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Energy Procedia 143 (2017) 154–160

Energy Procedia

www.elsevier.com/locate/procedia

World Engineers Summit – Applied Energy Symposium & Forum: Low Carbon Cities & Urban Energy Joint Conference, WES-CUE 2017, 19–21 July 2017, Singapore

Probabilistic and electricity saving analyses of Mist Coolers for Chiller System in a Hotel

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Abstract

This study investigates the optimization and electricity savings of an air-cooled chiller system with mist coolers. An air-cooled screw chiller having a nominal capacity of 282 kW was retrofitted with a mist cooler and dual condenser fan controls: fixed head pressure with constant speed and floating condensing temperature with variable speed. Comprehensive operating data logged at 5-min intervals were used to develop chiller models by random forest. The models served as a fitness function for genetic algorithm to predict the maximum coefficient of performance (COP) with optimal controlled variables under various operating conditions. Scatter plots of the optimal variables against the actual variables showed that to achieve the maximum COP, the set point of condensing temperature should be adjusted based on its measured value and the variation of heat rejection airflow rate. The probability of applying mist cooling was identified to be 70.91 – 78.33% for the simulated cooling load distribution of a hotel in a subtropical climate. Mist coolers with the optimal control brought electricity savings of 2.28 – 8.16%, depending on system configurations and condenser fan control modes. Fewer chillers in a system would result in more electricity savings from the optimal control while mist coolers complement the frequent full load operation of chillers to enhance electricity savings.

Peer-review under responsibility of the scientific committee of the World Engineers Summit – Applied Energy Symposium & Forum: Low Carbon Cities & Urban Energy Joint Conference.

Keywords: Air-cooled chiller; coefficient of performance; genetic algorithm; mist cooler; random forest

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Peer-review under responsibility of the scientific committee of the World Engineers Summit – Applied Energy Symposium & Forum: Low Carbon Cities & Urban Energy Joint Conference.
10.1016/j.egypro.2017.12.664

1. Introduction

Chillers are commonly used to provide cooling energy in commercial building and their operation takes up the major portion of the total electricity consumption [1]. Where water source is limited, air-cooled chillers are frequently used instead of water-cooled chillers with evaporative cooling towers. The compressor power of chillers increases with the temperature of heat rejection medium. To reduce the electricity consumption of air-cooled chillers, evaporative cooling is increasingly used to cool the condenser air with minimal water consumption. Kabeel et al. [2] investigated the performance of a novel air-cooled chiller using cold mist to cool condenser air. The coefficient of performance (COP)—defined as the cooling energy output over the electric power input—could increase by up to 91% compared with conventional air-cooled chillers. The additional increase in the COP came from a situation where the condensing air temperature dropped to below its wet bulb by the cold mist. Hao et al. [3] developed a seasonal energy efficient ratio to assess the energy saving potential of an air-cooled chiller with an evaporative condenser. It was found that the optimal pad thickness of the condenser depended on the face velocity of condenser air and climatic conditions. The savings varied by 2.4% - 14.0% based on climatic conditions of 31 main cities in China. Martínez et al. [4] carried out an experimental study on how a variable thickness of evaporative cooling pads influenced the energy performance of a split-type air-conditioner. The COP rose by 10.6% when the pad thickness was optimized to be about 100 mm. Yu and Chan [5] performed a simulation study on how mist precooling improved the COP of chiller systems operating for commercial buildings. Typical multivariate regression models and perfect mist cooling by condenser air were considered. The electricity savings were governed by the interaction between the chiller load, the control of condensing temperature and weather conditions. Previous studies seldom focus on the experimental analysis of chillers with variable speed control for condenser fans and the implementation of optimal control to maximize the COP under various operating conditions.

The aims of this study are to analyze the real operating characteristics of a mist-cooled chiller with advanced condenser fan control and to discuss how to implement the optimal control. Data-driven models with an optimization method were created to analyze how the controlled variables were optimized to maximize the COP. The probability of applying mist coolers with optimal control was investigated to ascertain electricity savings for a hotel in a subtropical climate. The significances of this study are to demonstrate how the COP of air-cooled chillers was maximized with optimal settings of variables and how the chiller system design interacted with the electricity savings resulting from the optimal operation.

