Design optimization, modelling, and performance evaluation of active chilled beam terminal units

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**Design Optimization, Modeling,
and Performance Evaluation of Active Chilled Beam
Terminal Units**

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Summary

In recent years, the concept of green building enjoys a great popularity throughout the world. Energy conservation and Indoor Environmental Quality (IEQ) improvement in buildings receive consistent attentions from all walks of life. As the core element to create comfortable and healthy indoor environmental conditions for human beings, Heating, Ventilation, and Air-Conditioning (HVAC) systems which are also the largest source of energy consumption are necessary to be well studied and developed. Among various HVAC schemes, using active chilled beam terminal units is a very superior option for next generation HVAC solutions, which was listed as one of the 15 most promising HVAC related technologies by American Council for Energy Efficient Economy (ACEEE) in 2009. Active chilled beam terminal units based HVAC systems originate in Scandinavia and have been adopted widely in Europe and to some extent in Australia. More recently, the systems are penetrating into North America and Asia. However, in depth investigations on the systems are still inadequate. Some technical difficulties have emerged in the existing engineering application and need to be resolved for a wider acceptance, especially in the emerging markets.

Therefore, this thesis tries to put some effort on this front, with the focus on the design optimization, modeling, and performance evaluation of active chilled beam terminal units for the tropical climate. The contributions of the thesis are briefly summarized as below:

- As a core part of active chilled beam terminal units, the secondary heat exchanger should not be a standard “off the shelf” product as it used to be. In order to maximize the cooling capacity while minimize the energy consumption, the circuit arrangement is optimized. An experimental comparison study of four fin-tube heat exchangers with different circuit numbers is conducted to determine the optimal circuit number. Then, tube connecting sequences of the circuits are investigated. Though a series of experiment-aided simulations taking the in-situ secondary air velocity profile

into consideration, optimal tube connecting sequences are proposed. With the optimized circuit arrangement, the performance of the secondary heat exchanger, as well as the terminal unit is substantially enhanced. More importantly, the findings and used methods will have significant effects on the design of future active chilled beam terminal units.

- An appropriate model of active chilled beam terminal units is indispensable in the system design, simulation, performance evaluation, as well as development of advanced control and optimization strategies. However, the issue has been so far overlooked by the research communities. In this work, a hybrid dynamic model of the terminal units with few unknown parameters is established by deriving the model using first principles and estimating the parameters experimentally. Through this approach, a reasonable compromise is made between capturing the exact underlying physics and suitability for engineering applications. Static and dynamic performances of the model are verified. As the first reported model of active chilled beam terminal units, it is expected to have a wide range of applications in the aforementioned aspects. In addition, the modeling technique can be extended to the other terminal units.
- When promoting the application of active chilled beam terminal units in different climates, inappropriate understanding of operating characteristics and efficiencies of the terminal units has probably been the essential obstacle. For example, in tropical regions, not only the sensible cooling capacity but also the latent cooling capacity of the terminal units should be matched with the counterparts of conditioned spaces to strictly avoid condensation. Nevertheless, the latent cooling capacity has never been involved in any studies. In order to address this issue, a series of simulations are carried out. The operating characteristics and efficiencies of the terminal units under variable air volume mode are revealed for the first time. The obtained result will be fundamental in designing and operating active chilled beam systems.

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Nomenclature

A	area or effective heat transfer area (m^2)
a	constant coefficient
b	constant coefficient
C	specific heat at constant pressure ($\text{J/kg}^\circ\text{C}$)
c	constant coefficient
D	diameter or characteristic length (m)
d	constant coefficient
dQ	local heat transfer rate (W)
dT	local temperature difference ($^\circ\text{C}$)
e	constant coefficient
f	Fanning friction factor
g	constant coefficient
h	heat transfer coefficient ($\text{W/m}^2^\circ\text{C}$)
i	constant coefficient
j	Colburn factor
k	thermal conductivity ($\text{W/m}^\circ\text{C}$)
L	length (m)
l	constant coefficient
M	mass (kg)
m	constant coefficient
n	constant coefficient
NTU	number of heat transfer unit
Nu	Nusselt number
P	power consumption (W)
P_{am}	ambient pressure (hPa)
P_d	waffle height (m)
P_f	fin pitch (m)
P_l	longitudinal tube pitch (m)
P_t	transverse tube pitch (m)
Pr	Prandtl number
Q	heat transfer capacity (W)
q	heat transfer capacity of a tube (W)
R	thermal resistance ($^\circ\text{C/W}$)

r	radius (m)
R_{air}	air flow resistance coefficient
Re	Reynolds number
RMS	root mean square error
s	sensitivity coefficient
T	temperature (°C)
T_n	constant coefficient
t	time (s)
U	total uncertainty
u	velocity (m/s)
$\dot{V}$	volume flow rate (m ³ /s)
W	moisture content (g/kg)
x	uncertainty source
x_f	projected fin pattern length for one-half wave length (m)
ΔP	differential pressure (Pa)

Subscripts

a	air or air side
at	air to tube
b	base surface
cal	calculated value
d	dew point
downs	downstream
e	elemental value
eq	equivalent value
f	fin
hA	heat transfer
in	inlet
ins	inside
l	lumped
lat	latent
max	maximum value
offc	off coil
out	outlet
outs	outside

pri	primary air
sec	secondary air
sen	sensible
static	static value
supply	supply air
t	tube
total	total value
tw	tube to water
ups	upstream
w	water or water side
zone	zone

Greek symbols

ρ	density (kg/m ³)
μ	dynamic viscosity (kg/ms)
η	efficiency
δ	fin thickness (m)
ε	heat exchanger effectiveness
ϕ	energy saving potential index

Abbreviations

AHRI	Air-Conditioning Heating, & Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CFD	Computational Fluid Dynamics
COP	Coefficient of Performance
ER	Entrainment Ratio
HVAC	Heating, Ventilation, and Air-Conditioning
IEQ	Indoor Environmental Quality
IES	Illuminating Engineering Society
NC	Noise Criterion
PID	Proportion Integration Differentiation
REHVA	Federation of European HVAC Associations
RH	Relative Humidity
SHR	Sensible Heat Ratio

TCC	Total Cooling Capacity
VAV	Variable Air Volume
VSD	Variable Speed Drive

Chapter 1. Introduction

1.1 Background

Since the first modern Heating, Ventilation, and Air-Conditioning (HVAC) system was invented in 1902 by Willis H. Carrier [1], HVAC systems have gradually become an indispensable part of people's daily life. The principal purpose of HVAC systems is to provide proper indoor environmental conditions for human thermal comfort [2]. With comfortable conditions, working and learning productivities of occupants can be maximized [3]. Additionally, the systems are also critical to maintain conditioned spaces healthy. Otherwise, diseases are inevitably caused with longtime environmental exposure [2], especially in tropical countries, where people spend more time staying indoors and the systems have to be operated all the year around. In order to achieve comfortable and healthy conditions, HVAC systems consume a significant portion of building energy. In those countries located in mild regions the proportion is about 30%, while in tropical countries the portion can be dramatically increased. Taking Singapore as an example, where the annual average temperature is 26.9 °C and the annual average relative humidity is 85%, HVAC systems take up to 52% of the total energy consumption in commercial buildings and 30% of that in residential buildings [4]. As a consequence, improving HVAC systems has huge potential for economic as well as environmental impacts on the society.

In reality, the development of energy efficient HVAC systems goes very fast in recent decades. Novel components, sub-systems, control and optimization strategies, and so on are constantly introduced into practice. Today's HVAC systems have become very profound and complex. In almost every HVAC application, there usually exist several options available to satisfy the same building programs or design intents. And then determining the optimal HVAC system becomes a challenging process involving many decision-making processes. Generally, attaining a good HVAC system often starts with the proper selection of indoor terminal units. The terminal units are desired to deliver even treat air in an effective and efficient manner and they also need to be properly matched with central equipment. Only then can indoor occupants and building owners alike be rewarded with superior comfort and health and lower energy usage [5].

There are many types of HVAC indoor terminal units, including diffusers, consoles, fan coils, blower coils, unit ventilators, active and passive chilled beams, radiant panels, etc. Among them, active chilled beam terminal units are a very competitive option. HVAC systems equipped with active chilled beam terminal units are considered as a promising candidate for the next generation HVAC systems. They can provide improved Indoor Environment Quality (IEQ) with tremendous energy saving potentials. In practice, they have been widely utilized in Europe for about twenty years and to some extent in Australia. Moreover, the interest in them is fueled in North America and Asia in recent years. They were even listed as one of the 15 most promising HVAC related technologies by American Council for Energy Efficient Economy (ACEEE) in 2009 [6].

1.2 Overview of active chilled beam systems

Active chilled beam terminal units are not new. Their predecessor is the floor and ceiling mounted induction units used in 1930's-1970. As shown in Fig. 1.1, core innovation of induction units was the use of high velocity jet nozzles to entrain room air across the secondary heat exchanger through which the induced room air were conditioned. Since only the primary air was recirculated through central air handling units, the size of air handling units and associated ductwork could be reduced. These space savings were quite valuable for high-rise skyscrapers, so induction units were widely deployed at that time.

Figure 1.1 Schematic diagram of an induction unit

However, induction units became less favored in the late 1960's-1970. The reasons included: 1) building stock moved away from the skyscraper profile, suitable application scenarios of induction units became less appealing; 2) some concerns of induction units on energy efficiency, maintenance issues, and initial cost appeared. For example, induction units usually operated with high pressures in the primary air plenum, and the high pressures required considerable higher fan energy consumptions than other terminal units. In addition, induction units allowed condensation on exposed surfaces of the heat exchanger, which required regular maintenance [7]. By the mid-1970's, induction units based HVAC systems were virtually replaced by Variable Air Volume (VAV) systems.

After almost 20 years' of silence and in the early 1990s, a variation of induction units once revived again, but in a more advanced form of active chilled beam terminal units. The changes included but not limited to:

- Improvements in the design of induction nozzles and terminal units;
- Dry condition operations of secondary heat exchangers;

- Integrations with latest central equipment (e.g. dedicated outdoor air systems, high temperature chillers, liquid desiccant dehumidifiers, etc.);
- Evolutions of ventilation, sensible, and latent loads and pertinent building programs and standards;

With those changes, the required inlet pressure to supply the primary air is lowered without any other penalty. And there is no need to clear and replace the heat exchanger or condensate water drainage pan. In short, the most negative concerns of induction units are alleviated in active chilled beam terminal units. However, the most important innovation, the use of induction nozzles to entrain the room air across the secondary heat exchanger, is kept.

In order to facilitate the understanding of modern active chilled beam terminal units, the structure of a typical 2-way discharge ceiling mounted terminal unit is visualized in Fig. 1.2. The working principle is also illustrated.

Figure 1.2 Schematic diagram of a 2-way discharge active chilled beam terminal unit

By maintaining a certain positive pressure in the primary air plenum, a specified amount of pretreated primary air is continuously forced through the induction nozzle through the mixing chamber and out into the conditioned space. The nozzle is designed in such a way that a negative pressure kernel is generated as the pressure is degraded when the primary air flows through the nozzle. The pressure kernel induces the secondary air through the secondary heat exchanger and into the mixing chamber. This induction of the

secondary air is called entrainment effect. Since the heat exchanger is imposed in the path of the secondary air, the secondary air is cooled. By adjusting the water temperature of the heat exchanger, it is operated at dry condition and consequently there is no condensate water. The primary air and secondary air are mixed before leaving the mixing chamber. Finally, the mixed air is discharged by the means of linear slots located along the outside edges of the terminal unit.

The primary air can be supplied by a conventional air treatment and distribution system, like the one shown in Fig. 1.3. The recirculation air is mixed with the outdoor fresh air before entering the air handling unit and the amount of the fresh air is controlled via a damper in order to meet ventilation requirements. In some cases there is no recirculation air for active chilled beam systems, while it is not feasible in tropical countries because of the high cost of handling the fresh air. A damper is also installed at the entrance of each zone or terminal unit which has two functions: 1) controlling the air volume flow rate supplied to the zone or terminal unit; 2) handling a partial operation environment such as during overtime or weekend usage of a particular area. Although such systems have the ability to turn down the primary air volume flow rate, they are generally set and operated in a constant air volume configuration for simplicity. In addition, a Variable Speed Drive (VSD) fan is installed to maintain the duct pressure. The primary air temperature is typically 10-15 °C, which is lower than that of conventional HVAC systems. Due to the entrainment effect and subsequent mixing with the secondary air, temperature of the supply air can be moderate without any degradation of indoor thermal comfort. It should be noted that relative humidity of the primary air needs to be low enough to handle the internal latent load. This is also the reason why active chilled beam systems are often utilized together with dehumidification technologies, especially for hot and humid applications. In summary, the primary air satisfies the entire latent cooling load and ventilation load and a small part of the sensible cooling load, usually around 35-45%.

Figure 1.3 Schematic diagram of a primary air system

A typical chilled water system is illustrated in Fig. 1.4. Without a dedicated high temperature chiller, there is an intermediate heat exchanger to produce the 14-18 °C chilled water for active chilled beam terminal units. This high temperature feature offers system designers many opportunities of adopting free cooling or low energy cooling technologies. Compared with the space air temperature, which is typically 24-26 °C, most of the sensible cooling load can be dissipated with the chilled water. The differential pressure across the supply and return pipes is regulated through a VSD pump. At the end of each branch, a modulating valve is configured to vary the chilled water volume flow rate. It produces a 3.5 to 4.5 °C swing in the secondary air temperature, which affects a 50-60% turndown in the terminal unit's sensible cooling capacity but without effect on the space ventilation and/or dehumidification. This is sufficient interior spaces (except conference areas) where sensible loads do not tend to vary significantly. In addition, the space temperature control can be accomplished by varying the secondary chilled water supply temperature. It should be noted that condensation should be strictly prevented in terms of cutting off the chilled water supply or increasing the chilled water supply temperature.

Figure 1.4 Schematic diagram of a chilled water system

Compared with conventional HVAC systems, active chilled beam systems offer several advantages that are briefly summarized as below:

- *IEQ improvement*: overall IEQ can be substantially improved for at least three reasons:
 - 1) Without fan in or near the occupied space, acoustic signature of active chilled beam systems is lowered. Where traditional overhead terminal units produce sound levels in the range of 35-40 Noise Criterion (NC), active chilled beam systems typically operate with sound levels under 20 NC;
 - 2) In some sense, the fresh air ventilation and sensible and latent cooling loads are decoupled by the primary air and chilled water respectively, so the fresh air supply, indoor temperature, and indoor relative humidity can be flexibly controlled;
 - 3) With the entrainment effect, more comfortable and uniform air velocity and temperature distributions, higher Air Diffusion Performance Index (ADPI), can be achieved.
- *Energy efficiency*: according to several energy retrofit projects in North America, the total power demand for active chilled beam systems is about 25%-30% less

than that for conventional VAV systems [6]. That means 8-10 LEED credits can be awarded through the optimizing energy performance section. This energy savings are also threefold:

- 1) The systems require less supply air flow than VAV systems for the same cooling load, so perpetual fan energy savings are created;
 - 2) With the higher chilled water temperature, chiller plants of the systems can operate at 15%-20% higher efficiency;
 - 3) The need for energy intensive reheat of over cooled supply air is essentially eliminated.
- *Space savings:* as same as induction units, active chilled beam systems afford designers an opportunity to replace large supply and return air ductworks with small chilled water pipes. That results in significant savings in terms of plenum space. In addition, smaller foot print of the primary air handling unit means increased usable floor space.

1.3 Motivations and objectives of the thesis

To fully utilize the potentials of active chilled beam systems, it is necessary for building practitioners to promote the application of the systems, particularly in tropical regions where the systems are operated all the year around and the benefits can be amplified. However, it is worthy to note that active chilled beam systems are originated in Scandinavian regions where the climate condition is much different from that in tropical regions. The existing active chilled beam systems may not be suitable to hot and humid conditions because of some technical difficulties such as insufficient cooling capacity, occurrence of condensation, etc. In addition, even in traditional applications in Scandinavian regions, some general difficulties still remain. In order to address the technical difficulties, in depth investigations are required. Yet, very few research works can be found in literature to address them, which affect the competitiveness of active chilled beam systems even directly hinder their wider applications. For instance,

- Even though being used for over twenty years, the optimal design of active chilled beam terminal units is not thoroughly considered. As a core part, the secondary heat exchanger is simply a standard “off the shelf” product without paying attention to

the particular application on the fin shape, fin spacing, pipe diameter, circuit arrangement, etc. As a result, the cooling capacity as well as energy performance of the conventional terminal units is not optimized or up to the standard for tropical applications.

- Modeling of active chilled beam terminal units is definitely a necessary, but there is no experimentally verified model available in the literature. That makes the design, simulation, and performance evaluation of active chilled beam systems as well as the development of advanced control and optimization strategies less pragmatic.
- Inappropriate understanding of the operating characteristics and efficiencies of active chilled beam terminal units, particularly of the latent cooling capacity, is also an essential obstacle to promote their applications. Particularly in tropical regions, without any idea on the latent cooling capacity, designing an active chilled beam system to strictly avoid condensation often makes the system operation more conservative.

In light of the strong demand for applying active chilled beam systems in tropical regions, the objective of this thesis is to customize an energy efficient tropical active chilled beam system. More specifically, there is a one to one correspondence between the main topics conducted in the thesis and the aforementioned difficulties need to be resolved:

- Optimize the design of active chilled beam terminal units to increase the cooling capacity as well as energy performance to suit to the hot and humid condition.
- Develop an appropriate model to describe active chilled beam terminal units, which is accurate and robust for engineering practices.
- Acquire a comprehensive understanding of the operating characteristics and efficiencies of active chilled beam terminal units to facilitate the applications.

1.4 Major contributions of the thesis

The major contributions of the thesis are accordingly summarized:

- The circuit arrangement of the secondary heat exchanger inside active chilled beam terminal units is optimized. An experimental comparison study of four fin and tube

heat exchangers with different circuit numbers is conducted to determine the optimal circuit number. Then, tube connecting sequences of the circuits are investigated. Though a series of experiment-aided simulations taking the in-situ secondary air velocity profile into consideration, optimal tube connecting sequences are proposed. With the optimized circuit arrangement, the performance of the secondary heat exchanger as well as the terminal unit is substantially enhanced. More importantly, the findings and used methods will have significant effects on the design of future active chilled beam terminal units.

- A hybrid dynamic model of the terminal units with few unknown parameters is established, which is deriving the model using first principles and estimating the parameters experimentally. Through this approach, a reasonable compromise is made between capturing the exact underlying physics and suitability for engineering applications. Static and dynamic performances of the model are experimentally verified. As the first reported model of active chilled beam terminal units, it is expected to have a wide range of applications in the aforementioned aspects. In addition, the modeling technique can be extended to the other terminal units.
- A series of simulations are carried out based on a static version of the dynamic model developed previously. The operating characteristics and efficiencies of the terminal units under variable air volume mode are revealed for the first time. Influences of different primary air and space conditions regarding the temperatures and relative humidities are also captured. The obtained result will be fundamental in designing and operating active chilled beam systems.

1.5 Organization of the thesis

The rest of this thesis is structured in 7 chapters:

Chapter 2 presents a comprehensive review of state of arts in the research and development of active chilled beam systems, which can be used as a context for understanding active chilled beam systems as well as the studies described in the following chapters.

Chapter 3 describes a self-designed 2-way discharge active chilled beam terminal unit and a self-constructed experimental setup, which is adopted to conduct a preliminary performance evaluation of the terminal unit. They will be the basis of the subsequent experimental studies.

Chapter 4 gives a primary study on circuit number of the secondary heat exchanger. With the experimental active chilled beam terminal unit and setup, four 2-rows fin and tube heat exchangers, containing 1 circuit, 2 circuits, 4 circuits, and 8 circuits respectively, are investigated under a wide range of chilled water volume flow rates. Given a nominal air side operating condition, thermodynamic and hydrodynamic characteristics on the chilled water side are compared. The heat transfer capacities are compared under three sets of criteria: identical chilled water volume flow rate, identical pressure drop, and identical pumping energy consumption. The heat exchanger effectiveness and performance index are also used as performance indicators. Based on the comparison, the optimal circuit number is selected.

