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ABSTRACT

This paper presents a simulation study on how to increase the coefficient of performance (COP) of an air-cooled screw chiller equipped with high static condenser fans. A thermodynamic chiller model was developed and validated using the operating data and specifications of the chiller. It includes an algorithm which makes use of a set point of condensing temperature to determine the number and speed of condenser fans staged to provide the heat rejection airflow required for any given operating condition. The simulation results show that reducing the condensing temperature as low as possible is incapable of maximizing the chiller COP when the rated condenser fan power is high by up to 77 W per kW cooling capacity. To minimize chiller power, the condensing temperature should be reset in response to the increase of condenser fan power and changes of chiller load and outdoor temperature. Depending on the load conditions, the chiller COP could increase by 1.7–84.8% when variable speed condenser fans and the optimum set point of condensing temperature are applied to existing air-cooled screw chillers. This study provides important insights into the low-energy design and operation of air-cooled chillers.

1. INTRODUCTION

Air-cooled chillers are widely used to provide cooling energy for air-conditioned buildings in the subtropical climate but with considerable electricity consumption (Yik *et al.*, 2001; Yu and Chan, 2005, 2006a, 2006b). It is worth considering how to improve their COP under various operating conditions—COP is defined as the cooling capacity over chiller power input, both in kW. Of particular concern is the method to increase the COP when the chillers need to work at part load for most of the operating time.

Some studies have indicated that the condensing temperature should be better controlled to increase the COP of air-cooled chillers (Briley, 2003; Brownell *et al.*, 1999; Chan and Yu, 2004; Manske *et al.*, 2001; Roper, 2000). The chillers currently operate under head pressure control (HPC) whereby the number of staged condenser fans is kept minimal in most operating conditions in order to limit the heat rejection airflow required to allow the condensing temperature to hover around its high set point of 45 °C. The condenser fan power under HPC can be minimized, but the compressor power remains high and the chiller COP decreases considerably when the chiller load drops in moderate outdoor temperatures. Floating condensing temperature control (CTC) is therefore suggested to be an alternative to HPC. The condensing temperature under CTC can drop and hover closely above the outdoor temperature via staging condenser fans as many as possible to enhance the heat rejection airflow in most operating conditions. This causes an increase in the fan power, but the compressor power can be minimized to maximize the chiller COP. The algorithm for implementing CTC is to adjust the set point of condensing temperature in response to changes of outdoor temperature (Yu and Chan, 2005, 2006b). Electronic expansion valves are a prerequisite for the successful implementation of CTC because the refrigerant flow has to be controlled at a wide range of differential pressures based on outdoor temperatures ranging between 10–35 °C (Love *et al.*, 2005; Yu and Chan, 2006a).

The chiller COP can be increased by lowering the condensing temperature only when the extent to which the compressor power can drop always exceeds the increase in fan power resulting from staging more condenser fans. CTC is generally applicable to air-cooled chillers containing low static condenser fans typically with a rated total power of 22 W per kW cooling capacity. For air-cooled chillers installed in an acoustic enclosure instead of an open space, high static condenser fans need to be used to cater for the additional pressure drop across the silencers. The rated total fan power then could be up to 77 W (350% of 22 W) per kW cooling capacity, which corresponds to 24% of the total rated compressor power. It is envisaged that the condensing temperature should be controlled at somewhere between its lower boundary and a high level of 45 ^{o}C in order to maximize the chiller COP. At present there is no detailed investigation into how to modulate the set point of condensing temperature to achieve the maximum chiller COP when the condenser fan power is comparatively high. It remains to be analyzed whether variable speed condenser fans, instead of the constant speed ones, facilitate the control of condensing temperature at its set point with reduced power.

