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Optimizing condenser fan control for air-cooled centrifugal chillers

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Abstract

The current design and operation of air-cooled condensers can cause a significant decrease in chiller performance under part load conditions. This paper demonstrates optimal condenser fan control to improve the coefficient of performance (COP) of air-cooled chillers. This control involves identifying the optimum set point of condensing temperature with the optimized power relationships of the compressors and condenser fans and enhancing the airflow and heat transfer area of the condensers. An example application of this control for an air-cooled centrifugal chiller indicated that the COP could increase by 11.4–237.2%, depending on the operating conditions. Such the increase of the COP results in a reduction of up to 14.1 kWh/m^2, or 27.3% in the annual electricity consumption per unit A/C floor area of chillers, given that the chillers serve an office building requiring an annual cooling energy per unit A/C floor area of 173.3 kWh/m^2. The simulation results of this study will give HVAC engineers a better understanding of how to optimize the design and operation of air-cooled chillers.

Keywords: Air-cooled chiller; Centrifugal compressor; Coefficient of performance; Electricity consumption

1. Introduction

Air-cooled chillers are commonly used in central cooling plants to provide comfort cooling for small to medium-scale buildings in the subtropical regions [1–3]. Yet their operation leads to considerable electricity consumption and the peak demand in the building sector. It is important to implement energy efficient measures for the chillers in order to effectively reduce the electricity demand for sustaining acceptable thermal comfort in buildings.

There are many research studies to improve the energy performance of chillers [4–10]. Using variable speed chillers and pumps is one of the possible means of enhancing their energy performance at part load operation, considering that their power consumption can drop considerably when running at lower speed [11–16]. However, all-variable speed chiller plants are still not popular at this moment because building owners may hesitate in taking an intensive investment to purchase the variable speed machines and the associated control and monitoring systems. There are some studies on providing variable flow to chillers by use of variable speed primary pumps in order to save pumping energy and facilitate the uneven load allocation for multiple chillers to maximize their aggregate coefficient of performance (COP) [11,17–21]. Yet concern has been expressed about the deterioration of evaporator performance and the complexity of the bypass and chiller staging controls under the variable flow condition [22–24]. Some studies opined that the chilled water temperature set point should be raised from the conventional level of 7 °C, so increasing the COP of chillers operating at part load with decreasing outdoor temperatures [4,5,19,20]. Other studies indicated that the performance of a chiller plant can be improved by using hybrid chillers with different types of compressors or using different energy sources and by allocating the chillers to operate at their optimum loading points [6,25,26]. HVAC engineers with expertise on chiller performance analysis are crucial for the successful implementation of the aforementioned techniques with varying degrees of complexity.

A more generic and direct approach to reducing the electricity consumption of chillers is to identify the change of their COP under various operating conditions and to investigate whether any deficient COP can be improved by altering their design and operational control. Compared with water-cooled

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Nomenclature

AU_{cd}	overall heat transfer coefficient of the condenser	kW/°C
C_{pa}	specific heat capacity of air, assumed to be 1.02 kJ/(kg °C)	
E	power input	kW
COP	chiller coefficient of performance	
m_r	mass flow rate of refrigerant	kg/s
m_w	mass flow rate of chilled water	kg/s
N_{cf}	number of staged condenser fans	
PLR	chiller part load ratio	
Q_{cd}	heat rejection	kW
Q_{cl}	cooling capacity	kW
T_{cd}	condensing temperature	°C
T_{cdae}	temperature of air entering the condenser or outdoor temperature	°C
T_{cdal}	temperature of air leaving the condenser	°C
T_{cdsc}	degree of subcooling	°C
T_{cdsp}	set point of condensing temperature	°C
T_{chws}	temperature of supply chilled water	°C
V_a	airflow provided by the staged condenser fans	m^3/s
$LMTD_{cd}$	log mean temperature difference at the condenser	°C
ρ_a	air density, assumed to be 1.2 kg/m^3	

Subscripts

cc	compressor
cd	condenser
cf	condenser fan
max	maximum
o	initial
op	optimum
tot	total

chillers, air-cooled chillers present a high possibility of COP improvement because the condensing temperature is intentionally maintained at around a high level of 50 °C under head pressure control (HPC), even when the outdoor temperature drops from a design level of 35 °C. Depending on the type of compressors, the compression efficiency could reduce considerably at off-design loads with a high condensing temperature. This results in high compressor power at partial load operation. Air-cooled condensers are usually designed with many constant speed condenser fans which produce variable airflow in steps to meet any given heat rejection and set point of condensing temperature. To reduce the condenser fan power, the heat rejection airflow is kept low in most operating conditions while controlling the condensing temperature at the high level under HPC. The use of variable speed condenser fans, on the other hand, allows the heat rejection airflow to regulate smoothly with reduced power at lower speed. It remains to be seen how the variable speed condenser fans help to precisely control the condensing temperature with an improved chiller COP.

Considering the deficiency of the traditional HPC, many studies have indicated the need to enhance the heat rejection airflow with better control of the condensing temperature in order to increase the chiller COP. The experimental results reported by Smith and King [27] illustrated that a 10% decrease in the overall power consumption of a reciprocating chiller rated at 35 kW is achieved by running the condenser fan at a higher speed when the outdoor temperature drops to below 25 °C. Yet they did not mention the actual fan control method. According to Roper's experimental findings [28], the power consumption of an air-cooled chiller can fall by as much as 20% when the condensing temperature is set at a lower value in relation to a high level of 50 °C used under HPC. He described a feedback control loop to monitor the difference between the outdoor temperature and the condensing temperature. This difference was then compared with its minimum value to determine whether to increase or reduce the fan speed. However, he did not give any elaboration of how this minimum value varies with the outdoor temperature and chiller load. For refrigeration systems with evaporative condensers, floating condensing temperature is a viable control strategy to enhance the COP of refrigeration systems [29,30]. The simulation of an industrial refrigeration system conducted by Manske et al. [29] affirmed that all fans of an evaporative condenser should be staged continuously to enable the condensing pressure to float at its lowest level, which is independent of the refrigeration load. The minimization of the system energy cost means controlling the condensing temperature as a linear function of the wet bulb temperature of the outdoor air. Yet Manske et al. [29] did not further explain how the set point of condensing temperature should be adjusted in response to any given outdoor temperature at part load operation. According to the discussion given by Briley [30] concerning the operation of industrial refrigeration systems at the lowest possible condensing temperature, it is possible to adjust the difference between the condensing temperature and the wet bulb temperature of outdoor air to be between 3–8 °C.

