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Abstract: This paper presents a performance comparison study of heat exchangers with different circuitry arrangements for active chilled beam (ACB) applications. Supported by a two-way discharge ACB terminal unit and a performance test pilot plant, four 2-rows plain fin and tube heat exchangers, namely, 1-circuit, 2-circuits, 4-circuits, and 8-circuits, are investigated under a wide range of chilled water volume flow rates. The water side thermal and hydraulic characteristics are evaluated. The heat transfer rates are compared under three sets of criteria: identical water volume flow rate, identical pressure drop, and identical pumping energy consumption. Meanwhile, the comparisons are also performed in terms of two non-dimensional parameters: heat exchanger effectiveness and performance index. The study shows that different circuitry arrangements should be preferred in different operation conditions and under different evaluation criteria, while the 2-circuits arrangement is the most comprehensive and reasonable option rather than the 1-circuit one that is widely used in all available ACB products.

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Sep 23, 2013

Dear Editor,

Kindly please find enclosed the manuscript: *Performance comparison of heat exchangers with different circuitry arrangements for active chilled beam applications*, by Can Chen et al., to be submitted as a Full Length Article to Energy and Buildings. All co-authors have seen and agree with the contents of the manuscript and there is no financial interest to report. We certify that the submission is not under review at any other publication.

This manuscript presents a performance comparison study of heat exchangers with different circuitry arrangements for active chilled beam (ACB) applications. Supported by a two-way discharge ACB terminal unit and a performance test pilot plant, four 2-rows plain fin and tube heat exchangers, namely, 1-circuit, 2-circuits, 4-circuits, and 8-circuits, are investigated under a wide range of chilled water volume flow rates. The water side thermal and hydraulic characteristics are evaluated. The heat transfer rates are compared under three sets of criteria: identical water volume flow rate, identical pressure drop, and identical pumping energy consumption. Meanwhile, the comparisons are also performed in terms of two non-dimensional parameters: heat exchanger effectiveness and performance index. The study shows that different circuitry arrangements should be preferred in different operation conditions and under different evaluation criteria, while the 2-circuits arrangement is the most comprehensive and reasonable option rather than the 1-circuit one that is widely used in all available ACB products.

We look forward to hearing from you soon.

Thank you very much for your time and consideration.

Yours sincerely,

Wenjian Cai

Reviewer 2

First and foremost, the authors would like to express their sincere gratitude and appreciation to the reviewer.

The uncertainty of the calculated value of Q has been revised by using a rigorous method. As suggested by the reviewer, the method outlined by Wheeler and Ganji (2010) has been adopted, which takes the sensitivity coefficients into consideration. Please refer to the revised manuscript Figure 3 and Table 2.

Some corrections have been performed to clarify this uncertainty analysis part better. The scatter in Q comes from two main error sources: slight different experimental conditions and measurement errors of transmitters. Due to the complex and unknown dependences, the effects of experimental conditions are difficult to be characterized or quantified. Furthermore, the experimental conditions summarized in Table 1 are almost same. As a result, this error source is ignored and the uncertainty analysis is focused on the main errors caused by the measurement transmitters. Table 2 has been simplified accordingly. Because each measurement is independent, the root of the sum of the squares method given in Wheeler and Ganji (2010) becomes feasible. It is consistent with the example given in the comment of the reviewer. The uncertainty of temperature difference is 0.5 °C, when the uncertainties for inlet and outlet temperatures are 0.3 °C and 0.4 °C respectively. The bias uncertainty values have been adopted instead of the relative ones for the temperatures and final Q and Table 2 has been revised as well.

Performance comparison of heat exchangers with different circuitry arrangements for active chilled beam applications

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Abstract: This paper presents a performance comparison study of heat exchangers with different circuitry arrangements for active chilled beam (ACB) applications. Supported by a two-way discharge ACB terminal unit and a performance test pilot plant, four 2-rows plain fin and tube heat exchangers, namely, 1-circuit, 2-circuits, 4-circuits, and 8-circuits, are investigated under a wide range of chilled water volume flow rates. The chilled water side thermal and hydraulic characteristics are evaluated. The heat transfer rates are compared under three sets of criteria: identical water volume flow rate, identical pressure drop, and identical pumping energy consumption. Meanwhile, the comparisons are also performed in terms of two non-dimensional parameters: heat exchanger effectiveness and performance index. The study shows that different circuitry arrangements should be preferred in different operation conditions and under different evaluation criteria, while the 2-circuits arrangement is the most comprehensive and reasonable option rather than the 1-circuit one that is widely used in all available ACB products.