Simulation is an expeditious means to identify the operating characteristics of air-cooled chillers with alternative condenser designs and to carry out an optimization study. The aim of this study is to investigate how to increase the coefficient of performance (COP) of an air-cooled screw chiller equipped with high static condenser fans. This paper first describes a thermodynamic model which simulates the operating performance of the chiller equipped with constant or variable speed condenser fans. The model was validated using a wide range of operating data and performance data in the chiller specifications. An algorithm was introduced to make use of a set point of condensing temperature to determine the number and speed of condenser fans staged at any given operating condition. Using the verified chiller model, an analysis was made on how the set point of condensing temperature should be adjusted to maximize the chiller COP for any given operating condition when there is a change in condenser fan power. A set of curves describing the optimum set point of condensing temperature was determined for a rated condenser fan power of 22 to 77 W per kW cooling capacity. In what follows, the potential increase in COP was assessed when variable speed condenser fans and the optimum set point of condensing temperature are applied to existing air-cooled screw chillers operating under HPC.

2. DESCRIPTIONS OF THE CHILLER MODEL

2.1 Details and Assumptions

The chiller model is based on an existing chiller which uses the refrigerant R134a and has a nominal cooling capacity of 1000 kW. The chiller comprises a shell-and-tube evaporator of the flooded type, an air-cooled condenser with constant speed condenser fans and four refrigeration circuits in parallel. Each refrigeration circuit includes one electronic expansion valve and one constant speed twin-screw compressor. The chiller operated under HPC and its operating variables were monitored by a building management system. The chiller COP at full load is 3.1 which complies with the performance data in the chiller specifications. The evaporator is designed to operate at an evaporating temperature of 3 ^{o}C at full load. The chilled water flowing through the chiller is maintained at 43.0 kg/s with a supply temperature of 7 ^{o}C and a temperature rise of 5.5 ^{o}C at full load. The cooling capacity can be controlled in 12 steps via a modulating sliding valve in each of the four compressors. The heat rejection capacity of the condenser is designed to control the condensing temperature at around 50 ^{o}C when the outdoor temperature is 35 ^{o}C. The heat rejection airflow is regulated from 18.9 through 94.5 m^3/s in steps of 18.9 m^3/s by staging five groups of condenser fans. Each group consists of four constant speed fans (of the low static type) rated at 1.1 kW each and provides the total airflow of 18.9 m^3/s.

With regard to the uncertainty in the measurement of variables, the temperatures of chilled water were measured by PT100 type temperature sensors calibrated to an uncertainty of ±0.15 ^{o}C at 0 ^{o}C and ±0.26 ^{o}C at 55 ^{o}C. The flow rate of chilled water was measured by a magnetic flow meter with an uncertainty of ±2% of actual flow. A power analyzer with an uncertainty of ±1% of reading was used to determine the chiller power. The root sum square error of chiller COP due to all the uncertainties of the individual variables was calculated to be 7.9–14.6% when the chiller operated at above half load. The highest uncertainty of chiller COP is up to 1.16% when the percentage error of each measured variable is considered independently.

With regard to the basic assumptions of the model, there was no heat exchange between the chiller and its surroundings. This means that heat rejection is the sum of cooling capacity and compressor power. The mass flow rate of the refrigerant was the same throughout the chiller and equal to the mass flow rate through the staged compressors. Pressure losses in the refrigerant pipelines were disregarded. The throttling of refrigerant at the expansion valve was assumed to be isenthalpic. The degree of subcooling (T_{cdsc}) and that of superheat (T_{evsh}) were assumed to be 8 and 3 ^{o}C respectively in all operating conditions, where their possible variations (T_{cdsc}: 1–6 ^{o}C; T_{evsh}: 4–8 ^{o}C) would cause up to 0.16% of uncertainty of chiller COP only (Chan and Yu, 2004).