Further to **Chapter 4**, tube connecting sequences of the heat exchanger circuits are explored in **Chapter 5**. Given the same air side operating condition, in-situ air velocity profile across the heat exchanger is measured and non-uniformities of the air flow caused by the entrainment effect are detected. Taking the air mal-distribution into consideration, thermodynamic performance of the heat exchanger is simulated with a tube to tube distributed parameter model. This simulation model is calibrated with experimental results by selecting appropriate correction factors to the heat transfer coefficients obtained via some published correlations. The tube connecting sequences are then optimized through a particle swarm optimization program for the maximum heat transfer capacity at different water side operating conditions. The potential pressure drop, manufacture difficulties, and material cost of the tube connecting sequences are qualitatively analyzed. Finally, new tube connecting sequences are proposed.

With respect to the active chilled beam terminal unit, **Chapter 6** obtains a dynamic model in a hybrid manner. The model encapsulates mechanical and thermal aspects of the confined air jet and the heat transfer process contained in the terminal unit and can be divided into two sub-models respectively. The description for the primary air, secondary

air, and mixing of them are together taken as the confined air jet sub-model. Another sub-model is the heat transfer description of the heat exchanger. The model is kept simple and practical, avoiding sophisticated jet flow as well as heat transfer theories. Thus, in deriving the model using first principles and estimating it experimentally, a reasonable compromise is made between capturing exact underlying physics and suitability for engineering applications. Unknown model parameters are identified by either a linear or nonlinear least-squares method. Performance of the model is then experimentally verified.

Based on a static version of the dynamic model developed in **Chapter 6**, **Chapter 7** reveals operating characteristics and efficiencies of the active chilled beam terminal unit. A series of realistic simulations are carried out under various primary air volume flow rates and various chilled water volume flow rates. Inherent correlations between Total Cooling Capacity (TCC), Sensible Heat Ratio (SHR), and energy saving potential are explored. The energy saving potential is newly defined as the chilled water sensible cooling capacity to the total sensible cooling capacity ratio. In addition, influences of different primary air conditions as well as space conditions are studied.

A conclusion of the thesis is given in **Chapter 8** and some potential future research directions are summarized as well.

Chapter 2. A review of research into active chilled beam systems

2.1 Introduction

As discussed in **Chapter 1**, active chilled beam systems are still a relatively new technology lacking in depth investigation and trial in tropical regions. Some technical difficulties have already emerged in some engineering practices and await further exploration. In order to figure out the difficulties and avoid major pitfalls, a comprehensive review on the research works into active chilled beam systems is presented in this chapter. It can also be used as a context for understanding the contributions of the studies described in this thesis.

For simplicity, state of art of research into active chilled beam systems can be distinguished into five classes:

- active chilled beam terminal units
- active chilled beam systems
- air flow patterns and indoor thermal comfort
- combinations with other systems
- applications

2.2 Active chilled beam terminal units

In order to attain the optimal design of active chilled beam systems, active chilled beam terminal units as the most critical part of the systems, should be optimized on all the aerodynamic, thermodynamic, and hydrodynamic aspects. To this end, the whole casing, induction nozzle, and secondary heat exchanger have to be in-depth studied. In addition, their performances need to be well captured, evaluated, and modeled at terminal unit level to further facilitate the optimizations. On this front, a few pertinent investigations have been carried out.

As same as the traditional induction units, using the entrainment effect of the induction nozzle to release the primary air and entrain the secondary air is the innovation of active chilled beam terminal units. The entrainment effect is a measure of the efficiency of the terminal units and ultimately the overall cooling capacity which can be achieved. That is

also the root cause of the benefits offered the systems. Therefore, passive control of air jet flow released from the induction nozzle is utilized to enhance the air entrainment effect. Nastase et al. [8-10] experimentally studied two turbulent 6-lobed nozzles with and without lobe deflection angles and compared them with a reference circular nozzle. Vertical mechanisms of the air flows as well as true nature of the air mixing and entrainment were revealed. It was found that the momentum flux transport role played by the streamwise structures was rendered more efficient and led to a spectacular enhancement in the entrainment effect in the initial region of the air jet. In other words, the amount of air being entrained in the lobed jet by the streamwise structure was drastically amplified by the double inclination of the nozzle exit boundary. In practice, Dadanco [11] has patented several special shaped nozzles, e.g. multiple lobed, oval, slot, horseshoe, etc. and implemented the technique in the products.

For an indicative measurement of the entrainment effect, Ruponen et al. [12] presented a novel method to experimentally measure the entrainment ratio, which was defined as the amount of secondary room air induced by per volume of primary air. The method used a single anemometer and a simple purpose built measurement venturi together with the primary air volume flow rate. The result showed that the rectangular venturi method produced reliable and consistent measurements.

The entrainment ratio is affected by a number of factors, i.e. nozzle arrangement, geometries of the casing, etc. Guan et al. [13] conducted a series of Computational Fluid Dynamics (CFD) simulations to address influences of the nozzle radius and spacing on the entrainment ratio. It showed that the nozzle radius was negatively correlated to the entrainment ratio, while the nozzle spacing was positively correlated to the entrainment ratio, and the former has higher influence. Cammarata et al. [14, 15] evaluated the entrainment ratio of three active chilled beam typologies with various geometries using CFD simulations.

Furthermore, Freitag et al. [16, 17] investigated influences of various Reynolds-averaged Navier-Stokes turbulence models on CFD simulations and the best turbulent model for the numerical investigation of flow features inside active chilled beam terminal units, namely free stream, deflected and wall-bounded flow, were found.

Another innovation of active chilled beam terminal units is using chilled water within the secondary heat exchanger to dissipate the sensible load. In order to capture the cooling capacity of the secondary heat exchanger, Clercq et al. [18] proposed a model, which accounted for nozzle type, circuit design, and heat exchanger length, etc. Parameters of the model were estimated and calibrated with experimental measurements. The result confirmed not only the measurement quality but also the accuracy and prediction power of the established capacity model in simulation programs.

With the purpose of establishing requirements, possibilities, and limitations for a well-functioning 2-pipe active chilled beam system for both cooling and heating of office buildings, Afshari et al. [19, 20] investigated the energy saving potential of an active chilled beam system incorporated with the 2-pipe heat exchanger. Simulations were performed to compare a conventional 4-pipe system and a 2-pipe one. The result showed that the energy consumption was 3% to 5% less in the 2-pipe system. Taking the advantage of low external air temperature for free cooling, together with transfer of energy, the energy consumption in the 2-pipe system was 5 % to 18% less.

In addition, performance evaluation and modeling of active chilled beam terminal units are definitely required for their applications. Betz et al. [21] listed some issues arising from the use of active chilled beam terminal units in energy models. Several softwares were reviewed that included a variety of approaches to modeling the terminal units, eQuest 3.64b, Trane TRACE, IES-VE, EnergyPlus v7.0, and TRNSYS 17. Pros and cons of the software were clarified. The topic was even identified as one of the key research needs by American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) in 2013 [22].

2.3 Active chilled beam systems

If an active chilled beam system is improperly designed, operated, or maintained, it may lead to unsatisfactory indoor thermal conditions, or wasted energy. In particular, active chilled beam terminal units are indoor air diffusion device and also air conditioning device, selecting, sizing, locating, operating, and maintaining the terminal units are vital to attaining the practical effectiveness at the system level. Up to now, a few studies have been involved in this class.

The general knowledge of the system operation can be found in references [23-27]. Some guidebooks of active chilled beam systems were developed for engineering practices by Federation of European HVAC Associations (REHVA) and ASHRAE [28, 29]. The guidebooks presented some more popular science knowledge on the working principle, benefits, control, installation, commissioning, operation, etc. Some extraordinary instructions and methodologies were illustrated with case studies.

Alexander et al. [30] presented a series of design considerations for active chilled beam systems, mainly about the duct design and working static pressures, terminal unit placement and room air distribution, chilled water side control, etc. Some unique insights were given. For example, increasing the end of run operating static pressure would be essentially favored, which ultimately required fewer terminal units to satisfy the sensible cooling load, and little penalties in terms of the fan energy and acoustic signature. Suitability of the systems for various spaces was also briefly discussed.

Loudermilk et al. [31, 32] analyzed the potential air distribution of active chilled beam systems in a qualitative manner. The local thermal comfort level was assessed by the draft risk. Accordingly, few design guidelines were given, i.e. the terminal units should be mounted 2 m or more above the designated occupied zone. The humidity control was considered as well. With a case study of four systems, it was proved that the dehumidification devices were often not a necessary, while an alternative to relax the space design humidity was preferred.

In view of a questionable design trend that active chilled beam systems are being designed with high air volume flow rates to match the increasing cooling loads while reduce the total number of terminal units and first costs, Livchark et al. [33] revealed the energy performance degeneration and other disadvantages behind this trend. Then it was recommended that the systems should be used with the minimum primary air volume flow rate. For such a purpose, the methods of increasing the secondary heat exchanger cooling capacity while maintaining the minimum primary air volume flow rate were elaborated via a detailed mathematical description of the secondary heat exchanger. If that was not possible, the systems were favored to be operated under variable air volume mode.

In order to manage indoor environment efficiently throughout the life cycle of buildings, performance based changes in terminal units and ductwork are required. As a result, Kosonen et al. [34] investigated the continuous and flexible organizational change of the work method and workspace, the flexibility to control the air flow pattern generated by active chilled beam terminal units, and the adaptability of the systems to the changes. Moreover, Trox [35] and Halton [36] individually developed adjustable outlet blades to handle the changes in the space layout. The indoor air flow patterns can be optimized, which provide considerable flexibilities of practical applications.

2.4 Air flow patterns and indoor thermal comfort

Comprehensive and systematic understanding of air flow patterns produced by active chilled beam terminal units is a solid basis for indoor thermal comfort design for the systems. To strictly avoid draught sensation in the occupied zone, the velocity entraining the zone can be accurately predicted and controlled in the design phase. The experimental data and models specified are needed to improve the accuracy of CFD simulations and subsequently attain the optimal thermal comfort. Although some preliminary results and conclusions may be derived with reference to other ceiling mounted diffusers, there should be some unique characteristics for active chilled beam terminal units because of the entrainment of the secondary air, etc.

For simplicity, the literatures can be reviewed along the air flow trajectory. The typical trajectory is illustrated in Fig 2.1. The air flow jet attaches to the ceiling, and then impinges on the ceiling wall corner, and then follows the vertical wall, and then impinges on the wall floor corner, and finally enters the occupied zone. The zone is pre-defined as the area 0.3 m from the internal wall, 1.0 m from the external wall, and 1.8 m from the floor [37].

Figure 2.1 Typical trajectory of air flow discharged from active chilled beam terminal units

Outside the occupied zone, the focus is the air flow pattern. Cao et al. [38-44] did a lot of pioneer studies on air flow behaviors. The air turbulence structure, air flow features, velocity distribution, and maximum velocity decay were all investigated. Impact of Reynolds number as well as turbulence intensity level was addressed. Each step along the trajectory was reflected by a model and then the model was experimentally verified. At first, structure and velocity field of the attached plane jet discharged from active chilled beam terminal units were reflected through the particle image velocimetry technology [38, 39]. It was proved that the jet would attach to the ceiling because of the Coanda effect and became fully turbulent in a short distance. The room air was constantly entrained into the jet and the jet grew in a certain rate. Then, the impingement of the attached plane jet on the ceiling wall corner was identified [40, 41]. The jet behaviors were found to be different from those obtained in a relatively low room. An efficient model was set up to predict the maximum air jet velocity decay after the jet impingement and a CFD tool, CFX 11.0, was shown to be effective to describe the process. With respect to the subsequent vertical downward jet, Cao et al. [42] proposed a free convection model by superposing a free convection velocity and an isothermal jet velocity. It was found the air velocities decreased quasi-linearly when approaching the floor level below the height of 1.7 m. The introduced model could be used for prediction of the maximum velocity of the

wall jet. As for the last wall floor corner, the air flow was measured and modeled in [43]. The result showed that the returning corner air flow reattached to the floor surface after separation from the wall. The maximum air velocity was very close to the floor.

Within the occupied zone, not only the air flow pattern but also draft risk, local thermal comfort, was desired to be investigated [45-58] such that the thermal comfort condition satisfies the related standards. Impacts of the work places layout, primary air volume flow rate, and heat load strength and distribution, etc. were all evaluated. Practical guidelines for minimizing the draught risk were given, i.e. placement and arrangement of the terminal units. For example, Fredriksson et al. [45] conducted some experiments in a mockup of an office room. Qualitative information about velocity and temperature distributions below active chilled beam terminal units was obtained by visualization. It was showed that the air flow pattern within the occupied zone behaved similarly to a two-dimensional plume but exhibited strong oscillations. Furthermore, air flows generated by the heat sources in the room might reverse the air flows generated by the terminal units. Koskela et al. [46, 47] studied the air flow pattern and thermal comfort of active chilled beam systems in a full-scale test room. The result indicated that the mean air velocity was high when the air flows discharged from two adjacent terminal units collided with each other and turned down into the occupied zone together. It was also found heat sources had a notable influence on the air flow pattern and draft risk. The mean air velocity might be high at the floor level because of a large scale circulation caused by thermal plumes of the heat sources. As a consequence, active chilled beam systems were difficult to fulfil the targets of the existing standards, especially with high cooling loads. However, an opposite conclusion was obtained by some other studies. Rhee et al. [48] evaluated the thermal uniformity of active chilled beam systems in terms of a comparative study with conventional air distribution systems in a full-scale test bed. Three performance indices were adopted: ADPI to evaluate the air diffusion performance, air velocity to evaluate the local discomfort due to cold draught, and vertical air temperature difference to evaluate temperature stratification. The result showed that the active chilled beam system was successful in providing the acceptable thermal uniformity, even with less air flow rate than the other conventional air distribution systems.

In addition, Dreau et al. [49] conducted a sensitivity analysis of four air distribution systems including an active chilled beam system to determine the parameters influencing their thermal performance the most. The air change rate, outdoor temperature, and temperature stratification had the largest effect on the cooling demand to maintain the same operative temperature. The active chilled beam system was found to be less energy effective, but the global thermal comfort level was always kept within the recommended range. Thermal comfort in the rooms with active chilled beam terminal units and chilled ceilings were also compared in references [50-52]. It was found that differences in the thermal conditions between the systems were not significant. The result was contrary to the expectation that operative temperature would be lower for chilled ceilings.

2.5 Combinations with other systems

The features of active chilled beam terminal units allow them to be combined with various systems, i.e. dedicated outdoor air systems, low energy cooling systems, air dehumidification systems, and air cleaning systems. Through the combinations, benefits of the systems can be maximized, including the overall energy efficiency increment, indoor environmental conditions improvement, space savings, etc. A few investigations have been carried out to discuss the combinations.

Dedicated outdoor air systems

Since active chilled beam systems are usually utilized without recirculation airs, they are simply combined with dedicated outdoor air systems, as an effective manner to manage high sensible cooling loads. For such combinations, Mumma et al. [59-61] did a series of studies on dedicated outdoor air systems to give full play to active chilled beam systems, including determination of supply air conditions, condensation avoidance, enhancing dehumidification ability, etc. As for the performance, Stein et al. [62] held a head to head competition between an active chilled beam terminal units plus dedicated outdoor air system and a variable air volume reheat system in California and concluded that the latter had much lower first cost and energy consumption but similar floor to floor height and this debatable conclusion then led to a series of heated discussions and arguments [63, 64].

Low energy cooling systems

It is known that temperature of the chilled water supplied to active chilled beam terminal units is generally 14-18 °C, which is much higher than the temperature of the chilled water produced by conventional vapor-compression chiller. As a result, active chilled beam terminal units are offered many opportunities to combine with solar and other low energy cooling systems. Fong et al. [65-67] formulated a solar hybrid air-conditioning system, using adsorption refrigeration, desiccant dehumidification, and several types of chilled water based indoor terminal units. Active chilled beam terminal units were one of them. Although they were proved to be a less energy efficient option to work together with solar adsorption refrigeration, technical feasibility of such a combination for space conditioning in the subtropical city and superior performance over vapor-compression chiller based systems were verified. Costelloe et al. [68] explored and confirmed a major potential for the combination between indirect evaporative cooling and active chilled beam terminal units in temperate climate.

Air dehumidification systems

Since the primary air is the sole source of the indoor air dehumidification, active chilled beam terminal units do not have much capability to manage high latent loads. This inability substantially limits their applications, especially in tropical regions. Consequently, the systems can be integrated with some air dehumidification systems. For example, Wahed et al. [69] integrated a thermally regenerated desiccant dehumidification system with active chilled beam systems to increase the dehumidification ability. Applicability of such a combination in Singapore, a typical tropical country, was shown.

Air cleaning systems

As indoor environmental conditions become more and more important, implementing novel air cleaning technologies to purify the secondary air is another attractive research area. Taipale et al. [70] conducted a study on the combination of an ionizer as an air cleaning method with an active chilled beam terminal unit and found that there would be a reduction of 6% to 15% in the secondary air volume flow rate. Ardkapan et al. [71] evaluated the possibility of integrating an electrostatic filter and an active chilled beam terminal unit. It was shown that adding the filter accelerated the removal rate of the particles by 2 h^{-1} . Nevertheless, the capacity of the terminal unit was reduced by 38%.

2.6 Applications

Active chilled beam systems are not a silver bullet for all; they are more suitable to some particular application scenarios than others, such as healthcare facilities, laboratories, service centers, and offices etc. In conjunction with the scenarios, few studies have been done to address the practical effectiveness of the systems.

Using 100% outdoor fresh air, recycling of the air between different rooms is prevented in active chilled beam systems, which subsequently decreases the cross-infection risk and improves the indoor air quality. Those features are very desirable for healthcare facilities. Devlin et al. [72] carried out some CFD simulations as well as full-scale prototype testing to verify the applicability of active chilled beam systems for them. The systems were found to be an appropriate solution for hospital wards. The fresh air could be delivered without much energy penalties or compromise to air change effectiveness, while the indoor thermal comfort was still maintained. Similar conclusions were also obtained via an energy performance simulation of an elderly nursing building [73].

Being equipment intensive facilities, laboratories generally have heavy sensible cooling loads [74, 75]. Barnet et al. [76] illustrated the potential energy savings for cooling and heating laboratories by using a design that combined active chilled beam terminal units with a ventilation system with dual energy recovery. The energy savings of about 50% was confirmed via an hourly simulation. Memarzadeh et al. [77] contained a numerical simulation and empirical validation of applying active chilled beam systems in a laboratory. Thermal comfort in the occupied zone as well as removal effectiveness of the gases and airborne particles was examined. A reduction of around 22% was estimated in annual energy cost.

In addition, Darwich et al. [78] explained the holistic HVAC design of a service center. The project was able to achieve 38% energy savings with respect to the ASHRAE/Illuminating Engineering Society (IES) Standard 90.1-2004 baseline on an energy cost basis for both electricity and steam via a series of energy efficiency measures. Within the design, active chilled beam systems were used in a hybrid manner, under variable air volume mode, and as the most important energy saving technology. Brzezinski et al. [79] implemented an active chilled beam system for the retrofit of a

converted historic warehouse and machine repair facility into offices and laboratories. It was shown that the building would save 31.5% in energy cost over ASHRAE/IES standard 90.1-2007.

With the engineering practices, some application related problems have been considered as well. For example, the condensation problem is an essential concern to extending active chilled beam systems in hot and humid tropical regions. Kosonen et al. [80] conducted a case study to investigate feasibility of the systems in Singapore. The result showed that the condensation is possible to be prevented if infiltration is minimized, supply air flow rate is sufficient to extract the humidity of the occupants and tuning of the automation system has been probably conducted. Furthermore, Frenger [81] patented a condensation avoidance technology named DrypacTM, which was a coating applied to the secondary heat exchanger of the terminal units to enable dry operation below dew point. The coating consisted of a mineral hygroscopic material, perlite, and a binding agent. In addition, Eurovent and Air-conditioning Heating, & Refrigeration Institute (AHRI) published some certification programs for active chilled beam terminal units [82, 83].

2.7 Summary

This chapter presented a comprehensive overall review of state of the art of the research into active chilled beam systems. Research focusing on the terminal units, the systems, air flow patterns and indoor thermal comfort, combinations with other systems, and applications had been carried out and progress had been made. For simplicity, a framework as depicted by Fig. 2.2 is used to categorize the main research results.

