PERFORMANCE IMPROVEMENT OF WATER-COOLED SCREW CHILLER UNDER PART LOAD OPERATION CONDITIONS BY THE PARALLELING THROTTLE MECHANISM

by

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Currently, water-cooled screw chiller is widely applied in commercial and industrial buildings, and the energy consumption of water chiller even could cover 70% of air-conditioning energy consumption in the most adverse operating conditions. Moreover, a number of chillers are running at a low rate under part-load state. In order to reduce the energy consumption of water-cooled screw chillers under part-load state, the paper presents the paralleling throttle mechanism. Through experimental study, under part-load state, it can be found that the by-pass tube diameter Ø16 has the most positive effect on the performance of water-cooled screw chiller in four by-pass tube diameters. The oblique access mode is better than the vertical access mode, while the spiral access mode has little influence on the chiller performance. Because of paralleling throttle using, the exergetic loss of the evaporator and chiller is able to be reduced by 3.4%-15.5% and 0-6.7%, respectively. Meanwhile, the coefficient of performance of chiller can be enhanced by 0.2%-1.6%, and discharge temperature can be reduced by 0.4-2.7 °C. In addition, the economic and environmental benefits of the advanced water-cooled screw chiller are more evident than the conventional water-cooled screw chiller's.

Key words: *water-cooled screw chiller, performance improvements, paralleling throttle mechanism, exergetic analysis*

Introduction

Generally, the air-conditioning energy consumption covers 30%-60% of building energy consumption [1], and the refrigeration source accounts for a large proportion of air conditioning energy consumption and is about 60%-70% [2]. In specially, water-cooled screw chiller is widely applied in commercial and industrial buildings to satisfy requirements of comfort and technique. Its energy consumption even could cover 70% of air-conditioning energy consumption in the most adverse operating condition [1]. As consequence, performance improvement of water-cooled screw chiller is necessary.

Based on previous studies, many researchers used physical and mathematical models of the chillers to forecast the performance of chillers under multi operation condition, for the sake of improving the chillers performance. Allen and Hamilton [3] according to actual measurement pa-

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rameters, set up a mathematical model for the performance prediction of the chiller under full load and partial load. Goto and Yoshida [4], set up a prediction model to simulate the variation of the chillers' performance parameters with the variation of external parameters. Later on, Liu *et al*. [5] developed a new steady model that can predict the performance of the screw compressor with and without vapor injection in a comparatively wide running condition scope and under part-load displacement condition with a fast running speed. Based on simulation, Fu *et al*. [6] found that the cooling capacity of screw liquid chiller can be increased apparently by adding the economizer. Lately, Zhao *et al*. [7] provided a path to improve part-load energy efficiency of screw chillers with economized screw compressors using intermediate gas injection.

Of course, there are some researchers that improve the performance of chillers by experiments. Lee *et al*. [8] obtained a water chiller model of variable flow rate by analyzing 2000 groups of experimental data. By experiments, we also known that different refrigerants as a substitute can make the chiller performance decrease in a screw chiller with shell-and-tube heat exchangers [9]. The flow state and flow rate of the refrigerant before entering the evaporator affects the heat transfer of refrigerant on the side of the evaporator, and the variable water-flow rate can reduce the power of unit 19.7% by thermodynamic analysis on the measured data and structural parameters [10, 11]. In addition, fine-turn operating variable could improve chiller system performance [12]. According to measuring the operation conditions of the water chiller in most buildings, Seo and Lee [13] found that most of the chiller is running at low load rate, and its low energy efficiency ratio has a serious impact on the energy consumption level of the building.

Owing to the building cooling load is time-varying [14], the chiller operates at part load conditions based on time-varying cooling loads [15, 16], and the energy consumption of chiller was significantly affected by the chiller *COP* low part load conditions [13]. On the basis of previous studies and available conditions, this paper researches performance improvement of the water-cooled screw chiller under part load operation conditions (the load ratio is 75% and 50%) by experiment. Based on the original water-cooled screw chiller, this paper adds a paralleling throttle mechanism to water-cooled screw chiller, and uses parameters of discharge temperature, cooling capacity, *COP* and exergetic loss to evaluate the performance of the advanced water-cooled screw chiller (ADWC). Furthermore, this paper adopts exergy analyses, economic and environmental analyses to prove the superiority of the ADWC.