2.2 Methods of Simulation

The chiller model was developed using TRNSYS (SEL, 2000). A subroutine given by Bourdouxhe *et al.* (1995) was included in the model to evaluate the thermodynamic properties of the refrigerant R134a. Each operation condition comprised seven inputs: outdoor temperature (T_{cdae}), cooling capacity (Q_{cl}), chilled water flow rate (m_w), the temperature of supply chilled water (T_{chws}), the degree of subcooling (T_{cdsc}), the degree of superheat (T_{evsh}) and the set point of condensing temperature (T_{cdsp}). T_{cdsp} was used to determine the number and speed of the staged condenser fans in order to control the condensing temperature close to its set point. The outputs were the operating variables of the evaporator, compressors, expansion valves and condenser, which were determined by solving over 30 algebraic equations through an iterative procedure. Due to the space limitation, details of the equations and the evaluation of operating variables were given elsewhere (Yu and Chan, 2006a, 2006b).

The compressors, expansion valves and condenser within the chiller model have to satisfy the mass balance of refrigerant and energy balance at the evaporator. The method of log mean temperature difference (LMTD) was used to model the evaporator and condenser and variations in their overall heat transfer coefficients at part load were considered. The model is sophisticated enough to simulate the steady-state behaviour of air-cooled screw chillers satisfying any given operating condition. The chiller COP is defined as the cooling capacity (Q_{cl}) over chiller power (E_{ch}) which is the sum of compressor power (E_{cc}) and condenser fan power (E_{cf}).

2.3 Algorithm for Controlling the Number and Speed of Condenser Fans Staged

The control of condensing temperature (T_{cd}) influences the heat rejection airflow required for any given cooling capacity and how the chiller power can be minimized to achieve maximum COP. To address this issue, the chiller model included an algorithm which makes use of a set point of condensing temperature (T_{cdsp}) to compute the number and speed of condenser fans staged. Based on Equation (1), Inequality (2) was established to show how T_{cd} can be controlled by T_{cdsp} for any given heat rejection (Q_{cd}). The required heat rejection airflow (V_a) is related to T_{cdsp} by Inequality (3) obtained by transposing Inequality (2).

$$Q_{cd} = V_a \rho_a C_{pa} (T_{cdal} - T_{cdae}) \tag{1}$$

$$T_{cdal} = \frac{Q_{cd}}{V_a \rho_a C_{pa}} + T_{cdae} < T_{cd} \le T_{cdsp} \tag{2}$$

$$\frac{Q_{cd}}{\rho_a C_{pa} (T_{cdsp} - T_{cdae})} < V_a \tag{3}$$

For the chiller studied, the required V_a was met by staging five groups of constant speed condenser fans step by step. Figure 1 shows how to calculate the number of condenser fans staged (N_{cf}) for any given Q_{cd} and T_{cdsp}. The total power of the staged condenser fans (E_{cf}) is equal to the rated power of a condenser fan ($E_{cf,ea}$) multiplied by N_{cf}.

The potential benefits of using variable speed condenser fans were analysed, considering that they can operate at lower speed with much reduced power while the condensing temperature is maintained at its set point. A modification was made on the current arrangement of staging five groups of constant speed condenser fans in order to continuously modulate the heat rejection airflow required for any given set point of condensing temperature. Accordingly, the condenser model contained four variable speed fans, each of which served one refrigeration circuit with one compressor. Each fan provided 23.6 m^3/s at the full speed of 15.8 rps and could operate down to 1.58 rps to deliver the minimum airflow of 2.36 m^3/s. It was assumed that all the condenser fans operated at the same speed and the variable speed drive accounted for 3% of the total power of condenser fans staged at any speed. The power of each fan at full speed varied from 5.5 to 19.3 kW in simulating the effect of using low to high static fans on the optimum set point of condensing temperature for maximum chiller COP.

Figure 2 shows that the fan law was applied to compute the number (N_{cf}), rotating speed (R_{cf}) and power (E_{cf}) of the staged condenser fans based on a given V_a. N_{cf} was equal to the number of staged compressors (N_{cc}) when the condenser fans operated at or below full speed. One more condenser fan would be staged if the condenser fans staged at full speed were not able to produce the required V_a.