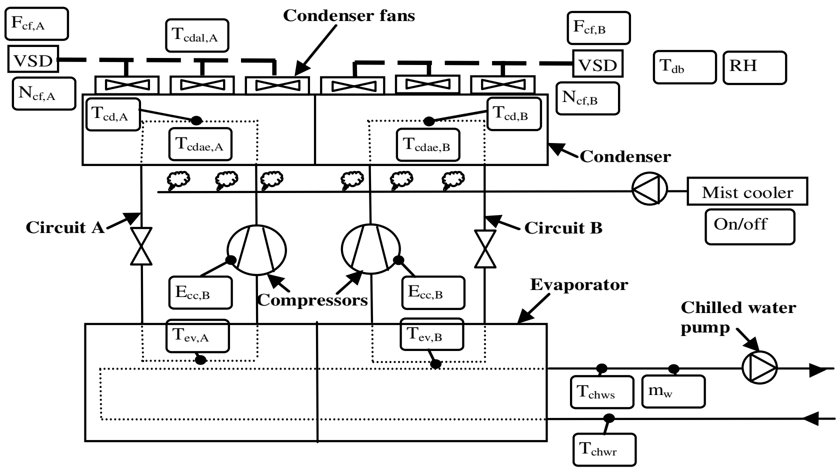
Nomenclature

COP	coefficient of performance	$Q_{cl,nom}$	nominal cooling capacity (282 kW)
c_{pa}	specific heat capacity of air (1.02 kJ/kg°C)	PLR	chiller part load ratio
c_{pw}	specific heat capacity of water (4.19 kJ/kg°C)	RH	relative humidity (%)
E_{cc}	compressor power (kW)	T_{cd}	saturated condensing temperature (°C)
$E_{cf,nom}$	total condenser fan power (14.4 kW)	T_{cdae}	temperature of air entering the condenser (°C)
E_{mist}	mist pump power (0.33 kW)	T_{cdal}	temperature of air leaving the condenser (°C)
E_{VSD}	power loss at variable speed drive (0.629 kW)	T_{chwr}	temperature of return chilled water (°C)
F_{cf}	power frequency of variable speed drive (Hz)	T_{chws}	temperature of supply chilled water (°C)
m_w	chilled water mass flow rate (kg/s)	T_{db}	dry bulb temperature of outdoor air (°C)
N_{cf}	number of condenser fans operating	T_{ev}	saturated evaporating temperature (°C)
$N_{cf,tot}$	total number of condenser fans (6)	T_{wb}	wet bulb temperature of outdoor air (°C)
Q_{cd}	heat rejection (kW)	V_a	heat rejection airflow rate (m^3/s)
Q_{cl}	cooling capacity (kW)	η_{mist}	mist cooler effectiveness

2. Chiller description

The chiller had a nominal capacity of 282 kW and a COP of 2.8 at a condenser air temperature of 35°C. It consisted of a flooded type evaporator and a condenser with 6 condenser fans. There were 2 identical refrigeration circuits. Each circuit contained one hermetic screw compressor with 3 capacity control steps and an electronic expansion valve for refrigerant flow control. The chiller was retrofitted with dual condenser fan controls: fixed head pressure with a set point of 45°C in the condensing temperature (the normal mode) and floating condensing temperature by controlling the fans at variable speed based on a set point adjusted by the dry bulb temperature (the VSD mode). Fig.1 shows the variables measured at 5-min intervals at the chiller components and the variables derived for performance analysis. The wet bulb temperature was evaluated from the empirical equations given in [6] based on the measured dry bulb temperature and relative humidity of outdoor air. The measurement was taken during the period of Aug 2015 – Mar 2016. 3905 sets of operating data were collected for the normal mode, 11451 sets for the VSD mode, 830 sets for the normal + mist mode, and 1317 sets for the VSD + mist mode.

The mist cooler operated with the chiller when 2 refrigeration circuits were activated, the outdoor temperature was above 20°C and the relative humidity was below 85%. The high-pressure pump for mist generation consumed a fixed power of 0.33 kW and a water flow rate of 2 L/min. The cooler produced fine mist at a size of 10 microns at around 450 mm in front of the condenser air intake to ensure complete evaporation by the air. The mist cooler effectiveness η_{mist} varies at 0 – 1 theoretically. 0 means that the mist cooler is switched off or no cooling effect. 1 means that the condenser air is cooled to its wet bulb. η_{mist} may exceed 1 when the mist carries over the condenser fins to provide an additional cooling effect.