Ge and Tassou [31] developed a model to simulate the performance of a refrigeration system for supermarket display cabinets and to investigate how to decrease the condensing pressure. Each cabinet was served by air-cooled chillers with multiple reciprocating compressors and condenser fans. Over 22% of the energy saving on a summer's day was obtained when the set point of condensing pressure dropped from 15.1 to 12 bars with frequent operation of more condenser fans. This saving was due to the situation where the degree of decrease in the compressor power exceeded the corresponding increase in the condenser fan power. The efficiency of the refrigeration system could be enhanced by resetting the condensing temperature according to the outdoor temperature. According to Gordon and Ng [32], the condensing temperature should be controlled at a lower level to save the compressor power of air-cooled chillers. The extent to which the condensing temperature can drop depends on the heat rejection capacity of an air-cooled condenser and how much airflow the condenser needs to be provided with in response to any given chiller load.

All the past studies have indicated a rather arbitrary approach to lowering the condensing temperature for chiller efficiency improvements. These may be applicable only to certain operating conditions with regard to a specific chiller or refrigeration equipment. There is, indeed, a lack of a generic method to determine optimal condenser fan control with a precise setting for the condensing temperature. It remains to be seen how the fan speed control interacts with the optimum trade-off between the compressor power and condenser fan power when the chiller operates at various outdoor temperatures and load conditions.

Computer simulation is an expeditious means to examine changes in the design and control of chiller components and hence to carry out an optimization study to achieve maximum chiller performance. There are numerous chiller models developed using different approaches and principles for different kinds of chiller performance study. However, very few models have sufficient capability for investigating the controllability of condensing temperature for air-cooled chillers. There are thermodynamic models for air-cooled chillers with reciprocating or screw compressors which are capable of investigating the steady-state behaviour of chiller COP under various operating conditions [33–38]. An algorithm is contained in these models to compute the number and speed of condenser fans staged to control the condensing temperature at a specified set point for any given cooling capacity. These models form a good basis to further develop optimum condenser fan control for air-cooled centrifugal chillers; something which is lacking in the existing chiller simulation studies.

The aim of this paper is to demonstrate how the COP of air-cooled centrifugal chillers can be improved by optimizing the control of condensing temperature and enhancing the airflow and heat transfer area of the condensers. First, the typical control of condenser fans by use of a condensing temperature will be explained. Second, a thermodynamic chiller model will be described to show how the condenser fan control interacts with the trade-off between compressor power and condenser fan power. An assessment will be made on the extent of the increase in chiller COP resulting from the optimum set point of condensing temperature and the enhanced condenser design. Discussion will be given on the possibility of using the Equal Marginal Performance Principal (EMPP) to achieve the optimized chiller performance. Following that, the cooling load profile of an office building will be considered to identify the annual electricity savings of chillers when applying the improved condenser fan control. The significance of this study is to present one more possible scheme to implement low-energy air-cooled chiller plants for air-conditioned buildings.

2. General methodology for controlling condenser fans by use of condensing temperature

According to the fundamental energy equation given by Eq. (1), for any given outdoor temperature (T_{cdae}), heat rejection airflow (V_a) has to vary to control the condensing temperature (T_{cd}) at a specified set point while meeting the required heat rejection (Q_{cd})—the sum of compressor power (E_{cc}) and cooling capacity (Q_{cl}). V_a is conventionally modulated step by step via staging different numbers of condenser fans at a constant speed. This kind of condenser fan staging has long been implemented under HPC, resulting in the imprecise control of condensing temperature.

To control T_{cd} at its set point, V_a has to comply with inequality (2) derived from Eq. (1). It is envisaged that using a high and fixed T_{cdsp} as under HPC is the simplest way to satisfy inequality (2) for any given operating condition, but it discounts the opportunity to optimize the trade-off between the compressor power and condenser fan power. The minimum V_a required is given by inequality (3) which is obtained by transposing inequality (2). For any Q_{cd}, the number of staged condenser fans (N_{cf}) is ascertained by using inequality (5) based on inequality (3) and the relationship between V_a and N_{cf} in Eq. (4). In Eq. (4), $V_{a,tot}$ is the total airflow produced by all condenser fans and $N_{cf,tot}$ is the total number of condenser fans. The total power of staged condenser fans (E_{cf}) is computed by N_{cf} multiplied by the rated power of each fan ($E_{cf,ea}$).

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

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

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

$$V_a = \frac{V_{a,tot}}{N_{cf,tot}} N_{cf} \tag{4}$$

$$\frac{N_{cf,tot}}{V_{a,tot} \rho_a C_{pa}} \frac{Q_{cd}}{(T_{cdsp} - T_{cdae})} < N_{cf} \tag{5}$$

The use of variable speed condenser fans, on the other hand, helps improve the controllability of the condensing temperature with reduced power [39]. They can vary V_a continuously based on any given set point of condensing temperature (T_{cdsp}). All of the variable speed condenser fans should operate at the same speed to provide equal heat rejection airflow. The rotating speed of each staged fan can be determined by Eq. (6), where R_{cfr} is the full speed of the fans. The total power input to the staged condenser fans (E_{cf}) is given by Eq. (7), where $E_{cf,ea}$ is the rated power of one condenser fan. The cube of the ratio of V_a to $V_{a,tot}$ based on the fan laws serves to explain why E_{cf} can drop considerably at reduced airflow with a low speed. Based on Eq. (7), it is possible to achieve a fan power saving even when the flow capacity (along with the rated power) of condenser fans is enlarged. In the simulation analysis, double-flow capacity will be considered for the variable speed condenser fans.

$$R_{cf} = \frac{V_a}{V_{a,tot}} R_{cfr} \tag{6}$$

$$E_{cf} = N_{cf} E_{cf,ea} \left(\frac{V_a}{V_{a,tot}} \right)^3 \tag{7}$$

Fig. 1 gives flow charts showing how the aforementioned control of condenser fans can be incorporated into the model