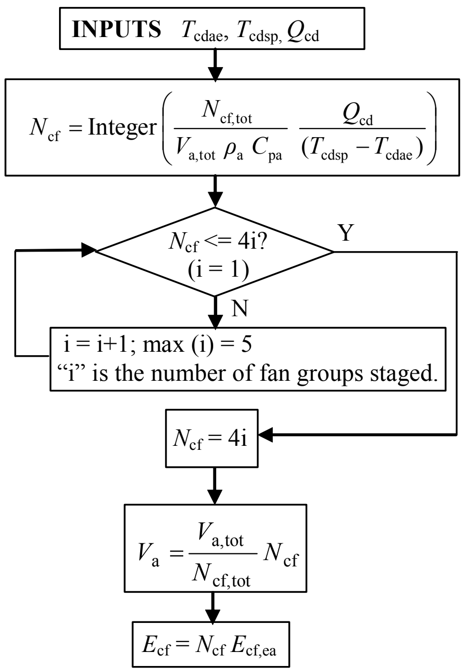

Figure 1: Procedure for determining the number and power of staged condenser fans at constant speed

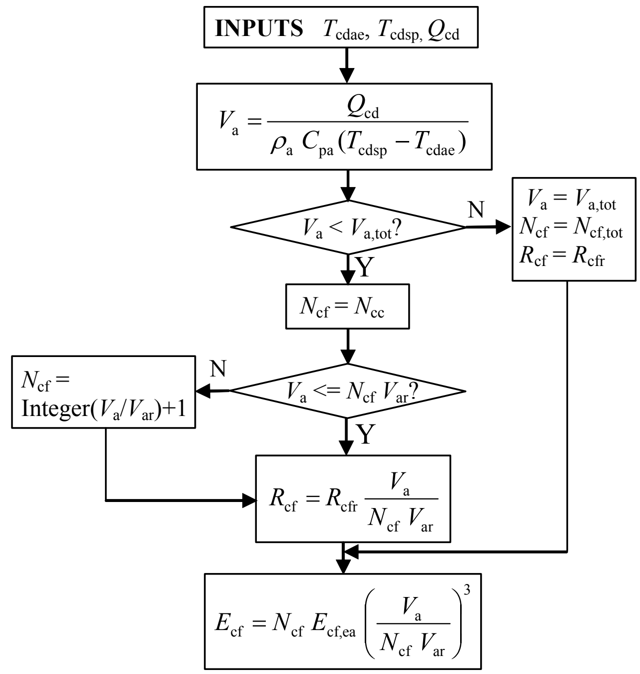

Figure 2: Procedure for evaluating the number, speed and power of staged condenser fans at variable speed

2.4 Validation of the Chiller Model

The chiller model was validated by the measured results of chiller COP calculated for over 1000 sets of operating conditions collected on an hourly basis. Table 1 shows how many hourly data were gathered in each group of operating conditions: combinations of outdoor temperatures (T_{cdae}) at 2 ^oC intervals and chiller part load ratios (PLRs) at 0.1 intervals. There are a total of 99 groups of operating conditions, and 46 of them relating mainly to low PLRs were absent from the validation because the chiller could operate frequently at above half load under the multiple chiller arrangement. The data collected in 53 groups of operating conditions were inputted into the chiller model. To simulate HPC, the set point of condensing temperature (T_{cdsp}) was kept at 45 ^oC in all the operating conditions. The inputs (m_w, T_{chws}, T_{cdsc} and T_{evsh}) were regarded as constants mentioned previously.

Figure 3 shows that the modelled COP agreed quite well with the measured COP. For 85% of data points, the uncertainty of chiller COP is less than 10%, and for 57% of these the uncertainty is even less than 5%. Overall, it is justifiable to use the chiller model to evaluate changes of the chiller COP at different ratings of condenser fan power under various operating conditions.