Experiments

Paralleling throttle mechanism and experimental set-up

The ADWC proposed in this study is comprised of a conventional water-cooled screw chiller and a specific throttle mechanism. The specific throttle mechanism parallelizes traditional throttle mechanism and by-pass tube. Figure 1(a) shows the schematic diagram of the ADWC, and fig. 1(b) displays the details of the paralleling throttle mechanism. Figure 2 illustrates three different access modes (vertical, oblique, and spiral access) of the by-pass tube. Table 1 details the information of the water-cooled screw chiller. In the case of vertical access, the by-pass tube is perpendicular to the main loop that is in front of evaporator. In the case of oblique access, the by-pass tube is bias insert to the main loop and the flow direction of refrigerant in the by-pass tube is similar to that of the main loop. In the case of spiral access, the angle of the spiral insertion with horizontal direction is less than 30°, and the refrigerant in the spiral tube is anti-clockwise to the main loop. In addition, four types of by-pass tube (outer diameter; 12, 16, 22 and 28) are used in tests. Figure 3 exhibits the arrangement of the experimental setup, which consists of indoor/outdoor chamber and control center.

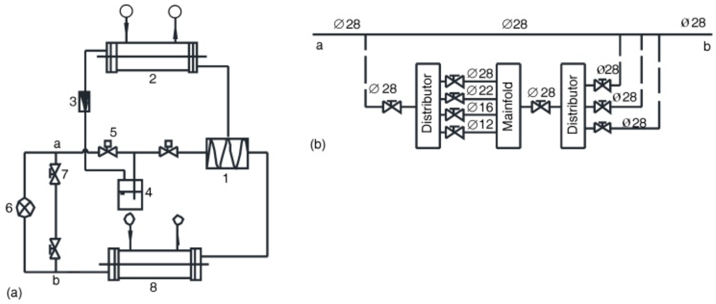

Figure 1. Schematic diagram of the ADWC; (a) syatem diagram of screw water-cooled chiller (b) diagram of by the paralleling throttle mechanism
1 – screw compressor, 2 – shell water-cooling condenser, 3 – non-return valve, 4 – sodium hypochlorite storage tank, 5 – solenoid valve, 6 – throttle mechanism, 7 – manual regulating valve, 8 – shell dry-evaporator

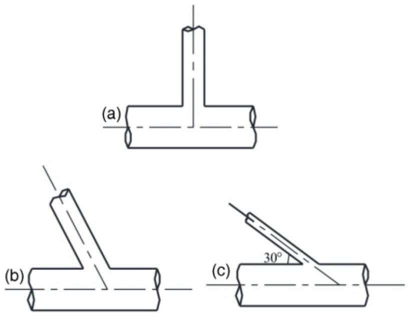

Figure 2. Access modes of throttle mechanism; (a) vertical access, (b) oblique access, and (c) spiral access

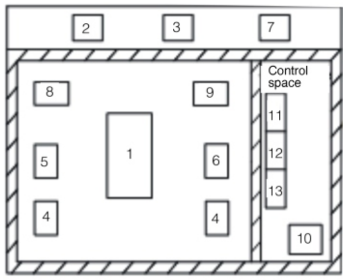

Figure 3. Arrangement of the experimental set-up
1 – water chiller unit, 2 – refrigerator, 3 – boiler, 4 – heat exchanger, 5 – cooling water tank, 6 – hot water tank, 7 – water replenishing system, 8 – cooling water pump, 9 – cold water pump, 10 – console, 11 – power cabinet, 12 – regulated power supply, 13 – electrical cabinet

Table 1. Parameters of water-cooled screw chiller

The model of screw water-cooled chill		LSN200Z
Cooling capacity	kW	220.0
Adjusting gear	%	0, 25, 50, 75, 100
Current source	V/Hz	380/50
Power	kW	42.5
Compressor	Model	Semi-hermetic screw compressor
	The amount of oil/L	18
Condenser	Model	2012-992L
	Heat transfer area/m^2	12.9
Throttle	Model	TEX-55-65.5
Evaporator	Model	2012-993F
	Heat transfer area/m^2	27.6
Refrigerant	Class	R134a
	Refrigerant charge amount/kg	55