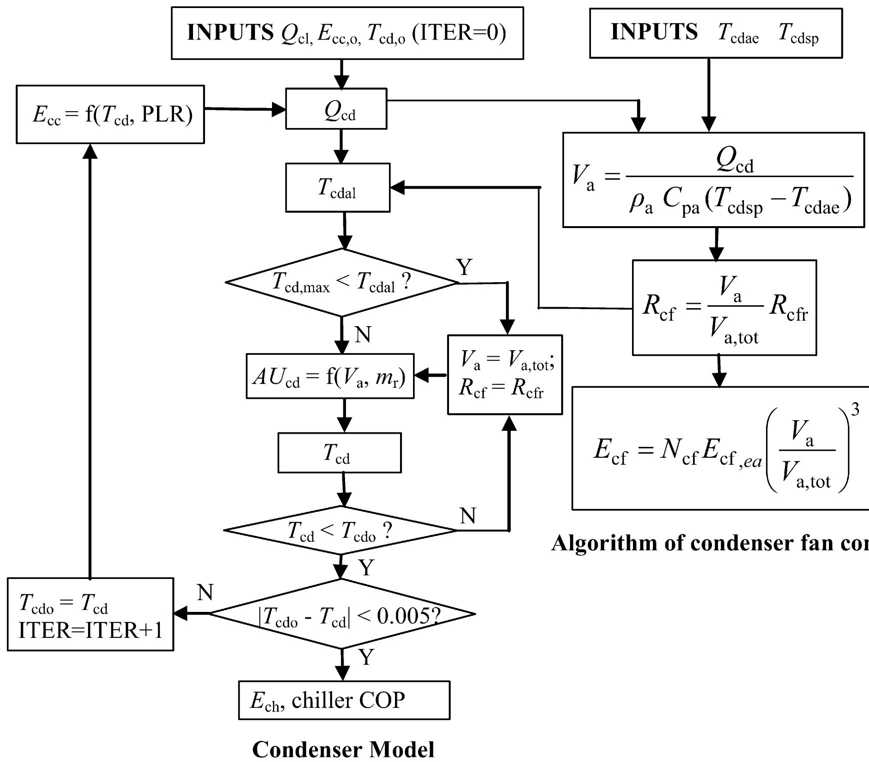

Fig. 1. Procedures for determining heat rejection airflow and the speed of staged condenser fans within an air-cooled condenser model.

of a typical air-cooled condenser to determine the condensing temperature (T_{cd}), heat rejection airflow (V_a) and the speed of staged condenser fans (R_{cf}). For any given cooling capacity (Q_{cl}), compressor power (E_{cc}) and heat rejection (Q_{cd}) can vary, depending on how T_{cd} is controlled at its set point (T_{cdsp}). Given this situation, the operating variables of chiller components are interacted with each other and their equations are solved altogether at a specified accuracy through an iterative procedure. The iterative procedure starts with an initial condensing temperature ($T_{cd,o}$) of 52 °C and the initial compressor power $E_{cc,o}$. Given a set point of condensing temperature (T_{cdsp}), V_a and R_{cf} can be determined using the equations for the condenser fan control. The overall heat transfer coefficient of the condenser (AU_{cd}) can be computed based on V_a and refrigerant flow rate (m_r). AU_{cd} is related to Q_{cd} and the log mean temperature difference (LMTD_{cd}) shown in Eqs. (8) and (9). While AU_{cd} increases with V_a, enhancing the heat transfer area A is a direct means to raise AU_{cd} and hence to lower T_{cd} for any given Q_{cd}. The effect of doubling the heat transfer area A will be studied in the simulation analysis.

$$Q_{cd} = AU_{cd}\mathrm{LMTD}_{cd} \tag{8}$$

$$\mathrm{LMTD}_{cd} = \frac{T_{cdal} - T_{cdae}}{\ln((T_{cd} - T_{cdae})/(T_{cd} - T_{cdal}))} \tag{9}$$

As Fig. 1 illustrates, there are two conditions where V_a needs to be maximized (i.e. $V_a = V_{a,tot}$) in order to evaluate all operating variables in a logical manner. The first condition is that the temperature of air leaving the condenser (T_{cdal}) exceeds the maximum condensing temperature ($T_{cd,max}$) of 52 °C. The second condition is that T_{cd} calculated in the condenser component is greater than the initial value $T_{cd,o}$. If the difference between T_{cd} and $T_{cd,o}$ lies within ±0.005 °C, all variables are evaluated with the required accuracy; otherwise the new values of T_{cd} and E_{cc} will be substituted for the previous values to proceed to the next iteration until the accuracy is met.

It is expected that for any given operating condition, the chiller COP varies, depending on the set point of condensing temperature for condenser fan control. Under head pressure control (HPC), the set point of condensing temperature (T_{cdsp}) is fixed at 45 °C, considering that the condensing temperature should hover above the outdoor temperature by 10–15 °C based on the typical condenser design. Condenser fans will be staged at low speed when the chiller is operating at part load with a low outdoor temperature. However, maintaining a fixed T_{cdsp} of 45 °C hinders the condensing temperature from reaching its lower boundary to minimize the compressor power when the outdoor temperature is low or when the overall heat transfer coefficient of the condenser can be enhanced at part load. It is worth adjusting the set point of condensing temperature (T_{cdsp}) at somewhere between 45 °C and the lower boundary in order to minimize the sum of compressor power and condenser fan power.

Most previous studies showed that the adjustment of T_{cdsp} depends solely on the outdoor temperature, i.e. $T_{cdsp} = T_{cdae} + C$, where C can be a constant of 3–10 °C. Yet this adjustment is applicable for air-cooled chillers having constant speed condenser fans with relatively low rated power. Referring to optimal control for cooling tower fans [40], the reset of T_{cdsp} for air-cooled chillers should depend on both the outdoor temperature and chiller part load ratio (PLR), as expressed by Eq. (10), where a_1 to a_6 are constant coefficients to be determined for individual chillers with their own condenser design and fan power rating. The inclusion of the terms with PLR is essential when a chiller contains an air-cooled condenser with high fan power rating.

$$T_{\mathrm{cdsp}} = T_{\mathrm{cdae}} + (a_1 + a_2\mathrm{PLR} + a_3\mathrm{PLR}^2 + a_4 T_{\mathrm{cdae}} + a_5 T_{\mathrm{cdae}}^2 + a_6 \mathrm{PLR}\,T_{\mathrm{cdae}}) \quad (10)$$

The lower boundary of condensing temperature is governed by the requirement of compressor lubrication and the heat rejection capacity of condensers. With regard to air-cooled reciprocating chillers, the condensing temperature should be above 20 °C in order to maintain lubricating oil with suitable viscosity to return to the compressors [34]. It is possible to identify the constraints on heat rejection capacity via a chiller model which is able to account for the possible change of the condenser's heat transfer characteristics at part load operation.

3. Example of applying optimal condenser fan control to air-cooled chillers

3.1. Description of the chiller model

To illustrate how the aforementioned control interacts with the compressor power and condenser fan power of a chiller, a thermodynamic model for air-cooled centrifugal chillers was developed using the simulation program TRNSYS version 15 [37]. The model considers mechanistic relations between chiller components. The log mean temperature difference (LMTD) method was used to model the heat transfer characteristics of the evaporator and condenser under the full load and part load conditions. The compressor and condenser have to satisfy the mass balance of refrigerant and energy balance at the evaporator. The model contains an algorithm to compute the required heat rejection airflow and the number and speed of condenser fans based on a set point of condensing temperature. The structure of the model is based on the thermodynamic models used extensively to investigate the energy performance of air-cooled reciprocating or screw chillers [33–36]. The way to simulate the capacity control of the inlet guide vanes is based on the steady-state model of a centrifugal compressor given in Refs. [38,39]. The model was validated by using the data of an existing air-cooled centrifugal chiller (rated at 1266 kW) operating for a wide range of ambient and load conditions. Details about the development and validity of the chiller model are given in Ref. [39].

