam Bhutta, Nasir Hayat, Muhammad Hassan Bashir, Ahmer Rais Khan, Kanwar Naveed Ahmad, and Sarfaraz Khan. Cfd applications in various heat exchangers design: A review. *Applied Thermal Engineering*, 32:1–12, 2012.
- [49] Anders Olsson, Göran Stemme, and Erik Stemme. Numerical and experimental studies of flat-walled diffuser elements for valve-less micropumps. *Sensors and Actuators A: Physical*, 84(1):165–175, 2000.
- [50] Raghvendra Gupta, David F Fletcher, and Brian S Haynes. On the cfd modelling of taylor flow in microchannels. *Chemical Engineering Science*, 64(12):2941–2950, 2009.
- [51] Kai-Shing Yang, Ing-Young Chen, Kuo-Hsiang Chien, and Chi-Chuan Wang. A numerical study of the nozzle/diffuser micropump. *Proceedings of the Institution*

- of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 222(3):525–533, 2008.
- [52] MM Noor, Andrew P Wandel, and Talal Yusaf. Analysis of recirculation zone and ignition position of non-premixed bluff-body for biogas mild combustion. *International Journal of Automotive and Mechanical Engineering*, 8:1176, 2013.
- [53] Th Frank, PJ Zwart, E Krepper, H-M Prasser, and D Lucas. Validation of cfd models for mono-and polydisperse air–water two-phase flows in pipes. *Nuclear Engineering and Design*, 238(3):647–659, 2008.
- [54] NK Chougule, GV Parishwad, PR Gore, S Pagnis, and SN Sapali. Cfd analysis of multi-jet air impingement on flat plate. In *Proceeding of the World Congress on Engineering, London, UK*, volume 3, 2011.
- [55] Yinhai Zhu and Peixue Jiang. Experimental and numerical investigation of the effect of shock wave characteristics on the ejector performance. *International Journal of Refrigeration*, 40:31–42, 2014.
- [56] Robert W. Fox, Alan T. McDonald, and Philip J. Pritchard. *Introduction to fluid mechanics*. Hoboken, N.J. : Wiley ; Chichester : John Wiley [distributor], 2009., 2009.
- [57] Help System ANSYS Workbench, Release 14.5. *Modeling Guide*. ANSYS, Inc.
- [58] TROX TECHNIK. Chilled beam design guide. 2009.

Appendix A

Simulation Report

Date

2016/12/15 16:01:55

Contents

[1. File Report](#)

[Table 1](#) File Information for CFX 2

[2. Mesh Report](#)

[Table 2](#) Mesh Information for CFX 2

[3. Physics Report](#)

[Table 3](#) Domain Physics for CFX 2

[Table 4](#) Boundary Physics for CFX 2

[4. Solution Report](#)

[Table 5](#) Boundary Flows for CFX 2

[Figure 1](#)

[Figure 2](#)

[Figure 3](#)

1. File Report

Table 1. File Information for CFX 2

Case	CFX 2
File Path	F:\CRP Projct Files\experiment model calibrate201505\experimental model with hood_files\dp0\CFX-2\CFX\Fluid Flow CFX_005.res
File Date	15 December 2016
File Time	03:11:05 PM
File Type	CFX5
File Version	14.5

2. Mesh Report

Table 2. Mesh Information for CFX 2

Domain	Nodes	Elements
Prim	26743	116477
Sec	40681	187153
All Domains	67424	303630

3. Physics Report

Table 3. Domain Physics for CFX 2

Domain - Prim	
Type	Fluid
Location	B672
Materials	
Air at 25 C	
Fluid Definition	Material Library
Morphology	Continuous Fluid
Settings	
Buoyancy Model	Non Buoyant
Domain Motion	Stationary
Reference Pressure	1.0000e+00 [atm]
Heat Transfer Model	Isothermal
Fluid Temperature	2.5000e+01 [C]
Turbulence Model	k epsilon
Turbulent Wall Functions	Scalable
Domain - Sec	
Type	Fluid
Location	B400
Materials	
Air at 25 C	
Fluid Definition	Material Library
Morphology	Continuous Fluid
Settings	
Buoyancy Model	Non Buoyant
Domain Motion	Stationary
Reference Pressure	1.0000e+00 [atm]
Heat Transfer Model	Isothermal
Fluid Temperature	2.5000e+01 [C]
Turbulence Model	k epsilon
Turbulent Wall Functions	Scalable
Domain Interface - Default Fluid Fluid Interface	
Boundary List1	Default Fluid Fluid Interface Side 1 1
Boundary List2	Default Fluid Fluid Interface Side 1
Interface Type	Fluid Fluid
Settings	
Interface Models	General Connection
Mass And Momentum	Conservative Interface Flux
Mesh Connection	GGI