Table 1: Data count in various groups of operating conditions

Outdoor temperature (T_{cdae}) (^oC)	Chiller part load ratio (PLR)								
	0.1–0.2	0.2–0.3	0.3–0.4	0.4–0.5	0.5–0.6	0.6–0.7	0.7–0.8	0.8–0.9	0.9–1
13–15	1	1	4	3	3	2	1	0	0
15–17	0	1	9	7	5	2	1	0	0
17–19	0	0	6	7	9	5	3	2	1
19–21	0	0	2	4	22	17	12	7	6
21–23	0	0	0	1	19	26	28	16	13
23–25	0	0	0	0	4	15	34	40	20
25–27	0	0	0	0	0	9	60	69	76
27–29	0	0	0	0	0	1	48	125	103
29–31	0	0	0	0	0	0	6	189	133
31–33	0	0	0	0	0	0	0	90	63
33–35	0	0	0	0	0	0	0	3	2

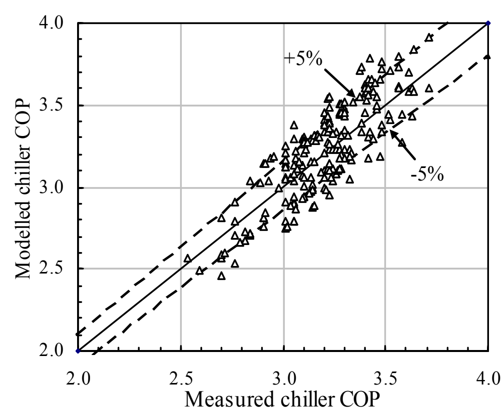

Figure 3: Comparison between the modelled and measured results of chiller COP

3. RESULTS AND DISCUSSION

3.1 Changes of Chiller COP at Different Condenser Fan Power Ratings

The model chiller was considered to have four variable speed condenser fans of the low to high static type, where their rated total power at full speed ranged from 22 kW to 77 kW (350% of 22 kW). Figures 4 and 5 show how the COP changed with various fan power ratings from 100 to 350% of 22 kW at different set points of condensing temperature (T_{cdsp}) when the chiller operated at a PLR of 0.25 or 1 with outdoor temperatures of 15 and 30 ^{o}C. Different T_{cdsp} could result in different numbers of condenser fan staged (N_{cf}) for any given operating condition (i.e. combination of outdoor temperatures and chiller part load ratios). When the chiller operated at a low PLR of 0.25, a lower T_{cdsp} would call for a higher N_{cf} in order to control the condensing temperature at slightly below T_{cdsp} while meeting the heat rejection. One more condenser fan was staged when T_{cdsp} dropped to a certain level depending on the outdoor temperature. A sudden fluctuation in the chiller COP occurred at a certain T_{cdsp} where there is a change in N_{cf}. This is because the total fan power could vary considerably when there are more fans operating at lower speed to provide the similar heat rejection airflow in relation to that T_{cdsp} instead of fewer fans operating at full speed.

When the low static condenser fans were applied (the case with 100% of total fan power), the maximum chiller COP always corresponded to a higher number of condenser fans staged, regardless of the operating conditions. The optimum T_{cdsp} for the maximum COP could be at a level where N_{cf} just increased by one from a lower number (see Figure 4), or at somewhere above the lower boundary of the condensing temperature when all the condenser fans needed to be staged (see Figure 5). The lower boundary of condensing temperature is given by outdoor temperature plus the LMTD of the condenser (i.e. T_{cdae} + $LMTD_{cd}$) (Chan and Yu, 2004). When the rated total fan power increased, the maximum COP might not be related to a higher N_{cf}. Instead a lower N_{cf} along with a higher T_{cdsp} could help minimize the sum of compressor power and condenser fan power.

When the chiller operated at full load (a PLR of 1), all the four condenser fans were staged throughout the entire range of set points of condensing temperature. As Figure 5 illustrates, the chiller COP remained unchanged when T_{cdsp} decreased to a certain level where all the fans started to operate at full speed to provide the maximum heat rejection airflow. The chiller COP was not maximized in most operating conditions when the maximum airflow took place. This indicates that the condenser fan power could drop significantly at lower fan speed with the reduced airflow, which outweighed the increased compressor power resulting from a higher T_{cdsp}.