Experimental procedure

In order to demonstrate the effect of paralleling throttle mechanism on the performance of the ADWC, a series of experimental tests are conducted. During the first run, the performance of the original chiller (conventional version) under full-load (the load ratio is 100%) and part-load (the load ratio is 75% and 50%) conditions is examined, as fig. 4. In the previous study, Zhang *et al*. [17, 18] investigated the relationship between the performance of water-cooled screw chiller and the operation parameters such as chilling/cooling water temperature and flow rate. In the present

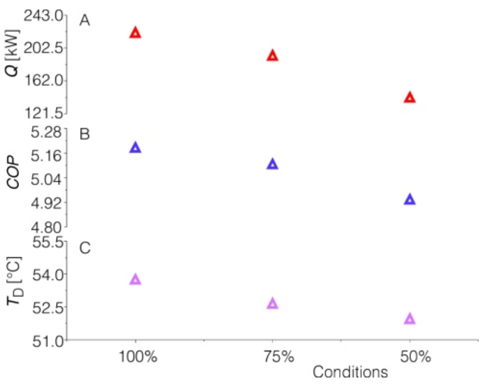

Figure 4. Performance parameters of original water-cooled screw chiller under different conditions

work, the effects of chilling/cooling water temperature and flow rate on the chiller performance are considered as tab. 2 shown. In accordance with tab. 2, in the experiments of the ADWC, the outlet temperature and flow rate of cooling water is kept at 30.0 ℃ and 32.0 m^3/h individually; the inlet temperature and flow rate of chilling water is set at 10.0 ℃ and 22.3 m^3/h, respectively. Referring to GB/T18430.1 -2007 and GB/T10870-2014, the steady-test duration is kept at 70 minute. In this research, the record interval is 6 seconds. In the part-load tests, the water-cooled screw chiller should be restarted before shifting different by-pass tubes and access modes. The measuring instruments utilized in this study are detailed in tab. 3.

Table 2. Performance parameters of the original water-cooled screw chiller under full load condition in different cooling and cooled water systems

Cooling water		Cooled water		Cooling capacity	*COP*	Discharge temperature
Outlet temperature	Flow	Inlet temperature	Flow			
°C	m^3/h	°C	m^3/h	kW		°C
30.0	45.8	7.0	30.0	200.7	4.77	51.8
30.0	45.8	10.0	30.0	219.9	5.15	51.9
30.0	45.8	10.0	22.3	220.8	5.14	52.7
30.0	32.0	10.0	22.3	222.0	4.99	53.7

Table 3. Measuring instrument

Parameters	Name	Range	Remarks
Pressure	Pressure sensors	0~2.1 MPa	Model: SEPRA2112
Temperature	Temperature sensor	–40~400 ℃	Platinum resistor(PT100); Accuracy: 0.1 ℃
Flow	Orifice flowmeter	0~70 m^3/h	Accuracy: 0.01 m^3/h
Power	Instrument of electric parameter	0~1500 kW	Accuracy: 0.04 kW
Time	Stopwatch	0~24 h	Accuracy: 0.01 s

Results and discussion

In this section, we employ discharge temperature, cooling capacity, and *COP* to analyze the energy performance of the water-cooled screw chiller, and adopt the exergetic analysis to examine the feasibility of various improving approaches. The cooling capacity and *COP* can be calculated by eqs. (1) and (2), respectively. The exergetic loss for each component of the chiller can be calculated by eqs. (3)-(6).

$$Q \quad C\rho q_{\mathrm{v}}(T_{\mathrm{IN}} \quad T_{\mathrm{OUT}}) \tag{1}$$

$$COP \quad \frac{Q}{N_{\mathrm{n}}} \tag{2}$$

where Q is the cooling capacity, C – the specific heat capacity, ρ – the density, q_{v} – the thermal flux per unit volume, T_{IN} – the inlet temperature, T_{OUT} – the outlet temperature, COP – the coefficient of performance, and N_{n} – the input power.

