Operating characteristics and efficiencies of an active chilled beam terminal unit under variable air volume mode

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Abstract: Appropriately designing and maintaining temperature and relative humidity in a given space is essential for active chilled beam systems, where condensation should be strictly prevented. As a consequence, the Total Cooling Output Capacity (TCOC) of an active chilled beam system should be matched with the total cooling load of the applied space, as well as the Sensible Heat Ratio (SHR) of the system with the SHR of the space. From such a perspective, this paper for the first time explored the operating characteristics of a 2-way discharge active chilled beam terminal unit. Based on an experimentally verified model of the unit, a series of realistic simulations were carried out under various primary air volume flow rates and various chilled water volume flow rates. Inherent correlations between the TCOC and SHR were revealed. In the meanwhile, the operating efficiencies of the unitwere also measured by an energy saving potential index ε , which is defined as the ratio of chilled water sensible cooling output capacity to the total sensible cooling output capacity. In addition, influences of different primary air and space conditions on the operating characteristics and efficiencies were studied. The results obtained in this study are expected to facilitate 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.

Keywords: Active chilled beam, Variable air volume, Operating characteristics, Total cooling output capacity, Sensible heat ratio, Energy saving potential index

Nomenclature

A	constant coefficient
A_a	constant coefficient
A_w	constant coefficient
B	constant coefficient
C	constant coefficient
C_a	air specific heat (J/(kg℃))
C_w	water specific heat (J/(kg℃))
ER	entrainment ratio
e	constant coefficient
k	constant coefficient
m	constant coefficient
n	constant coefficient
P	pressure (hPa)
Q	cooling output capacity (W)

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R	constant coefficient
RH	relative humidity
T	temperature (℃)
T_n	constant coefficient
$\dot{V}_a$	air volume flow rate (m^3/s)
$\dot{V}_w$	water volume flow rate (L/s)
W	moisture content (g/kg)
ε	energy saving potential index
ρ	density (kg/m^3)
Subscripts	
a	air
am	ambient
d	dew point
in	inlet
lat	latent
offc	off coil
pri	primary
s	space
sec	secondary
sen	sensible
tot	total
w	water

1. Introduction

Over the past few years, active chilled beam systems have become a popular alternative to all-air HVAC systems. Their adoption has been significantly fueled because of the space saving, energy conservation, indoor thermal comfort improvement, etc. [1-4]. However, these benefits are quite easy to be diminished even eliminated due to over-conservative, non-optimized system designs and/or operations. For example, in the design phase, system designers may attempt to increase primary air exceeding the space latent load and ventilation requirement considering transient infiltration, purge demand, tendency to put a safety margin, etc. It then leads to such application where the primary air provides sufficient cooling while all the chilled water control valves are closed throughout the entire summer [5]. In this case, active chilled beam terminal units become expensive and the inherited benefits are essentially declined. In the operating phase, most active chilled beam systems have the ability to turn down the primary air, but they may be set at constant air volume mode. That means the primary air is inevitably oversized at part-load conditions and above declination could be further exacerbated. As a consequence, problematic or even erroneous deployments of active chilled beam systems do happen in practice and the perception of those benefits is sometimes just that, a perception, albeit misguided. For instance, Stein et al. [6] held a head to head competition between an active chilled beam 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. This debatable conclusion then led to a series of heated discussions and arguments [7, 8]. Kosonen et al. [9] presented a case study to investigate the feasibility of active chilled beam systems in

Singapore. It was shown that condensation is possible to be prevented if the infiltration is minimized, the primary air is sufficient to extract the humidity of people, and tuning of the automation system is conducted probably. However, four 2.1m active chilled beam terminal units were installed in a 20m^2 office room. Cost/benefit of such an application was questionable.