Derived variables

$COP = Q_{cl}/(E_{cc} + E_{cf} + E_{mist})$
$E_{cf} = (N_{cf}/N_{cf,tot})E_{cf,nom}$ (normal mode)
$E_{cf} = (F_{cf}/F_{cf,tot})^3 E_{cf,nom} + E_{VSD}$ (VSD mode)
$E_{mist} = 0.33$ (mist on) or $= 0$ (mist off)
$Q_{cl} = m_w c_{pw}(T_{chwr} - T_{chws})$
$Q_{cd} = Q_{cl} + E_{cc}$
$PLR = Q_{cl}/Q_{cl,nom}$
$V_a = Q_{cd}/[\rho_a\, c_{pa}(T_{cdal} - T_{cdae})]$
$T_{wb} = f(T_{db}, RH)$ [10]
$\eta_{mist} = (T_{db} - T_{cdae})/(T_{db} - T_{wb})$

Fig. 1. Schematic diagram of the chiller with measured and derived variables.

3. Chiller model development and optimization

Random forest (RF) [7] was used to develop chiller models under four modes (normal, VSD, normal + mist and VSD + mist). RF builds an ensemble of decision trees based on random sets of inputs. The importance of each input is identified during the split of decision trees. Compared with multivariate regression, RF helps correct overfitting to the training data set in creating the models when the data set is huge with uneven distribution. In each mode, the operating data sets were split randomly in a ratio of 70:30 for training and testing the model. The models were created by using the "randomForest" package (version 4.6-12) [8] in statistical software platform R (version 3.3.1) [9]. Details of the model development are given in [10]. The output of the models was COP while the inputs were T_{db}, T_{wb}, RH, T_{chws}, PLR, T_{ev}, T_{cd}, $T_{cdal} - T_{cdae}$, V_a for all modes, together with N_{cf} for the normal mode, F_{cf} for the VSD mode, N_{cf} and η_{mist} for the normal + mist mode and F_{cf} and η_{mist} for the normal + mist mode.

The validated RF models served as fitness functions for genetic algorithm (GA) to predict the maximum COP along with the optimal controlled variables of T_{ev}, T_{cd}, $T_{cdal} - T_{cdae}$, V_a, N_{cf}, F_{cf} and η_{mist}. The package "GA" [11] was used under the statistical software platform R (version 3.3.1). Based on the initial fitness value of COP by the RF models, the genetic operator carried out parents selection, crossover and mutation to compute another fitness value of COP with the optimized values imputed by the RF models. The next fitness value would become the final maximum COP when the convergence was met.

Considering the uneven spread of data sets, unreliable prediction may result from a global optimal search in the entire search space. Two types of data dependent constraints were considered in the GA computation. The deterministic constraints involved limiting the search space in a rectangular region based on the 10 nearest neighboring points for a given set of inputs. This would avoid extrapolating unreasonable results from the data outside the region. As the RF models were purely data-driven models, data dependent constraints were applied to ensure the physical relationships of the predicted variables. Any predicted variable had to lie between its corresponding minimum and maximum values in the data set. The inequality $T_{cdae} \leq T_{cdal} \leq T_{cd}$ had to comply with for heat rejection to take place. For two successive iterations 1 and 2, if $T_{cd,1} \leq T_{cd,2}$, then $V_{a,1} \geq V_{a,2}$.

4. Simulation of hourly cooling load distribution of hotel

EnergyPlus [12] was used to simulate hourly cooling load distribution of a reference hotel in Hong Kong. Details of the hotel are given in [13]. The hotel had a gross floor area of 52020 m^2 with 24 floors. The simulated peak cooling demand was 5202 kW and the annual cooling energy required was 7500140 kWh for 6187 hrs. Table 1 shows the percentage frequencies in a matrix of hourly wet bulb temperatures and hourly building cooling loads as ratios to the peak demand. 34.39% of the building load ratios were in the lowest range of 0 – 0.1 which covered the entire range of wet bulb temperatures. The higher ranges of building load ratios tended to correlate with the higher wet bulb temperatures.