Figure 2.2 Current state of research into active chilled beam systems

It can be observed that several issues of active chilled beam terminal units have drawn much attention, such as air flow patterns and indoor thermal comfort, while issues like design, performance evaluation, and modeling of the terminal units have been largely ignored by the research community. As illustrated in the figure, if origin of all the research can be represented by the dot at center of the pentagon while the research that is necessary to attain predictable optimal active chilled beam systems can be represented by the boundary, insufficient of the research can be found in all the five aspects. Thus, there is still a long way to go for building professionals and practitioners to make active chilled beam systems efficient and effective, and feasible in various conditions.

Chapter 3. Experimental active chilled beam terminal unit and setup

3.1 Introduction

To be sure, there exist a variety of active chilled beam terminal units for various applications. For instance, from the perspective of air discharge direction, a 4-way discharge terminal unit is usually preferable to a small personal office than a 2-way discharge one. When taking into account the other design options, such as the nozzle configuration, the secondary heat exchanger, and so on, variations of active chilled beam terminal units would become huge. As a consequence, it is unrealistic to present and investigate all the terminal units in this thesis. Instead, only a 2-way discharge active chilled beam terminal unit is focused on in the following studies.

Besides the terminal unit, a proper experimental setup is indeed needed for at least two reasons. Experimental verification of CFD simulations used to optimize the terminal unit is still lacking without the setup. On the other hand, the setup is necessary to make up the shortfall of CFD simulations. For example, it is quite difficult and time-consuming to conduct a CFD simulation that simultaneously contains the entrainment effect of the nozzles and the heat transfer process of the heat exchanger. In contrast, such an experiment is relatively practical and acceptable. Therefore, an experimental setup is constructed.

In this chapter, the experimental active chilled beam terminal unit is introduced in **Section 3.2**. It is followed by a description of the experimental setup in **Section 3.3**. **Section 3.4** gives a short summary for this chapter.

3.2 Experimental active chilled beam terminal unit

The experimental active chilled beam terminal unit is a ceiling mounted 2-way unit, customized with local tropical climates regarding the fresh air supply, sensible cooling capacity, and latent cooling capacity, etc. It is optimized using a series of CFD simulations. Particular attention is paid to the aerodynamic performance, especially of geometries of the casing and nozzle. The CFD simulations are not the focus of this thesis, so they are ignored. For simplicity, the detailed design of the terminal unit is directly

given in **Appendix A: Design of a 2-way discharge active chilled beam terminal unit** and the corresponding prototype is introduced here.

A prototype of the casing is shown in Fig. 3.1. The secondary heat exchanger supporters are made from 2 mm galvanized steel, while the remaining parts are all made from 0.8 mm galvanized steel. The face dimensions of the terminal unit are of 0.6 m×1.2 m and the height is 0.3 m.

Figure 3.1 Prototype of the casing of the experimental active chilled beam terminal unit

A prototype of the rubber nozzle is given in Fig. 3.2. The nozzle is made from fire retardant material and designed to be leak proof. Compared with metal nozzles which are directly punched on the nozzle plate, the rubber nozzles are superior in terms of acoustic performance. Thirty circular rubber nozzles of 9 mm inner diameter are evenly distributed on each side of the nozzle plate. In reality, the inner diameter can be varied to offer various fresh air supply, sensible cooling capacity, and latent cooling capacity, etc.

Figure 3.2 Prototype of the induction nozzle

A prototype of the secondary heat exchanger employed in the active chilled beam terminal unit is illustrated in Fig. 3.3. On the whole, the heat exchanger is a standard one mechanically expanded copper tube plus aluminum fins and the total 16 tubes are configured in the form of 2-rows staggered tube layout. Since the heat exchanger is a research focus in this thesis, several different heat exchangers are investigated. To make it easier to follow what's happening in the following chapters, the details of these heat exchangers will be presented in the corresponding chapters.

Figure 3.3 Prototype of the secondary heat exchanger

3.3 Experimental setup

A schematic diagram and the physical counterpart of the experimental setup are separately presented in Figs. 3.4 and 3.5. The setup consists of two physical loops: an air loop and a chilled water loop. The air loop is provided to blow air into the primary air plenum, force it through the nozzles, and entrain the room air through the secondary heat exchanger. The chilled water loop is designed to supply chilled water from a self-contained chiller system to the secondary heat exchanger.

Figure 3.4 Schematic diagram of the experimental setup

Figure 3.5 The experimental setup

In the air loop, two SAER ELETTROPOMPE 350 W centrifugal fans are installed in series to keep a certain gauge pressure in the primary air plenum. A Dwyer series MS Magnesense® differential pressure transmitter is used to measure the gauge pressure with error of $\pm 1\%$. Two intelligent vortex precession flowmeters are adopted for different measurement spans of the primary air volume flow rate with error of $\pm 1.5\%$ and the corresponding secondary air volume flow rate are measured by a TSI model 8710 DP-CALCTM micromanometer with error of $\pm 3\%$. Both the primary and secondary air states are assumed to be equal to the room state, which are also acquired by the TSI model 8710 DP-CALCTM micromanometer.

In the chilled water loop, a 750 W water circulating pump is placed in the cycle to circulate the chilled water between the chiller system and the heat exchanger. A float flowmeter is equipped for the measurement of the chilled water volume flow rate with error of $\pm 1.6\%$. The heat exchanger inlet and outlet chilled water temperatures are measured by two PT1000 platinum resistance temperature transmitters with error of $\pm 0.3\text{ }^{\circ}\text{C}$ and a Yokogawa EJA series differential pressure transmitter is installed between inlet and outlet ports of the heat exchanger.

The self-contained chiller system provides the chilled water by a vapor compressor cycle and stores it in an insulated water bath with a 4 kW immersion electrical heater. The desired temperature of the stored chilled water is controlled by a SHIMADEN Proportion Integration Differentiation (PID) controller. All motors, including the fans and pump, are equipped with VSD, so the fluid volume flow rates can be adjusted as needed. All the water pipes and fittings and temperature transmitters are thermally insulated properly to minimize the heat loss and measurement inaccuracy.

3.4 Summary

This chapter covered a general description of the experimental active chilled beam terminal unit and setup. Although the terminal unit is a customized one, subsequent explorations are still intended to provide some results, conclusions, and methods with generality. As for the experimental setup, it can be observed that most of the operating parameters can be varied as needed except from the inlet states of the primary air and secondary air. Although flexibility of experimental studies is restricted, aerodynamic, thermodynamic, and hydrodynamic performances of the terminal unit can be attained preliminarily. In addition, the studies originate from different point of views and then different experimental conditions or experimental procedures are required. Therefore, all the pertinent information was ignored in this chapter while the details will be presented in the corresponding chapters.

Chapter 4. A primary study on the heat exchanger circuit number for active chilled beam terminal units

4.1 Introduction

As suggested in **Chapter 2**, the secondary heat exchanger is an important concern of active chilled beam terminal units. It should not be a standard “off the shelf” component but has to be customized in conjunction with the terminal units. The customization can occur on many issues typically: fin shape, fin spacing, heat exchanger circuits, pipe diameter, etc. Once these issues are well addressed, performance of the heat exchanger, as well as performance of the terminal units, can be substantially enhanced. Considering that the sophisticated entrainment effect inside active chilled beam terminal units contributes a sensitive and unknown air side configuration for the heat exchanger, it is very challenging to optimize the fin shape or spacing. Conversely, optimization of the circuit arrangement is more practical. In reality, that is a basic but effective method to improve the heat exchanger. If the heat exchanger circuits are properly branched and joined, both the thermodynamic and hydrodynamic performances can be enhanced. More importantly, circuit arrangements can be adjusted without much effect on the air side operation or the heat exchanger compactness and complexity in structure.

Up to now, there has been a spate of interest in the technique. In the aspect of principle, Guo et al. [84] provided an idea that the high effectiveness of counter flow heat exchangers could be owing to the most uniform local temperature difference between two flowing fluids compared with other heat exchangers. The so-called uniformity principle of temperature difference was proposed, which became an important guideline for the optimization of heat exchanger circuit arrangements. Furthermore, Guo et al. [85] conducted a theoretical analysis and an experimental confirmation with thirteen types of heat exchangers to prove the uniformity principle. Cabezas-Gomez et al. [86, 87] also addressed the same confirmation numerically with some new flow arrangements. In the aspect of application, Wang et al. [88] carried out an experimental study including six 1-circuit and two 2-circuits heat exchangers to investigate the effect of circuit arrangements on the performance of wavy finned condensers. Liang et al. [89, 90] attempted a numerical and experimental study on the refrigerant circuit of evaporators. In addition,

Miura et al. [91] explored the effect of circuit arrangements on the pressure drop of thirty two plate heat exchangers.

Based on the above reality, this chapter presents a primary study on the heat exchanger circuit arrangement regarding the circuit number for active chilled beam terminal units. Supported by the two-way discharge terminal unit and experimental setup illustrated in **Chapter 3**, four 2-rows fin and tube heat exchangers, containing 1 circuit, 2 circuits, 4 circuits, and 8 circuits respectively, are investigated under a wide range of chilled water volume flow rates. Given a nominal air side operating condition, the chilled water side thermodynamic and hydrodynamic characteristics are evaluated. The heat transfer capacities are compared under three sets of criteria: identical chilled water volume flow rate, identical pressure drop and identical pumping energy consumption. To facilitate the understanding from a viewpoint closely related to heat exchanger theories, the heat exchanger effectiveness and performance index are also used as performance indicators [92]. It is showed that different circuit numbers should be preferred in different operating conditions and under different evaluation criteria, while the 2-circuits arrangement should be the most comprehensive and reasonable option rather than the 1-circuit one, which is widely used in all the market available active chilled beam terminal units.

The rest of this chapter is organized as follows: in **Section 4.2**, a theoretical analysis is given to reveal the fundamentals behind the study; a complete experimental investigation concerning the experimental procedure and conditions, assessment criteria and indicators, and uncertainties analysis and repeatability test of the experiments is provided in **Section 4.3**; it is followed by performance comparisons and discussions in **Section 4.4**; a summary of this chapter is drawn in **Section 4.5**.

4.2 Theoretical analysis

Prior to the experimental investigation, some fundamentals are briefly discussed in this section. That is not a complete description of thermodynamic and hydrodynamic characteristics of air water fin and tube heat exchanger, in general, but still contains a wealth of useful information, which is helpful to understand the experimental results.

The heat exchanger is firstly characterized by heat transfer capacity. The rate of heat transferred from the secondary air to the chilled water through a finite element can be simply written as:

$$dQ = hAdT \quad (4.1)$$

where dQ , h , A and dT are the local heat transfer capacity, heat transfer coefficient, heat transfer area, and temperature difference respectively.

The overall thermal resistance of such a finite element can be divided into four major parts: chilled water side convection, tube conduction, contact conduction (between the tube and fin) and air side convection thermal resistances. Except from the chilled water side convection thermal resistance, all the rest ones can be integrated together as a lumped thermal resistance R_l .

$$\frac{1}{hA} = \frac{1}{h_w A_w} + R_l \quad (4.2)$$

where h_w and A_w are the heat transfer coefficient and area on the chilled water side respectively.

Since the chilled water is driven mechanically by external forces, the heat transfer mechanism is forced convection. Referring to dimensional analysis, Nusselt number Nu so as to the force convection heat transfer coefficient for a single phase flow can be calculated by Reynolds number Re and Prandtl number Pr as:

$$Nu = \frac{h_w D_{t.ins}}{k_w} = a Re^b Pr^c = a \left(\frac{\rho_w u_w D_{t.ins}}{\mu_w} \right)^b \left(\frac{C_w \mu_w}{k_w} \right)^c \quad (4.3)$$

where $D_{t.ins}$ the tube inside diameter; k_w , ρ_w , u_w , μ_w and C_w are the chilled water conductivity, density, velocity, absolute viscosity and specific heat respectively; a , b and c are unknown constant coefficients. Although these properties depend on the water temperature, they are always evaluated at a bulk temperature thus avoiding iteration. Moreover, they are assumed to be constant, because variation of the water bulk temperature can be ignored.

If the water is measured by its volume flow rate $\dot{V}_w$, the water side heat transfer coefficient is then rewritten with an integrated constant coefficient d :

$$h_w = d * \dot{V}_w^b \quad (4.4)$$

$$d = \frac{k_w}{D_{t.ins}} a \left(\frac{\rho_w D_{t.ins}}{\mu_w} \right)^b \left(\frac{C_w \mu_w}{k_w} \right)^c \left(\frac{4}{\pi D_{t.ins}^2} \right)^b \quad (4.5)$$

Substituting Eqs. (4.2) and (4.4) into Eq. (4.1) results the following equation:

$$dQ = \left(\frac{1}{d * \dot{V}_w^b * \pi D_{t.ins} L_e} + R_l \right)^{-1} dT \quad (4.6)$$

where L_e is the length of the finite element. From Eq. (4.6), it can be observed that the local heat transfer capacity is affected by two variables: the chilled water volume flow rate and local temperature difference. If the total heat transfer capacity is considered in an integral form of Eq. (4.6), it will be affected by both the water volume flow rate and temperature difference distributions. However, these distributions are generally coupled in the sense that their variations by some design changes have opposite effect on the total heat transfer capacity. For example, using a multi-circuits heat exchanger instead of a 1-circuit one leads to higher and more uniform temperature differences but lower water volume flow rates in the circuits. Consequently, it is difficult to maximize the heat transfer capacity theoretically because it in essence involves the optimization of partial differential equations with some constraints, which is still an unsolved problem. That is also the reason why an experimental comparison is desired in this study.

Pressure drop is another important characteristic of the heat exchanger. The total pressure drop ΔP_w includes friction, curvature, flow velocity profile distortion and inherent static pressure drops. It can be expressed by an integrated Fanning friction factor f .

$$\Delta P_w = \frac{4fL_w}{D_{t.ins}} \frac{1}{2} \rho_w \left(\frac{4\dot{V}_w}{\pi D_{t.ins}^2} \right)^2 + \Delta P_{w.static} \quad (4.7)$$

where L_w and $\Delta P_{w.static}$ are the tube length and static pressure drop respectively. From Eq. (4.7), it can be seen that the chilled water volume flow rate and the water circuit length are important factors of the total pressure drop. Without the static pressure drop, increasing the heat exchanger circuit number will exponentially decrease the pressure drop. For example, comparing a 2-circuits heat exchanger with a 1-circuit one, the chilled water path length will be halved, as well as the chilled water volume flow rate, in each

circuit. Then the total pressure drop will be $1/8$. Nevertheless, the static pressure drop has to be taken into consideration and the Fanning friction factor is usually not a constant, which changes with Reynolds number, as well as the water volume flow rate.

4.3 Experimental investigation

Experimental procedure and conditions

As the main concern of this study is to investigate the optimal circuit number, four 2-rows fin and tube heat exchangers, containing 1 circuit, 2 circuits, 4 circuits and 8 circuits respectively, are tested. The circuit arrangements are schematically illustrated in Fig. 4.1. For these heat exchangers, this only difference can be easily attained by different headers.

Figure 4.1. Conventional 1-circuit (1) and multiple-circuits (2, 3, and 4) arrangements

The heat exchangers are in sequence implemented in the experimental terminal unit. With the experimental setup, the primary air plenum gauge pressure is specified at 85 Pa, a general operating condition. That results in constant primary and secondary air volume flow rates. The heat exchangers have exact the same air side configuration, so all uncertainties caused by the entrainment effect can be eliminated. During the experiments, all the other independent variables, including the room environmental temperature and

chilled water inlet temperature are tried to keep constant, except from the chilled water volume flow rate. The experimental parameters setting and their variations are summarized in Table 4.1.

Table 4.1 Summary of experimental parameters setting

Parameter	Set point	Practical range	Unit
Environmental temperature	24	23.7-24.1	℃
Secondary air volume flow rate	360	355-365	m ³ /h
Chilled water inlet temperature	14	13.8-14.1	℃
Chilled water volume flow rate	144-504	144-504	L/h

Besides the chilled water volume flow rate, the chilled water inlet and outlet temperatures and pressure drop are recorded. All the experimental data are obtained under steady state, so attention is focused on thermal equilibrium of the heat exchangers. Since the secondary air outlet temperature is not available, the thermal equilibrium is only determined via the chilled water loop. Once the chilled water outlet temperature is within ± 0.1 ℃ limits for 20 minutes, the steady state is confirmed.

Assessment criteria and indicators

As stated in the theoretical analysis section, the heat exchanger is basically characterized by the heat transfer capacity and pressure drop. However, it must be noted that, when the 1-circuit arrangement is replaced by the multiple-circuits arrangement, the objective is to increase the heat transfer capacity using a smaller amount of pumping power. Thus, such an assessment concerning pumping energy consumption is considered essential, while the pressure drop is intermediate measurement for the pumping energy consumption. In order to provide a comprehensive performance comparison, three widely used constraints are adopted: identical chilled water volume flow rate, identical pressure drop and identical pumping power. In addition, the heat exchanger effectiveness and performance index are used. Their characteristics are briefly displayed in the following:

- Heat transfer capacity (identical chilled water volume flow rate): the identical water volume flow rate means identical pumping energy consumed in a predefined water supply system.

- Heat transfer capacity (identical pressure drop): the pressure drop measurement is helpful if there is a pressure limitation, but there is no direct consideration of pumping energy consumption.
- Heat transfer capacity (identical pumping energy consumption): this is a direct pumping energy consumption consideration but limited to the heat exchanger itself.
- Effectiveness: it is non-dimensional and reflects the heat exchanger effectiveness. It agrees with the heat transfer capacity compared at identical chilled water volume flow rate.
- Performance index: it is non-dimensional and considers the total pumping energy consumption, while it enlarges the effect of the pressure drop in some sense.

The pressure drop is measured directly, but it is not the case for the other parameters. Therefore, the following equations are used to calculate the heat transfer capacity, as well as associated performance indicators. The heat transfer capacity Q equals to either the heat decrement of the air or heat increment of the water.

$$Q = -C_a \rho_a \dot{V}_{sec} (T_{sec} - T_{zone}) \quad (4.8)$$

$$Q = C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in}) \quad (4.9)$$

where $\dot{V}_{sec}$ is the secondary air volume flow rate; T_{sec} and T_{zone} are the secondary air off coil temperature and zone temperature respectively; $T_{w.out}$ and $T_{w.in}$ are the chilled water outlet and inlet temperatures respectively.

Assuming that the water pump efficiency is kept 80%, the pumping energy consumption P_w can be then calculated by:

$$P_w = \frac{\dot{V}_w \Delta P_w}{0.8} \quad (4.10)$$

The heat exchanger effectiveness ε is calculated from experimental observations using:

$$\varepsilon = \frac{Max[(T_{w.out} - T_{w.in}), (T_{zone} - T_{sec})]}{T_{zone} - T_{w.in}} \quad (4.11)$$

The performance index COP_w is defined as the ratio of heat transferred between the fluids to the pumping energy consumption.

$$COP_w = \frac{Q}{P_w} \quad (4.12)$$

Uncertainty analysis and repeatability test

In order to verify experimental results and evaluate reliability and accuracy of the measurements, it is necessary to carry out an uncertainties analysis and repeatability test and show that all the experimental data are within reasonable uncertainty limits. In reality, the uncertainties mainly come from two error sources: unfixed experimental conditions and measurement errors of the corresponding transmitters. Due to complex and unknown dependences, effects of the experimental conditions are difficult to be characterized or quantified. Furthermore, the experimental conditions summarized in Table 4.1 are almost constant. As a result, this error source is ignored and the uncertainty analysis is focused on the major errors caused by the transmitters.

Both independent and dependent variables experience uncertainties. The uncertainties occurred for independent variables are directly estimated from the accuracies of the transmitters. For the dependent variables, the uncertainties are obtained via the accuracies of multiple independent transmitters and the principle of propagation of uncertainty. Denoting the errors in the individual variables by x_n , error estimation of dependent variables U is made using the following equation:

$$U = \left[(s_1 x_1)^2 + (s_2 x_2)^2 + \dots + (s_n x_n)^2 \right]^{1/2} \quad (4.13)$$

where s_i is the sensitivity coefficient. The total uncertainties of independent and dependent variables are presented in Table 4.2.