In reality, the community is still constantly seeking effective methodologies to 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 knowing the operating characteristics and efficiencies of a system is definitely necessary, particularly on latent cooling output capacity [10]. 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 latent cooling output capacity of a system is oversized, the space humidity level becomes unnecessarily low, which results in a lot of energy wasted on treating and transporting the primary air. On the contrary, without sufficient latent cooling output capacity or designing a high humidity level may lead to some condensation issues, which conversely decreases the sensible cooling output capacity of the system due to the condensation avoidance actions. In addition, an exorbitant humidity level may cause growth of fungus and bacteria. Only when the system is perfectly matched with the conditioned space in terms of both sensible and latent cooling output capacities can indoor temperature and relative humidity be simultaneously and accurately controlled. Taking direct expansion air conditioning systems as an example, which has the same concern, Li et al. [11] studied the TCOC 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. [12] extended the study to various space conditions and depicted the system operating characteristics in a more intuitive form. Li et al. [13] carried out a further study to model and predict the system for humidity control purpose. These research results were indeed utilized in developing proper control algorithms [14]. As for active chilled beam systems, there exist a few proactive studies from the point view of latent cooling output capacity. For example, Loudermilk et al. [15, 16] explored the humidity behaviors of spaces served by active chilled beam systems and forth brought some discussions on the system design and operation, but the studies were just in the form of case studies with some prescribed settings. Alexander et al. [17] and Setty [18] listed some application issues for active chilled beam systems, also including the humidity related ones. It was suggested that using energy wheels and wraparound style heat pipes can enhance the system dehumidification ability in humid climates. Additionally, Wahed [19] integrated a thermally regenerated desiccant dehumidification system with an active chilled beam system in Singapore.

Beyond that, it should be noted that in active chilled beam systems the ventilation requirement and latent load of a space 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 the system efficiency. Livchak [5] defined a parameter that represents this kind of important performance of active chilled beam terminal units and called it coil output to primary airflow ratio. As implied by the name, it is the amount of cooling output capacity produced by the chilled water per volume of the primary air used. The parameter can be a measurement of the efficiency of active chilled beam terminal units' design rather than of the systems' operation. In practice, coefficient of performance is the worldwide acknowledged measurement of HVAC components and systems, which is directly related to cooling output capacity instand of fluid volume flow rate. As a result, efficiencies of active chilled beam systems are not well captured until now. Kosonen et al. [20] simulated the sensible and latent cooling output capacities and energy consumption of active chilled beam systems in different cities of Europe and Asia. The sensible cooling output capacity shared by the primary air and chilled water were presented separately. However, the quantities were more synonymous with the space loads than with the capacities of the system. In other words, the study didn't reveal the system characteristics or efficiencies.

In the present paper, the inherent operating characteristics and efficiencies of a 2-way discharge active chilled beam terminal unit were studied in depth. The study was 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 laboratory, it is almost impossible to experimentally capture their influences. As a compromise scheme, this study was 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. The TCOC was no longer the only performance index. The sensible cooling output capacity and latent cooling output capacity were assessed in terms of the SHR. Furthermore, the sensible cooling output capacities provided by the primary air and chilled water were distinguished using a newly defined energy saving potential index ε, which is also the measurement of the system efficiency. The operating constraints between these key operational parameters were reported and sensitivity of the active chilled beam terminal unit to actual primary air and space conditions were evaluated.

This paper is organized as below: in Section 2, the active chilled beam systems, where all the simulations were carried out, are described. Section 3 presents the simulation model and performance indexes. It is followed by Section 4 about the simulation results and discussions. Section 5 draws a conclusion.

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. 1 and the corresponding air treatment processes are depicted in a psychrometric chart in Fig. 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. Generally, 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 theair 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 primary air and the temperature is supposed to be raised 1 ℃ 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 cooling coil as secondary air. Since condensation is strictly avoided, the secondary cooling coil only extracts the sensible heat. The secondary air is cooled from state 2 to state 6 when it passes through the coil. 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.