The chiller model is capable of investigating the steady-state behaviour of chiller COP under various operating conditions when the design and control of air-cooled condensers are changed. With the control algorithm of condenser fans, the model can search for the optimum set point of condensing temperature from 20 to 45 °C at small intervals of 0.05 °C to minimize the sum of compressor power and condenser fan power for any given operating condition—a combination of chiller part load ratios and outdoor temperatures. The coefficients given in Eq. (10) can then be identified based on the optimum set point for each operating condition.

The model consists of five inputs: outdoor temperature (T_{cdae}), cooling capacity (Q_{cl}), chilled water flow rate (m_{w}), the temperature of supply chilled water (T_{chws}) and the degree of subcooling (T_{cdsc}). T_{cdae} and Q_{cl} are readily available based on the load profile of chillers. m_{w}, T_{chws} and T_{cdsc} can be considered as constants for a given chiller's nominal capacity. The outputs are the operating variables of the chiller components and the chiller COP. The annual electricity consumption of chillers in a cooling plant can be calculated by the model, given the cooling load profile of a building and the schedule of staging the chillers.

It should be noted that none of the existing building energy simulation programs can perform the energy analysis of air-cooled chillers with various designs and controls of condensers. This is because these programs usually model the energy performance of chillers using regression curves based on specific combinations of outdoor temperatures and load conditions. Such curves disregard the possible changes of chiller COP under part load conditions with varying outdoor temperatures and different controls of condensing temperature.

3.2. Improved part load performance due to optimal condenser fan control and enhanced condenser design

Drawing on the thermodynamic model, a variation in chiller COP was investigated under various operating conditions with different controls of condensing temperature and improved condenser design. Fig. 2 shows the typical part load performance curves when the chiller operated under HPC (the base case). The capacity control of the inlet guide vanes in the compressor resulted in the maximum COP at a part load ratio of 0.77 to 0.83 for a constant outdoor temperature. This is contrary to the maximum chiller COP at full load with regard to air-cooled chillers with reciprocating or screw compressors at constant speed [33–36]. The chiller COP dropped considerably when the chiller load reduced from such part load ratios. This is due to

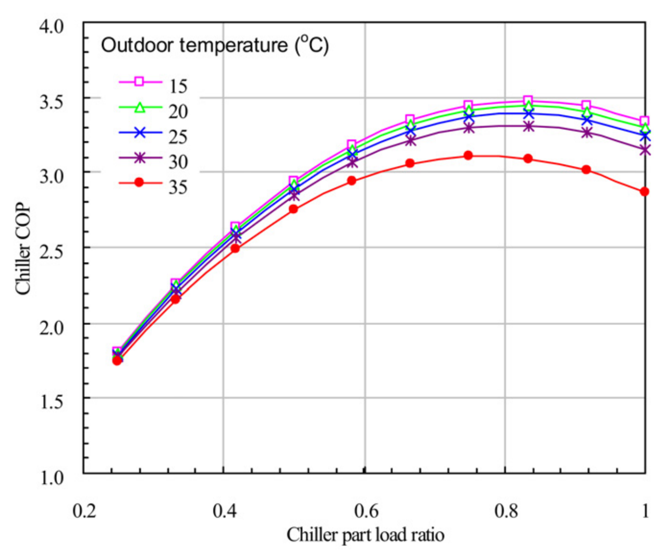

Fig. 2. Part load performance curves of the chiller under head pressure control (the base case).

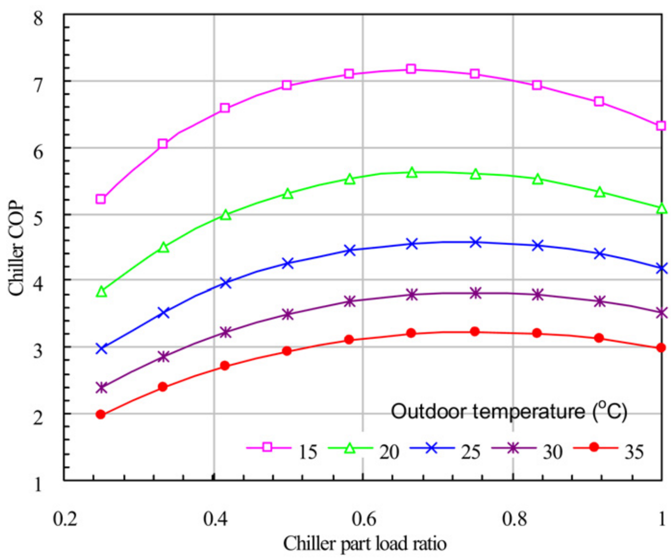

Fig. 3. Part load performance curves of the chiller with optimum set point of condensing temperature.

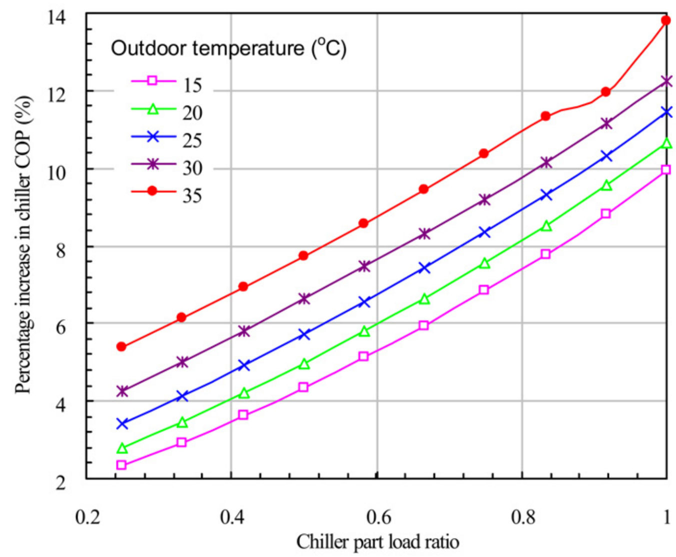

Fig. 4. Percentage increase in chiller COP when doubling the heat transfer area of the condenser in relation to the base case.

the significant drop in the compression efficiency during off-design operation, particularly with a high condensing temperature. HPC hindered the improvement of chiller COP at a low load when the outdoor temperature dropped from a design level of 35 °C.