Considering that air-cooled chillers operate typically with the highest COP near the full load, multiple-chiller systems are frequently used for commercial buildings to enhance their high load operation. In this analysis, 2 configurations were examined for the system serving the hotel. One was 6 chillers with a nominal capacity of 867 kW each and another was 8 chillers with a nominal capacity of 651 kW each. The chiller loading patterns of the 2 configurations are given in Tables 2 and 3. Each chiller would carry frequently the higher loads under the sequential chiller loading. Based on the frequency distribution of building cooling loads, the frequency distribution of chiller loads would be identified for the probabilistic analysis of the cooler application.

Table 1. The matrix of hourly wet bulb temperatures and hourly building load ratios.

Wet bulb temperature (°C)	Building load ratio (a ratio to 5202 kW) 0 – 0.1	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 – 1.0	Sub-total
8.4 - 10.6	0.52	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.52
10.7 - 12.9	2.44	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2.44
13.0 - 15.2	3.83	0.02	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	3.85
15.3 - 17.5	9.88	0.11	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	9.99
17.6 - 19.8	10.13	0.44	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	10.57
19.9 - 22.1	6.04	4.33	0.08	0.00	0.00	0.00	0.00	0.00	0.00	0.00	10.46
22.2 - 24.4	0.65	7.63	1.81	0.06	0.00	0.00	0.00	0.00	0.00	0.00	10.15
24.5 - 26.7	0.02	1.44	9.50	2.34	0.48	0.05	0.00	0.00	0.00	0.00	13.84
26.8 - 29.0	0.71	0.00	1.41	17.12	10.51	0.58	0.89	0.27	0.10	0.00	31.58
29.1 - 31.3	0.18	0.00	0.00	0.36	5.61	0.32	0.00	0.02	0.08	0.05	6.61
Sub-total	34.39	13.96	12.80	19.88	16.60	0.95	0.89	0.29	0.18	0.05	100.00

Table 2. Chiller loading pattern of the 6-chiller configuration.

Building cooling load X (kW)	No. of chillers operating	Range of PLRs
0 < X ≤ 867	1	0.03 – 1
867 < X ≤ 1734	2	0.5 – 1
1734 < X ≤ 2601	3	0.67 – 1
2601 < X ≤ 3468	4	0.75 – 1
3468 < X ≤ 4335	5	0.8 – 1
4335 < X ≤ 5202	6	0.83 – 1

Table 3. Chiller loading pattern of the 8-chiller configuration.

Building cooling load X (kW)	No. of chillers operating	Range of PLRs
0 < X ≤ 651	1	0.04 – 1
651 < X ≤ 1302	2	0.5 – 1
1302 < X ≤ 1953	3	0.67 – 1
1953 < X ≤ 2604	4	0.75 – 1
2604 < X ≤ 3255	5	0.8 – 1
3255 < X ≤ 3906	6	0.83 – 1
3906 < X ≤ 4557	7	0.86 – 1
4557 < X ≤ 5208	8	0.88 – 1

5. Results and discussion

5.1 Model validation

The accuracy of the RF models was examined by the robust coefficient of determination R_{L1}^2 given by Eq. (1) [14], where y_i is the actual COP and $\hat{y}$ is the corresponding estimated COP. Table 4 summarizes the results which ascertained the model accuracy as all values of R_{L1}^2 were above 0.8 in the presence of outliers.

$$R_{L1}^2 = 1 - \left(\sum_{i=1}^{n} | y_i - \hat{y} | \div \sum_{i=1}^{n} | y_i - \text{median } y | \right)^2 \quad (1)$$

Table 4. Results of R_{L1}^2 in different modes.

Mode	R_{L1}^2		% of outliers in sample size	
	Training set	Testing set	Training set	Testing set
Normal	0.9810	0.9091	12.26	15.36
VSD	0.9933	0.9653	7.20	12.44
Normal + mist	0.9607	0.8052	7.38	11.45
VSD + mist	0.9789	0.8781	5.50	9.11

Table 5. Rankings of variables importance.