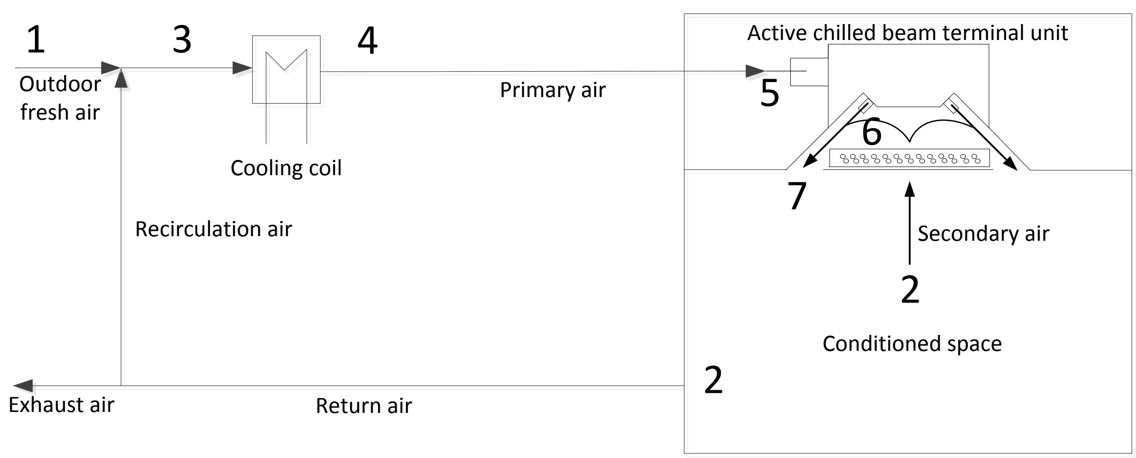

Fig. 1 Schematic diagram of an active chilled beam system combined with a conventional air handling unit

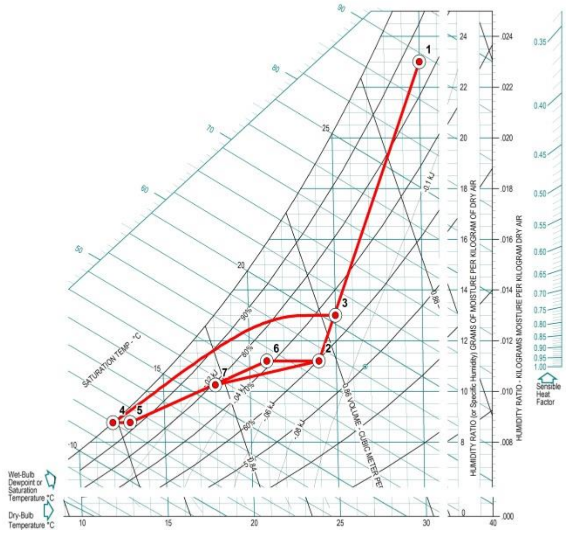

Fig. 2 Psychrometric chart of an active chilled beam system combined with a conventional air handling unit

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. 3 and the air treatment processes are depicted in a psychrometric chart in Fig. 4. The figures can be interpreted in a similar manner as Figs. 1 and 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.

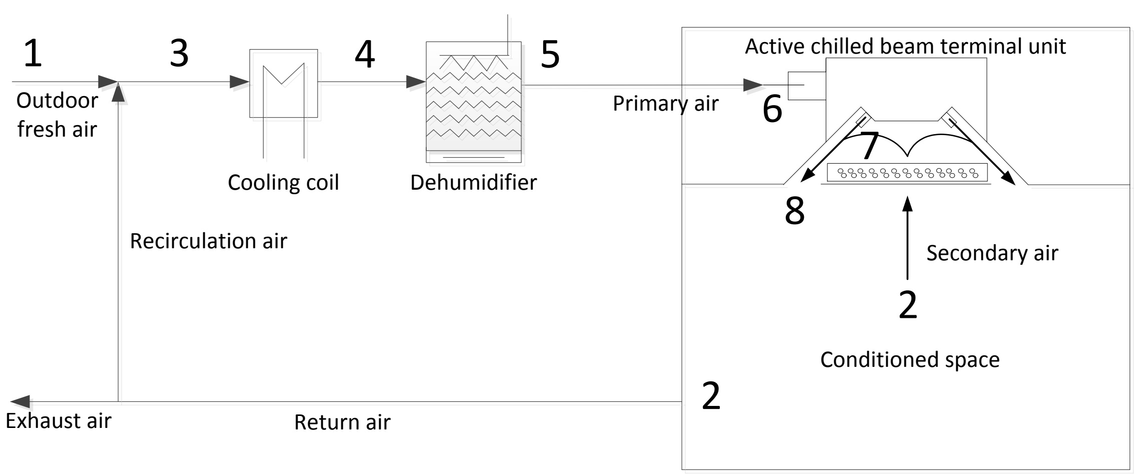

Fig. 3 Schematic diagram of an active chilled beam system combined with an air handling unit and a dehumidifier