As Fig. 3 illustrates, the chiller COP could increase in various degrees under different operating conditions when the set point of condensing temperature was optimized to minimize the sum of compressor power and condenser fan power. To achieve the maximum chiller COP, the optimum set point of condensing temperature ($T_{cdsp,op}$) should increase with the chiller part load ratio (PLR) from 0.2 to 1 for a constant outdoor temperature (T_{cdae}). Eq. (11) is a mathematical function of $T_{cdsp,op}$ for the chiller with the given condenser design and fan power rating.

$$T_{cdsp,op} = \begin{cases} 20\,^{\circ}\mathrm{C} \text{ for } T_{cdae} < 15\,^{\circ}\mathrm{C} \\ (T_{cdae} - 3.2211\mathrm{PLR}^2 + 11.113\mathrm{PLR} \\ \quad + 2.1587)\,^{\circ}\mathrm{C} \text{ for } 15\,^{\circ}\mathrm{C} \leqslant T_{cdae} \end{cases} \tag{11}$$

By comparing the part load performance curves in Fig. 2 with those in Fig. 3, it is possible to identify the extent to which the COP could rise when the optimum set point of condensing temperature was applied. The increase of the chiller COP could vary from 3.4 to 188%. Such an increase is small but noticeable at full load and is prominent at low chiller loads with low outdoor temperatures. The maximum chiller COP occurred at a part load ratio of 0.70 to 0.77, which is slightly different from the optimum range of part load ratios under HPC.

The potential benefits of enhancing the condenser capacity were studied. Fig. 4 shows the percentage increase in chiller COP under various operating conditions when the heat transfer area of the condenser was doubled while HPC was still used. The increase could vary by 2.3–13.8%, depending on the outdoor temperatures and load conditions. It is interesting to see that the percentage increase of chiller COP is highest at full load at the designed outdoor temperature when doubling the heat transfer area of the condenser. This is contrary to the behaviour of the improved COP resulting from the optimum set point of condensing temperature. Indeed, the increased heat transfer area of the condenser helps enhance the heat rejection and, in turn, the refrigeration effect via increasing the subcooling effect of the condenser. For any given cooling capacity, the refrigerant flow rate can drop due to the increased refrigeration effect, resulting in the reduction in both the compressor power and condenser fan power. The enhancement of heat rejection capacity is greater when the chiller carries larger loads with a higher refrigerant flow rate.

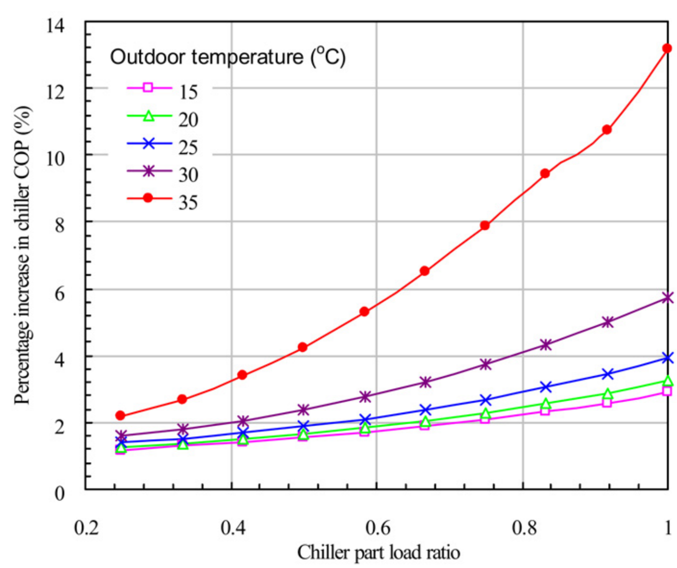

Fig. 5. Percentage increase in chiller COP when doubling the flow capacity of condenser fans in relation to the base case.

As Fig. 5 illustrates, when the airflow capacity of the condenser was doubled, the chiller COP under HPC could increase by 0.95–13.2%. Such a percentage increase is due only to the reduced power of the condenser fans which varied in proportion

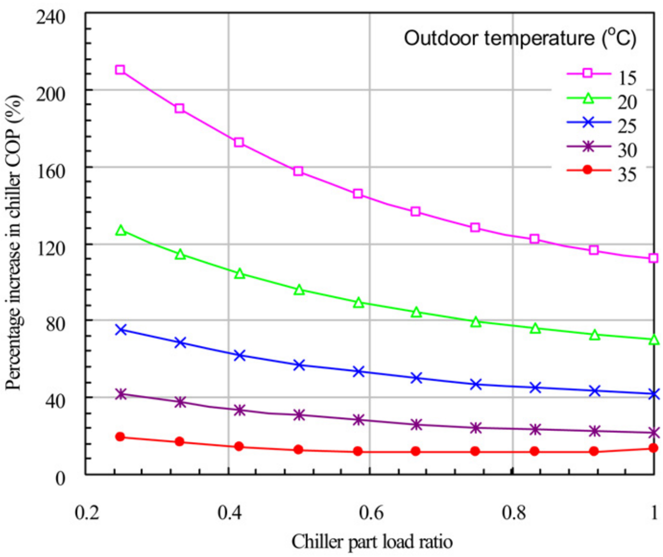

Fig. 6. Percentage increase in chiller COP when applying double-heat transfer area and optimum set point of condensing temperature in relation to the base case.

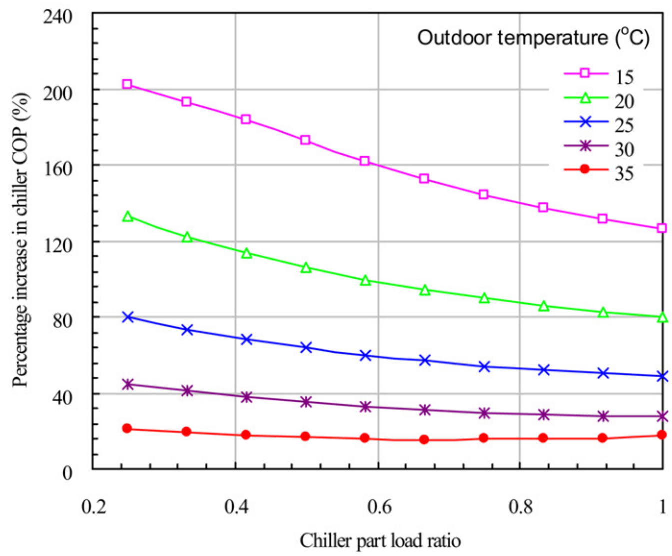

Fig. 7. Percentage increase in chiller COP when applying double-flow capacity and optimum set point of condensing temperature in relation to the base case.

to their speed cubed while producing the same heat rejection airflow. Indeed, the fan power dropped by 67–84% from the fan power found in the base case, though the double-flow capacity led to a doubling of the nominal power of the condenser fans. The compressor power remained unchanged in relation to the base case as long as HPC was applied with a fixed set point of 45 °C in the condensing temperature.