The exergetic loss of evaporator is:

$$Ex_{\mathrm{eva}} \quad [(h_{\mathrm{IN}} \quad h_{\mathrm{OUT}}) \quad T_{\mathrm{SUR}}(s_{\mathrm{IN}} \quad s_{\mathrm{OUT}})] \quad \frac{T_{\mathrm{SUR}}}{T_{\mathrm{e}}} \quad 1 \quad q_{\mathrm{e}} \tag{3}$$

The exergetic loss of condenser is:

$$Ex_{\mathrm{con}} \quad [(h_{\mathrm{IN}} \quad h_{\mathrm{OUT}}) \quad T_{\mathrm{SUR}}(s_{\mathrm{IN}} \quad s_{\mathrm{OUT}})] \quad \frac{T_{\mathrm{SUR}}}{T_{\mathrm{c}}} \quad 1 \quad q_{\mathrm{c}} \tag{4}$$

The exergetic loss of throttle mechanism is:

$$Ex_{\mathrm{th}} \quad [(h_{\mathrm{IN}} \quad h_{\mathrm{OUT}}) \quad T_{\mathrm{SUR}}(s_{\mathrm{IN}} \quad s_{\mathrm{OUT}})] \tag{5}$$

The exergetic loss of compressor is:

$$Ex_{\mathrm{com}} \quad (h_{\mathrm{IN}} \quad h_{\mathrm{OUT}}) \quad T_{\mathrm{SUR}}(s_{\mathrm{IN}} \quad s_{\mathrm{OUT}}) \quad \frac{1}{2}(c_{\mathrm{IN}}{}^{2} \quad c_{\mathrm{OUT}}{}^{2}) \quad \mathrm{g}(z_{\mathrm{IN}} \quad z_{\mathrm{OUT}}) \tag{6}$$

The total exergetic loss of the chiller can be calculated by eq. (7).

$$Ex_{\mathrm{to}} \quad Ex_{\mathrm{eva}} \quad Ex_{\mathrm{con}} \quad Ex_{\mathrm{th}} \quad Ex_{\mathrm{com}} \tag{7}$$

where Ex is the exergetic loss per unit mass in each component or entire chiller, h_{IN} – the inlet enthalpy of refrigerant, h_{OUT} – the outlet enthalpy of refrigerant, s_{IN} – the inlet entropy of refrigerant, s_{OUT} – the outlet entropy of refrigerant, c_{IN} – the inlet velocity of refrigerant, c_{OUT} – the outlet velocity of refrigerant, z_{IN} – the inlet height of refrigerant, z_{OUT} – the outlet height of refrigerant, g – the acceleration due to gravity, T_{SUR} – the environmental temperature, T_{e} – the evaporating temperature, T_{c} – the condensing temperature, q_{e} – the cooling capacity per unit mass of refrigerant, and q_{c} – the heating capacity per unit mass of refrigerant.

In a current research, Jiang *et al*. [19] found that, thermal parameter analyses could effectively guide the future research and improvement of chiller performance. Under the part-load state of chiller (the load ratio is 75% and 50%), the exergrtic loss of evaporator accounts for the largest proportion of the total exergetic loss of chiller, which can be 87% and 63%, respectively, fig. 5. Thus, there may be a huge potential of energy saving in evaporator of the chiller.

Effect of bypass-tube diameter

Under the load-ratio of 75%, in the cases of oblique access and vertical access, with the bypass-tube diameter increasing, the performance of ADWC becomes more sensitive to the opening-degree change of valve on the bypass line. Meanwhile, in the case of spiral access, it has a different phenomenon. The performance of ADWC seems obscure to the opening-degree change of valve on the bypass line. Figure 6 details these trends.

In the case of vertical access, the performance of ADWC tends to be improved first and then decline with the increase in opening-degree of valve on the bypass line. In the case of oblique access, the performance of ADWC tends to be improved first and then decline with the increasing valve opening-degree except the diameter of 12. When the bypass-tube diameter is 12,

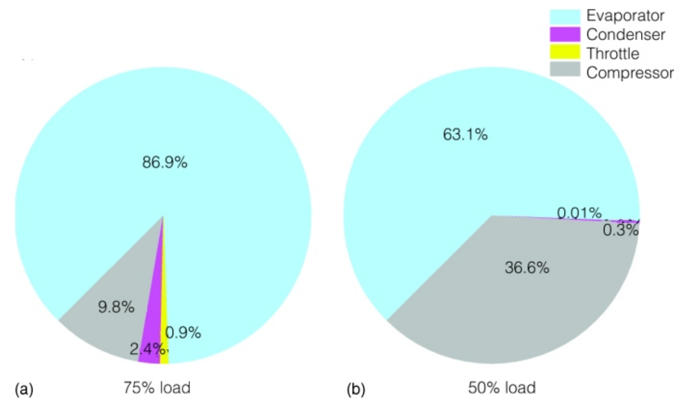

Figure 5. The percentage of exergetic loss of screw chiller main parts under part load state