3.2 Optimum Set Point of Condensing Temperature at Different Condenser Fan Power Ratings

As Figures 4 and 5 illustrate, the optimum T_{cdsp} for maximum COP tended to rise when the rated total condenser fan power increased. Figure 6 shows how the optimum T_{cdsp} varied at different fan power ratings under various operating conditions. There is no definite set point of condensing temperature to maximize the chiller COP in all operating conditions. This confirms that HPC with a fixed T_{cdsp} of 45 ^{o}C cannot optimize the chiller COP, particularly in moderate outdoor temperatures or part load conditions. It is found that the optimum T_{cdsp} can be expressed as a function of outdoor temperature and chiller part load ratio, regardless of the type of condenser fans. When low static condenser fans were used, the optimum T_{cdsp} had the modest fluctuation across the entire range of part load ratios. This is because it is desirable to have higher number and speed of condenser fans staged in most operating conditions to minimize the chiller power. When high static condenser fans were applied, their power was comparatively high and their number and speed might need to reduce in certain operating conditions in order to maximize the chiller COP. This results in the greater fluctuation in the optimum T_{cdsp} when the chiller part load ratio increased from 0.083.

It is possible that the optimum T_{cdsp} related directly to the chiller part load ratio when all the condenser fans were staged at a PLR of above 0.8. The optimum T_{cdsp} has to be raised for a chiller containing high static condenser fans at variable speed. This highlights the need to present the optimum T_{cdsp} in terms of different functions of outdoor temperature and chiller part load ratio when different condenser fan power ratings are considered. Following the variety of optimum T_{cdsp}, a unique set of curves for the optimum T_{cdsp} should be established for a given condenser capacity design with its own heat transfer characteristics under various operating conditions.

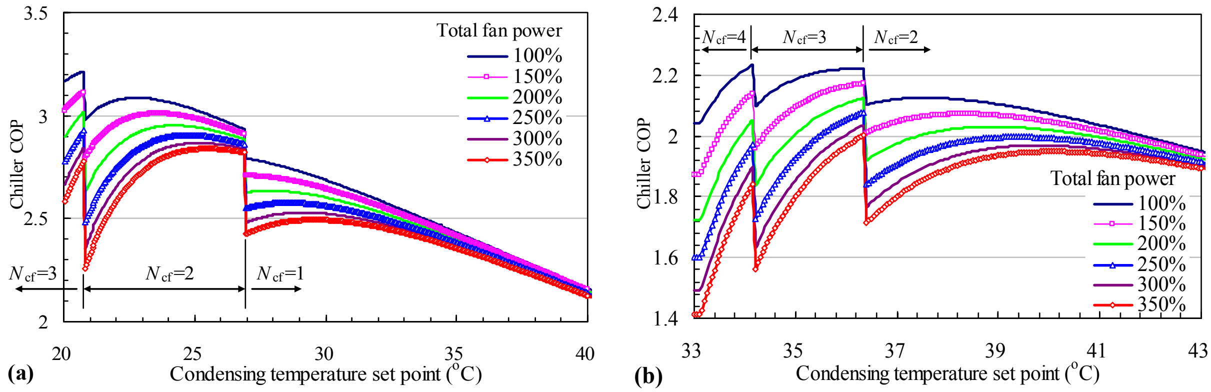

Figure 4: Variations in chiller COP at a PLR of 0.25 with outdoor temperatures of (a) 15 °C and (b) 30 °C.