When the double-heat transfer area and optimum set point of condensing temperature were implemented together, the chiller COP could increase by 11.4–237.2%, as shown in Fig. 6. For a constant outdoor temperature, the percentage increase of COP generally rose when the chiller load dropped from a full load. When the outdoor temperature dropped, the condenser had ample heat rejection capacity to further lower the condensing temperature, resulting in the considerable increase in the chiller COP. It is noted that the optimum set point of condensing temperature, when determined based on the original condenser design, is still applicable to the case where the heat transfer area of the condenser was doubled.

Similar to doubling the heat transfer area, the double-flow capacity of the condenser fans could result in a 15.5–229.1% increase in the chiller COP when the optimum set point of condensing temperature took place, as shown in Fig. 7. The percentage increase of the chiller COP at full load in the double-flow case is higher than that in the double-heat transfer area case. It is noted that the optimum set point of condensing temperature should be re-adjusted when there is a change in the flow capacity and power rating of condenser fans. When the flow capacity was doubled, the set point of condensing temperature had to drop by 0.25–3.6 °C, depending on the ambient and load conditions, in order to achieve the maximum chiller COP.

Overall, when the enhanced condenser design is used together with the optimum set point of condensing temperature, the chiller COP could increase by at least 50% in moderate outdoor temperatures ranging between 11 and 25 °C. This range accounts for 48% of the total cooling hours for office buildings and for 54% of the total cooling hours for hotels, based on local weather conditions [41].

Using the condensing temperature reset to implement the optimal condenser fan control calls for the monitoring of the outdoor temperature and chiller load. A chiller part load ratio is usually not monitored directly by a chiller microprocessor because the signal of chilled water flow rate is not received by the microprocessor. If a chiller runs with its nominal flow for all operating conditions, it is possible to determine the part load ratio by using the difference between the temperatures of supply and return chilled water which are monitored variables. Considering that all the temperature variables are measured by thermistors with a typical uncertainty of ±0.1 °C in the entire measurement range, the combined uncertainty of $T_{cdsp,op}$ given by Eq. (11) could be calculated to be ±0.08 °C at 45 °C to ±0.57 °C at 20 °C. Fig. 8 shows how the uncertainty of $T_{cdsp,op}$ influences the maximum COPs at the various operating conditions given in Fig. 3. It is acceptable that the maximum COP deviates by up to 1.4% if the measurement uncertainty is considered in the condensing temperature reset for condenser fan control.

4. Implementation of improved condenser fan control using the Equal Marginal Performance Principle (EMPP)

To perform the condensing temperature reset for optimal condenser fan control, it is essential to have a dedicated controller which calculates the optimum set point based on signals of outdoor temperatures and the temperature differences of chilled water, and operates the fans at the right speed to meet that set point. While this temperature reset control is an add-on for typical condenser fan operations, it would demand more sophisticated control and instrumentation techniques to verify whether the chiller performance is optimized at the given fan speed. To counter this, Hartman [42] proposed the EMPP which takes into account system components and their power relation-

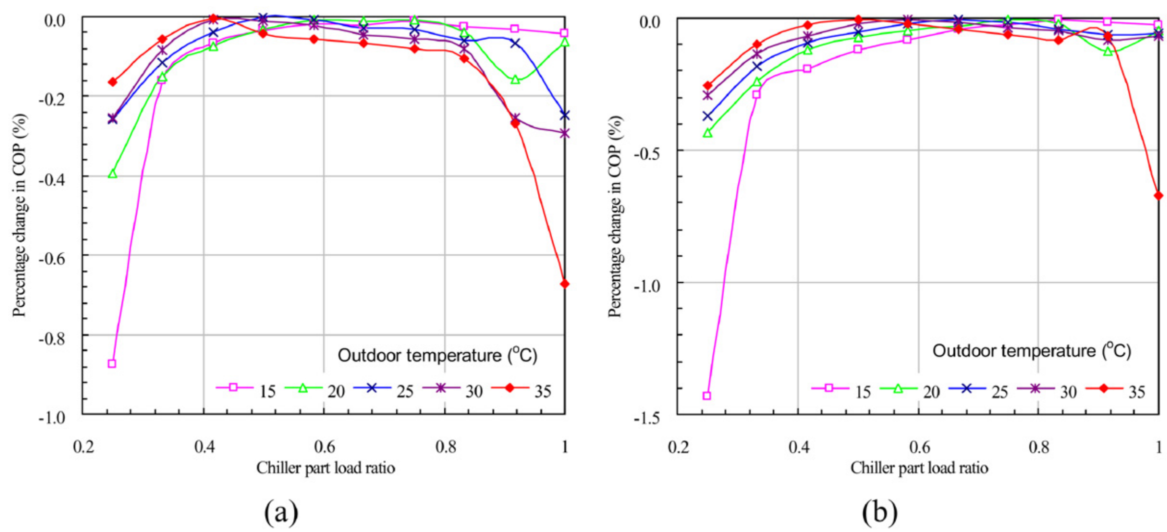

Fig. 8. Percentage change of maximum COP at various operating conditions due to the measurement uncertainty of the optimum condensing temperature set point (a) with negative error and (b) with positive error.

ship as a whole to ensure the optimal operation of any HVAC system. Using EMPP would eliminate instrumentation issues arising from temperature or pressure control which could diminish the potential COP improvements. Under EMPP, the energy performance of any system operating with multiple modulating components is optimized when the change in system output (called the marginal system output) per unit energy input is the same for all individual components in the system.