Table 4.2 Summary of experimental variables' uncertainties in the water loop

Variable	Uncertainty value
Chilled water inlet temperature	± 0.3 (°C)
Chilled water outlet temperature	± 0.3 (°C)
Chilled water volume flow rate	± 1.6 (%) (± 0.0036 L/s)
Heat transfer rate	$\pm 69.3 \sim \pm 250.1$ (W)
Pressure drop	± 300 Pa ($\Delta p < 3$ kPa); ± 650 Pa (3 kPa $\leq \Delta p \leq 10$ kPa); ± 2 kPa (10 kPa $< \Delta p \leq 50$ kPa)

Figs. 4.2 and 4.3 illustrate experimental results of the heat transfer capacity and pressure drop repeatability tests for the 2-circuits arrangement heat exchanger. It can be easily observed that the practical heat transfer capacity should be reasonable. The uncertainty limits of $\pm 10\%$ are sufficient for 95% confidence level, so that occurrence of the invalid 5% experimental data can be treated as small probability event. The measured pressure drop is also almost all within the uncertainty limits. Therefore, the reliability and accuracy of the experiment results can be partially confirmed.

Figure 4.2 Heat transfer capacity repeatability test of the 2-circuits heat exchanger

Figure 4.3 Pressure drop repeatability test of the 2-circuits heat exchanger

4.4 Experimental results and discussions

Fig. 4.4 presents variations of the heat transfer capacity of the heat exchangers with different water circuits under different water volume flow rates. For any specified circuit arrangement, the increase of water volume flow rate yields a larger heat transfer coefficient, while the temperature difference distribution is almost unchanged, thus, a higher heat transfer capacity is achieved with reference to Eq. (4.6). For different circuit arrangements at a fixed water volume flow rate, the water is split into multiple circuits. Multiple-circuits arrangements contribute a reduced water volume flow rate in each circuit but a larger and more uniform temperature difference distribution. They conversely affect the heat transfer capacity and the effects cannot be quantified accurately. For example, 1-circuit arrangement has the highest heat transfer capacity when the water volume flow rate is less than 0.078 L/s, while 2-circuits arrangement has the highest heat transfer capacity when the water volume flow rate is more than 0.078 L/s. Comparing the heat transfer capacity at an identical chilled water volume flow rate, 1-circuit arrangement or 2-circuits arrangement should be recommended in different operating conditions.

Figure 4.4 Variations of the heat transfer capacity for different water circuits

Fig. 4.5 shows variations of the pressure drop of the heat exchangers with different water circuits under different water volume flow rates. As indicated in Eq. (4.7), the pressure drop increases with the increase of water volume flow rate for any given circuit

arrangements, due to the same water path length and static pressure drop. Comparing different circuit arrangements at the same water volume flow rate, multiple-circuits arrangements can decrease the length of each water path, as well as the water volume flow rate through it. It leads to lower pressure drops. The pressure drop of 8-circuits arrangement is lowest, followed by 4-circuits, 2-circuits and finally 1-circuit ones, while there is a reduction limitation contributed by the static pressure drop and friction factor adjustment with Reynolds number. Fig. 4.6 is derived to show variations of the heat transfer capacity under different pressure drop. It can be seen that 8-circuits arrangement has the highest heat transfer capacity when the pressure drop is lesser than 7.5 kPa; otherwise, 2-circuits one has the highest capacity. Thus, 8-circuit arrangement or 2-circuits arrangement is favored.

Figure 4.5 Variations of the pressure drop for different water circuits

Figure 4.6 Variations of the heat transfer capacity for different water circuits under different pressure drops

Fig. 4.7 illustrates variations of the heat transfer capacity of the heat exchangers with different water circuits under different pumping energy consumption. For any individual circuit arrangement, increase of the pumping energy consumption means the increase of water volume flow rate, as well as associated pressure drop, based on Eq. (4.10), while it improves the heat transfer capacity. For different circuit arrangements under same pumping energy consumption, multiple-circuits arrangements have to use higher water volume flow rate due to the relative lower pressure drop. As a result, multiple-circuits arrangements have the possibility to improve the heat transfer capacity according to Fig. 4.4, which depends on how much the water volume flow rate can be increased. As shown in Fig. 4.7, 4-circuits or 8-circuits arrangement has the highest heat transfer capacity.

Figure 4.7 Variations of the heat transfer capacity for different water circuits under different pumping energy consumptions

Figs. 4.8 and 4.9 present variations of the effectiveness and performance index of the heat exchangers with different water circuits under different water volume flow rates respectively. The characteristics are extremely similar to the ones shown in Figs. 4.4 and 4.5.

Figure 4.8 Variations of the effectiveness for different water circuits

Figure 4.9 Variations of the performance index for different water circuits

Summarizing above experimental results in Table 4.3, it can be concluded that different circuit numbers are recommended with respect to different operating conditions and evaluation criteria.

Table 4.3 Summary of circuit number recommendations

Heat transfer capacity (identical water volume flow rate)	Heat transfer (identical pressure drop)	Heat transfer capacity (identical pumping energy consumption)	Effectiveness	Performance index
1-circuit	8-circuits	4-circuits	1-circuit	8-circuits
($\dot{V}_w < 0.078 \text{ L/s}$)	($\Delta P < 7.5 \text{ kPa}$)	($P_{pump} < 0.015 \text{ W}$)	($\dot{V}_w < 0.078 \text{ L/s}$)	
2-circuits	2-circuits	8-circuits	2-circuits	
($\dot{V}_w \geq 0.078 \text{ L/s}$)	($\Delta P \geq 7.5 \text{ kPa}$)	($P_{pump} \geq 0.015 \text{ W}$)	($\dot{V}_w \geq 0.078 \text{ L/s}$)	

In practice, the chilled water side operating condition of active chilled beam systems is given in terms of water volume flow rate and pressure drop, which are typically between 0.03-0.12 L/s and lower than 30 kPa respectively [28, 93-95]. Since there is not a widely recognized optimum operating condition, it is nearly impossible to assure the best circuit

number accordingly. To be comprehensive, the 2-circuits arrangement is the most reasonable selection among these options in the full range of operating conditions. It can achieve competitive even higher heat transfer capacity and heat exchanger effectiveness, considerable pressure drop and pumping energy consumption reduction and improved performance index compared with the 1-circuit arrangement. Compared to the 4-circuits arrangement or 8-circuits one, it can offer significant heat transfer capacity and heat exchanger effectiveness enhancement with little penalty of pressure drop, pumping energy consumption and performance index.

4.5 Summary

In this chapter, a primary study on the secondary heat exchanger circuit number was presented. In the form of an experimental comparison, four 2-rows fin and tube heat exchangers with different circuit numbers were investigated under different water volume flow rates. Combining with a basic theoretical analysis, the thermodynamic and hydrodynamic performances were in-depth discussed using the heat transfer capacity, pressure drop, pumping energy, heat exchanger effectiveness and performance index. It was found that the 2-circuits arrangement could offer a competitive heat transfer capacity with a considerable lower pressure drop compared with the widely used 1-circuit arrangement. This unexpected conclusion meant a great deal of potential improvement for the existing active chilled beam terminal units. More importantly, it was confirmed that performance of the secondary heat exchanger could be enhanced via an advisable heat exchanger circuit arrangement, which would have significant and practical influences on the heat exchanger design for the terminal units in the future.

Chapter 5. Further study on the heat exchanger circuit connecting sequences for active chilled beam terminal units

5.1 Introduction

In the last chapter, the heat exchanger circuit number has been determined as two, while the best tube connecting sequences for the circuits are still yet to be resolved. In fact, the tube connecting sequences essentially affect the heat exchanger performance, particularly in the case of air mal-distribution. If the air passing through the heat exchanger is not uniformly distributed, performance of the heat exchanger may dramatically degenerate. For example, Chwalowski et al. [96] experimentally observed that the evaporator capacity degenerated as much as 30% due to a substantial air mal-distribution. More recently, Kaern et al. [97] showed that coefficient of performance of a residential air-conditioning system decreased as much as 43% because of the same problem. Rather than actively controlling the fluids supplied to the heat exchanger [98-100], this kind of performance degradation can be recovered in a passive manner as long as the tubes are well connected and the fluids are appropriately paired at each location. That is also confirmed effective via a series of studies by Domanski et al. [101-103]. Lee et al. [104] also examined an experiment-aided simulation study to optimize the refrigerant circuit arrangement of a roof top air conditioning unit with in-situ air velocity profile and found that nearly 8% heat transfer capacity increment could be achieved.

In addition, more and more applications of advanced algorithms [105-107] to optimize the heat exchanger circuit arrangement suggested the potential benefits. For instance, Wu et al. [107, 108] proposed an improved generic algorithm containing one-dimensional representation string, correction operator, knowledge-based generation, greedy crossover, greedy mutation, and all previous populations based selection to optimize the heat transfer capacity or the material cost. It was showed that 2.8-7.4% heat transfer capacity increment could be obtained or the joint tube length could be reduced by 0-40% for the same heat transfer capacity. Yashar et al. [109] employed a dual-mode knowledge-based and symbolic learning based evolutionary algorithm incorporated into an intelligent system for the heat exchanger design. The result also demonstrated the ability to generate circuit

architectures with the heat transfer capacities superior to those conventionally designed heat exchangers by 2.6-6.5%.

Back to active chilled beam terminal units, the secondary air across the heat exchanger is driven by the sophisticated entrainment effect, so the air velocity profile should be more or less non-uniform. Up to now, this air mal-distribution has never been analyzed, let alone the resultant heat exchanger performance influences. As a consequence, this follow-up study tries to address a quantitative analysis on these problems and then determine the best tube connection sequences. Different from the previous study on the circuit number, there are 16! types of tube connecting sequences for the heat exchanger. It is definitely impossible to test all the variations even a small part of them, so this study is implemented in the form of experiment-aided simulations. Given the same air side operating condition in last chapter, in-situ air velocity profile across the heat exchanger is measured and non-uniformities of the air flow are detected. Taking the air mal-distribution into consideration, heat transfer performance of the heat exchanger is simulated with a tube to tube distributed parameter model. This simulation model is verified with experimental results by selecting appropriate correction factors to the heat transfer coefficients obtained via some published correlations. The heat exchanger tube connecting sequences are then optimized through a particle swarm optimization program for maximum heat transfer capacity in different water side operating conditions. The potential pressure drop, manufacture difficulties, and material cost are also qualitatively analyzed. Finally, a new water circuit arrangement is proposed, which can provide better performances than the existing one on all the aspects.

This chapter is organized as below: in **Section 5.2**, a tube by tube distributed parameter heat exchanger model and associated solution procedure are developed; in **Section 5.3**, the experimental procedure and results are presented; the simulation model is verified in **Section 5.4** and hence forth brings the tube connecting sequences determination; a summary is finally drawn in **Section 5.5**.

5.2 Simulation model

This simulation model is a tube by tube distributed parameter one. Only then can the tubes be distinguished and individually analyzed. The main concern is to simulate the heat

transfer capacities of the tubes. Therefore, each tube and assembling fins are herewith together selected as a control volume as shown in Fig. 5.1. Without loss of generality, the following assumptions are applied when modeling:

1. The heat exchanger is always at dry condition.
2. The flow and thermal properties of the air and chilled water are independent of temperature.
3. The heat conduction between neighboring tubes is neglected.
4. The tubes are adiabatic in the parts of return bends and branch joints.

Figure 5.1 A control volume: a tube with fins

The first assumption is reflecting the general operation requirement of any active chilled beam terminal unit. Condensation on the heat exchanger surface is strictly avoided. The second one is to simplify the modeling process. The flow and thermal properties variances of the air and chilled water are very small because their temperature variances are limited. Similarly, the heat conduction ignored in the third assumption is also quite small because of the limited temperature difference between the tubes. The last assumption is very common in developing a heat exchanger simulation model, which is actually an acceptable approximation to the practical situation.

Equations of each control volume

According to energy conservation, the governing equation for each control volume can be expressed as:

$$Q_w = -Q_a = Q_{hA} \quad (5.1)$$

where Q_w and Q_a are the rates of enthalpy increase on the chilled water and air sides; Q_{hA} is the heat transfer rate from the air to the chilled water and also the heat transfer capacity of the heat exchanger.

Among them, the first two variables are calculated based on their definitions.

$$Q_w = C_w V_w \rho_w (T_{w.out} - T_{w.in}) \quad (5.2)$$

$$Q_a = C_a V_a \rho_a (T_{a.out} - T_{a.in}) \quad (5.3)$$

where C_w , V_w and ρ_w are the specific heat at constant pressure, volume flow rate, and density of the chilled water; $T_{w.out}$ and $T_{w.in}$ are the outlet and inlet temperatures of the chilled water; C_a , V_a , ρ_a , $T_{a.out}$ and $T_{a.in}$ are the counterparts on the air side.

The third one can be computed via arithmetic average temperature difference method, log mean temperature difference method as well as $\varepsilon-NTU$ method. To avoid iterative calculation, $\varepsilon-NTU$ method which only depends on the inlet conditions is selected. Considering that the minimum heat capacity is always on the air side in practical operation, the number of heat transfer units (NTU) in $\varepsilon-NTU$ method is defined as:

$$NTU = \frac{hA}{C_a V_a \rho_a} \quad (5.4)$$

where h and A the overall heat transfer coefficient and associated area.

Then, the heat exchanger effectiveness ε can be obtained by the $\varepsilon-NTU$ relation for unmixed-unmixed cross flow [110].

$$\varepsilon = 1 - \exp \left\{ \frac{NTU^{0.22}}{C^*} \left[\exp(-C^* NTU^{0.78}) - 1 \right] \right\} \quad (5.5)$$

where

$$C^* = \frac{C_a V_a \rho_a}{C_w V_w \rho_w} \quad (5.6)$$

With this heat exchanger effectiveness, the third heat transfer capacity is determined by:

$$Q_{hA} = \varepsilon Q_{\max} = \varepsilon C_a V_a \rho_a (T_{a.in} - T_{w.in}) \quad (5.7)$$

However, the total heat transfer coefficient hA has to be acquired before the calculations of Eqs. (5.4-5.6). In terms of overall heat transfer resistance, the total heat transfer coefficient is represented by:

$$\frac{1}{hA} = \frac{1}{h_w A_w} + \frac{D_{t,outs}}{2k_t A_t} \ln \frac{D_{t,outs}}{D_{t,ins}} + \frac{D_{f,outs}}{2k_f A_f} \ln \frac{D_{f,outs}}{D_{t,outs}} + \frac{1}{\eta_a h_a A_a} \quad (5.8)$$

where h_w and A_w are the heat transfer coefficient and area inside the tubes of the heat exchanger; k_t and A_t are the heat conductivity and heat transfer area of the tubes; k_f and A_f are the heat conductivity and heat transfer area of the assembling fins; η_a , h_a and A_a are the surface efficiency, heat transfer coefficient and area outside the fins; $D_{t,ins}$, $D_{t,outs}$ and $D_{f,outs}$ are the inside and outside diameters of the tubes and the total diameter including the fins.

Parameters concerning the heat exchanger dimensions can be measured or obtained from the manufacturer, while the heat transfer coefficients have to be evaluated from appropriate published correlations in conjunction with corrections. On the chilled water side, the following correlation is used for the Nusselt number Nu and subsequently for the chilled water side heat transfer coefficient when $Re < 2300$.

$$Nu_w = 3.66 + \frac{0.0668 Re_w Pr_w (D_{t,ins} / L_t)}{1 + 0.04 [Re_w Pr_w (D_{t,ins} / L_t)]^{2/3}} \quad (5.9)$$

where Re_w is the Reynolds number on the chilled water side; Pr_w the Prandtl number on the chilled water side; L_t is the length of the tubes.

In the case of $Re > 3000$, the Gnielinski's equation [111] is adopted.

$$Nu_w = \frac{(f_w / 2)(Re_w - 1000)Pr_w}{1 + 12.7(f_w / 2)^{1/2}(Pr_w^{2/3} - 1)} \quad (5.10)$$

where f_w is the Fanning friction factor that can either be obtained from Moody chart or for smooth tubes from correlation:

$$f_w = \frac{1}{4} (0.79 \ln(Re_w) - 1.64)^{-2} \quad (5.11)$$

In the remaining transition flow situations, a linear interpolation method is employed between the two boundary values respectively obtained via Eq. (5.9) and Eq. (5.10).

On the air side, in order to get the heat transfer coefficient, the Colburn factor is calculated via the correlation of sine wavy fin under dry surface condition developed by Youn et al. [112].

$$j_a = 0.2199 \text{Re}_a^{-0.318} \left(\frac{P_f}{D_{f,outs}} \right)^{0.309} \left(\frac{x_f}{P_d} \right)^{0.163} \quad (5.12)$$

where P_f , P_d and x_f are the fin pitch, waffle height and projected fin pattern length for one-half wave length respectively.

In addition, the following non-dimensional groups are necessary:

$$\text{Re} = \frac{\rho v D}{\mu} \quad (5.13)$$

$$\text{Nu} = \frac{h D}{k} \quad (5.14)$$

$$j = \frac{\text{Nu}}{\text{Re Pr}^{1/3}} \quad (5.15)$$

The surface efficiency can be written in terms of the fin efficiency η_f , fin surface area A_f and total surface area A_{total} , as follows:

$$\eta_a = 1 - \frac{A_f}{A_{total}} (1 - \eta_f) \quad (5.16)$$

where $A_{total} = A_f + A_b$ is the total surface area not a projected one. For simplicity, the fin efficiency is calculated by using the approximation method as described by Schmidt [113].

$$\eta_f = \frac{\tanh(m r_{f,outs} \theta)}{m r_{f,outs} \theta} \quad (5.17)$$

where

$$m = \sqrt{\frac{2h_a}{k_f \delta}} \quad (5.18)$$

$$\theta = \left(\frac{r_{eq}}{r_{f,outs}} - 1 \right) \left[1 + 0.35 \ln \left(\frac{r_{eq}}{r_{f,outs}} \right) \right] \quad (5.19)$$

and

$$\frac{r_{eq}}{r_{f.outs}} = 1.27 \frac{X_M}{r_{f.outs}} \left(\frac{X_L}{X_M} - 0.3 \right)^{1/2} \quad (5.20)$$

$$X_L = \sqrt{(P_t / 2)^2 + P_t^2 / 2} \quad (5.21)$$

$$X_M = \frac{P_t}{2} \quad (5.22)$$

Constraints between the control volumes

Constraints between the control volumes are actually the connection conditions between them. Since the heat exchanger with either inner divergences or confluents is not considered here, the simulated heat exchanger is just consisted by a series of control volumes from inlet to outlet on the chilled water side. Even if the heat exchanger may have multiple circuits with multiple inlets and outlets, the series configuration for each circuit is kept unchanged and then these circuits can be analyzed separately. The chilled water volume flow rates through them are equal, while for each tube the chilled water outlet temperature serves as the inlet temperature of the next tube.

$$\dot{V}_{w.downs} = \dot{V}_{w.ups} \quad (5.23)$$

$$T_{w.in.downs} = T_{w.out.ups} \quad (5.24)$$

On the air side, the connection conditions are a bit more complicated due to the staggered arrangement. The air volume flow rate is one-half of the sum of the air volume flow rates applied to the two upstream neighboring control volumes and the inlet temperature is the weighted average value based on the volumes flow rates.

$$\dot{V}_{a.downs} = \frac{\dot{V}_{a.ups.1} + \dot{V}_{a.ups.2}}{2} \quad (5.25)$$

$$T_{a.in.downs} = \frac{(T_{a.out.ups.1} \dot{V}_{a.ups.1} + T_{a.out.ups.2} \dot{V}_{a.ups.2})}{2\dot{V}_{a.downs}} \quad (5.26)$$

Solution procedure

Taking above equations of control volumes and their constraints together as a whole, it is actually difficult to develop an effective algorithm to solve those coupled nonlinear conversation equations in a short time. Conventional finite difference method [114-118] is

still time-consuming even if some methods are used to decouple the equations and relax the nonlinearities, particularly in the case of a large number of control volumes. In the present study, there are 16 control volumes. Thus, particle swarm optimization is utilized in solving the proposed model, which is faster and more robust to initial values. More information about the algorithm is presented in **Appendix B: Particle swarm optimization**.

Logical flow chart of the model solution procedure is illustrated in Fig. 5.2. It begins with a set of predetermined inlet parameters, including inlet temperature and volume flow rate of the chilled water and inlet temperature and volume flow rate distribution on the front row of the air. By assuming the heat transfer capacities for the tubes, the chilled water outlet temperatures can be calculated based on Eq. (5.2) and Eq. (5.24) from the first to the last tube along the chilled water flow path. In a similar manner, the outlet temperatures of air can be derived based on Eq. (5.3) and Eqs. (5.25-5.26) from the first to the last row. Then, the heat transfer capacities can be validated by submitting the inlet conditions into the $\varepsilon-NTU$ model. As long as the convergence condition is not satisfied, the assumed heat transfer capacities will be adjusted via the particle swarm optimization algorithm.