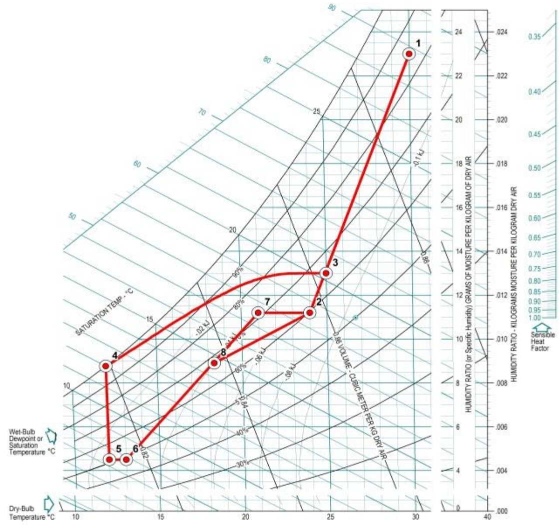

Fig. 4 Psychrometric chart of an active chilled beam system combined with an air handling unit and a dehumidifier

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

3.1 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 output 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 [21] on the basis of Reference [22].

$$W_{pri} = \frac{A * B * 10^{\left(\frac{CT_{offc}}{T_{offc}+T_n}\right)}}{P_{am} - B * 10^{\left(\frac{CT_{offc}}{T_{offc}+T_n}\right)}} \tag{1}$$

where W_{pri} the primary air moisture content, T_{offc} is the off coil temperature, P_{am} is the ambient pressure, A, B, C 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. (1) is replaced by:

$$W_{pri} = \frac{A * B * 10^{\left(\frac{CT_{offc}}{T_{offc}+T_n}\right)} * RH_{pri}}{P_{am} - A * 10^{\left(\frac{CT_{offc}}{T_{offc}+T_n}\right)} * RH_{pri}} \tag{2}$$

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

$$T_{pri} = T_{offc} + 1 \tag{3}$$

3.2 Secondary air model

With respect to the secondary air, it is induced by the primary air and then treated by the heat exchanger. This is actually a very complex process, so the secondary air model given below is directly simplified from a dynamic hybrid model reported elsewhere [23].The model was kept simple and practical, avoiding sophisticated jet flow theories as well as heat transfer theories. In deriving the model using first principles and estimating it experimentally, a reasonable compromise was made between capturing exact underlying physics and suitability for engineering applications. Accuracy of the model was also proven.

At first, 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*:

$$ER = m\left(\dot{V}_{pri}^{2} R\right)^{n} \tag{4}$$

$$\dot{V}_{sec} = \dot{V}_{pri} ER \tag{5}$$

where *m*, *n* and *R* are the unknown constant coefficients need to be estimated. The secondary air moisture content is exact the same as that of the space air, while its temperature T_{sec} is decreased and can be represented as below:

$$
\begin{aligned}
&T_{sec} = \frac{K_1 T_s + K_2 T_{w.in}}{K_3} \\
&K_1 = C_a \rho_a \dot{V}_{sec} - \frac{C_w \rho_w \dot{V}_w k_w \dot{V}_w^{e_w} A_w}{C_w \rho_w \dot{V}_w + \frac{k_w \dot{V}_w^{e_w} A_w}{2}} \left(\frac{1}{2} - \frac{C_a \rho_a \dot{V}_{sec}}{k_a \dot{V}_{sec}^{e_a} A_a} \right) \\
&K_2 = \frac{C_w \rho_w \dot{V}_w k_w \dot{V}_w^{e_w} A_w}{C_w \rho_w \dot{V}_w + \frac{k_w \dot{V}_w^{e_w} A_w}{2}} \\
&K_3 = C_a \rho_a \dot{V}_{sec} + \frac{C_w \rho_w \dot{V}_w k_w \dot{V}_w^{e_w} A_w}{C_w \rho_w \dot{V}_w + \frac{k_w \dot{V}_w^{e_w} A_w}{2}} \left(\frac{1}{2} + \frac{C_a \rho_a \dot{V}_{sec}}{k_a \dot{V}_{sec}^{e_a} A_a} \right)
\end{aligned} \tag{6}
$$

where T_s is the space temperature, $T_{w,in}$ and $\dot{V}_w$ are the chilled water inlet temperature and volume flow rate, C_a, ρ_a, C_w and ρ_w are the specific heat and density of air and chilled water, k_a, e_a and A_a are the heat transfer related constant coefficients need to be estimated as well on the secondary air side, k_w, e_w and A_w are the corresponding counterparts on the chilled water side.