Keywords: Active chilled beam, Heat exchanger, Circuitry arrangement, Thermal performance, Hydraulic performance, Performance comparison

Nomenclature

A	effective heat transfer area (m^2)
A_t	tube section area (m^2)
A_w	water side effective heat transfer area (m^2)
C	empirical constant
C_a	air specific heat ($\text{J/kg}^\circ\text{C}$)
C_w	water specific heat ($\text{J/kg}^\circ\text{C}$)
d	inside tube diameter (m)
dQ	local heat transfer rate (W)
dT	local temperature difference ($^\circ\text{C}$)
e	sensitivity coefficient
f	lumped Fanning friction factor
h	overall heat transfer coefficient ($\text{W/m}^2^\circ\text{C}$)
h_w	water side convection heat transfer coefficient ($\text{W/m}^2^\circ\text{C}$)
K	integrated constant
k	water thermal conductivity ($\text{W/m}^\circ\text{C}$)
L	water path length (m)
m	empirical constant
Nu	Nusselt number
n	empirical constant

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P_{pump}	pumping energy consumption (W)
Pr	Prandtl number
Q	heat transfer rate (W)
R_a^*	integrated thermal resistance (°C/W)
Re	Reynolds number
T_{sec}	outlet secondary air temperature (°C)
$T_{w,in}$	inlet chilled water temperatures (°C)
$T_{w,out}$	outlet chilled water temperatures (°C)
T_{zone}	zone environment temperature (°C)
u	water velocity (m/s)
$\dot{V}_{sec}$	secondary air volume flow rate (m ³ /s)
$\dot{V}_w$	air volume flow rate (m ³ /s)
W	total uncertainty
x	uncertainty source
ΔP	water fluid pressure drop (Pa)
ΔP_{static}	static pressure drop (Pa)
ε	heat exchanger effectiveness
η	performance index
μ	water absolute viscosity
ρ	water density (kg/m ³)
ρ_a	air density (kg/m ³)

1. Introduction

Active Chilled Beam (ACB) combining with Dedicated Outdoor Air System (DOAS) is an alternative HVAC scheme to currently popular all-air Variable Air Volume (VAV) system [1-5]. It has been successfully utilized in Europe for more than 20 years and there are increasing interests in this kind of integrated system in North America and Asian countries in the past decade. With the entire latent cooling load and ventilation load treated by the primary air and the most sensible cooling load handled by the secondary chilled water, the energy consumption can be substantially reduced. According to several energy retrofit projects in America [1], the total power demand for ACB is about 25%-30% less than that for conventional VAV system. These energy savings are mainly attained in the following aspects: 1) ACB requires less air flow than VAV for the same cooling load, so perpetual fan energy savings are created; 2) with higher chilled water temperatures, the dedicated chiller for ACB can operate at 15%-20% higher efficiency than that for VAV; 3) the need for energy-consuming reheat of cooled supply air in VAV is eliminated. In addition, ACB offers some other benefits, including better indoor thermal comfort by

flexible temperature and humidity controls, space saving via the replacement of large air ductwork with small water pipework, less air drafts and low acoustic signature, and investment reduction etc.

In order to further explore and make full use of the benefits of ACB, several related studies have been conducted in recent years. Kenneth et al. [2, 3] provided some design guidelines when selecting, sizing, locating ACB terminal unit, which suggested the space design humidity levels could be partially relaxed and the primary air volume flow rate could be further reduced. Andrey et al. [4] and Darren et al. [5] did similar considerations to improve ACB effectiveness and maximize use of the chilled water for sensible cooling load. Nastase et al. [6-8] studied a kind of novel nozzle using inclining lobes for air diffuser applications, including ACB, to enhance the air mixing and entrainment. Mike et al. [9] proposed a simplified method to measure the air entrainment ratio of ACB terminal unit. Cao et al. [10-12] explored some turbulent air flow behaviors of the attached plane jet discharged from ACB terminal unit using Particle Image Velocimetry (PIV). Fred et al. [13] reported the differences of the design of actual ACB versus how they are modeled in existing energy simulation software, which are usually ignored due to acceptable simulation results. It is noted, however, there is no study can be found so far in the existing literatures that deals with the heat exchanger issues of ACB terminal unit.