Figure 5.2 Logical flow chart of the model solution procedure

Suppose there are in total n tubes, the searching space of heat transfer capacities is n -dimensional. A population of m candidate solutions, $Q_i = (q_{1i} \ q_{2i} \ \cdots \ q_{ni}) \ i=1 \ 2 \ \cdots \ m$, here dubbed particles, are firstly defined. Movements of these particles are influenced by their own best positions as well as the population's best position and expected toward the best position in the searching space. These so-called best positions are judged according to the fitness of the particles, which can be calculated via the designated objection function.

$$fitness = \sum (Q_i - Q_{i,cal})^2 \quad (5.27)$$

5.3 Experimental investigation

The fin and tube heat exchanger employed in the experimental terminal unit is the standard one mechanically expanded copper tubes plus aluminum fins, which has been selected in the last chapter. The total 16 tubes are configured in the form of 2-rows staggered tube layout and the tube connecting sequences are defined in Fig. 5.3. The tubes are numbered from 1 to n from the front row to the back row and from the bottom column to the top column. The fins are of sine wavy geometry. Compared with plain fin, air side heat transfer coefficient for sine wavy fin is higher, but with a normally prohibitive friction factor penalty. Detailed geometric parameters of the heat exchanger are tabulated in Table 5.1.

Figure 5.3 Heat exchanger schematic drawing (unit: mm)

Table 5.1 Summary of heat exchanger parameters

Geometric parameters	Values
Tube number in each row	8
Transverse tube spacing	31.75 mm
Longitudinal tube spacing	27.5 mm
Outer tube diameter	12.75 mm
Tube thickness	0.35 mm
Fin thickness	0.15 mm
Fin pitch	4.08 mm
Waffle height	3 mm
Projected wave length	13.75 mm
Heat exchanger length	1065 mm
Heat exchanger width	269.8 mm
Heat exchanger height	55 mm

Air velocity distribution measurement

With the experimental setup given in **Chapter 3**, the secondary air volume flow rate can be measured, while the secondary air velocity distribution cannot be captured. Thus, an extra single-point air velocity transducer is used to scan over the flow domain. Compared to the widely used particle image velocimetry technique, this method may be less accurate, time consuming, and unable to reflect instantaneous flow structures. However, it has to be pointed out that the secondary air velocity distribution to be described is driven by the entrainment effect in a passive manner. In this case, if the particle image velocimetry method is utilized, a large environmental chamber capable of containing the whole terminal unit as well as sufficient space for the secondary air to be induced has to be built. Moreover, the chamber has to be fully filled with tracking particles. For the ease of measurement, the single- point air velocity transducer method is obviously a priority.

A TSI air velocity transducer Model 8475 is adopted, which is suitable for universal direction measurement with a spherical probe. The field selected velocity range is set from 0.05 m/s to 1m/s and the accuracy is 3% of the reading plus 1% of the selected full scale range. As illustrated in Fig. 5.4, the transducer is horizontally attached on the casing. The probe is about 15 mm above the heat exchanger. In reality, there is a grille over the

heat exchanger. To eliminate its influences on the velocity distribution, it is removed during the experiments. In addition, the secondary air is assumed to pass the heat exchanger vertically.

Figure 5.4 Air velocity measurement

It should be noted that the terminal unit is facing upwards in the experiments. Then the cooling process of the secondary air increases the air velocity because of the variation of air density, which is a contrast to the practical situation where the terminal unit is facing downwards. In order to alleviate this influence, the secondary air velocity distribution is measured under an isothermal condition. The primary air is supplied at the conditioned-zone temperature while the chilled water supply is shut down. The primary air plenum gauge pressure is still specified at 85 Pa. In addition, the existing air-conditioning system is turned off for a still environment.

Based on the symmetry of the terminal unit, the secondary air velocity distribution is assumed to be symmetrical as well. As a consequence, the flow domain to be scanned is selected as shown in Fig. 5.5. Within this domain, total 15*6 (90) measurement points are defined. To obtain the air velocity distribution accurately, the distances between any two

adjacent measurement points should be small enough to reflect variations of the air velocities while big enough to minimize influences of the probe itself. The probe diameter is about 3 mm and then the distances are set at 10 mm, 20 mm, 30 mm, 32.5 mm, 34.9 mm, or 40 mm. Since the air velocity gradients near to the edges may be bigger, the measurement points there are denser.

Figure 5.5 Air velocity measurement points (unit: mm)

Output time constant of the transducer is set at its maximum value, 10 seconds, and the reading frequency is 2. Each velocity value is averaged over a two minutes period. The 90 measurement points are scanned twice and the averaged value is regarded as valid.

Air velocity distribution

A three-dimensional velocity distribution map, Fig. 5.6, is obtained by overlaying the measurements on the map and linearly interpolating the velocities between points within the scanned domain. The rest part is derived based on the symmetry property. The map shows that there is a considerable amount of variation in the air velocity distribution passing through the heat exchanger. The highest air velocity reaches 0.75 m/s while the lowest one is 0.36 m/s and the system uncertainties are 4.3% and 5.8% respectively. In

addition, this result confirms what is suspected: the air velocity gradient near to the edges is bigger.

Figure 5.6 Velocity distribution map (unit: m/s)

To capture influences of this air mal-distribution on the heat exchanger heat transfer performance and integrate this information into the tube connecting sequences optimization, the three-dimensional air velocity map is transformed into a two-dimensional velocity profile. The simulation model interprets the heat exchanger in a tube to tube way, so the two-dimensional velocity profile is obtained by laterally averaging the velocities. An integral average is utilized here rather than an arithmetic average of the experimental results, as the measurement points are not equally distributed. The resulting profile shown in Fig. 5.7 is therefore arranged as the air velocity distribution passing the heat exchanger.

Figure 5.7 Two-dimensional velocity profile (unit: m/s)

5.4 Simulation investigation

Simulation model verification

The simulation model established in the simulation model section is implemented on the investigated heat exchanger. The same operating conditions, including the air velocity distribution, the inlet temperatures of the air and chilled water, are applied. To avoid overmuch simulation cases, the chilled water volume flow rates are set at 0.06, 0.1, and 0.14 L/s. The resultant heat transfer capacities are simulated and the simulation results are illustrated in red in Fig. 5.8.

Figure 5.8 Simulated heat transfer capacities

It can be seen that there exist some errors between the model predicted heat exchanger capacities and experimentally measured ones. These errors are actually caused by many factors: 1) there are some inaccurate experimental measurements, although they are alleviated via curve fitting of the experimental results; 2) the heat transfer coefficient correlations adopted in the simulation model inevitably under or over predict actual performance due to the different enhanced geometries, particularly in the case of air mal-distribution, low air speed, sine wavy fin with large tube, etc.; 3) the assumptions defined in the modeling process may introduce some minor errors; 4) some significant errors may come from the fact that the secondary air velocity distribution used in the simulation is measured under an isothermal condition, while the air cooled by the heat exchanger would sink in the experiments and the velocities would be increased.

In order to remedy the errors so that this simulation model can be used to analyze local heat transfer capacities of the tubes and optimize the tube connecting sequences, correction factors are directly brought for the air side and chilled water side heat transfer coefficients with the reference to [104], correction factors are brought for the main error sources, the air side and chilled water side heat transfer coefficients, to improve accuracy of the simulation model. An iterative approach is used to determine correction factors

during this process. After that, Eq. (5.8) is incorporated with the correction factors e_w and e_a as below:

$$\frac{1}{hA} = \frac{1}{e_w h_w A_w} + \frac{D_{t.outs}}{2k_t A_t} \ln \frac{D_{t.outs}}{D_{t.ins}} + \frac{D_{f.outs}}{2k_f A_f} \ln \frac{D_{f.outs}}{D_{t.outs}} + \frac{1}{\eta_a e_a h_a A_a}$$

An iterative approach is used to determine correction factors during this process. The correction factors for the air side and chilled water side take turns to be increased with a step increment of 0.05 until the difference between the calculated heat transfer capacity and the measured one is less than 5 W. Since there should be additional error due to the change of secondary air velocity distributions on the air side as analyzed above in the fourth bullet, the maximum value of the correction factor for the chilled water side is limited to 1.2 while the extra correction accomplished by the air side factor is assumed to be dedicated to this additional error. Given the three typical chilled water volume flow rates, the heat transfer capacities of tuned simulation results and experimental ones are exactly matched in Table 5.2.

Table 5.2 Summary of model correction factors

Chilled water volume flow rate (L/h)	Heat transfer coefficient correction factor		Heat transfer capacity (W)	
	Air side	Water side	Model predicted	Experimental
216	1.15	1.1	624	623
360	1.30	1.2	758	761
504	1.45	1.2	850	852

In order to highlight general characteristics of the heat transfer capacity variation caused by the non-uniform air velocity distribution and achieve a better understanding on the performance of active chilled beam terminal units, one of the simulation cases for the chilled water volume flow rate 0.1 L/s is chosen for analysis. Fig. 5.9 shows distribution of the heat transfer capacities for each tube. It is seen that the heat transfer capacity varies greatly for the tubes. The maximum and minimum heat transfer capacities are 32 W and 63 W respectively.

Figure 5.9 Heat transfer capacity distribution

Tube connecting sequences optimization

Without any ideas about the optimal tube connecting sequences or the maximum heat transfer capacity, the original tube connecting sequences and heat transfer capacities of the tubes can't be fairly assessed. As a result, the tube connecting sequences are to be optimized in this section. Before that, each tube of the heat exchanger has to be clearly identified. A one-dimensional integer string is developed to represent the tube connecting sequences. Taking the original tube connecting sequences shown in Fig. 5.3 as an example, it can be represented by:

$$X = \{16, 15, 8, 7, 14, 13, 6, 5, 12, 11, 4, 3, 10, 9, 2, 1\}$$

The first 8 nodes are for the circuit 1, while the remaining 8 ones are for circuit 2. In this way, any circuitry arrangement can be explained accordingly.

Figure 5.10 Logical flow chart of the circuit optimization procedure

The particle swarm optimization is redefined in a discrete form in **Appendix B: Particle swarm optimization** and it is employed for the optimization achieving the maximum heat transfer capacity. Comparing to traditional genetic algorithm, it can be performed more easily without selection, crossover, or mutation, and the generation directly evolves according to existing optimum solutions. By introducing concept of basic swap sequence, addition, and subtraction of two particle positions can be uniquely interpreted. Therefore, the population evolves in a slight different way. Logical flow chart of the optimization is illustrated in Fig. 5.10. It begins with a set of randomly initialized circuitry arrangements as particles and basic swap sequences as their velocities. These particles are then submitted into the verified simulation model and the heat transfer capacities are calculated. As long as the maximum generation number is not reached, the

particles and the velocities are updated. The fitness function for obtaining the maximum heat transfer capacity is:

$$fitness = \frac{1}{Q}$$

The optimization in each chilled water volume flow rate is performance 10 times with 50 designs per generation and 200 generations. The best performed designs are selected and the results are summarized in Table 5.3. It can be observed that the heat transfer capacities in various operating conditions can be improved by 1.9-3%. These increments seem to be a bit low. There are two main reasons: the investigated heat exchanger is an air-water heat exchanger rather than an air refrigerant one, so the maximum temperature difference between the fluids is just 10 °C. On the other hand, the air velocity distribution across the heat exchanger is quite non-uniform, but the velocities are relatively small. The heat transfer capacity improvement potential via the optimization is finally limited.

Table 5.3 Optimized circuit arrangements and the performances

Chilled water volume flow rate (L/h)	Optimized design	Heat transfer capacity (W)	improvement
216	{12,14,13,16,7,6,2,5,10,11,15,9,1,4,8,3}	642	3%
360	{11,15,14,16,1,8,4,7,9,10,13,12,3,2,6,5}	774	1.7%
504	{12,10,16,6,15,1,3,13,8,2,11,14,4,9,7,5}	876	2.8%

In this case, the heat exchanger tube connecting sequences have to be determined in the other aspects, including the pressure drop, manufacture difficulties, material cost, and so on. Above tube connecting sequences are relatively complex and there are some overlong and crossover tube bends and branch joints, so they obviously consume higher pressure losses. For the same reason, their manufacture difficulties and material costs are inevitably increased. Back to the original tube connecting sequence, the structure is simple, which achieves a bit lower heat transfer capacity but lower pressure loss, lesser manufacture difficulties and material cost. Furthermore, a new one is proposed in Fig. 5.11. The heat transfer capacities of the present tube connecting sequence are superior to that of the original one under various chilled water volume flow rates and the comparison

is summarized in Table 5.4. In comparison, circuit entrances and circuit exits of the proposed one are closer to each other, so the heat exchanger header can be substantially simplified. Then the pressure drop across the header is considerably reduced and the material cost for this header can be saved by 80%. The pressure drops and material costs of the other tubes and bends are kept same. The manufacture difficulties of them are almost equal.

Figure 5.11 Proposed circuit arrangement

Table 5.4 Proposed circuit arrangement and its performance

Chilled water volume flow rate (L/h)	Proposed design	Heat transfer capacity (W)	improvement
216		630	1.0%
360	{13,14,15,16,8,7,6,5,12,11,10,9,1,2,3,4}	770	1.2%
504		859	1.1%

5.5 Summary

In the present study, the heat exchanger circuit arrangement for active chilled beam terminal units was further investigated in terms of tube connecting sequences. Provided the fixed primary air plenum gauge pressure, the secondary air mal-distribution was for the first time found and the air velocity varied from 0.36 m/s to 0.75 m/s. The resultant heat transfer capacities of the individual tubes were also non-uniform, from 32 W to 63 W. The optimized circuit arrangements were able to increase the capacities by 1.9-3% under various chilled water volume flow rates while carried some penalties on the potential pressure drops, manufacture difficulties, and material costs. Therefore, a simple circuit

arrangement was proposed, which was proven to be better and more comprehensive. This refinement of the circuit arrangement has some practical significances for active chilled beam terminal units. Without any doubts, there can be a variety of operating conditions besides the ones considered in the simulation. This study is not intended to provide a definitive conclusion of the optimal heat exchanger circuit arrangement, tube connecting sequences, but to illuminate possible performance effects and considerations in this aspect. To be sure, the result obtained here are helpful for understanding and refining active chilled beam terminal units and the method used in the present study is easily extended to any other relevant terminal units.

Chapter 6. A hybrid dynamic modeling of active chilled beam terminal units

6.1 Introduction

Except from the optimization of the heat exchanger conducted in the last two chapters, modeling of active chilled beam terminal units is also an urgent research need. It is necessary to correctly design and implement the terminal units as part of building mechanical systems and to the development of control and optimization strategies, while there has been no dedicated study on this issue until now. Such a weakness has probably been the biggest obstacle to promote applications of active chilled beam systems. Fortunately, in order to obtain an appropriate model of the terminal units, there have been some references on the entrainment effect and heat transfer process, which may be incorporated into the modeling.

The entrainment effect within the terminal units is essentially a turbulent merging confined jet, which is very complicated. In most cases, it has to be explored with the assistance of massive CFD simulations and experiments. Morton et al. [119] explained the entrainment effect as an incorporation of fluid from the surrounding into the jet by turbulent eddies generated by the shear existing between the two regions. Since the turbulent energy dissipation can't be measured, some empirical hypotheses are employed to close the equation system in jet models. Based on the entrainment hypothesis and spreading hypothesis, Enjalbert et al. [120] and Wang [121] individually derived the entrainment effect models for single turbulent jet from conservations of momentum and mass. Given the Reichardt's hypothesis, the model becomes more powerful and flexible. Hodgson et al. [122] and Wang et al. [123] demonstrated some models of complex processes involving complicated boundary conditions and multiple jets by the method of superposition of particular solutions. All these models required the details of system configurations, such as nozzle dimensions, arrangements and so on. More unfortunately, the independent variables of these models, spreading constant and virtual point, should be determined empirically with a lot of experiments as well. As a result, the models are generally useful in active control of the jets. Targeted at modeling of an existing terminal unit rather than at developing innovative terminal units, such kind of confined air jet

models are unsuitable. As a consequence, the following model captures the entrainment effect via an empirical model.

With respect to the heat transfer process, a wide range of models is currently available. Lebrun [124] and Brandemuehl [125] developed two theoretical heat exchanger models for ASHRAE HVAC Toolkits. The models required dimensions of the fin and the tube thickness, diameter, and spacing as inputs in order to calculate the heat transfer coefficients but presented insufficient robustness. Stoecher [126] proposed an empirical model, which could predict the performance of heat exchangers regardless of the type of fluid in the tubes with a given set of constants. Its defect was that for different fluids different sets of constants were needed. Either theoretical or empirical modeling approach has some inherent disadvantages, while a hybrid modeling approach takes advantages of both theoretical and empirical modeling approaches. It can offer acceptable accuracy and robustness with simple models. The model structures are derived from first principles, while the unknown parameters are identified by catalog or experiment data. A variety of models have been achieved in this way. Braum et al. [127] established an effective model for heat exchangers via introducing the concept of air saturation specific heat. Rabehl et al. [128] relaxed some of the assumptions and complications in Braum's model and captured the effect of geometry on performance. Furthermore, the model was simplified to several unknown lumped parameters without any geometry descriptions in references [129-133].

It should be noted that most of the existing heat exchanger models referred above are usually static models. They are not sufficient in many cases. For example, compared to controlling the fluid flow rate, controlling the exit temperature of the cold fluid with dynamic models is better to maximize the cooling capacity of heat exchangers. Meanwhile, the condensation can be strictly avoided if the transient behavior can be predicted in active chilled beam terminal units. In a word, a dynamic model is more convenient in the terminal units. The first study on the dynamic description of fin and tube heat exchangers was attempted by Shah [134]. General formulations of different problems, novel approaches and various techniques and so on were addressed step by step [135-137]. However, these dynamic models were developed for computer simulation, which were too complex to be applied for control and optimization purposes. Wang et al. [138] presented a nonlinear dynamic model and designed a nonlinear controller, but the model

performance was compromised because of ignoring several important heat transfer properties, such as variations of the heat transfer coefficients, heat storage of the heat exchanger and so on. Jin et al. [139] showed another dynamic model and the model was able to offer acceptable accuracy and robustness with five or six unknown parameters. In references [140-142], similar dynamic models were also developed for automotive waste heat recovery systems, borehole and ground applications. Constrained by the active chilled beam terminal units, the secondary air outlet temperature and the heat exchanger temperature are very difficult to be measured. Thus, these available modeling techniques [138-142] become unfeasible. The following model, however, tries to maintain the important characteristics and achieve a new dynamic model with less information.

Based on the active chilled beam terminal unit previously developed, a hybrid dynamic model is proposed, which is the first reported model of the terminal unit. It is divided with two sub-models. The models for the primary air, induced secondary air and thermal and mechanical mixing of them are lumped together as one sub-model. Another sub-model is a thermal description of the heat exchanger. Due to the necessary simplification of the entrainment effect, it is described by an empirical function. In contrast, tremendous effort is invested on the detailed modeling of the heat exchanger based on incomplete information. Using the heat transfer mechanism and energy balance principle, a dynamic heat exchanger model with no more than five parameters that represent the lumped geometric terms is developed. The unknown model parameters are identified by either a linear or nonlinear least-squares method with the experimental results from the pilot plant. Meanwhile, the experimental data and simulation results are compared to validate the model and illustrate its effectiveness.

The rest part of this chapter is structured in 3 sections: **Section 6.2** devotes to the model development of the active chilled beam terminal unit; in **Section 6.3**, the model estimation methods, linear or nonlinear least-squares estimations, are introduced; the model is identified and verified via experiments in **Section 6.4**; that is followed by a brief summary of this chapter in **Section 6.5**.

6.2 Model development

Without loss of generality, some assumptions are adopted for ease of mathematical modeling:

1. The mixing of the primary and secondary air is considered instantaneous and homogenous.
2. The tubes are adiabatic in the part of return bends and branch joints.
3. The heat exchanger is always under dry condition.
4. The heat storage associated with moisture in the secondary air is ignored.
5. The flow and thermal properties of the fluids are supposed to be independent of temperature.
6. The temperature field of the heat exchanger is assumed to be evenly distributed.