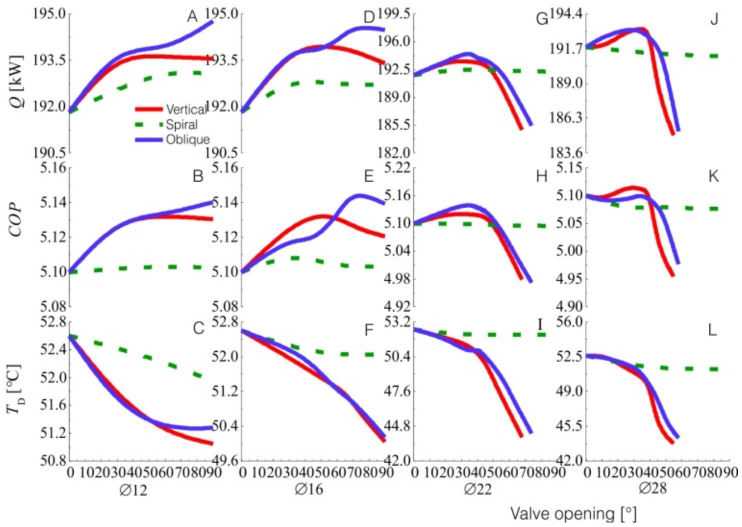

Figure 6. Effect of performance parameters by different diameters and access modes of by-pass tube under load ratio 75% condition

the performance of ADWC is promoted constantly with the increase in opening-degree of valve. Above all bypass-tube diameters, the diameter of 16 has the most positive effect on the performance enhancement of ADWC. Compared with the conventional chiller under load-ratio 75%, when the bypass-tube diameter is 16, the cooling capacity and *COP* of the ADWC increases 0.6%-1.5% and 0.2%-1.0%, respectively. In the meantime, the discharge temperature of ADWC reduces by 0.4-1.6 ºC. The similar phenomena can be observed under load-ratio 50%, the cooling capacity and *COP* of the ADWC increases 0.2%-1.1% and 0.5%-1.6%, respectively. In the meantime, the discharge temperature of ADWC reduces by 0.9-2.7 ºC, as fig. 7 described.

The essence of phenomenon that little bypass-tube diameters has an impactful influence on performance of water-cooled chiller could be not certain now. Through the tests, the performance of ADWC with bypass-tube diameter 28 and 22 can be promoted only in lower opening-degree of valve on the bypass line, while the performance of ADWC with bypass-tube diameter 16 and 12 is rarely limited by the opening of the valve. It could be supposed that the performance of ADWC is affected by refrigerant flow rate via the bypass-tube.

We can notice from the exergetic analysis results that, the exergetic loss of evaporator and total ADWC system drops dramatically due to bypass-tube use. Moreover, the exergetic loss of

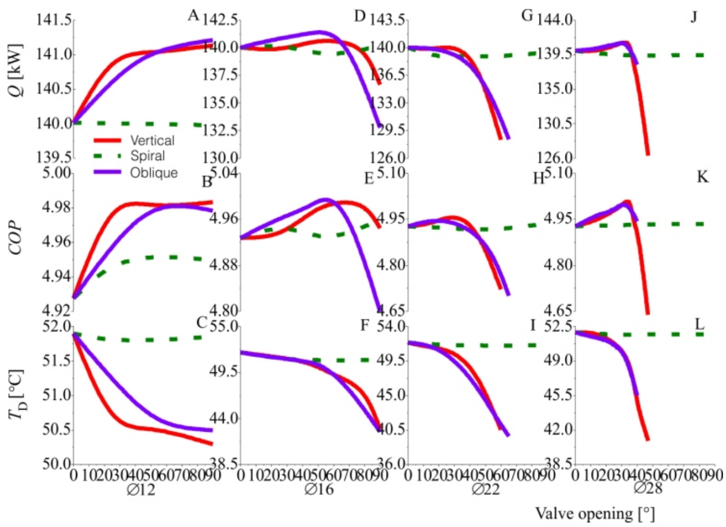

Figure 7. Effect of performance parameters by different diameters and access modes of by-pass tube under load ratio 50% condition