In reality, heat exchanger for HVAC applications has intensified over the last years [14-16]. As an essential component of ACB terminal unit which has a big effect on the system performance, fin and tube heat exchanger should be also exploited further. Among several heat exchanger improvement technologies currently available, circuitry adjustment is an effective one, which can be realized without any effect on the heat exchanger compactness and complexity in structure. If the water circuits can be properly branched and joined, heat exchanger thermal and hydraulic performances can be substantially enhanced at same time. Many research results in this field are available in the literatures. Guo et al. [17] provided an idea that for counterflow heat exchanger high effectiveness could be owing to the most uniform local temperature differences between the two flowing fluids compared with other heat exchangers. The so-called uniformity principle of temperature difference was proposed, which became an important guideline for circuitry adjustment. Guo et al. [18] successfully conducted theoretical analysis and experimental confirmation of the uniformity principle with thirteen types of heat exchangers. Cabezas-Gomez et al. [19, 20] addressed

same confirmations theoretically and numerically with new flow arrangements as well. Moreover, the circuitry adjustment method is extended to other applications. Wang et al. [21] finished an experimental study including six 1-circuit and two 2-circuits heat exchangers to investigate the effect of circuitry on the performance of wavy finned condensers. Liang et al. [22, 23] attempted a numerical and experimental study of refrigerant circuitry of evaporator. Miura et al. [24] tested thirty two plate heat exchanger with different circuitry arrangements experimentally to explore the effect of circuitry arrangement on the pressure drop. Additionally, more and more applications of advanced algorithms [25-27] to optimize the heat exchanger circuitry arrangement have suggested the potential benefits of excellent circuitries.

Considering that the sophisticated entrainment effect inside ACB contributes a sensitive and unknown air side configuration for the heat exchanger, the investigation on circuitry adjustment is more important to be introduced into ACB terminal unit. It can be easily implemented without effect on the air side. That is the focus of the present study. In this paper, the authors presented a research result on the selection a proper water circuitry arrangement for ACB applications through an experimental comparison on four different circuitry arrangements. The heat transfer rates are compared under three sets of criteria: identical 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 [28]. A conclusion which supports the adoption of 2-circuits heat exchanger is finally obtained.

This paper is organized as below: In Section 2, ACB working principle is briefly described for a general understanding; In Section 3, a theoretical analysis is given to reveal some fundamental theories behind the study; The complete experimental setup, testing procedure and the experimental uncertainties analysis are provided in Section 4; In Section 5, the resultant performance comparisons and discussion are presented; Finally a conclusion is drawn in Section 6.

2. ACB working principle

Schematic diagram of an ACB terminal unit is depicted in Fig. 1. It consists of a primary air plenum, a number of uniquely designed nozzles, a mixing chamber and a fin and tube heat exchanger. A fixed pressure is maintained in the primary plenum, so that a certain amount of primary air can be forced through

the nozzles, through the mixing chamber and out into conditioned-zone. The nozzles are designed in such a way that a negative pressure kernel is generated in the region beyond each nozzle exit. It induces the flow of zone air named secondary air through the heat exchanger and into the mixing chamber. This induction of the secondary air by virtue of nozzle design is called entrainment effect, the better the design the stronger it is. The secondary air will be cooled when it moves through the heat exchanger. The resultant mixed air supplies into the zone through linear slots on the edges of the unit.

Generally, the primary air is continuously supplied by a DOAS. The typical temperature is 10 -15 °C with low relative humidity. It satisfies the entire latent cooling load, ventilation load and a small part of sensible cooling load. Although most ACB do have the ability to turn down primary air volume flow rate, they are usually setup and operate in a constant air volume configuration for simplification. The chilled water supplied by a dedicated chiller is at an evaluated temperature, typically 14 -16 °C, as compared with the zone air temperature, which is typically 23-25°C. As a consequence, most of the sensible cooling load can be handled by the secondary chilled water. Zone temperature control is primarily accomplished by varying the secondary chilled water flow rate or its supply temperature to the terminal units without effect on space ventilation and/or dehumidification. It typically produces a 3.5 to 4.5 °C swing in the secondary air temperature, which affects a 50-60% turndown in the unit's sensible cooling capacity. This is sufficient for the control of interior spaces (except conference areas) where sensible loads do not tend to vary significantly.