Mode	Ranking		
	1st	2nd	3rd
Normal	PLR	T_{cd}	N_{cf}
VSD	T_{cd}	PLR	F_{cf}
Normal + mist	PLR	V_a	T_{cd}
VSD + mist	T_{cd}	PLR	T_{chws}

Table 5 shows the top three significant input variables in developing the RF models. The PLR depended on how the building cooling load was handled by the operating chillers and its optimization involved the whole system including chillers and auxiliary pumps. With regard to the individual chiller, the interaction between the condensing temperature and heat rejection airflow rate influenced strongly the maximum COP. Using mist did not change significantly the model characteristics under the VSD mode but altered the ranking of T_{cd} and V_a under the normal mode. Indeed, the cooler effectiveness influenced the V_a required to maintain the T_{cd} under a fixed set point control.

5.2 *Optimization analysis*

Figs. 2 and 3 are scatter plots showing how the T_{cd} and V_a should be adjusted under the normal + mist mode and the VSD + mist mode to achieve the maximum COP. In many operating conditions, the T_{cd} was required to drop by increasing the V_a. The adjustment of the set point of T_{cd} was based on its actual value and the variation of V_a. The η_{mist} could not be optimized directly and its variation depended on the V_a and the $T_{db} - T_{wb}$.

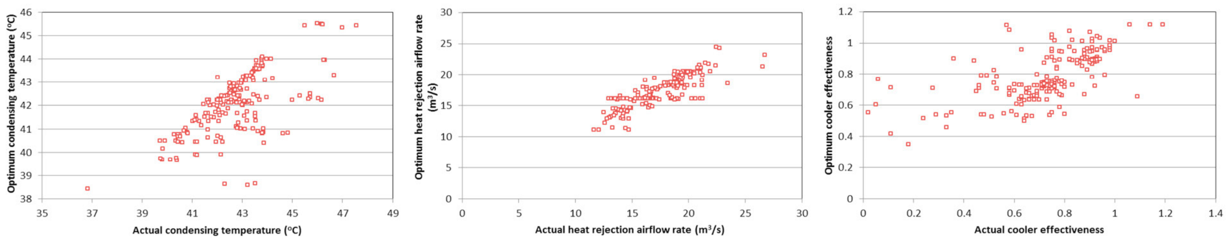


Fig. 2. Optimum vs actual variables under the normal + mist mode.

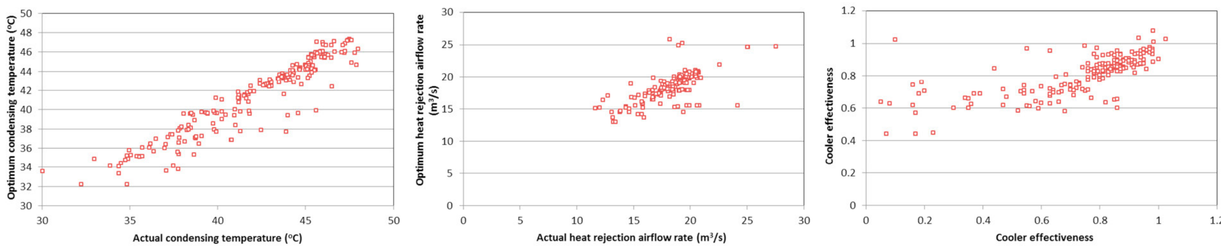
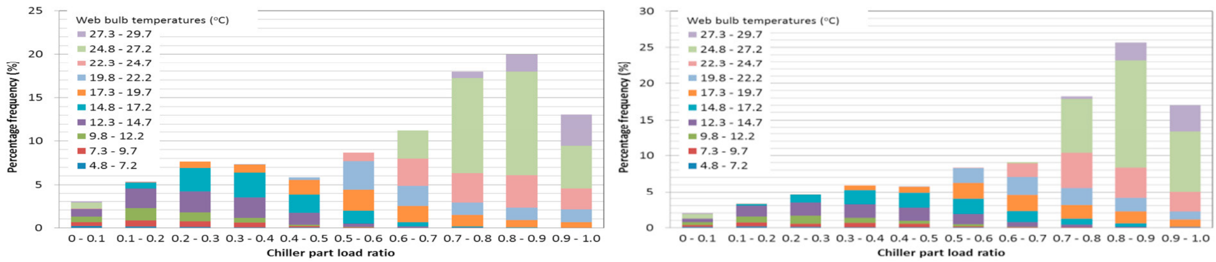


Fig. 3. Optimum vs actual variables under the VSD + mist mode.