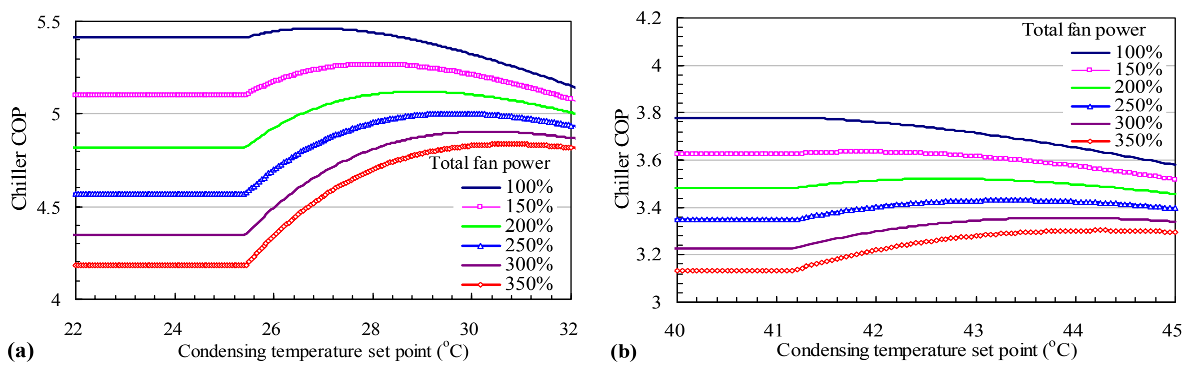

Figure 5: Variations in chiller COP at a PLR of 1 with outdoor temperatures of (a) 15 °C and (b) 30 °C

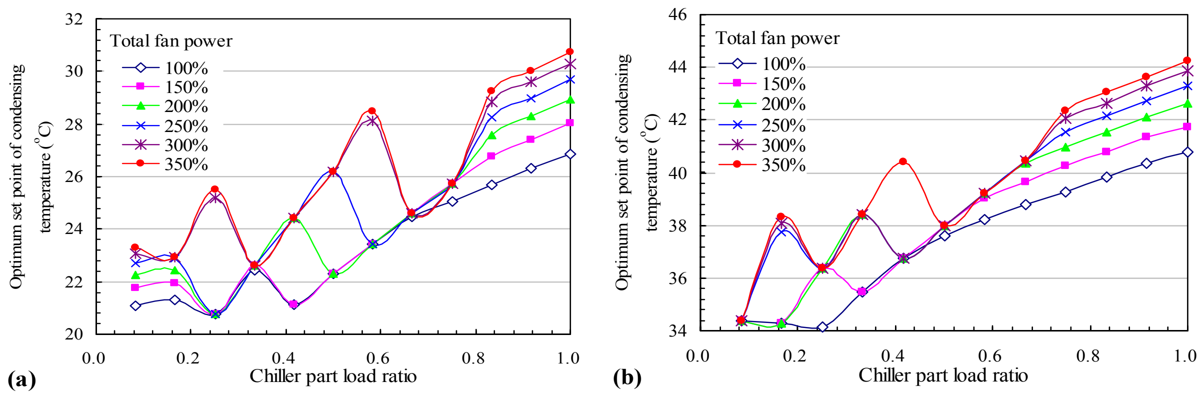

Figure 6: Optimum set point of condensing temperature at different combinations of chiller part load ratios and outdoor temperatures (a) 15 °C and (b) 30 °C

3.3 Increased Chiller COP with the Use of Variable Speed Condenser Fans and Optimum Set Point of Condensing Temperature

After studying the optimum set point of condensing temperature, it is worth assessing the extent to which the COP can increase when the optimum set point and variable speed condenser fans are applied to air-cooled screw chillers operating with constant speed fans under HPC. Table 2 shows a comparison of COP between the existing chiller and the hypothetical chiller with the enhanced condenser features under various operating conditions, with regard to the use of high-static condenser fans having a total power of 77 kW for both the chillers. The enhanced condenser features helped increase the chiller COP in varying degrees under all operating conditions, except the conditions in which the compressor power was unchanged while there was 3% of the additional fan power imposed by the variable speed drive. When the chiller operated at half load or above, the COP could increase by 1.7–84.8%, depending on the outdoor temperatures. The improvement in COP tends to be greater when the outdoor temperature is lower. There is a reduction of 4.5–148.9 kW in the chiller power along with the increased COP.