3 Theoretical analysis

Prior to the experimental investigation, some fundamentals behind it are briefly discussed. The description is not a complete description of the thermal and hydraulic characteristics, in general, but still contains a wealth of useful information which is helpful to understand the following experimental results.

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

$$dQ = hAdT \quad (1)$$

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

$$\frac{1}{hA} = \frac{1}{h_w A_w} + R_a^* \quad (2)$$

Referring to dimensional analysis, Nusselt number as well as the convection heat transfer coefficient for a single phase flow can be calculated by Reynolds number Re and Prandtl number Pr as:

$$Nu = \frac{h_w d}{k} = C Re^m Pr^n = C \left(\frac{\rho u d}{\mu} \right)^m \left(\frac{C_w \mu}{k} \right)^n \quad (3)$$

Although these properties depend on the water temperature, they are always evaluated at bulk temperature thus avoiding iteration. Moreover, they are assumed to be constant, because the water bulk temperature variation can be ignored. If the water is measured by its volume flow rate $\dot{V}_w$, the water side heat transfer coefficient is rewritten as:

$$h_w = K \dot{V}_w^m \quad (4)$$

$$K = \frac{k}{d} C \left(\frac{\rho d}{\mu} \right)^m \left(\frac{C_w \mu}{k} \right)^n \left(\frac{1}{A_t} \right)^m \quad (5)$$

Substituting Eqs. (2) and (4) into Eq. (1), results the following equation

$$dQ = \left(\frac{1}{K \dot{V}_w^m A_w} + R_a^* \right)^{-1} dT \quad (6)$$

From Eq. (6), it can be observed that local heat transfer rate is affected by two variables: chilled water volume flow rate $\dot{V}_w$ and local temperature difference dT . If the total heat transfer rate of a heat exchanger is considered in an integral form of Eq. (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 change have opposite effect on the total heat transfer rate. For example,

using multi-circuits heat exchanger instead of single circuit one leads to higher and more uniform temperature difference but lower water volume flow rate as well as heat transfer coefficient. Consequently, it is difficult to theoretically maximize the heat transfer rate because it involves the optimization of partial differential equations with some constraints, which is still an unsolved problem.

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

$$\Delta P = \frac{4fL}{d} \frac{1}{2} \rho \left(\frac{\dot{V}_w}{A_t} \right)^2 + \Delta P_{static} \quad (7)$$

where L is the water path length and ΔP_{static} is the static pressure drop. From Eq. (7), it can be directly observed that water volume flow rate and the water circuit length are important factors of the total pressure drop. Moreover, the Fanning friction factor is usually not a constant, which changes with Reynolds number as well.

4. Experimental investigation

4.1 Experimental setup

The experiments are conducted on a specially designed two-way discharge ACB terminal unit as shown in Picture 1. Its face dimensions are of 0.6m×1.2m and twenty nine 9mm inner diameter circular rubber nozzles are equally distributed on each side. The plain fin and tube heat exchangers are measured at 0.3m×1.1m×0.055m. The aluminum fin thickness and pitch are 0.5mm and 4.35mm respectively. There are two stagger tube rows totally 16 half inch copper tubes. The horizontal distance between center lines of tubes is 31.7mm and the vertical distance between center lines of rows is 27.3mm. For simplicity, four different circuitry arrangements investigated in the present study are mainly distinguished by their water circuit numbers. As schematically illustrated in Fig. 2, the four 2-rows plain fin and tube heat exchangers contain 1, 2, 4 and 8 circuits respectively. The only difference is their various tube connections that can be attained by different headers.

An experimental setup is developed to test ACB terminal unit performance. As presented in Picture 2, it mainly consists of two physical loops: air loop and water loop. The air loop is provided to blow air into the ACB primary air plenum and force it through the nozzles, so that secondary air can be induced across the heat exchanger. The water loop is designed to supply chilled water from a dedicated chiller system to ACB heat exchanger.

In the air loop, two SAER ELETTROPOMPE 350W 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 is measured by a TSI model 8710 DP-CALCTM micromanometer with error of $\pm 3\%$. The micromanometer covers the whole heat exchanger via a customized hood and the air volume flow rate reading is automatically calculated with the velocity measurements of a dedicated velocity matrix. Meanwhile, the zone temperature is measured by a temperature probe integrated with this micromanometer. In the water loop, a 750W water circulating pump is placed in the cycle to circulate 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 $\pm 0.3^\circ\text{C}$ 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 chilled water by a vapor compressor cycle and stores it in an insulated water bath with a 4kW immersion heater. The desired temperatures of the stored chilled water as well as the inlet chilled water are controlled by a SHIMADEN PID controller. All motors, including the fans and pump, are equipped with VSDs, 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.