Assumption 1 is quite general for air mixing and assumptions 2-5 are good approximations to the practical situation. For example, the tube bends and joints are well thermal insulated in the experiments; the dry condition is strictly kept during the operation of active chilled beam terminal units; maximum moisture content of the secondary air is about 3% by mass so that its heat storage is small; the flow and thermal properties variances of the fluids are also very small because the fluids' temperature variances are limited. The last assumption is adopted simply because the heat exchanger is considered as a whole in the present study.

Confined air jet

Generally, air ducts or diffusers in HVAC systems are modeled by their flow resistances. For the flow of a particular fluid through a flow passage at a particular volume flow rate, the energy expended to sustain the flow appears as the drop in pressure along the flow passage, the higher the flow resistance of the passage, the higher the pressure drop for the same volume flow rate. Here, the resistance of the primary air offered by its total path is denoted as R_{air} . This flow resistance depends on many factors, including outlet area of the nozzles and outlet shape, nozzle length, nozzle arrangement, air properties and so on, while it is treated as a constant during the normal operating conditions of the terminal unit. Then the primary air volume flow rate $\dot{V}_{pri}$ can be calculated with the following equation.

$$\dot{V}_{pri} = \sqrt{\frac{\Delta P_a}{R_{air}}} \quad (6.1)$$

where ΔP_a is the gauge pressure in the primary air plenum.

The entrainment effect is measured by entrainment ratio (ER), which is the most important assessment criterion of the terminal unit. ER is defined as the ratio of the secondary air volume flow rate $\dot{V}_{sec}$ to the primary air volume flow rate $\dot{V}_{pri}$.

$$ER = \frac{\dot{V}_{sec}}{\dot{V}_{pri}} \quad (6.2)$$

As mentioned in the introduction, an empirical relationship is adopted to reflect the entrainment effect avoiding sophisticated jet flow theories. Although the entrainment ratio is defined by the air volume flow rates, the primary air plenum pressure is actually the only manipulated variable in this confined air jet sub-model. Therefore, the entrainment effect is described by:

$$ER = g (\Delta P_a)^i \quad (6.3)$$

where g and i are unknown constant coefficients.

Since the instantaneous and homogenous mixing of the primary and secondary air forms the supply air to the zone with assumption 2, the volume flow rate $\dot{V}_{supply}$ and temperature T_{supply} are calculated as:

$$\dot{V}_{supply} = (1 + ER) \dot{V}_{pri} \quad (6.4)$$

$$T_{supply} = \frac{T_{pri} + ER * T_{sec}}{(1 + ER)} \quad (6.5)$$

where T_{pri} and T_{sec} are the primary air temperature and the outlet temperature of the secondary air.

It is known that the cooling capacity supplied to the zone is evaluated by the supply air volume flow rate and temperature. Based on Eqs. (6.4-6.5), it can be seen that the gauge pressure is the only variable determines the supply air volume flow rate. Except from the

primary air temperature, the supply air temperature depends on the outlet temperature of the secondary air as well. That is described by the following heat exchanger sub-model.

Heat exchanger

Different from the widely used finite element analysis method, the heat exchanger is treated as the only control volume here. Otherwise, the model has to be described by partial differential equations. Without spatial variation considerations, the model can be represented in the form of differential equations, which is better for the subsequent control and optimization applications. Fig. 6.1 shows schematic diagram of a simplified heat exchanger. The secondary air is forced by the entrainment effect with inlet temperature T_{zone} and volume flow rate $\dot{V}_{sec}$ and its outlet temperature decreases to T_{sec} . Similarly, the chilled water flows is forced by the pump with inlet temperature $T_{w.in}$ and volume flow rate $\dot{V}_w$ and the outlet temperature increases to $T_{w.out}$.

Figure 6.1 Schematic diagram of a simplified heat exchanger

Both the secondary air and chilled water are driven mechanically by external forces, so the heat transfer mechanisms are forced convections. As same as Eqs. (4.4-4.5), forced convection heat transfer coefficient h for a single phase flow can be calculated by Reynolds number and Prandtl number and then:

$$h = d\dot{V}^b \quad (6.6)$$

$$d = \frac{k}{D} a \left(\frac{\rho D}{\mu} \right)^b \left(\frac{C\mu}{k} \right)^c \left(\frac{1}{A} \right)^b \quad (6.7)$$

where k is the fluid thermal conductivity; D is the fluid characteristic length; a , b and c are unknown constant coefficients; ρ is the fluid density; μ is the fluid absolute viscosity, C is the fluid specific heat and A is the flow area.

In the secondary air loop, the heat capacity transferred from the secondary air to the heat exchanger can be expressed in form of Newton's law of cooling.

$$Q_{at} = h_a A_a \eta_a (T_a - T_t) \quad (6.8)$$

$$h_a = d_a V_{sec}^{b_a} \quad (6.9)$$

where d_a and b_a are the counterparts of constants d and b in Eq. (6.7) for the secondary air loop.

Rather than inlet temperature difference or log mean temperature difference, the bulk temperature difference is adopted in Eq. (6.8). The bulk temperature of the secondary air is its average temperature.

$$T_a = \frac{T_{zone} + T_{sec}}{2} \quad (6.10)$$

The resultant heat increment rate of the secondary air is calculated by:

$$Q_a = C_a \rho_a \dot{V}_{sec} (T_{sec} - T_{zone}) \quad (6.11)$$

In the same manner, the following formulas can be used to denote the chilled water loop.

$$Q_{tw} = h_w A_w (T_t - T_w) \quad (6.12)$$

$$h_w = d_w \dot{V}_w^{b_w} \quad (6.13)$$

$$T_w = \frac{T_{w.in} + T_{w.out}}{2} \quad (6.14)$$

$$Q_w = C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in}) \quad (6.15)$$

where d_w and b_w are the counterparts of constants d and b in Eq. (6.7) for the chilled water loop.

The operation of such heat exchangers can be considered as three interacting dynamic processes describing the energy storage of the secondary air, heat exchanger, and chilled water respectively. However, for the sake of developing a model that is simple and practicable, two of the processes are neglected and only the energy storage of the heat exchanger temperature is kept as an intermediate. Hence, the simplified model will be of the form.

$$M_t C_t \frac{dT_t}{dt} = Q_{at} - Q_{tw} \quad (6.16)$$

$$0 = Q_{at} + Q_a \quad (6.17)$$

$$0 = Q_{tw} - Q_w \quad (6.18)$$

Refer to [143], the heat exchanger temperature dynamic is the only source of the dynamics of the secondary air outlet temperature and the chilled water outlet temperature. The dynamic relationship between these variables can be obtained by taking derivative to both sides of Eqs. (6.17) and (6.18),

$$\frac{dT_{sec}}{dt} = \frac{h_a A_a \eta_a}{C_a \rho_a \dot{V}_{sec} + \frac{1}{2} h_a A_a \eta_a} \frac{dT_t}{dt} \quad (6.19)$$

$$\frac{dT_{w.out}}{dt} = \frac{h_w A_w}{C_w \rho_w \dot{V}_w + \frac{1}{2} h_w A_w} \frac{dT_t}{dt} \quad (6.20)$$

The model in Eqs. (6.16-6.18) then can be reduced to two dynamic processes on the secondary air outlet temperature and chilled water outlet temperature. Substitute Eqs. (6.8-6.11) and Eq. (6.19) into Eq. (6.16) and rearrange the equation properly,

$$\begin{aligned} M_t C_t \left(\frac{C_a \rho_a \dot{V}_{sec}}{d_a \dot{V}_{sec}^{b_a} A_a \eta_a} + \frac{1}{2} \right) \frac{dT_{sec}}{dt} = & - \left[C_a \rho_a \dot{V}_{sec} + \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} \left(\frac{1}{2} + \frac{C_a \rho_a \dot{V}_{sec}}{d_a \dot{V}_{sec}^{b_a} A_a \eta_a} \right) \right] T_{sec} \\ & + \left[C_a \rho_a \dot{V}_{sec} - \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} \left(\frac{1}{2} - \frac{C_a \rho_a \dot{V}_{sec}}{d_a \dot{V}_{sec}^{b_a} A_a \eta_a} \right) \right] T_{zone} + \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} T_{w.in} \end{aligned} \quad (6.21)$$

Similarly, corresponding equation for the chilled water loop can be obtained.

$$\begin{aligned}
M_t C_t \left(\frac{C_w \rho_w \dot{V}_w}{d \dot{V}_w^{b_w} A_w} + \frac{1}{2} \right) \frac{dT_{w.out}}{dt} = & - \left[C_w \rho_w \dot{V}_w + \frac{C_a \rho_a \dot{V}_{sec} d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{C_a \rho_a \dot{V}_{sec} + \frac{d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{2}} \left(\frac{1}{2} + \frac{C_w \rho_w \dot{V}_w}{d \dot{V}_w^{b_w} A_w} \right) \right] T_{w.out} \\
& + \left[C_w \rho_w \dot{V}_w - \frac{C_a \rho_a \dot{V}_{sec} d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{C_a \rho_a \dot{V}_{sec} + \frac{d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{2}} \left(\frac{1}{2} - \frac{C_w \rho_w \dot{V}_w}{d \dot{V}_w^{b_w} A_w} \right) \right] T_{w.in} + \frac{C_a \rho_a \dot{V}_{sec} d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{C_a \rho_a \dot{V}_{sec} + \frac{d_a \dot{V}_{sec}^{b_a} A_a \eta_a}{2}} T_{zone}
\end{aligned} \quad (6.22)$$

Reviewing above heat exchanger sub-model, it can be observed that its air side operation depends on the confined air jet in terms of the secondary air volume flow rate, while the chilled water inlet temperature and volume flow rate on the water side could be independent variables to regulate the secondary air outlet temperature and finally adjust the cooling capacity.

6.3 Model estimation

Least-squares method is widely used to estimate the unknown parameter of linear models, which can be represented by:

$$\Psi X = \Gamma + \Delta \quad (6.23)$$

where Ψ , Γ are the measurements, X is the unknown model parameter to be estimated and Δ is the measurement error. The best estimated X^* of X can be found using a standard least-squares method as:

$$X^* = (\Psi^T \Psi)^{-1} \Psi^T \Gamma \quad (6.24)$$

For the confined air jet sub-model, the primary air resistance and the description function of the entrainment effect can be estimated in this way. Eq. (6.1) is rearranged to be linear form.

$$\dot{V}_{pri}^2 * R_{air} = \Delta P_a \quad (6.25)$$

If experiments are conducted N times for different primary air plenum pressures, Eq. (6.25) is then considered in the form of Eq. (6.23).

$$X = R_{air}, \Psi = \begin{bmatrix} (\dot{V}_{pri}^2)_1 \\ \vdots \\ (\dot{V}_{pri}^2)_N \end{bmatrix}, \Gamma = \begin{bmatrix} (\Delta P_a)_1 \\ \vdots \\ (\Delta P_a)_N \end{bmatrix}$$

Similarly, Eq. (6.3) is rewritten to be:

$$\ln(g) + i \ln(\Delta P_a) = \ln(ER) \quad (6.26)$$

and then,

$$X = \begin{bmatrix} \ln(g) & i \end{bmatrix}, \Psi = \begin{bmatrix} 1 & (\ln(\Delta P_a))_1 \\ \vdots & \vdots \\ 1 & (\ln(\Delta P_a))_N \end{bmatrix}, \Gamma = \begin{bmatrix} (\ln(ER))_1 \\ \vdots \\ (\ln(ER))_N \end{bmatrix}$$

The estimation procedure of the heat exchanger sub-model is separated into two steps: steady state estimation and transient state estimation. In steady state, the heat transfer processes are balance and then based on Eqs. (6.8-13) and (6.15), the following equation can be established.

$$\frac{1}{d_a A_a \eta_a \dot{V}_{sec}^{b_a}} + \frac{1}{d_w A_w \dot{V}_w^{b_w}} = \frac{T_{zone} - \frac{T_{w.in} + T_{w.out}}{2}}{C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in})} - \frac{1}{2C_a \rho_a \dot{V}_{sec}} \quad (6.27)$$

In normal engineering applications, the rule of thumb value $b_a = b_w = 0.5$ is often adopted. Then, Eq. (6.27) has only two aggregated unknown parameters. Repeat the experiment N times and rewrite them in the form of Eq. (6.23).

$$X = \begin{bmatrix} \frac{1}{d_a A_a \eta_a} \\ \frac{1}{d_w A_w} \end{bmatrix}, \Psi = \begin{bmatrix} \left(\frac{1}{\dot{V}_{sec}^{b_a}} \right)_1 & \left(\frac{1}{\dot{V}_w^{b_w}} \right)_1 \\ \vdots & \vdots \\ \left(\frac{1}{\dot{V}_{sec}^{b_a}} \right)_N & \left(\frac{1}{\dot{V}_w^{b_w}} \right)_N \end{bmatrix}, \Gamma = \begin{bmatrix} \left(\frac{T_{zone} - \frac{T_{w.in} + T_{w.out}}{2}}{C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in})} - \frac{1}{2C_a \rho_a \dot{V}_{sec}} \right)_1 \\ \vdots \\ \left(\frac{T_{zone} - \frac{T_{w.in} + T_{w.out}}{2}}{C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in})} - \frac{1}{2C_a \rho_a \dot{V}_{sec}} \right)_N \end{bmatrix}$$

If the parameters b_a and b_w are regarded as unknown parameters, Eq. (6.27) has total four unknown parameters. Since this equation becomes nonlinear, nonlinear least-squares method, widely known as Levenberg-Marquardt method, is used by defining an objective function as:

$$f(x) = \sum_{i=1}^N r_i^2(x) = \sum_{i=1}^N \left(\frac{T_{zone} - \frac{T_{w.in} + T_{w.out}}{2}}{C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in})} - \frac{1}{2C_a \rho_a \dot{V}_{sec}} - \frac{1}{d_a A_a \eta_a \dot{V}_{sec}^{b_a}} - \frac{1}{d_w A_w \dot{V}_w^{b_w}} \right)^2 \quad (6.28)$$

where $r(x)$ are the residuals, $f(x)$ is the sum of squares of the residuals, and $x = [d_a A_a \eta_a \quad d_w A_w \quad b_a \quad b_w]$ is the unknown parameter vector.

After the steady state estimation, the model time constant is the last unknown parameter to be estimated. Consequently, Eq. (6.22) is rewritten as:

$$\theta \frac{dT_{w.out}(t)}{dt} = \Theta(t) \quad (6.29)$$

where

$$\begin{aligned} \theta &= M_t C_t \left(\frac{C_w \rho_w \dot{V}_w}{d_w \dot{V}_w^{b_w} A_w} + \frac{1}{2} \right) \\ \Theta(t) &= - \left[C_w \rho_w \dot{V}_w + \frac{C_a \rho_a \dot{V}_{sec}(t) d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{C_a \rho_a \dot{V}_{sec}(t) + \frac{d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{2}} \left(\frac{1}{2} + \frac{C_w \rho_w \dot{V}_w}{d_w \dot{V}_w^{b_w} A_w} \right) \right] T_{w.out}(t) \\ &+ \left[C_w \rho_w \dot{V}_w - \frac{C_a \rho_a \dot{V}_{sec}(t) d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{C_a \rho_a \dot{V}_{sec}(t) + \frac{d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{2}} \left(\frac{1}{2} - \frac{C_w \rho_w \dot{V}_w}{d_w \dot{V}_w^{b_w} A_w} \right) \right] T_{w.in}(t) + \frac{C_a \rho_a \dot{V}_{sec}(t) d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{C_a \rho_a \dot{V}_{sec}(t) + \frac{d_a \dot{V}_{sec}^{b_a}(t) A_a \eta_a}{2}} T_{zone} \end{aligned}$$

Starting from this initial steady state at $t=0$, step inputs of the secondary air volume flow rate or the inlet temperatures are given for testing. Assuming that the intercept of the tangent to the step response that has the largest slope with respect to the horizontal axis gives H and integrating the differential terms in Eq. (6.29) from $t=0$ to $t=\tau$ ($\tau \leq H$) results in the following equation:

$$\theta \int_0^{\tau} \frac{dT_{w.out}(t)}{dt} dt = \int_0^{\tau} \Theta(t) dt \quad (6.30)$$

With the sampling interval of T_s , and $\tau = nT_s$. When the perturbation of the input variable is very small, the following linear form equation is achieved.

$$\theta [T_{w.out}(nT_s) - T_{w.out}((n-1)T_s)] = \int_{(n-1)T_s}^{nT_s} \Theta(t) dt \approx \frac{T_s}{2} [\Theta(nT_s) - \Theta((n-1)T_s)] \quad (6.31)$$

and

$$X = [\theta], \Psi = \begin{bmatrix} T_{w.out}(T_s) - T_{w.out}(0) \\ \vdots \\ T_{w.out}(nT_s) - T_{w.out}((n-1)T_s) \end{bmatrix}, \Gamma = \begin{bmatrix} \frac{T_s}{2} [\Theta(T_s) - \Theta(0)] \\ \vdots \\ \frac{T_s}{2} [\Theta(nT_s) - \Theta((n-1)T_s)] \end{bmatrix}$$

6.4 Experimental results and discussions

As the main concern is to model the active chilled beam terminal unit, it is desired to be accurate in various operating conditions. Except from the primary air temperature, all the other independent variables, including the gauge pressure in the primary air plenum, chilled water inlet temperature and volume flow rate are tried to be varied during the experiments. Although the environmental temperature can't be controlled, the experiments are conducted in different weather conditions so that the environmental temperature is adjusted in a passive way. Most of the typical operating conditions are covered and summarized in Table 6.1.

Table 6.1 Summary of experimental parameters setting

Parameter	Range	Unit
Environment temperature	25.0-29.0	℃
Primary air plenum pressure	20-260	Pa
Chilled water volume flow rate	50-400	L/h
Chilled water inlet temperature	13.6-18.5	℃

All the experimental data, including the primary air plenum pressure, primary and secondary air volume flow rates, environment temperature, chilled water volume flow rate, and its inlet and outlet temperatures, are recorded under steady-state besides the model dynamic property considerations. Besides the thermal equilibrium of the heat exchanger, the attention is also focused on the stabilization of the secondary air volume flow rate. The micromanometer reading for the secondary air is valid only if its variation is within ± 10 L/h for 5 minutes.

As same as the model development, the model is estimated and verified in terms of two sub-models. The heat exchanger is the only dynamic element in the whole model and therefore it is more reasonable and convenient to evaluate it as a whole.

Static performance

For the confined air jet sub-model, totally 40 data sets are collected consisting of different primary air plenum gauge pressures, primary air volume flow rates, and secondary air volume flow rates. Randomly select 28 data sets for model fitting, while the rest 12 ones are used for model validation. The primary air resistance and the description function of the entrainment effect are estimated.

$$\dot{V}_{pri}^2 * 0.003997 = \Delta P_a$$

$$ER = 1.563 * \Delta P_a^{0.107}$$

The model fitting and validation results are given in Figs. 6.2-6.5. Given the confidence level of 95%, it can be observed that the maximum model errors are about 5%. The inconsistency of the experimental results shown in the red circles in Figs. 6.2 and 6.4 is due to the different flowmeters.

Figure 6.2 Experiment fitting for the primary air resistance

Figure 6.3 Model validations for the primary air resistance

Figure 6.4 Experiment fitting for the entrainment effect

Figure 6.5 Model validation for the entrainment effect

For the heat exchanger sub-model, totally 41 data sets are obtained, including the environment temperature, pressure air plenum gauge pressure, chilled water inlet temperature and chilled water volume flow rate. Since the cooling coil model is relatively complex, more data are adopted for model validation. About 50% of the data, 21 sets, are used for model fitting and the rest 20 ones for model validation.

Table 6.2 Summary of heat transfer model parameters and their performances

Model	b_a	b_w	$d_a A_a \eta_a$	$d_w A_w$	RMS
Two-parameters	0.5	0.5	11.6279	13.1406	14.5786
Four-parameters	0.4465	0.3426	25.7086	22.4594	17.5950

With the least and nonlinear squares estimation produces defined in model estimation section, the unknown parameters of the heat exchanger model are summarized in Table 6.2. The model fitness is included as well, which is used to compare the two-parameters model and the four-parameters model. The fitness of a model is evaluated through the root mean square error (RMS), which is defined by:

$$\text{RMS} = \sqrt{\frac{\sum_{i=1}^N (\mathcal{Q}_{cal} - \mathcal{Q}_m)^2}{N}}$$

where N is the number of fitted points, $\mathcal{Q}_{cal}$ is the heat transfer capacity of point i calculated by the model, $\mathcal{Q}_m$ is the heat transfer capacity of point i measured from experiments.

