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Cellulose-pad water cooling system with cold storage

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ABSTRACT

The present study develops a cooling system using water as the working medium which is cooled at night by cellulose-pad cooling tower (CWCT) and stored for cooling application at daytime. That is, it utilizes the natural energy drawn from diurnal ambient air temperature difference. A cooling system was built and tested. It is found that the coefficient of performance of CWCT for heat dissipation of water at night, COP_{nt}, is between 3.8 and 11 and varies linearly with the evaporation temperature glide D_G (difference between cold water temperature in the storage tank and wet-bulb temperature of ambient air). The COP for room cooling at daytime run with air cooler in a room, COP_{day}, is between 8.8 and 12.6. For day cycle operation, the measured overall cooling COP_o is 5.1. COP_o is expected to reach 9.4 at room temperature 45 °C.

Système de refroidissement d'eau de milieu dispersant-cellulose avec entreposage frigorifique

Mots clés : Refroidissement évaporatif ; Système de refroidissement d'eau de milieu dispersant à cellulose ; Refroidissement de milieu dispersant-cellulose ; Système de refroidissement d'eau

1. Introduction

Air conditioning system consumes majority of energy and the improvement of energy efficiency is quite important. For example, residential type vapor-compression air conditioner has COP (coefficient of performance) between 3.0 and 4.5. The CFC refrigerants used in air conditioner also cause global warming. Water is thus the most natural refrigerant and is used in LiBr–H_2O absorption system, a sophisticated machine using heat as the driving energy source.

It is noticeable that the ambient air temperature varies all day, at a lower temperature at night and higher at daytime. The largest diurnal ambient air temperature variation depends on

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Nomenclature

a	ratio of air/water contact area to tower volume [m^{-1}]
C_p	specific heat capacity of water [kJ kg^{-1} K^{-1}]
COP_{day}	COP of room cooling in daytime mode [$=E_{room}/E_{day}$]
COP_{nt}	COP of cold storage in nighttime mode [$=E_d/E_{nt}$]
D_a	cooling tower temperature approach of CWCT $= T_{wout} - T_{WB}$, K
D_{dn}	difference between daytime dry-bulb temperature (T_{DBd}) and nighttime wet-bulb temperature (T_{WBn}), $= T_{DBd} - T_{WBn}$, K
D_G	evaporation temperature glide $= T_{wi} - T_{WBi}$, K
E_d	total cold energy storage [kWh]
E_{day}	total energy consumption in daytime mode [kWh]
E_{max}	cold energy storage limit [J]
E_{nt}	total energy consumption in nighttime mode [kWh]
E_{room}	total energy removed from room [kWh]
G_a	mass flux of air through CWCT [kg s^{-1} m^{-2}]
H	height of cellulose-pad water cooling tower [m]
H_d	heat dissipating rate of the cellulose-pad cooling tower (CWCT) at night [W]
J_a	mass flux of water through CWCT [kg s^{-1} m^{-2}]
K	mass transfer coefficient
L	water mass flow rate through CWCT [kg s^{-1}]
m_{air}	air mass flow rate through CWCT [kg s^{-1}]
M_t	water storage in tank [kg]
P_{nt}	average power consumption of CWCT [W]
P_{day}	average power consumption in daytime mode [W]
Q_c	average cooling rate in daytime mode [W]
T_a	ambient temperature [°C]
T_{ai}	initial ambient temperature [°C]
T_{DBd}	average daytime dry-bulb temperature [°C]
T_{room}	room temperature [°C]
T_{ri}	initial room temperature in daytime or nighttime mode [°C]
T_{rf}	final room temperature in daytime or nighttime mode [°C]
T_{WB}	wet-bulb temperature of air through CWCT [°C]
T_{WBi}	initial wet-bulb temperature of air through CWCT, °C]
T_{WBn}	average nighttime wet-bulb temperature [°C]
T_{wi}	initial water temperature in tank [°C]
T_{wiD}	initial water temperature in tank in daytime mode [°C]
T_{wiN}	initial water temperature in tank in nighttime mode [°C]
T_{wfD}	final water temperature in tank in daytime mode [°C]
T_{wfN}	initial water temperature in tank in nighttime mode [°C]
T_{winlet}	inlet water temperature of CWCT [°C]
T_{wf}	final water temperature in tank [°C]
T_{wout}	CWCT outlet water temperature [°C]
t_{day}	operating time of daytime mode [hr]
t_{nt}	operating time of nighttime mode [hr]
V	volume of CWCT [m^3]

the season, location, and climate. If water is used as the working medium which is cooled at night by ambient air and stored for cooling application at daytime, the energy efficiency may be feasible for cooling applications. Fig. 1 shows a typical summer diurnal ambient temperature variation in Taipei City, located in humid subtropical area. The difference (D_{dn}) between average daytime dry-bulb temperature (T_{DBd}) and average nighttime wet-bulb temperature (T_{WBn}), i.e. $D_{dn} = T_{DBd} - T_{WBn}$, varies every