Reviewing the manipulating variables in Eq. (6), the chilled water volume flow rate is the control input to regulate the space temperature and its inlet temperature is an operation variable need to be prescribed with the following equations [21]. 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 operation efficiency. Suppose a safety margin of 1℃, the temperature can be determined via the space dew point temperature $T_{d.s}$.

$$
T_{d.s} = T_n * \left[\frac{C}{\log_{10}\left(10^{\left(\frac{CT_s}{T_s + T_n} \right)} * RH_s \right)} - 1 \right]^{-1} \tag{7}
$$

$$
T_{w.in} = T_{d.s} + 1 \tag{8}
$$

3.3 Performance indexes

As only the primary air can provide the latent cooling output capacity, then in steady state, that capacity Q_{lat} can be obtained with the primary air and space statuses. From the polynomial curve fit to the table 2.1 of reference [24],

$$
Q_{lat} = -\left(2500.8 - 2.36T_s + 0.0016T_s^2 - 0.00006T_s^3 \right) \rho_a \dot{V}_{pri} \left(W_{pri} - W_s \right) \tag{9}
$$

To evaluate Eq. (9), the space moisture content W_s is proactively calculated in a similar manner as Eq. (1) and Eq. (3).

$$W_s = \frac{A * B * 10^{\left(\frac{CT_s}{T_s + T_n}\right)} * RH_s}{P_{am} - A * 10^{\left(\frac{CT_s}{T_s + T_n}\right)} * RH_s} \tag{10}$$

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

$$Q_{sen} = -C_a \rho_a \dot{V}_{pri} \left(T_{pri} - T_s\right) - C_a \rho_a \dot{V}_{sec} \left(T_{sec} - T_s\right) \tag{11}$$

With above equations, the performance indexes can be easily obtained. The TCOC comes first. According to the definition, it is the sum of the sensible cooling output capacity and latent cooling output capacity.

$$Q_{tot} = Q_{sen} + Q_{lat} \tag{12}$$

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

$$\mathrm{SHR} = \frac{Q_{sen}}{Q_{tot}} \tag{13}$$

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 output capacity handled by the chilled water to the total one.

$$\varepsilon = \frac{Q_w}{Q_{sen}} \tag{14}$$

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

$$Q_w = -C_a \rho_a \dot{V}_{sec} \left(T_{sec} - T_s\right) \tag{15}$$

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 [21] 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. The active chilled beam terminal unit investigated in the present study was a specially designed 2-way discharge unit. It consisted of a primary air plenum, twenty nine 9mm inner diameter circular nozzles evenly distributed on each side, a mixing chamber, and a 2-row fin and tube heat exchanger. The testing were performed on a pilot plant that consists of two physical loops. The air loop was designed to blow primary air into the terminal unit primary plenum and force it through the nozzles, while the water loop was used to circulate chilled water between the heat exchange and a dedicated chiller system. More details about the terminal unit, experimental setup, and identification of the unknown efficiencies were reported in the reference [23]. The results were briefly summarized in Table 1.

Table 1 Summary of unknown parameters

a	b	R	$c_a A_a$	e_a	$c_w A_w$	e_w
1204.1	0.107	0.003997	698	0.5	788	0.5

The simulation model was implemented and solved in the Matlab environment. In total, there were 13 sets of simulations carried out under different combinations of typical primary air and space conditions. The combinations were recorded in Table 2. In each simulation, the primary air volume flow rate and the secondary chilled water volume flow rate were 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 could 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 was progressively increased from 0.015m^3/s to 0.05m^3/s with a step increment of 0.005m^3/s. As for the chilled water volume flow rate, it was also carefully determined within a reasonable range from 0.02 L/s to 0.04 L/s with a step increment of 0.002L/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 was desirable, while the pressure drop caused by the high volume flow rate should be taken into account.

Table 2 Summary of simulation conditions

Set No.	T_{pri} (℃)	RH_{pri} (%)	T_s (℃)	RH_s (%)
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