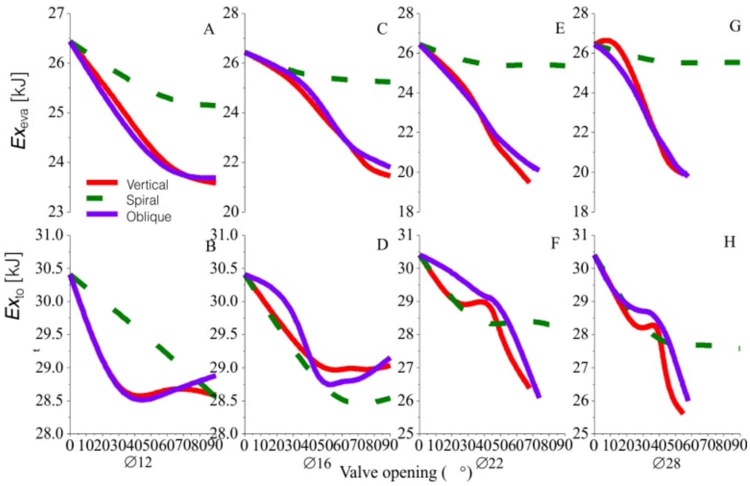

Figure 8. Effect of exergetic loss of evaporator and ADWC under load ratio 75% condition

evaporator is found to decrease constantly with the increase in opening-degree of bypass-tube valve, no matter what the bypass-tube diameter is (figs. 8 and 9). Specifically, the exergetic loss of evaporator and total ADWC system is 3.4%-15.4% and 4.1%-5.4% lower than the convectional water-cooled systems, respectively, when the bypass-tube diameter is 16, and so do others diameters. Besides, the load ratio 50% of ADWC has the similar phenomena too, the exergetic loss of evaporator and total ADWC system decreases by 4.6%-12.8% and 0-6.7%, individually.

Effect of access mode of by-pass tube

As fig. 6 like, under load ratio of 75%, at the same bypass-tube diameter, the bypass-tube of vertical and oblique access modes have obviously effects on performance of ADWC, while the bypass-tube of spiral access has petty influence on the performance of ADWC. With the exception of the bypass-tube diameter of 28, the performance improvement of ADWC with the oblique access mode is better than ADWC with the vertical access mode and *vice versa*. Mean-

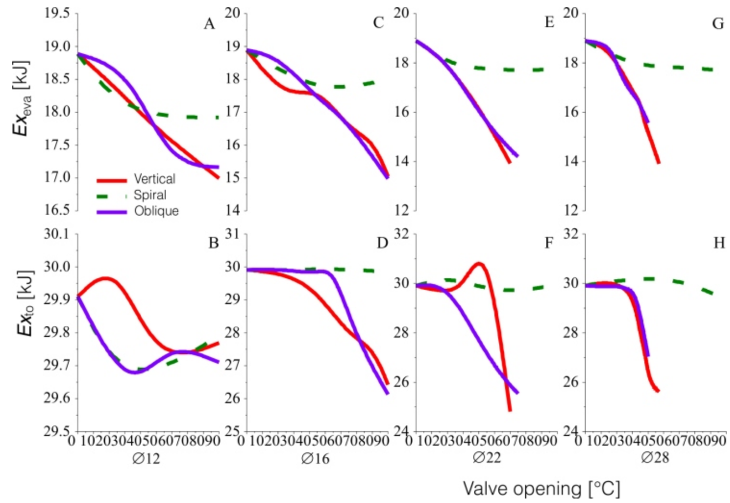

Figure 9. Effect of exergetic loss of evaporator and ADWC under load ratio 50% condition

while, at spiral access mode, the performance of ADWC can be improved a little at proper by-pass-tube diameter, otherwise, the performance of ADWC can be dropped. Specifically, at the oblique access mode, the cooling capacity and the *COP* are increased by 0.7%-1.6% and 0.1%-1.0%, respectively, while the discharge temperature is decreased by 1.3-1.9 ºC. So do other access modes have similar trend. It is also such alike phenomenon of ADWC under load-ratio 50% as fig. 7 description, the cooling capacity and *COP* of the ADWC increases 0-1.1% and 0.6%-1.6%, respectively. In the meantime, the discharge temperature of ADWC reduces by 0.4-1.8 ºC.