5.3 *Probability of using mist coolers and electricity savings estimation*

Fig. 4 illustrates the frequency distribution of chiller part load ratios for the 6-chiller and 8-chiller configurations. Based on the weather and operating criteria, the probability of using mist coolers was 70.91% for the 6-chiller configuration and 78.33% for the 8-chiller configuration. Table 6 shows the annual electricity consumption and average COP of chillers in the system serving the hotel under different modes and configurations. Extra electricity savings of 2.28 – 8.16% were achieved by using mist coolers with optimal control under the normal and VSD modes. The 6-chiller configuration gave higher electricity savings under the VSD and VSD + mist modes. Extra savings by mist coolers were higher in the normal mode and the 8-chiller configuration with more mist cooling time.



(a) 6-chiller configuration (b) 8-chiller configuration

Fig. 4. Frequency distribution of chiller part load ratios.

Table 6. Annual electricity consumption and annual average COP of the chiller system serving the hotel.

Mode		Normal (base)	VSD	Normal + mist	VSD + mist	Normal + mist + optimal control	VSD + mist + optimal control
	Consumption (kWh)	3076128	2414568	2992156	2344441	2896425	2259641
6-chiller	(% saving w.r.t. base)	(-)	(21.51)	(2.73)	(23.79)	(5.84)	(26.54)
	Average COP	2.44	3.1	2.5	3.2	2.58	3.32
	Consumption (kWh)	3001530	2431044	2918507	2359646	2825132	2274296
8-chiller	(% saving w.r.t. base)	(-)	(20.97)	(5.12)	(23.29)	(8.16)	(26.07)
	Average COP	2.5	3.08	2.57	3.18	2.65	3.30

6. Conclusions

This study discusses how to reduce the electricity consumption of an air-cooled chiller system by using mist coolers with optimal control. The chiller studied was equipped with screw compressors and had a coefficient of performance (COP) of 2.8 at a condenser air temperature of 35°C. It was retrofitted with dual condenser fan controls: fixed head pressure with constant speed (the normal mode) and floating condensing temperature with variable speed (the VSD mode). Mist coolers were installed with the chiller to cool condenser air (i.e. the mist mode) when two refrigeration circuits were activated, the outdoor temperature was above 20°C and the relative humidity was below 85%. A data acquisition panel was newly added to log operating data at 5-min intervals for detailed performance analysis. EnergyPlus was used to simulate the hourly cooling load distribution of a hotel in a subtropical climate. 6-chiller and 8-chiller configurations were considered for the system serving the hotel in order to evaluate probability distribution of applying mist coolers under various operating conditions.

Based on comprehensive sets of operating data, random forest (RF) models were developed for the normal mode, the VSD mode, the normal + mist mode and the VSD + mist mode. Their accuracy was ensured by a robust coefficient of determination of above 0.8. The RF models indicated that the interaction between the condensing temperature and heat rejection airflow rate influenced strongly the maximum COP. An optimization study was conducted by using genetic algorithm with the RF models as a fitness function. To achieve maximum COP, the adjustment of the set point of condensing temperature was based on its actual value and the variation of heat rejection airflow rate. The cooling effectiveness of mist coolers could not be optimized directly and its variation depended on the heat rejection airflow rate and the difference between the dry bulb and wet bulb temperatures. The probability of using mist coolers was 70.91% for the 6-chiller configuration and 78.33% for the 8-chiller configuration. Mist coolers with the optimal control reduced the electricity consumption of chillers by 2.28 – 8.16%, depending on system configurations and condenser fan control modes. The 6-chiller configuration brought higher electricity savings from the optimal control while mist coolers gave a higher energy saving potential for the 8-chiller configuration with more operating hours near the full load.

Acknowledgements

This study was supported by a grant from the College of Professional and Continuing Education, an affiliate of The Hong Kong Polytechnic University (Project A/C: 4.8C.xx.EZ72) and a grant from the Research Grant Council of HKSAR (Project A/C: B-Q31R).

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