Table 2: COP of the existing chiller (HPC case) and the hypothetical chiller (CTC case)

Outdoor temperature (T_{cdae}) (^{o}C)	Capacity control steps in terms of chiller part load ratio (PLR)													
	0.5		0.58		0.67		0.75		0.83		0.92		1.0	
	HPC	CTC	HPC	CTC	HPC	CTC	HPC	CTC	HPC	CTC	HPC	CTC	HPC	CTC
15	2.21	3.79	3.01	4.04	2.99	4.27	2.94	4.50	2.84	4.74	2.69	4.97	3.68	5.20
20	2.59	3.38	2.57	3.59	2.51	3.80	2.41	4.01	3.12	4.22	3.11	4.43	3.07	4.63
25	2.20	2.83	2.14	3.01	2.62	3.19	2.62	3.36	3.05	3.52	3.09	3.68	3.11	3.83
30	2.17	2.64	2.44	2.80	2.50	2.97	2.77	3.13	2.84	3.29	3.12	3.45	3.21	3.60
35	2.13	2.31	2.31	2.45	2.49	2.59	2.68	2.72	2.86	2.86	2.99	2.99	3.11	3.11

4. CONCLUSIONS

This paper shows how to increase the COP of an air-cooled screw chiller containing high static condenser fans with increased power. The thermodynamic model of the chiller was developed using TRNSYS and validated using a wide range of operating data and specifications of the chiller. An algorithm was introduced to make use of a set point of condensing temperature to determine the number and speed of condenser fans staged at any given operating condition. The model was used to investigate how the COP varies at different set points of condensing temperature when the chiller with different condenser fan power ratings operated under various load and ambient conditions.

The simulation results highlight the importance of including a chiller load in addition to outdoor temperature to reset the condensing temperature, considering the changing trade-off between the increased condenser fan power and decreased compressor power under various load conditions. This present analysis also ascertains that the rated total condenser fan power is one of the factors influencing the optimum set point of condensing temperature. For an air-cooled chiller using low static condenser fans rated at 22 W per kW cooling capacity, the set point of condensing temperature should be kept low in order to maximize the chiller COP. Yet reducing the condensing temperature as low as possible is incapable of minimizing the chiller power when the rated fan power is high by up to 77 W per kW cooling capacity. Depending on the operating conditions, the COP could increase by 1.7–84.8% when variable speed condenser fans and the optimum set point of condensing temperature are applied to air-cooled screw chillers operating with high static fans under HPC. The likely increase of COP helps judge if the implementation of variable speed condenser fans with the reset of condensing temperature is cost effective via engineering economics. This study provides important insights into the development of low-energy air-cooled chillers.

NOMENCLATURE

C_{pa}	specific heat capacity of air	(1.02 kJ/kg ^{o}C)	R_{cfr}	full speed of condenser fans	(15.8 rps)
E	power input	(kW)	T_{cd}	Condensing temperature	(^{o}C)
$E_{cf,ea}$	rated power of one condenser fan	(kW)	T_{cdae}	outdoor temperature	(^{o}C)
m_{w}	mass flow rate of chilled water	(kg/s)	T_{cdal}	temperature of air leaving the condenser	(^{o}C)
N_{cc}	number of staged compressors	(-)	T_{cdsc}	degree of subcooling	(^{o}C)
N_{cf}	number of staged condenser fans	(-)	T_{cdsp}	set point of condensing temperature	(^{o}C)
PLR	chiller part load ratio (Q_{cl}/Q_{cr})	(-)	T_{chws}	temperature of supply chilled water	(^{o}C)
Q_{cd}	heat rejection	(kW)	T_{evsh}	degree of superheat	(^{o}C)
Q_{cl}	cooling capacity	(kW)	V_{a}	airflow by staged condenser fans	(m^{3}/s)
Q_{cr}	nominal cooling capacity	(kW)	V_{ar}	rated airflow of a variable speed condenser fan	(23.6 m^{3}/s)
R_{cf}	rotating speed of staged condenser fans	(rps)	ρ_{a}	air density	(1.2 kg/m^{3})

Subscripts

cc	compressor	ev	evaporator
cd	condenser	max	maximum
cf	condenser fan	op	optimum
ch	chiller	tot	total

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