Figures 8 and 9 can confirm that exergetic loss of evaporator and total ADWC system is dropped significantly by using these access modes of bypass-tube. What is more, at all kinds of access modes of bypass-tube, the exergetic loss of evaporator decreases constantly with the open degrees of valve increasing, no matter what the part-load condition is. Besides, ADWC with oblique access mode is better than the vertical and spiral access mode that makes the exergetic loss of evaporator and total ADWC system go down. Specifically, as the ADWC with oblique access mode working operates under 75% load condition, the exergy loss of evaporator and total ADWC system is decreased by 10.4%-16.0% and 4.4%-5.8%, respectively, and so do others access modes. It is obviously that the load ratio 50% of ADWC has the similar phenomena too, the exergetic loss of evaporator and total ADWC system decreases by 4.4%-10.4% and 0-0.7%, individually.

In order to clearly show the superiority of the combination of the by-pass tube diameters with access modes, comparing the maximum improvement performance of the ADWC under thirteen conditions. As shown in fig. 10, whatever the ADWC operates under 75% or 50% load conditions, the combination of by-pass tube diameter 16 with oblique access mode is better than others combinations that have positive effect on performance of ADWC. Specifically, when the load ratio is 75%, the cooling capacity is increased by 1.5%, and the *COP* is increased by 1.0%, and the discharge temperature is decreased by 1.6 ºC, and the promotion for load ratio of 50% is more obvious.

Approximated economic and environmental analyses

Currently, the Chinese market price of a conventional type of water-cooled screw chiler (for a 220 kW scale) is about 130,000 RMB yuan, while the ADWC costs more 5,000 RNB yuan than the conventional one. Under part-load state (the load-ratio is 75% and 50%), the operating costs of ADWC is less 8.9 RMB yuan and 13.2 RMB yuan per day than conventional type of water-cooled screw chiller, individually. The dynamic payback period is calculated as 2.7 years (the load-ratio is 75%) and 1.3 years (the load-ratio is 50%) severally. Here the discount rate is taken as 8.0%. Table 4 shows the thermoeconomic performance of the ADWC under part load conditions.

The paper uses the cap of emission to approximately assess the environmental effect of the proposed ADWC. The mathematical descriptions of the approximately environmental analysis are given by eqs. (8) and (9). The cap of emissions for CO_2 and SO_2 are 295.3 10^{-6} kg/kJ and 2.8 10^{-6} kg/kJ individually.

$$\Delta M_i \quad \Delta E m_i \tag{8}$$

$$\Delta E \quad E_{\mathrm{N}} \quad E_{\mathrm{O}} \tag{9}$$

where ΔM_i is the difference of emissions reduction of i gas, ΔE – the difference of energy, m_{i} – the cap of emission of i gas, E_{N} – the energy of advanced water-cooled screw chiller, and E_{O} – the energy of conventional water-cooled screw chiller.

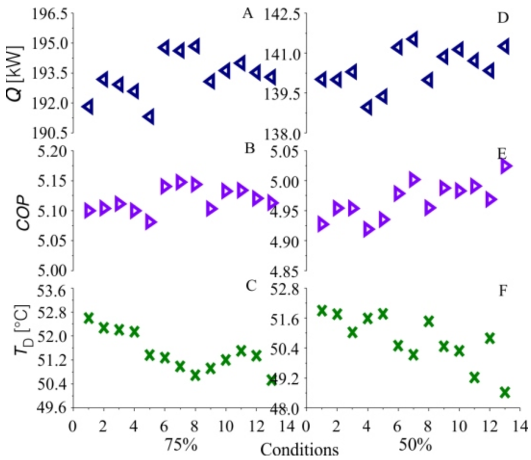

Figure 10. The ADWC performance of adjustment modes under load ratio 75% and 50% states

Table 4. Thermoeconomic performance of the water-cooled screw chillers under part load

Name	Value	Value
Refrigeration power consumption rate, [kW]	37.24	28
Load rate, [%]	75	50
COP	5.15	5.00
Cooling season, [d]	180	180
Power consumption saved per hour, [kWhh^{-1}]	0.37	0.55
Referable price of electricity, RMB yuan·[kWh^{-1}]	1	1
Additional initial investment, RMB* yuan	5.000	5.000
Discount rate, [%]	8.0	8.0
Dynamic payback period, years	2.7	1.3

The calculated results indicate that, when the load-ratio is 75%, the emission-reduction of CO_2 and SO_2 for ADWC in a whole cooling season can reach 1699.2 kg and 16.1 kg, respectively, while the emission-reduction can reach 2525.9 kg and 24.0 kg under load of 50%. Thus, the environmental prospect of the proposed ADWC may be bright in the approximately environmental view.