A. Operating characteristics and efficiencies

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

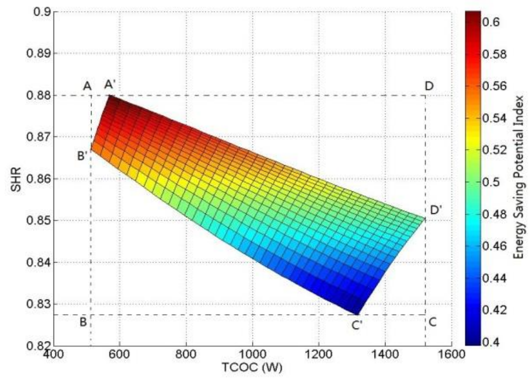

Fig. 5 Simulation result of set 1

With reference to [12], the simulation result was represented by plotting the TCOC and SHR as x-axis and y-axis in the same diagram, Fig. 5. The energy saving potential index ε was depicted via the color bar. In the figure, the primary air and chilled water volume flow rates were 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 were all increased with the increasing of the chilled water volume flow rate and vice versa. At each chilled water volume flow rate, the TCOC was increased while another two parameters were decreased with the increasing of the primary air volume flow rate. The maximum and minimum values of the TCOC were 1520W and 512 W and the SHR counterparts were 0.827 and 0.88. Provided with these limits, it would be easy to have a common false impression that the TCOC and SHR can be freely combined within the limits, but actually the TCOC and SHR were correlated and mutually constrained with a trapezoid A'-B'-C'-D'. That meant the preferred operating range should be A'-B'-C'-D' rather than A-B-C-D. In addition, there was a positive correlation property between the SHR and energy saving potential index ε. When the SHR was increased from 0.827 to 0.88, the index ε was also increased from 0.40 to 0.61. It was consistent with the general application guideline that active chilled beam systems are an effective means to manage large sensible load.

B. Influences of the primary air temperature

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

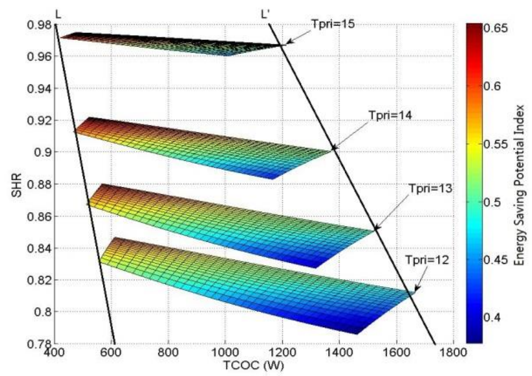

Fig. 6 Simulation results of sets 1-4

Fig. 6 collected the simulation results of sets 1-4. Given any particular primary air temperature, the simulation result was described by a colorful trapezoid as same as the one shown in Fig. 5 and the characteristic trapezoids could be analyzed in the same way. For simplicity, these analyses were ignored here. It was observed that the characteristic trapezoids were constrained by the straight lines L and L', while the trapezoids had different positions and color distributions. These influences could be interpreted from the following four aspects:

1. As the primary air temperature increasing, the characteristic trapezoid was shifted to the left. In other words, the TCOC was decreased. For example, the maximum TCOC was changed from 1657W, to 1520W, then to 1369W, and finally to 1206W. This influence was greater at relatively high temperatures. For instance, when the temperature was increased from 14℃ to 15℃, the variation of TCOC was 163W, while it was only 137W when the temperature was increased from 12℃ to 13℃.
2. The characteristic trapezoid was lifted up and the SHR was improved when the primary air temperature was increased. The maximum value of SHR was raised from 0.85, to 0.88, then to 0.92, and finally to 0.97. As same as the influence on the TCOC, the influence on the SHR was greater at high temperatures.
3. With the increasing of the primary air temperature, dominant tone of the characteristic trapezoid gradually changed from blue to red. The maximum energy saving potential index ε was 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 was reducing the trapezoid size, so as to applicable ranges of the terminal unit. Furthermore, at a high primary air temperature (e.g., 15℃), the characteristic appearance deviated from the rest ones, i.e., a trapezoid shape became less obvious.