The model fitting and validation results of the heat exchanger sub-models are successively illustrated in Figs. 6.8-6.11. The maximum model errors are also about 5% with 95% confidence levels.

Figure 6.6 Experiment fitting by two-parameter model for the heat exchanger

Figure 6.7 Experiment fitting by four-parameter model for the heat exchanger

Figure 6.8 Model validation by two-parameter model for the heat exchanger

Figure 6.9 Model validation by four-parameter model for the heat exchanger

Dynamic performance

According to the estimation procedure defined above, the time constant is directly identified from the experimental step responses without any normalization processes. The effects of steady state errors would be introduced into this process. In order to minimize the effect, particular attention is paid on the data selection. The steady states of the

experimental setup and the model are tried to be as close as possible before and after the step input. 2 sets of step responses for the primary air chamber gauge pressure change, from 106 Pa to 40 Pa and from 20 Pa to 50 Pa, are selected. This selection is also due to the fact that the active chilled beam terminal unit is normally configured in the form of modulating air loop and on-off water loop. Using the time constant estimation procedure, the unknown parameter $M_t C_t$ is identified to be 5946 J/ °C. The model fitting results are summarized in Figs. 6.10 and 6.11, where red lines mean model simulation outputs and blue lines represent experiment outputs.

Figure 6.10 Experiment fitting for the time constant with a primary air chamber pressure drop

Figure 6.11 Experiment fitting for the time constant with a primary air chamber pressure increase

The model validation is conducted under the real-time working conditions. The time varying chilled water inlet temperature and primary air plenum gauge pressure are given in Fig. 6.12. The environment temperatures before and after the experiment are 24.8 °C and 25.1 °C, while it is assumed to be a constant 25 °C in the model simulation. The practical system outputs in terms of the chilled water outlet temperature and heat transfer rate are compared to the model prediction results. For comparison, the number of the unknown parameters is considered and the time constant is also estimated by the compositions and the specific heat of them, 2674 J/°C. The result are given in Figs. 6.13-6.16.

Figure 6.12 Time varying the chilled water inlet temperature and the primary air plenum pressure

Figure 6.13 Dynamic performance of two-parameter model with $M_i C_i$ estimated by heat exchanger compositions

Figure 6.14 Dynamic performance of four-parameter model with $M_i C_i$ estimated by heat exchanger compositions

Figure 6.15 Dynamic performance of two-parameter model with $M_i C_i$ estimated by experiments

Figure 6.16 Dynamic performance of four-parameter model with $M_i C_i$ estimated by experiments

From the comparison of blue circles in Figs 6.13-6.16, the dynamic performance of the model with estimated $M_i C_i$ value should be better. That means something else ignored in the $M_i C_i$ value calculation can also store heat, such the heat exchanger structure, unit casing and so on. Meanwhile, it should be pointed out that the smaller calculated $M_i C_i$ value seems to get a slower dynamic response from the figures, which is quite strange. The reason may be that the final dynamic responses contain the effect of time-varying chilled water inlet temperature and the faster dynamic response with a smaller $M_i C_i$ value may be cancelled with it.

6.5 Summary

Blending the first principles and experimental results in a hybrid manner, an accurate but robust model of the active chilled beam terminal unit was established based on limited information. It was integrated by two sub-models for the confined air jet and heat exchanger respectively. From the experimental results, the following conclusions can be summarized:

1. The confined air jet sub-model is very simple, avoiding the sophisticated jet flow theories, but practical and accurate in a wide operation range, confined within errors of $\pm 5\%$.

2. For the heat exchanger, the the steady state performances of both two-parameters model and four-parameters model are both satisfied, within $\pm 5\%$ error ranges.
3. $M_i C_i$ value estimated by the linear-squares method can precisely reflect the transient performance of the model. It provides better performance than the value estimated by the compositions of the heat exchanger.
4. Totally, the two-parameters model for the heat exchanger seems to be better than the four-parameters one. Although its steady state accuracy is a little worse than that of four-parameter model, it is much better in the dynamic performance test.

The significance of the proposed model is that it has broad potential to apply to real-time control and optimization applications. For example, to avoid condensation, a dynamic tracking control scheme to control the off coil temperature of the secondary air above the dew point temperature can be well done.

Chapter 7. Operating characteristics and efficiencies of active chilled beam terminal units

7.1 Introduction

So far, the HVAC research community is constantly seeking effective methodologies to optimally utilize active chilled beam systems in different climates. An optimum system generally requires some specific parameters and ends up in many cases to be an iterative selection and design process. It is also urged to weight cost/benefit of various solutions on a project by project basis. Before that, appropriately understanding the system operating characteristics and efficiencies are definitely necessary, particularly on latent cooling capacity [21]. Different from all-air HVAC systems, where space dehumidification is only a by-product of space cooling, it is a critical concern in active chilled beam systems. If the system latent cooling capacity is oversized, the space humidity level becomes unnecessarily low and then results in a lot of energy wasting on treating and transporting the primary air. On the contrary, without sufficient latent cooling capacity or designing a high humidity level may lead to some condensation issues, which conversely decrease the system sensible cooling capacity due to the condensation avoidance actions as well. In addition, an exorbitant humidity level may cause growth of fungus and bacteria. Only when the system perfectly matched with the conditioned space in terms of both sensible and latent cooling capacities, as well as TCC and SHR, can indoor temperature and relative humidity be simultaneously and accurately controlled. Taking direct expansion air conditioning systems as an example, Li et al. [144] studied the TCC and SHR of an experimental direct expansion air conditioning system under different combinations of compressor and supply fan speed, but only at a fixed space condition. Xu et al. [145] extended the study to various space conditions and depicted the system operating characteristics in a more intuitive form. Li et al. [146] carried out a further study to model and predict the system for humidity control purpose. These research results were indeed utilized in developing proper system control algorithms [147].

Beyond that, it should be noted that in active chilled beam systems the space ventilation requirement and latent load can only be satisfied by the primary air, while the sensible load can be accommodated via either the primary air or the chilled water. Since using the

chilled water to handle the sensible load is much more energy efficient, quantity of the sensible load shared by the chilled water can be an indirect measurement of system efficiency.

In the present chapter, inherent operating characteristics and efficiencies of a 2-way discharge active chilled beam terminal unit are in depth studied. The study is focused on variable air volume mode for at least two reasons: active chilled beam systems operating at constant air volume mode have significant limitations in adjusting the cooling output to manage the variable space load. Sooner or later they will evolve as variable air volume systems. On the other hand, the system design and performance assessment should be verified under part-load conditions besides the peak load conditions. Because of the self-regulating property of active chilled beam terminal units, not only the primary air condition but also the space condition affects the system operation. Without a standard thermal insulation lab, it is almost impossible to experimentally capture their influences. As a compromise scheme, this study is implemented in the form of a series of simulations based on an experimentally verified model. Incorporated with an air handling unit or an air handling unit plus a liquid desiccant dehumidifier, the active chilled beam terminal unit is simulated under various combinations of the primary air and chilled water volume flow rates. As mentioned earlier, the TCC is no longer the only performance index. The sensible cooling capacity and latent cooling capacity are assessed in terms of the SHR. Furthermore, the sensible cooling capacities provided by the primary air and chilled water are distinguished using a newly defined energy saving potential index, which is also the efficiency measurement. The operating constraints between these key operational parameters are reported and sensitivity of the active chilled beam terminal unit to actual primary air and space conditions are evaluated.

This chapter is organized as below: in **Section 7.2**, the active chilled beam systems, where all the simulations are carried out, are described. **Section 7.3** presents the simulation model and performance indexes. It is followed by **Section 7.4** about the simulation results and discussions. **Section 7.5** draws a conclusion.

7.2 System description

In the most of existing applications, active chilled beam terminal units are used together with conventional air handling units. Schematic diagram of such a combination is shown in Fig. 7.1 and the corresponding air treatment processes are depicted in a psychrometric chart in Fig. 7.2. For ease of understanding, the processes can be explained as below.

$1 \cup 2 \rightarrow 3$: The outdoor fresh air at state 1 is firstly mixed with the recirculation air at state 2 in a certain ratio and then the resultant air is at state 3. Sometimes, this mixing process is not used in active chilled beam systems as the systems tend to have full fresh air. If energy consumption on the fresh air treatment is very high, the recirculation air should be used and the fresh air ratio should be minimized as long as it is sufficient for the conditioned space.

$3 \rightarrow 4 \rightarrow 5$: The mixed air is assumed to be cooled and mechanically dehumidified via a cooling coil from state 3 to saturated state 4. Although the air leaving the coil may not be saturated because of the air bypass, such an assumption reflects the most application situations and also simplifies the following study. The treated air is then pumped to the conditioned space as the primary air and the temperature is supposed to be raised 1°C due to the heat gains along supply fan, air ducts, etc. It can be seen that dehumidification of the mixed air is closely coupled with the cooling. In other words, it is impossible to control the primary air temperature and relative humidity independently.

$5 \cup (2 \rightarrow 6) \rightarrow 7$: The primary air at state 5 is finally driven through the nozzles and leads to entrainment effect. As a consequence, a certain amount of air in the conditioned space can be induced through the secondary heat exchanger as the secondary air. Since condensation is strictly avoided, the secondary heat exchanger only extracts the sensible heat. The secondary air is cooled from state 2 to state 6. The primary air together with the secondary air is supplied into the conditioned space through linear slots on edges of the active chilled beam terminal unit.

Figure 7.1 Schematic diagram of an active chilled beam system combining with a conventional air handling unit

Figure 7.2 Psychrometric chart of an active chilled beam system combining with a conventional air handling unit

For comparison, a liquid desiccant dehumidifier is utilized to decouple the temperature and relative humidity of the primary air. It essentially extends the applicability of active chilled beam systems, particularly in hot and humid conditions. Schematic diagram of such a system is shown in Fig. 7.3 and the air treatment processes are depicted in a psychrometric chart in Fig. 7.4. The figures can be interpreted in a similar manner as Figs. 7.1 and 7.2. The only difference is that the primary air leaving the cooling coil is driven through a dehumidifier and its relative humidity is significantly reduced. During this

dehumidification process, the primary air temperature may be increased, decreased, or even kept the same. That depends on the operation of the dehumidifier. In the present study, the primary air temperature is assumed to be unchanged.

Figure 7.3 Schematic diagram of an active chilled beam system combining with an air handling unit and a dehumidifier

Figure 7.4 Psychrometric chart of an active chilled beam system combining with an air handling unit and a dehumidifier

7.3 Simulation model and performance indexes

From above described working principles of the systems, the simulation model contains two parts, for the primary air and secondary air respectively. As a result, the model is

separately derived based on fundamentals of energy and mass conservations of them. It is then followed by the performance indexes.

Primary air model

Without loss of generality, a primary air model should address three parameters, the primary air volume flow rate, temperature, and relative humidity. As mentioned earlier, the active chilled beam terminal unit is studied under variable air volume mode and the primary air volume flow rate is selected as the control input. In addition, the primary air temperature is defined as an operation condition, so this model is focused on the coupling between the primary air temperature and relative humidity.

For the active chilled beam system combining with a conventional air handling unit but without any dehumidifier, the primary air leaving the cooling coil becomes saturated with 100% relative humidity. Since the latent cooling capacity is generally calculated based on the mass conservation of the moisture content in the space, this saturated status has to be represented by the moisture content. Then the air temperature and moisture content can be correlated by the following formula, which is developed by Vaisala [148] on the basis of Reference [149].

$$W_{pri} = \frac{l * m * 10^{\left(\frac{nT_{offc}}{T_{offc} + T_n}\right)}}{P_{am} - m * 10^{\left(\frac{nT_{offc}}{T_{offc} + T_n}\right)}} \quad (7.1)$$

where W_{pri} is the primary air moisture content; T_{offc} is the off coil temperature; P_{am} is the ambient pressure; l , m , n and T_n are the constant coefficients which depend on the applicative temperature range. In the present study, they are set as 621.99, 6.12, 7.59 and 240.73 respectively.

With the liquid desiccant dehumidifier, the primary air moisture content can be independently controlled in response to its relative humidity RH_{pri} . Then, Eq. (7.1) is replaced by:

$$W_{pri} = \frac{l * m * 10^{\left(\frac{nT_{offc}}{T_{offc} + T_n}\right)} * RH_{pri}}{P_{am} - m * 10^{\left(\frac{nT_{offc}}{T_{offc} + T_n}\right)} * RH_{pri}} \quad (7.2)$$

Taking the heat gains along the supply fan, air ducts, etc. into consideration, the primary air temperature T_{pri} becomes:

$$T_{pri} = T_{offc} + 1 \quad (7.3)$$

Secondary air model

Referring to Eqs. (6.1-6.3), the secondary air volume flow rate $\dot{V}_{sec}$ depends on the primary air volume flow rate $\dot{V}_{pri}$ and can be calculated via the entrainment ratio ER :

$$ER = g \left(\dot{V}_{pri}^2 R_{air} \right)^i \quad (7.4)$$

$$\dot{V}_{sec} = \dot{V}_{pri} ER \quad (7.5)$$

where g , i and R_{air} are the unknown constant coefficients need to be estimated. The secondary air moisture content is exact same as that of the space air, while its temperature is decreased and can be represented by a static version of Eq. (6.21):

$$T_{sec} = \frac{K_1 T_{zone} + K_2 T_{w,in}}{K_3} \quad (7.6)$$

where,

$$\begin{aligned} K_1 &= C_a \rho_a \dot{V}_{sec} - \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} \left(\frac{1}{2} - \frac{C_a \rho_a \dot{V}_{sec}}{d_a \dot{V}_{sec}^{b_a} A_a \eta_a} \right) \\ K_2 &= \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} \\ K_3 &= C_a \rho_a \dot{V}_{sec} + \frac{C_w \rho_w \dot{V}_w d_w \dot{V}_w^{b_w} A_w}{C_w \rho_w \dot{V}_w + \frac{d_w \dot{V}_w^{b_w} A_w}{2}} \left(\frac{1}{2} + \frac{C_a \rho_a \dot{V}_{sec}}{d_a \dot{V}_{sec}^{b_a} A_a \eta_a} \right) \end{aligned}$$

where T_{zone} is the zone temperature; $T_{w,in}$ and $\dot{V}_w$ are the chilled water inlet temperature and volume flow rate; C_a and ρ_a are the specific heat and density of air; d_a , b_a and A_a are the heat transfer related constant coefficients need to be estimated on the secondary air side; C_w , ρ_w , d_w , b_w and A_w are the corresponding counterparts on the chilled water side.

Reviewing the manipulating variables in Eq. (7.6), the chilled water volume flow rate is usually the control input to regulate the space temperature and its inlet temperature is a prescribed operation variable. As long as there is no condensation, the chilled water inlet temperature is desired to be set as low as possible to share more space sensible load so

that to maximize the system efficiency. With a proper safety margin, 1 °C, the temperature is determined via the space dew point temperature T_d .

$$T_d = T_n * \left[\frac{n}{\log_{10} \left(10^{\left(\frac{nT_{zone}}{T_{zone} + T_n} \right)} * RH_{zone} \right)} - 1 \right]^{-1} \quad (7.7)$$

$$T_{w.in} = T_d + 1 \quad (7.8)$$

Performance indexes

As only the primary air can provide the latent cooling capacity, then in steady state, that latent cooling capacity Q_{lat} can be obtained with the primary air and space statuses. From the polynomial curve fit to the table 2.1 in the reference [150],

$$Q_{lat} = - \left(2500.8 - 2.36T_{zone} + 0.0016T_{zone}^2 - 0.00006T_{zone}^3 \right) \rho_a \dot{V}_{pri} (W_{pri} - W_{zone}) \quad (7.9)$$

To evaluate Eq. (7.9), the space moisture content W_{zone} is proactively calculated in a similar manner as Eqs. (7.1) and (7.3).

$$W_{zone} = \frac{l * m * 10^{\left(\frac{nT_{zone}}{T_{zone} + T_n} \right)} * RH_{zone}}{P_{am} - m * 10^{\left(\frac{nT_{zone}}{T_{zone} + T_n} \right)} * RH_{zone}} \quad (7.10)$$

Apart from the primary air, the secondary air can offer the sensible cooling capacity as well. In order to address the sensible cooling capacity Q_{sen} , the secondary air status has to be taken into account. Then, based on the space energy conservation,

$$Q_{sen} = -C_a \rho_a \dot{V}_{pri} (T_{pri} - T_{zone}) - C_a \rho_a \dot{V}_{sec} (T_{sec} - T_{zone}) \quad (7.11)$$

With above equations, the performance indexes can be easily obtained. The TCC Q_{total} comes first. According to the definition, it is the sum of the sensible cooling capacity and latent cooling capacity.

$$Q_{total} = Q_{sen} + Q_{lat} \quad (7.12)$$

Then, the SHR can be obtained as the ratio of the sensible cooling capacity to the TCC.

$$\text{SHR} = \frac{Q_{sen}}{Q_{total}} \quad (7.13)$$

In addition, it is known that the most efficient active chilled beam system is the one that makes full use of the chilled water to satisfy the space sensible load. In order to enhance the benefits of active chilled beam systems, the primary air has to be minimized to satisfy the minimum latent load and ventilation requirement in the given space while the use of the chilled water has to be maximized to cool the space. As a result, an important energy saving potential measurement can be defined as the ratio of the sensible cooling capacity handled by the chilled water to the total one.

$$\phi = \frac{Q_w}{Q_{sen}} \quad (7.14)$$

where the sensible cooling capacity provided by the chilled water Q_w is exactly the one afforded by the secondary air.

$$Q_w = -C_a \rho_a \dot{V}_{sec} (T_{sec} - T_{zone}) \quad (7.15)$$

7.4 Simulation results and discussions

In above simulation model, the unknown constant coefficients describing the moisture laden air and saturated air can be found in the reference [148] and specific heats and densities of the air and chilled water in the secondary air model can also be found in relevant handbooks, while the remaining unknown constant coefficients need to be experimentally estimated. Fortunately, they have been estimated in **Chapter 6**. The results are briefly summarized in Table 7.1 with minor unit conversions.

Table 7.1 Summary of the unknown parameters

g	i	R_{air}	$d_a A_a \eta_a$	b_a	$d_w A_w$	b_w
1204.1	0.107	0.003997	698	0.5	788	0.5

The simulation model is implemented and solved in the Matlab environment. In total, there are 13 sets of simulations carried out under different combinations of typical primary air and space conditions. The combinations are recorded in Table 7.2. In each simulation, the primary air volume flow rate and the secondary chilled water volume flow rate are varied as control inputs. To supply sufficient fresh air while avoid overmuch

noise and maintain proper indoor air distribution, the primary air volume flow rate can neither be too low nor too high. Considering the investigated active chilled beam terminal unit and its nozzle configuration, the primary air volume flow rate is progressively increased from 0.015 m³/s to 0.05 m³/s with a step increment of 0.005 m³/s. As for the chilled water volume flow rate, it is also carefully determined within a reasonable range from 0.02 L/s to 0.04 L/s with a step increment of 0.002 L/s. A high chilled water volume flow rate with a high Reynolds number and a high heat transfer coefficient between the chilled water and the heat exchanger surface is desirable, while the pressure drop caused by the high volume flow rate should also be taken into account.

Table 7.2 Summary of simulation conditions

Set No.	T_{pri} (°C)	RH_{pri} (%)	T_{zone} (°C)	RH_{zone} (%)
1	13	100	24	55
2	12	100	24	55
3	14	100	24	55
4	15	100	24	55
5	13	100	23	55
6	13	100	25	55
7	13	100	26	55
8	13	100	24	50
9	13	100	24	60
10	13	100	24	65
11	13	40	24	55
12	13	60	24	55
13	13	80	24	55

Operating characteristics and performance

Group A (set 1): without any comparison, there is only one set of simulation in this group. This group aims to reveal the inherent operating characteristics and efficiencies of the active chilled beam terminal unit with fixed primary air and space conditions.