*6.2 RMB = 1 $US

Conclusions

The paper proposes an advanced water-cooled screw chiller that contains paralleling throttle mechanism, and makes research on the effects that different diameters and access modes of by-pass tube have on the performance of water-cooled screw chiller under part-load state (the load ratio is 75% and 50%). Through an experimental study, the conclusions can be drawn as follow.

- The by-pass tube diameter 16 has the most positive effect on the performance of water-cooled screw chiller under part-load states. The cooling capacity and *COP* of the ADWC increases 0.2%-1.5% and 0.2%-1.6% severally. The discharge temperature of ADWC reduces by 0.4-2.7 ºC.
- The oblique access mode is better than the vertical access mode, while the spiral access mode has little influence on the chiller performance. The cooling capacity and *COP* of the ADWC increases 0-1.6% and 0.1%-1.6%, respectively. The discharge temperature of ADWC reduces by 0.4-1.9 ºC.
- Total exergetic loss of the ADWC has been clearly reduced due to the bypass paralleling throttle mechanism, indicating thermodynamic advantages.
- Compared with the conventional water-cooled screw chiller, the economic and environmental benefits of the advanced water-cooled screw chiller are evident.

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Appendix – Error analysis

The maximum error associated with the pressure, temperature, flow, and electric power is listed in tab. 3. The propagation of uncertainties associated with the calculated cooling capacity and *COP* can be evaluated:

$$\delta Q \quad \rho C\sqrt{(T_{\mathrm{IN}} \quad T_{\mathrm{OUT}})^2(\mathrm{d}\dot{V})^2 \quad (\dot{V})^2[(\mathrm{d}T_{\mathrm{IN}})^2 \quad (\mathrm{d}T_{\mathrm{OUT}})^2]} \tag{A1}$$

$$\delta_{COP} \quad \frac{\rho C}{N_n}\sqrt{(T_{\mathrm{IN}} \quad T_{\mathrm{OUT}})^2(\mathrm{d}\dot{V})^2 \quad (\dot{V})^2[(\mathrm{d}T_{\mathrm{IN}})^2 \quad (\mathrm{d}T_{\mathrm{OUT}})^2] \quad \frac{(\dot{V})(T_{\mathrm{IN}} \quad T_{\mathrm{OUT}})}{N_n}^2 (\mathrm{d}N_n)^2} \tag{A2}$$

Calculated using eqs. (A1) and (A2), the maximally relative errors for the cooling capacity and the *COP* are 1.1% (the absolute value is 1.5 kW) and 1.1% (the absolute value is 0.05), respectively. The results demonstrate that the uncertainty values of both the measured and the calculated data are acceptable in engineering applications.

Nomenclature

C	– the specific heat capacity, [$kJkg^{-1}K^{-1}$]
COP	– coefficient of performance, [–]
c	– velocity of refrigerant, [ms^{-1}]
E	– energy, [kJ]
Ex	– exergetic loss, [kJ]
g	– the gravitational acceleration, [ms^{-2}]
h	– enthalpy, [$kJkg^{-1}$]
M	– the emissions reduction, [kga^{-1}]
m	– the cap of emissions, [$kgkJ^{-1}$]
N_n	– input power, [kW]
Q	– cooling capacity, [kW]
q_v	– thermal flux per unit volum, [kJm^{-3}]
s	– entropy, [$kJkg^{-1}$]
T	– temperature, [°C]
$\dot{V}$	– the refrigerant volume flux, [m^3s^{-1}]
z	– the relative height, [m]

Greek symbols

Δ	– difference
δ	– error
ρ	– density, [kgm^{-3}]

Subscripts

c	– condensing
com	– compressor
con	– condenser
D	– discharge
e	– evaporating
eva	– evaporator
i	– kind of gas
IN	– inlet
N	– advanced screw water-cooled chiller
O	– original screw water-cooled chiller
OUT	– outlet
SUR	– surroundings
th	– throttle
to	– total

Acronym

ADWC – advanced water-cooled chiller

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