4.2 Experimental procedure

As the main concern of this study is to investigate and compare the water side performances of different circuitry arrangements, the air loop is fixed at a common chamber pressure operation point, which results in

constant primary and secondary air volume flow rates as well. The heat exchangers have exact same air side configuration, so all the uncertainties caused by the entrainment effect can be eliminated. Except from the chilled water volume flow rate, all other independent variables, including the secondary air volume flow rate, the zone environment temperature, and the inlet chilled water temperature are tried to keep constant during the experiments.

The experimental parameters setting and their variations are summarized in Tab. 1, by rejecting some unexpected environmental changes.

All the experimental data should be recorded under steady-state, so the attention is focused on the thermal equilibrium of the heat exchangers. Since the outlet secondary air temperature cannot be available, the thermal equilibrium is only determined via the chilled water loop. Once the outlet chilled water temperature is within $\pm 0.1^{\circ}\text{C}$ limits for 20 minutes, the steady state is confirmed.

As stated in the theoretical analysis part, a heat exchanger is typically characterized by the heat transfer rate and pressure drop. However, it must be noted that, when the 1-circuit arrangement is replaced by multiple-circuits arrangements, the objective is to increase the heat duty using a smaller amount of pumping power. Thus, such an assessment concerning pumping energy consumption is considered essential. The pressure drop is just intermediate measurement for the pumping energy consumption. Unfortunately, there exist massive conflicts between them. In order to provide a comprehensive performance comparison, the three widely used constraints are adopted: identical flow rate, identical pressure drop, identical pumping power as well as the heat exchanger effectiveness and performance index. Their characteristics are briefly displayed in the following:

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

During the experiments, the pressure drop can be measured directly and the following equations are used to calculate the heat transfer rate as well as associated performance indicators. The heat transfer rate equals to either the heat decrement of the air or heat increment of the water.

$$Q = -C_a \rho_a \dot{V}_{\text{sec}} (T_{\text{sec}} - T_{\text{zone}}) \quad (8)$$

$$Q = C_w \rho_w \dot{V}_w (T_{w.out} - T_{w.in}) \quad (9)$$

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

$$P_{\text{pump}} = \frac{\dot{V}_w \Delta P}{0.8} \quad (10)$$

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

$$\varepsilon = \frac{\text{Max}[(T_{w.out} - T_{w.in}), (T_{\text{zone}} - T_{\text{sec}})]}{T_{\text{zone}} - T_{w.in}} \quad (11)$$

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

$$\eta = \frac{Q}{P_{\text{pump}}} \quad (12)$$

4.3 Uncertainty analysis and repeatability test

Up to now, there is no testing result can be found in the existing literatures or datasheets from the manufactures. However, in order to verify the experimental system and evaluate reliability and accuracy of the measurements, it is necessary to carry on some uncertainties analysis and repeatability tests and show

that all sets of 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 the complex and unknown dependences, the effects of experimental conditions are difficult to be characterized or quantified. Furthermore, the experimental conditions summarized in Tab. 1 are almost same. 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 some uncertainties. The uncertainties occurred for independent variables can be directly estimated from the accuracies of the transmitters. For the dependent variables, the uncertainties can be 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 W can be made using the following equation:

$$W = \left[(e_1 x_1)^2 + (e_2 x_2)^2 + \dots + (e_n x_n)^2 \right]^{1/2} \quad (13)$$

The total uncertainties of independent and dependent variables during the measurements are presented in Tab. 2.

Figs. 3 and 4 illustrate the experimental results of repeatability test for 2-circuits circuitry arrangement. It can be easily observed that the practical heat transfer rate 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.

5. Testing results and discussion

Fig. 5 presents the variation of heat transfer rate of the heat exchangers with different water circuits under different water volume flow rate. For any specified circuitry arrangement, the increase of water volume flow rate yields a larger water volume flow rate, while the temperature difference distribution is almost unchanged, thus, a higher heat transfer rate is achieved with reference to Eq. (6). For different circuitry 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 rate and the effects cannot be quantified accurately. For example, 1-circuit arrangement has the highest heat transfer rate when the water volume flow rate is less than 0.078L/s, while 2-circuits arrangement has the highest heat transfer rate when the water volume flow rate is more than 0.078L/s. Comparing the heat transfer rate at an identical volume flow rate, 1-circuit arrangement or 2-circuits arrangement should be recommended in different operation conditions.