With regard to the chiller studied, the output is the cooling capacity (Q_{cl}) and the components consuming power are the compressor and condenser fans. One prerequisite for using the EMPP is to develop an algorithm for the system that relates output as a function of the power input of the two components. Based on the simulation results, it is possible to generate a set of formulae to represent the output (Q_{cl}) by the compressor power (E_{cc}) and condenser fan power (E_{cf}) at different outdoor temperatures, as summarized in Table 1. According to EMPP, the system is optimized or the maximum (marginal) COP takes place when the partial derivative of the output with respect to each power component is equal to each other (i.e. $\partial Q_{cl}/\partial E_{cc} = \partial Q_{cl}/\partial E_{cf}$). Using the optimized relationship between the compressor power and fan power, the actual fan power for maximum COP was ascertained for each operating condition. The fan rotating speed was then identified based on the fan law expressed in Eq. (12). Table 1 gives the fan speed required to be controlled at various operating conditions under EMPP. The implementation of EMPP relies on the new demand-based control which performs improved operation of HVAC systems based on optimized power relationships rather than using some calculated temperature or pressure set points which may not be directly related to system optimization. Although the EMPP is simple to apply and generic for most HVAC systems or equipment, it can be difficult or time consuming to determine the system output in terms of power relationships and to derive the marginal COP for each component if there is interdependence in the performance of the system components.

$$\text{Fan speed} = (\text{Design full speed}) \times (\text{actual fan power as a ratio to the rated power})^{1/3} \qquad (12)$$

5. Potential benefits from the improved control

The cooling load profile of a reference office building in Hong Kong was considered in order to assess the potential electricity savings when the enhanced condenser design and the improved control of condenser fans were applied to air-cooled centrifugal chillers. Detailed features of the building are given in Refs. [41,43]. The building has 40 storeys and a gross floor area (GFA) of 51 840 m^2 and its air-conditioned areas account for 82.6% of the GFA. Fig. 9 is a histogram showing how many hourly data of building cooling loads were collected in various ranges of outdoor temperatures. The data were expressed as ratios to the peak building cooling load of 6389 kW. There are 2834 cooling hours which account for 90.5% of the total office hours (3131 h a year). The annual cooling energy for the building is 7 423 883 kWh based on local weather conditions of the test reference year (TRY) in 1989. TRY is considered representative of the prevailing weather conditions in Hong Kong for building energy analysis [44]. As Fig. 9 illustrates, data of higher building load ratios were generally gathered at higher outdoor temperatures with a narrower range. For about 60% of the total cooling hours the building load ratios are 0.5 or below. The chillers needed to operate frequently for the building load ratio of 0.1 to 0.2 with a wide range of 11 to 25 °C in the outdoor temperature.

To meet the peak building cooling load, the chiller plant was designed with six identical air-cooled centrifugal chillers rated at 1124 kW each. There is a single-loop pumping system with a differential pressure by-pass pipe to control the amount of chilled water flowing from the operating chillers to cooling coils of the airside equipment. There are six constant speed pumps, each dedicated to one chiller to provide a constant chilled water flow of 47 l/s. Each pump has a rated power of 31.3 kW.

Table 1
Expression of system output and optimum condenser fan power under EMPP

Outdoor temperature (°C)	Cooling capacity (% of max.)	System output formulae	Optimum fan speed (% of full speed)	Optimum fan power (% of rated power)	Maximum COP
15	100	$Q_{cl} = 246.8E_{cc}^{-0.2046}E_{cf}^{0.7981}$	96.34	89.42	6.31
15	75	$Q_{cl} = 246.8E_{cc}^{-0.2046}E_{cf}^{0.7981}$	82.59	56.33	7.10
15	50	$Q_{cl} = 246.8E_{cc}^{-0.2046}E_{cf}^{0.7981}$	69.59	33.71	6.92
15	25	$Q_{cl} = 246.8E_{cc}^{-0.2046}E_{cf}^{0.7981}$	49.13	11.86	5.21
20	100	$Q_{cl} = 5371.7E_{cc}^{-1.3793}E_{cf}^{1.7351}$	99.06	97.19	5.08
20	75	$Q_{cl} = 5371.7E_{cc}^{-1.3793}E_{cf}^{1.7351}$	84.84	61.07	5.61
20	50	$Q_{cl} = 5371.7E_{cc}^{-1.3793}E_{cf}^{1.7351}$	71.77	36.97	5.31
20	25	$Q_{cl} = 5371.7E_{cc}^{-1.3793}E_{cf}^{1.7351}$	57.49	19.00	3.84
25	100	$Q_{cl} = 377.9E_{cc}^{-0.5443}E_{cf}^{1.2294}$	99.69	99.07	4.19
25	75	$Q_{cl} = 377.9E_{cc}^{-0.5443}E_{cf}^{1.2294}$	87.27	66.47	4.58
25	50	$Q_{cl} = 377.9E_{cc}^{-0.5443}E_{cf}^{1.2294}$	73.60	39.87	4.25
25	25	$Q_{cl} = 377.9E_{cc}^{-0.5443}E_{cf}^{1.2294}$	59.09	20.63	2.98
30	100	$Q_{cl} = 56.4E_{cc}^{-0.0534}E_{cf}^{0.9893}$	99.74	99.21	3.51
30	75	$Q_{cl} = 56.4E_{cc}^{-0.0534}E_{cf}^{0.9893}$	89.91	72.69	3.82
30	50	$Q_{cl} = 56.4E_{cc}^{-0.0534}E_{cf}^{0.9893}$	76.11	44.08	3.50
30	25	$Q_{cl} = 56.4E_{cc}^{-0.0534}E_{cf}^{0.9893}$	61.44	23.19	2.40
35	100	$Q_{cl} = 14.6E_{cc}^{0.2699}E_{cf}^{0.8241}$	100	100	2.98
35	75	$Q_{cl} = 14.6E_{cc}^{0.2699}E_{cf}^{0.8241}$	92.80	79.91	3.23
35	50	$Q_{cl} = 14.6E_{cc}^{0.2699}E_{cf}^{0.8241}$	78.29	47.99	2.94
35	25	$Q_{cl} = 14.6E_{cc}^{0.2699}E_{cf}^{0.8241}$	63.34	25.41	1.98

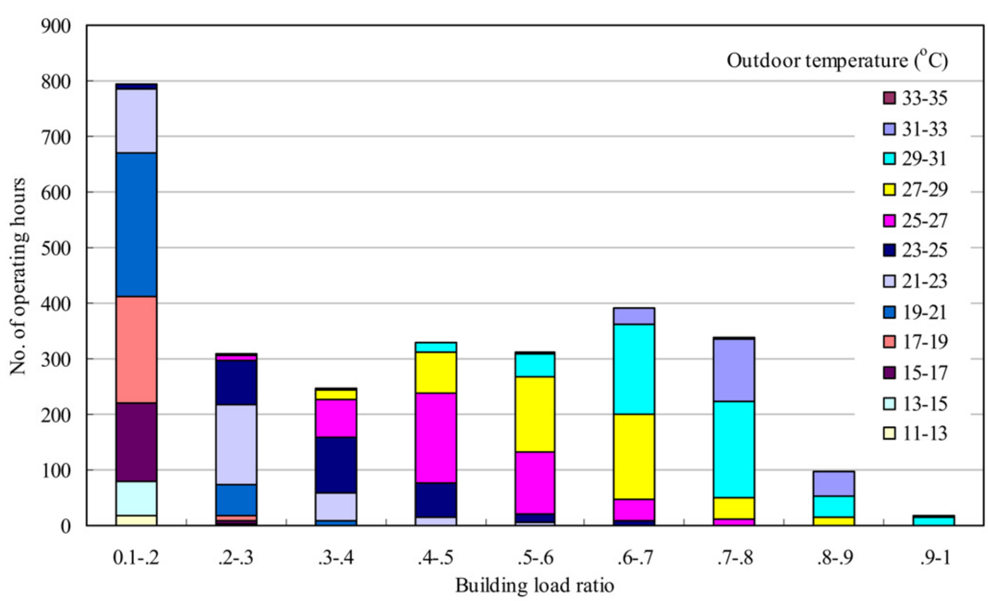

Fig. 9. Frequency distribution of hourly building load ratios in different ranges of outdoor temperatures.