C. Influences of the space temperature

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

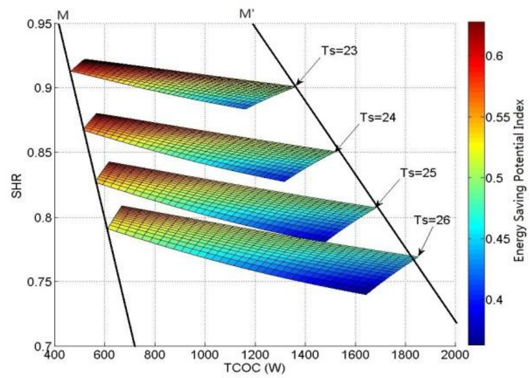

Fig. 7 Simulation results of sets 1 and 5-7

The simulation results of group C were illustrated in Fig.7. As same as Fig 6, this figure was interpreted according to the positions, tones, and sizes of the characteristic trapezoids. As a result, similar conclusions were obtained from the same four aspects. Simply put, decreasing the space temperature had 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 had competitive influences on the TCOC but slight smaller ones on the SHR and index ε. For example, with the same temperature variations of 4℃, variation of the minimum TCOC was 139W and the responding value in Fig. 6 was 136W, but variation of the minimum SHR were 0.143 and 0.174 and variation of the minimum index ε were 0.0565 and 0.0696. In addition, unlike those characteristic trapezoids shown in Fig. 6, shapes of the trapezoids were all maintained.

D. Influences of the space relative humidity

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

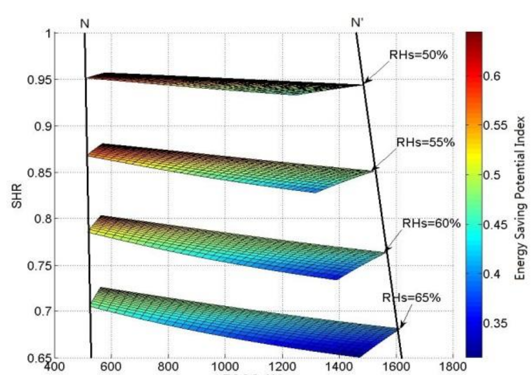

Fig. 8 Simulation results of sets 1 and 8-10

The simulation results of group D were described in Fig.8. Comparing Fig. 8 with Fig. 6 and Fig 7, decreasing the space relative humidity had analogous influences as increasing the primary air temperature and decreasing the space temperature, while it had minimum influences on the TCOC but maximum influences on the SHR and index ε. For example, the straight lines N and N' were almost perpendicular to the TCOC axis and the SHR and index ε varied about 0.08 and 0.05 with a variation of the space relative humidity of 5%.

E. 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 was predefined at 40%, 60%, 80%, and 100%. Since the primary air was no longer saturated, the simulations of sets 11- 13 were conducted with the active chilled beam terminal unit in conjunction with both the air handling unit and the liquid desiccant dehumidifier.

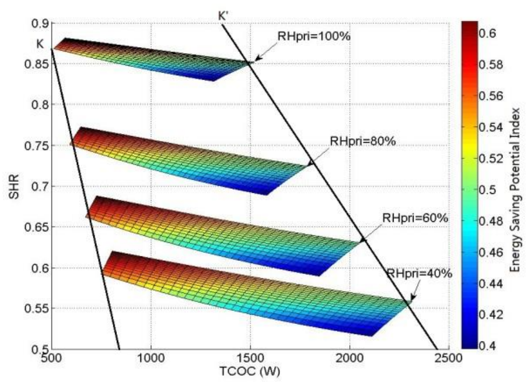

Fig. 9 Simulation results of sets 1 and 11-13

The simulation results of Group E were presented in Fig. 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 was 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 was greatly reduced to 0.52. To show this consequence better, all the 13 sets of simulations were summarized in Fig. 10. It could be seen that the polygon A-B-C-D-E was 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 was enlarged into the polygon A-B-F-G-H-D-E when introducing the liquid desiccant dehumidifier.

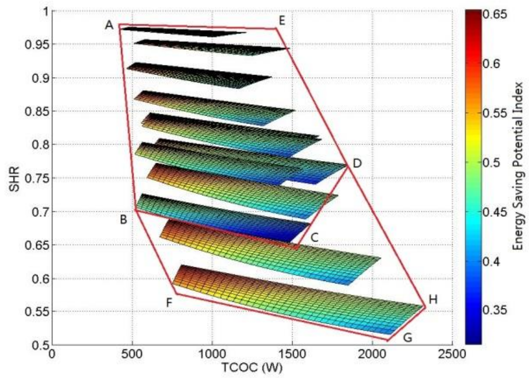

Fig. 10 Simulation results of sets 1-13

5. Conclusion

In this paper, a series of simulation studies on the inherent operating characteristics and efficiencies of a 2-way discharge active chilled beam terminal unit were investigated under variable air volume mode. Given fixed primary air and space conditions, the TCOC, SHR, and energy saving potential index ε 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 findings obtained here 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 07, 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, this paper 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.

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