Table 4. Boundary Physics for CFX 2

Domain	Boundaries	
Prim	Boundary - PrimIn	
	Type	INLET
	Location	PrimIn
	<i>Settings</i>	
	Flow Direction	Zero Gradient
	Flow Regime	Subsonic
	Mass And Momentum	Static Pressure
	Relative Pressure	1.0000e+02 [Pa]
	Turbulence	Zero Gradient
	Boundary - Default Fluid Fluid Interface Side 1 1	
	Type	INTERFACE
	Location	F681.672, F684.672, F686.672, F688.672, F690.672, F692.672, F694.672, F696.672, F698.672, F700.672, F702.672, F704.672, F706.672, F708.672, F710.672, F712.672, F714.672, F716.672, F718.672, F720.672, F722.672, F724.672, F726.672, F728.672, F730.672, F732.672, F734.672, F736.672, F738.672, F740.672, F742.672, F744.672, F746.672, F748.672, F750.672, F752.672, F754.672, F756.672, F758.672, F760.672, F762.672, F764.672, F766.672, F768.672, F770.672, F772.672, F774.672, F776.672, F778.672, F780.672, F782.672, F784.672, F786.672, F788.672, F790.672, F792.672, F794.672, F796.672, F798.672, F800.672, F802.672, F804.672, F806.672, F808.672, F810.672, F812.672, F814.672, F816.672, F818.672, F820.672, F822.672, F824.672, F826.672, F828.672, F830.672, F832.672, F834.672, F836.672
	<i>Settings</i>	
	Mass And Momentum	Conservative Interface Flux
	Turbulence	Conservative Interface Flux
	Boundary - Prim Default	
	Type	WALL
	Location	F673.672, F674.672, F677.672, F838.672, F839.672, F840.672, F841.672, F948.672, F949.672, F950.672, F951.672, F953.672
	<i>Settings</i>	
	Mass And Momentum	No Slip Wall
	Wall Roughness	Smooth Wall
	Boundary - PrimWall	
	Type	WALL
	Location	WallInPrim
	<i>Settings</i>	

	Mass And Momentum	No Slip Wall
	Wall Roughness	Smooth Wall
Sec	Boundary - Default Fluid Fluid Interface Side 1	
	Type	INTERFACE
	Location	F245.400, F247.400, F249.400, F251.400, F253.400, F255.400, F257.400, F259.400, F261.400, F263.400, F265.400, F267.400, F269.400, F271.400, F273.400, F275.400, F277.400, F279.400, F281.400, F283.400, F285.400, F287.400, F289.400, F291.400, F293.400, F295.400, F297.400, F299.400, F301.400, F303.400, F305.400, F307.400, F309.400, F311.400, F313.400, F315.400, F317.400, F319.400, F321.400, F323.400, F325.400, F327.400, F329.400, F331.400, F333.400, F335.400, F337.400, F339.400, F341.400, F343.400, F345.400, F347.400, F349.400, F351.400, F353.400, F355.400, F357.400, F359.400, F361.400, F363.400, F365.400, F367.400, F369.400, F371.400, F373.400, F375.400, F377.400, F379.400, F381.400, F383.400, F385.400, F387.400, F389.400, F391.400, F393.400, F395.400, F397.400, F399.400
	<i>Settings</i>	
	Mass And Momentum	Conservative Interface Flux
	Turbulence	Conservative Interface Flux
	Boundary - Outlet	
	Type	OPENING
	Location	Outlet
	<i>Settings</i>	
	Flow Direction	Normal to Boundary Condition
	Flow Regime	Subsonic
	Mass And Momentum	Opening Pressure and Direction
	Relative Pressure	0.0000e+00 [Pa]
	Turbulence	Zero Gradient
	Boundary - SecIn	
	Type	OPENING
	Location	SecIn
	<i>Settings</i>	
	Flow Regime	Subsonic
	Loss Coefficient	2.3000e+01
	Mass And Momentum	Entrainment
	Relative Pressure	0.0000e+00 [Pa]

Pressure Option	Static Pressure
Turbulence	Zero Gradient
Boundary - Sec Default	
Type	WALL
Location	F401.400, F402.400, F412.400, F413.400, F414.400, F415.400, F416.400, F417.400, F418.400, F420.400, F421.400, F422.400, F423.400, F424.400, F425.400, F426.400, F428.400, F890.400, F891.400, F892.400, F894.400, F895.400, F896.400
<i>Settings</i>	
Mass And Momentum	No Slip Wall
Wall Roughness	Smooth Wall
Boundary - SecWall	
Type	WALL
Location	WallInSec
<i>Settings</i>	
Mass And Momentum	No Slip Wall
Wall Roughness	Smooth Wall

4. Solution Report

Table 5. Boundary Flows for CFX 2

Location	Type	Mass Flow	Momentum		
			X	Y	Z
Default Fluid Fluid Interface Side 1 1 (Prim)	Boundary	0.0000e+00	0.0000e+00	0.0000e+00	0.0000e+00
Default Fluid Fluid Interface Side 1 (Sec)	Boundary	0.0000e+00	0.0000e+00	0.0000e+00	0.0000e+00
Outlet (Sec)	Boundary	-1.4838e-01	4.4186e-05	3.4623e-04	5.2061e-04
Prim Default (Prim)	Boundary	0.0000e+00	-1.3433e+00	-3.3690e+01	-3.8886e-03
PrimIn (Prim)	Boundary	4.0677e-02	-7.6406e-01	5.9790e-04	2.8597e-04
PrimWall (Prim)	Boundary	0.0000e+00	2.1057e+00	3.3245e+01	3.2923e-03
Sec Default (Sec)	Boundary	0.0000e+00	8.7941e-03	-3.1024e-02	-1.5273e-04
SecIn (Sec)	Boundary	1.0770e-01	-2.4161e-03	-3.2585e-01	-9.4651e-04
SecWall (Sec)	Boundary	0.0000e+00	-5.9825e-03	8.0011e-01	7.2717e-04

Figure 1.

Figure 2.

Figure 3.