Fig. 6 shows the variation of pressure drop of the heat exchangers with different water circuits under different water volume flow rate. As indicated in Eq. (7), the pressure drop increases with the increase of water volume flow rate for any given circuitry arrangements, due to same water path length and static pressure drop. Comparing different circuitry arrangements at a 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 drop. 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. 7 is derived to show the variation of heat transfer rate under different pressure drop. It can be seen that 8-circuits arrangement has the highest heat transfer rate when the pressure drop is lesser than 7.5kPa; otherwise, 2-circuits one has the highest one. Thus, 8-circuit arrangement or 2-circuits arrangement is favored.

Fig. 8 illustrates the variation of heat transfer rate of the heat exchangers with different water circuits under different pumping energy consumption. For any individual circuitry arrangement, increase of pumping energy consumption means the increases of water volume flow rate as well as associated pressure drop based on Eq. (10), while it improves the heat transfer rate. For different circuitry 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 rate according to Fig. (5), which depends on how much the water volume flow rate can be increased. As shown in Fig. 8, 4-circuits or 8-circuits arrangement has the highest heat transfer rate.

Figs. 9 and 10 present the variation of effectiveness and performance index of the heat exchangers with different water circuits under different water volume flow rate respectively.

Summarizing above experimental results in Tab. 3, it can be concluded that different circuitry arrangements are recommended with respect to different operation conditions and evaluation criteria.

In practice, the ACB system operation condition is given in terms of the water volume flow rate and the pressure drop, which are typically between 0.03-0.12 L/s and lower than 30kPa respectively [29-32]. Since there is not a widely recognized optimum operation condition, it is nearly impossible to assure the best circuitry arrangement accordingly. To be comprehensive, 2-circuits arrangement is the most reasonable selection among these options in the full range of operation conditions. It can achieve competitive even higher heat transfer rate and heat exchanger effectiveness, considerable pressure drop and pumping energy consumption reduction, and improved performance index compared with 1-circuit arrangement. Compared to 4-circuits arrangement or 8-circuits one, it can offer significant heat transfer rate and heat exchanger effectiveness enhancement with little penalty of pressure drop, pumping energy consumption and performance index.

6. Conclusions

In this work, four 2-rows plain fin and tube heat exchangers with different water circuits were the experimental investigated under different water volume flow rate. Comprehensively speaking, 2-circuits arrangement is the best selection rather than the 1-circuit one that is widely used in all available ACB products. This unexpected conclusion means a great deal of potential improvement for the existing ACB terminal units. More importantly, it is confirmed that heat exchanger thermal and hydraulic performances could be simultaneously enhanced via advisable water circuitry arrangement, which will have significantly practical influence on the heat exchanger design for ACB applications in the future.

Acknowledgements

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Fig.1. Schematic diagram of an ACB terminal unit (Lateral Cross Section View)

Picture 1 Two-way discharge ACB terminal unit

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

Picture 2 Pilot plant of ACB test

Fig.3. Heat transfer rate repeatability test of 2-circuits heat exchanger

Fig.4. Pressure drop repeatability test of 2-circuits heat exchanger

Fig.5. Variation of heat transfer rate for different water circuits

Fig.6. Variation of pressure drop for different water circuits

Fig.7. Variation of heat transfer rate for different water circuits under different pressure drop

Fig.8. Variation of heat transfer rate for different water circuits under different pumping energy

Fig.9. Variation of effectiveness for different water circuits

Fig.10. Variation of performance index for different water circuits

Table 1 Summary of experimental parameters setting

Parameter	Set point	Practical range	Unit
Environment temperature	24	23.7-24.1	°C
Secondary air volume flow rate	360	355-365	m ³ /h
Inlet chilled water temperature	14	13.8-14.1	°C
Chilled water volume flow rate	144-504	144-504	L/h

Table 2 Summary of experimental variables' uncertainties

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

Table 3 Summary of circuitry arrangement recommendations

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

***Highlights (for review)**

- This paper gives the first reported research about heat exchanger issues for ACB.
- Four different circuitry arrangements are investigated.
- The effectiveness of circuitry adjustment to ACB terminal unit is confirmed.
- The unexpected conclusion indicates much potential improvement of ACB system.