While air-cooled centrifugal chillers operate with maximum COP at a part load ratio of 0.71 to 0.84, it is not desirable to frequently operate the chillers at such part load ratios. This is because more pumping energy will be incurred along with the frequent part load operation, which tends to offset any energy savings of chillers operating closely at their maximum COP. To meet the changing building cooling load, conventional chiller sequencing was still implemented so all the chillers are operating at the same load, and no additional chillers start to operate until each of the running chillers is operating at full load. The schedule of staging chillers and their possible loading ranges were then determined, as shown

Table 2
Range of chiller part load ratios at different numbers of operating chillers under chiller sequencing

Building load ratio (BLR)	Number of cooling hours	Number of operating chillers (N_{ch})	Total capacity of operating chillers (kW)	Chiller part load ratio (PLR)
$0 < BLR \leqslant 0.18$	746	1	1124	0.29–1
$0.18 < BLR \leqslant 0.35$	479	2	2248	0.5–1
$0.35 < BLR \leqslant 0.53$	568	3	3372	0.67–1
$0.53 < BLR \leqslant 0.70$	624	4	4496	0.75–1
$0.70 < BLR \leqslant 0.87$	405	5	5620	0.8–1
$0.87 < BLR \leqslant 1$	12	6	6744	0.83–0.95

Table 3
Energy performance of chillers with various energy efficient measures

Case	Average chiller COP	Normalized annual electricity consumption of chillers (kWh/m^2)	Electricity savings w.r.t. base case (%)
1 (base case)	3.36	51.60	–
2 (M1)	4.07	42.60	17.4
3 (M2)	3.41	50.88	1.4
4 (M3)	3.60	48.16	6.7
5 (M1 + M2)	4.62	37.51	27.3
6 (M1 + M3)	4.42	39.23	24.0

in Table 2. For a given N_{ch}, the lower limit of the range of chiller part load ratios was calculated by the total cooling capacity at $(N_{ch} - 1)$ over the total cooling capacity at that N_{ch}. When more chillers are staged to meet higher building cooling loads, each of them can operate more frequently at higher loads.

The annual electricity consumption of the chillers and pumps was calculated, based on the building cooling load profile and the schedule of staging chillers. The consumption is normalized by the total air-conditioned floor area of the building (i.e. 42 840 m^2) in terms of kWh/m^2. Table 3 shows the annual electricity consumption of the chillers under different cases. Case 1 refers to the base case where all the chillers with the conventional condenser design operated under HPC. Cases 2 to 6 refer to the individual and mixed use of energy efficient measures applied to all the chillers: (M1) optimum set point of condensing temperature; (M2) double-flow capacity of the condenser fans; (M3) double-heat transfer area of the condensers. The annual electricity consumption of pumps is 5.85 kWh/m^2 for cases 1 to 6 where the same schedule of staging chillers was applied.

Based on the electricity savings of the chillers, optimizing the set point of condensing temperature (M1) is a prominent approach to improving the energy performance of the chiller plant, enabling the average chiller COP to rise from 3.36 to 4.07. The average chiller COP is defined as the annual cooling energy of a building in kWh divided by the annual electricity consumption of chillers in kWh. If the flow capacity of the condenser fans was doubled with a fixed condensing temperature set point of 45 °C, the normalized annual electricity consumption of chillers reduced slightly by 1.4% or 0.72 kWh/m^2. This reduction is due to the decreased power of condenser fans running at low speed in most operating conditions in order to produce the minimum airflow required to meet the high set point. Doubling the heat transfer area of the condensers could achieve an electricity saving of 6.7% or 3.44 kWh/m^2 even when the condensing temperature was maintained at a high level of 45 °C.

When the optimum set point of condensing temperature was applied together with the enhanced condenser design, there is a decrease of up to 27.3% or 14.1 kWh/m^2 in the annual electricity consumption of the chillers. This electricity saving correlates closely to the increase in the chiller COP shown in Fig. 7. It is worth noting that in cases 5 and 6 the overall electricity savings resulting from two energy efficient measures can be equal to or even greater than the sum of the electricity savings contributed by the individual measures. In general, it is hard to achieve ultra-electricity savings when applying two or more energy efficient measures together because combining energy efficient measures tends to diminish their individual ability to decrease electricity consumption. Overall it is highly desirable to implement the optimum set point of condensing temperature together with the double-flow capacity of condenser fans to magnify their potential electricity savings for an air-cooled chiller plant.

6. Conclusions

This paper presents the impact of optimizing the condenser fan control and condenser design to enhance the performance of air-cooled chillers under various operating conditions. A general methodology has been explained to determine the optimal condensing temperature set point for condenser fan controls. An example application of the improved control to an air-cooled centrifugal chiller model showed that the COP could increase by 11.4–237.2%, depending on the ambient and load conditions. It is possible to further improve the chiller COP by increasing the airflow capacity of condenser fans and the heat transfer area of air-cooled condensers, along with the optimal fan control. The use of the Equal Marginal Performance Principle to achieve optimized chiller performance has been discussed.

The cooling load profile of an office building was considered to identify the electricity savings of chillers resulting from the low-energy condenser fan control and condenser design. It is estimated that the double-flow capacity of condenser fans together with the optimum set point of condensing temperature enables the annual electricity consumption per unit A/C floor area of chillers to drop by 27.3% or 14.1 kWh/m^2, with regard to the building requiring an annual cooling energy per unit A/C floor area of 173.3 kWh/m^2. The findings of this research provide important insights into how to improve the COP of air-cooled chillers operating for a building cooling load profile. It remains to be seen how life-cycle-cost analysis can be used to determine if the electricity savings are worth the increased initial costs for implementing the low-energy condenser features to an air-cooled chiller plant.

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