Figure 7.5 Simulation result of set 1

With reference to [145], the simulation result is represented by plotting the TCC and SHR as x-axis and y-axis in the same diagram, Fig. 7.5. The energy saving potential index ϕ is depicted via the color bar. In the figure, the primary air and chilled water volume flow rates are gradually increased along the directions from A' to D' (B' to C') and from B' to A' (C' to D') respectively. At each primary air volume flow rate, the three performance indexes are all increased with the increasing of the chilled water volume flow rate and vice versa. At each chilled water volume flow rate, the TCC is increased while another two parameters are decreased with the increasing of the primary air volume flow rate. The maximum and minimum values of the TCC are 1520 W and 512 W and the SHR counterparts are 0.827 and 0.88. Provided with these limits, it would be easy to have a common false impression that the TCC and SHR can be freely combined within the limits, but actually the TCC and SHR are correlated and mutually constrained with a trapezoid A'-B'-C'-D'. That means the preferred operating range should be A'-B'-C'-D' rather than A-B-C-D. In addition, there is a positive correlation property between the SHR and energy saving potential index ϕ . When the SHR is increased from 0.827 to 0.88, the index ϕ is also increased from 0.40 to 0.61. It is consistent with the general application guideline that active chilled beam systems are an effective means to manage large sensible load.

Influences of the primary air temperature

Group B (sets 1-4): 4 sets of simulations with the primary air temperature at 12 °C, 13 °C, 14 °C, and 15 °C are investigated in this group, so this group is to address influences of the primary air temperature on the terminal unit operating characteristics and efficiencies.

Figure 7.6 Simulation results of sets 1-4

Fig. 7.6 collects the simulation results of sets 1-4. Given any particular primary air temperature, the simulation result is described by a colorful trapezoid as same as the one shown in Fig. 7.5 and the characteristic trapezoids can be analyzed in the same way. For simplicity, these analyses are ignored here. It is observed that the characteristic trapezoids are constrained by the straight lines L and L', while the trapezoids have different positions and color distributions. These influences can be interpreted from the following four aspects:

1. As the primary air temperature increasing, the characteristic trapezoid is shifted to the left. In other words, the TCC is decreased. For example, the maximum TCC is changed from 1657 W, to 1520 W, then to 1369 W, and finally to 1206 W. This influence is greater at relatively high temperatures. For instance, when the temperature is increased from 14 °C to 15 °C, the variation of TCC is 163 W, while it is only 137 W when the temperature is increased from 12 °C to 13 °C.

2. The characteristic trapezoid is lifted up and the SHR is improved when the primary air temperature is increased. The maximum value of SHR is raised from 0.85, to 0.88, then to 0.92, and finally to 0.97. As same as the influence on the TCC, the influence on the SHR is greater at high temperatures.
3. With the increasing of the primary air temperature, dominant tone of the characteristic trapezoid gradually changes from blue to red. The maximum energy saving potential index ϕ is varied from 0.58, to 0.61, then to 0.63, and finally to 0.65.
4. The last consequence of increasing the primary air temperature is reducing the trapezoid size, so as to applicable ranges of the terminal unit. Furthermore, at a high primary air temperature (e.g., 15 °C), the characteristic appearance deviates from the rest ones, i.e., a trapezoid shape becomes less obvious.

Influences of the space temperature

Group C (sets 1 and 5-7): in this group, the space temperature is varied at 23 °C, 24 °C, 25 °C, and 26 °C within the thermal comfort zone. Therefore, this group of simulations tries to establish influences of the space temperature.

Figure 7.7 Simulation results of sets 1 and 5-7

The simulation results of group C are illustrated in Fig.7.7. As same as Fig 7.6, this figure is interpreted according to the positions, tones, and sizes of the characteristic trapezoids. As a result, similar conclusions are obtained from the same four aspects. Simply put, decreasing the space temperature has analogous influences on the operating characteristics and efficiencies of the terminal unit as increasing the primary air temperature. However, comparing to the primary air temperature, the space temperature has competitive influences on the TCC but slight smaller ones on the SHR and index ϕ . For example, with the same temperature variations of 4 °C, variation of the minimum TCC is 139 W and the responding value in Fig. 7.6 is 136 W, but variation of the minimum SHR are 0.143 and 0.174 and variation of the minimum index ϕ are 0.0565 and 0.0696. In addition, unlike those characteristic trapezoids shown in Fig. 7.6, shapes of the trapezoids are all maintained.

Influences of the space relative humidity

Group D (sets 1 and 8-10): this group of simulations tries to discover influences of the space relative humidity. The space relative humidity is set at 50%, 55%, 60%, and 65%.

Figure 7.8 Simulation results of sets 1 and 8-10

The simulation results of group D are described in Fig.7.8. Comparing Fig. 7.8 with Figs. 7.6 and 7.7, decreasing the space relative humidity has analogous influences as

increasing the primary air temperature and decreasing the space temperature, while it has minimum influences on the TCC but maximum influences on the SHR and index ϕ . For example, the straight lines N and N' are almost perpendicular to the TCC axis and the SHR and index ϕ vary about 0.08 and 0.05 with a variation of the space relative humidity of 5%.

Influences of the primary air relative humidity

Group E (sets 1 and 11-13): in order to provide some insight into influences of the primary air relative humidity on the operating characteristics and efficiencies of the terminal unit, it is predefined at 40%, 60%, 80%, and 100%. Since the primary air is no longer saturated, the simulations of sets 11- 13 are conducted with the active chilled beam terminal unit in conjunction with both the air handling unit and the liquid desiccant dehumidifier.

Figure 7.9 Simulation results of sets 1 and 11-13

The simulation results of Group E are presented in Fig. 7.9. Apart from the similar influences as same as the other operation conditions, the most important uniqueness of the influences caused by the primary air relative humidity is that applicable range of the active chilled beam terminal unit is substantially enlarged at a low primary air relative humidity (e.g., 40% or 60%). The low limit of SHR is greatly reduced to 0.52. To show

this consequence better, all the 13 sets of simulations are summarized in Fig. 7.10. It can be seen that the polygon A-B-C-D-E is the applicable range of the active chilled beam in conjunction with a single air handling unit with whatever specific primary air and space air conditions, while the range is enlarged into the polygon A-B-F-G-H-D-E when introducing the liquid desiccant dehumidifier.

Figure 7.10 Simulation results of sets 1-13

7.5 Summary

In this chapter, a series of simulation studies on the inherent operating characteristics and efficiencies of the active chilled beam terminal unit were investigated under variable air volume mode. Given fixed primary air and space conditions, the TCC, SHR, and energy saving potential index ϕ were correlated in a colorful trapezoid. With respect to various primary air and space conditions, the characteristic trapezoid was varied in terms of position, tone, and shape. The obtained findings are expected to achieve a better understanding of the active chilled beam terminal unit, so as to the designs, the operating principles, and the control strategies of active chilled beam systems for an improved indoor thermal environment. For example, as SHR of a common conditioned-space is from 0.6 to 0.7, the active chilled beam systems can be applied together with the liquid desiccant dehumidifier. If the SHR is lower than 0.5, some extra equipment have to be required to enhance the system dehumidification capability.

It is sure that there are a variety of active chilled beam terminal units. Also, there can be many system configurations, operating conditions, and space conditions besides the ones assumed. However, the study is intended to provide a general method to present the operating characteristics and efficiencies of any active chilled beam systems rather than to offer any definitive conclusions about for all active chilled beam systems. With simple extensions, it is easy to discover analogous characteristics for other active chilled beam terminal units in the same method.

Chapter 8. Conclusions and future work

8.1 Conclusions

If properly designed, operated and maintained, active chilled beam systems will have significant improvements on thermal comfort and healthy of indoor occupants as well as on energy and cost efficiency of buildings, especially for tropical regions. It is therefore highly desirable to develop an energy efficient active chilled beam systems for tropics to fulfill the benefits. The present thesis has addressed the need with following contributions:

- The circuit number of the secondary heat exchanger was determined via an experimental comparison with four 2-rows fin and tube heat exchangers with different circuit numbers. Combining with a basic theoretical analysis, the thermodynamic and hydrodynamic performances were investigated under different water volume flow rates. The performance indicators included the heat transfer capacity, pressure drop, pumping energy, heat exchanger effectiveness, and performance index. It was found that different circuit numbers should be preferred in different operating conditions and under different evaluation criteria, while the 2-circuits arrangement should be the most comprehensive and reasonable option rather than the 1-circuit one. The 2-circuits arrangement could offer a competitive heat transfer capacity with a considerable lower pressure drop compared with the widely used 1-circuit arrangement. And then the tube connecting sequences were obtained through a series of experiment-aided simulations. Provided the fixed primary air plenum gauge pressure, the secondary air mal-distribution was for the first time found and the air velocity varied from 0.36m/s to 0.75m/s. The resultant heat transfer capacities of the individual tubes were also non-uniform. The optimized circuit arrangements were able to increase the capacities by 1.9-3% under various chilled water volume flow rates while carried some penalties on the potential pressure drops, manufacture difficulties, and material costs. Therefore, a simple circuit arrangement was proposed, which was proven to be better and more comprehensive. It was found that thermodynamic and hydrodynamic performances of the heat exchanger as well as performance of active chilled beam terminal units could be substantially enhanced with the circuit optimization.

- Combining the first principles and experimental results in a hybrid manner, an accurate but robust model of the active chilled beam terminal unit was established based on limited information. A reasonable compromise was made between capturing exact underlying physics and suitability for engineering applications. The model was integrated by two sub-models for the confined air jet and heat exchanger respectively. Static accuracy of the model was confirmed within $\pm 5\%$ and dynamic accuracy was also satisfied. The model was feasible in a wide operation range.
- Based on a static version of the dynamic model, a series of realistic simulations were conducted to investigate the operating characteristics and efficiencies under variable air volume mode. In addition, influences of different primary air as well as space conditions are studied. The TCC, SHR, and energy saving potential index were found to be correlated in a colorful trapezoid. With respect to various primary air and space conditions, the characteristic trapezoid was varied in terms of position, tone, and shape. Without the liquid desiccant dehumidifier, the minimum SHR of active chilled beam systems could achieve was about 0.65, while it became 0.5 with the dehumidifier.

8.2 Future work

Despite all the achievements in this thesis, they are insufficient to form a complete tropical energy efficient active chilled beam system. It would be very desirable and valuable to make further efforts on this area. In order to throw some light, few recommendations are given as follows:

1. There is an intuitive notion of the global development of a row of air jets discharged by the induction nozzles inside active chilled beam terminal units, from an early behavior as individual jets merging into a later behavior similar to a two dimensional jet. For the individual jets, the entrainment ratio is higher, while that is lower for the two dimensional jet. So then the interesting “how to make full use of the former and avoid the later” question arises. This question can be interpreted as “how to determine the nozzle size and pattern of the nozzles” as the critical distance for measurement is constrained by the

dimension of discharge chamber. In order to maximize the entrainment effect of active chilled beam terminal units, the question has to be well solved.

2. In general, active chilled beam systems are operated at low primary chamber pressures due to the understanding that high pressures always produce excessive acoustic signature. However, Alexander et al. [31] claimed that increasing the end of run operating static pressure would be essentially favored, which ultimately required fewer terminal units to satisfy the sensible cooling load, and little penalties in terms of the fan energy and acoustic signature. With the factually problematic idea, it is necessary to figure out what are the optimal operating conditions of active chilled beam systems. That is important in the system design and operation phases.
3. As derived in **Chapter 7**, the combination of active chilled beam systems with liquid desiccant dehumidification systems is very necessary for the applications in tropical region. Thus, the performance should be evaluated experimentally, not limited to the simulation result. More importantly, the evaluation should be extended to the whole system including both active chilled beam systems and liquid desiccant dehumidification systems. That is the first step to promote the combination into practice.

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1. Can Chen, Wenjian Cai, Youyi Wang, Chen Lin, Performance comparison of heat exchangers with different circuitry arrangements for active chilled beam applications, *Energy and Buildings*, 79 (2014) 164-172.
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Appendix A Design of a 2-way discharge active chilled beam terminal unit

Appendix B Particle swarm optimization

Particle Swarm Optimization (PSO) is an evolutionary search algorithm inspired by the social behavior of a bird flock or fish school. It is originally attributed to Kennedy and Eberhart [151]. So far, it has been extensively applied in many areas [152]. PSO is feasible to many different problems as there are few even no assumptions for the problem being optimized. For example, it can be implemented without gradient, which is generally required by common optimization algorithms. Iteration operation of PSO is also very intuitive and easy to understand. Nevertheless, as a stochastic search algorithm, PSO cannot guarantee an optimal solution.

Suppose that the problem being optimized can be mathematically defined by a designated objective function $f: R^n \rightarrow R$. That means the searching space is n -dimensional and continuous but there is only one continuous optimization objective. Let $X = (x_1 \ x_2 \ \cdots \ x_n)$ be a candidate solution in the form of a vector and let y be the optimization objective. The problem becomes:

$$\begin{aligned} \text{optimize } y &= f(X) \\ X &\in U(B_{lo} \ B_{up}) \end{aligned}$$

where B_{lo} and B_{up} are the lower and upper boundaries of the searching space.

Generally, there are four steps to apply a standard PSO for above problem:

Particle swarm initialization: predefine the number of candidate solutions, i.e. m , and randomly initialize the m candidate solutions, $X_i = (x_{1i} \ x_{2i} \ \cdots \ x_{ni}) \ i=1 \ 2 \ \cdots \ m$, within the searching space boundaries. Swarm is defined by a population of candidate solutions, here dubbed particles. If $X_i = (x_{1i} \ x_{2i} \ \cdots \ x_{ni}) \ i=1 \ 2 \ \cdots \ m$ are treated as the locations of the particles in the space, another group of vectors $V_i = (v_{1i} \ v_{2i} \ \cdots \ v_{ni}) \ i=1 \ 2 \ \cdots \ m$ can be defined as the velocities of the particles. Then, statuses of the particles can be completely captured.

Particle swarm evaluation: to compare the fitness of each particle, the entire particle swarm should be put into the designated objective function $y = f(X)$. According to the function outputs, the particles, the candidate solutions, can be evaluated to be “good” or “bad”. Let $P_i = (p_{1i} \ p_{2i} \ \cdots \ p_{ni})$ $i=1 \ 2 \ \cdots \ m$ be the best known position of the i^{th} particle up to the current iteration and let $G = (g_1 \ g_2 \ \cdots \ g_m)$ be the best known position of the entire swarm up to the current iteration. Then, some so-called best positions can be obtained via simple comparisons.

$$P_i = \text{best}(P_{i1} \ P_{i2} \ \cdots \ P_{ik}) \ k=1 \ 2 \ \cdots$$

$$G = \text{best}(P_1 \ P_2 \ \cdots \ P_m)$$

where k is the current iteration number.

Particle swarm evolution: as long as the termination criteria is not met, Then, the particles constantly move in the searching space according to the following mathematical formulae over the particle's position and velocity.

$$X_{i(k+1)} = X_{ik} + V_{i(k+1)}$$

$$V_{i(k+1)} = \omega V_{ik} + c_1 r_1 (P_{ik} - X_{ik}) + c_2 r_2 (G_k - X_{ik})$$

where $i=1,2,\dots,m$. ω is the inertia coefficient which is a constant in the interval $[0, 1]$; c_1 and c_2 are the learning rates which are nonnegative constants; r_1 and r_2 are the randomly generated constants in the interval $[0, 1]$. All those parameters are used to control the efficiency of the PSO algorithm, which can be also tuned via another overlaying optimization algorithm. It can be observed that the movement of the particles is influenced by both the local and global best positions. All the positions of the particles are constantly updated and they are guided toward the global best position. With the movement of the particle swarm, the problem being optimized can be interpreted as finding the best position in the searching space.

Termination of the evaluation and evolution: PSO is a stochastic searching algorithm, so it is actually difficult to accurately specify the convergence of the solution. For simplicity, the typical program termination criteria is listed as below:

- The maximum generation number is reached;
- The fitness of a best individual is better than the predefined value;
- The population fitness is approaching some limit.

It should be noted that the PSO described above is only applicable for continue problems. For discrete problems, the particle swarm optimization can be initialized, evaluated, and terminated in the same method as long as the problem is properly defined and mathematically represented, while the key difficulty is how to implement the particle swarm evolution, more specifically, how to interpret the plus, minus and multiplication sign in the equation. For this reason, a method based on swap operator and swap sequence is presented. Suppose that the candidate solution of the discrete problem being optimized can be represented by a solution sequence with n nodes,

$$X = (x_1 \ x_2 \ \cdots \ x_n)$$

Then a swap operation of changing node a and b can be defined as $SO(a \ b)$. Then a new solution can be obtained by acting the operator $SO(a \ b)$ on the previous solution.

$$X^* = X + SO(a \ b)$$

Here, the plus sign can be interpreted in a different manner, as a swap operation. A concrete example can be given below:

$$X = (1 \ 4 \ 3 \ 2 \ 5)$$

$$SO(1 \ 4)$$

$$X^* = X + SO(1 \ 4) = (1 \ 4 \ 3 \ 2 \ 5) + SO(1 \ 4) = (2 \ 4 \ 3 \ 1 \ 5)$$

Since the plus sign changes to be a swap operation of two nodes, the minus sign will means the same operation. That can be simply proved as below:

$$X^* = X - SO(a \ b)$$

then,

$$X = X^* + \text{SO}(a \ b)$$

adding another swap operation on both sides,

$$X + \text{SO}(a \ b) = X^* + \text{SO}(a \ b) + \text{SO}(a \ b) = X^*$$

Obviously,

$$X - \text{SO}(a \ b) = X + \text{SO}(a \ b)$$

In this way, one or more such swap operators can be added to be a swap sequence. For example, successively implementing m swap operators is equivalent to implementing a swap sequence.

$$\text{SS} = (\text{SO}_1 \ \text{SO}_2 \ \cdots \ \text{SO}_m)$$

For such a swap sequence, the order of those swap operators is important as the swap sequence acting on a solution means all the swap operators should be acted on the solution in order. That can be clarified via the following formula.

$$X^* = X + \text{SS} = X + \text{SO}_1 + \text{SO}_2 + \cdots + \text{SO}_m$$

It can be easily observed that the same solution may be obtained by different swap sequences. To distinguish them, a basic swap sequence which has the least swap operators in the set of swap sequences that can produce the same solution is defined. For example, the originate solution $X = (5 \ 1 \ 4 \ 2 \ 3)$, while the new solution is $X^* = (1 \ 2 \ 3 \ 4 \ 5)$. There exists a basic swap sequence SS , so that,

$$X^* = X + \text{SS}$$

The construction of SS should swap the nodes in X according to X^* from left to right as follows:

$$X^*(1) = X(2) = 1, \text{ so the first swap operator is } \text{SO}_1(1 \ 2),$$

$$X_1 = X + \text{SO}_1(1 \ 2) = (5 \ 1 \ 4 \ 2 \ 3) + \text{SO}_1(1 \ 2) = (1 \ 5 \ 4 \ 2 \ 3);$$

$X^*(2) = X_1(4) = 2$, so the second swap operator is $SO_2(2\ 4)$,

$$X_2 = X_1 + SO_2(1\ 2) = (1\ 5\ 4\ 2\ 3) + SO_2(2\ 4) = (1\ 2\ 4\ 5\ 3);$$

$X^*(3) = X_2(5) = 3$, so the third swap operator is $SO_3(3\ 5)$,

$$X_3 = X_2 + SO_3(3\ 5) = (1\ 2\ 4\ 5\ 3) + SO_3(3\ 5) = (1\ 2\ 3\ 5\ 4);$$

$X^*(4) = X_3(5) = 4$, so the fourth swap operator is $SO_4(4\ 5)$,

$$X_4 = X_3 + SO_4(4\ 5) = (1\ 2\ 3\ 5\ 4) + SO_4(4\ 5) = (1\ 2\ 3\ 4\ 5);$$

$X^*(5) = X_4(5) = 5$ and that is the last node of the solution sequence.

In summary, the basic swap sequence can be $SS = SO_1 + SO_2 + SO_3 + SO_4$. It can be observed that the number of the swap operators in the basic swap sequence should be smaller than that of the nodes in the solution sequence. In above example, it is 4.

With the concepts of the swap operator and basic swap sequence, the particle swarm evolution can be easily interpreted as below:

$$X_{i(k+1)} = X_{ik} + SS_{i(k+1)}$$

$$SS_{i(k+1)} = \omega SS_{ik} + \alpha * (P_{ik} - X_{ik}) + \beta * (G_k - X_{ik})$$

where α and β are random numbers between 0 and 1. The multiplication operations here mean all the basic swap sequence should be maintained with the corresponding probabilities α and β .