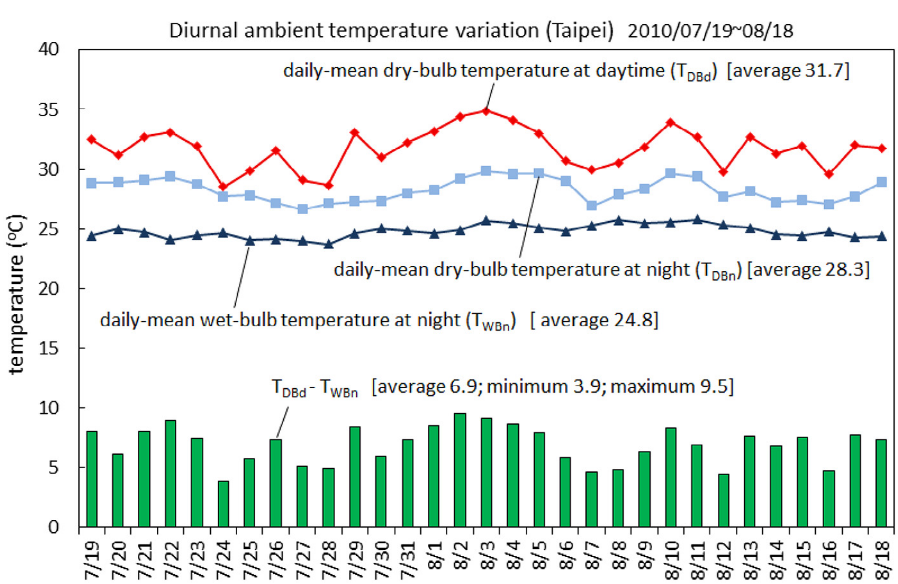

Fig. 1 – A typical diurnal ambient temperature variation in Taipei City.

day. The monthly mean is 6.9 °C with minimum 3.9 °C and maximum 9.5 °C. It may be feasible to design a water cooling system with storage which cools the water at night and stored for cooling application at daytime.

Conventional water cooling tower can be used for water cooling by ambient air. But, the volume and the power consumption are large. Cellulose pad has a much better water evaporative cooling performance than conventional cooling tower. It is widely used in room air cooling in dry regions or in agriculture.

Many researchers studied the performance improvement of air conditioners using water cellulose pad to cool the air before entering the condenser (Goswami et al., 1993) or to cool the warm water from the condenser (Hu and Huang, 2005). Franco et al. (2010, 2014) studied the cellulose evaporative cooling pads to provide cooled air for greenhouse application. The water cellulose pad is used to cool the air or water directly for various applications.

In the present study, we studied a water cooling system using cellulose pads to cool the water at night and store in a storage tank. The cold water is used for cooling purpose at daytime. It is a concept of natural energy utilization.

2. Design of experimental cellulose-pad water cooling system

The concept of the cellulose-pad water cooling system is shown in Fig. 2. Water is circulated through the cellulose-pad cooling tower to dissipate heat to the ambient at night. The cooled water is stored in a storage tank with thermal insulation. The cold water is circulated to an indoor air cooler to cool a room at daytime. An experimental system is designed and built for test in the present study.

2.1. *Design of cellulose-pad water cooling tower (CWCT)*

2.1.1. *Heat and mass transfer correlation for design of CWCT*

The conventional water cooling tower using packing material is widely used in air conditioning system. The phenomena of heat and mass transfer in cooling tower are extensively studied.

Dowdy and Karabash (1987), Hu and Huang (2005), and Franco et al. (2010, 2014) studied experimentally the heat and mass transfer for the design of cellulose-pad water cooling tower. Hu and Huang (2005) derived an empirical correlation, Eq. (1), for a fundamental cell unit of cellulose pad in the dimension 0.3 m (width) × 0.3 m (height) × 0.15 m (thickness) and further developed a design procedure for different sizes of cellulose-pad water cooling tower with height H.

$$\frac{KaV}{L} = 2.2899H\left[\frac{J_a}{G_a}\right]^{-0.3389} \quad (1)$$

where K is the mass transfer coefficient, a is the ratio of air/water contact area to tower volume (V), L is the water mass flow rate, J_a and G_a are the mass flux of water and air, respectively. Eq. (1) is used in the heat and mass transfer calculation of CWCT for a given flow rate of water and air and the cooling tower temperature approach D_a which is defined as

$$D_a = T_{wout} - T_{WB} \quad (2)$$

where T_{wout} is cooling tower outlet water temperature, T_{WB} is the air wet-bulb temperature.

2.1.2. *Water storage tank and heat dissipating rate estimation*

From Fig. 1, there is a difference of 6.9 °C between the wet-bulb temperature of air at night (T_{WBn}) and that of the daytime (T_{DBd}). The lowest temperature limit at outlet water flow of cooling tower will be the wet-bulb temperature of air at night (T_{WBn}). The highest temperature limit at inlet water flow of the cellulose-pad cooling tower will be the ambient air temperature at daytime (T_{DBd}) which is usually higher than the room temperature (T_{room}). The cold energy storage limit is thus:

$$E_{max} = M_t C_p (T_{DBd} - T_{WBn}) \quad (3)$$

For experimental purpose, we designed a cold storage tank containing 1000 kg of water. Hence, the cold energy storage limit is E_{max} = 29 MJ (8 kWh). The heat dissipating rate of the cellulose-pad cooling tower at night (H_d) is between 1000 W and 2000 W (assuming 5–10 hours operation). We take 1500 W as the design. That is, H_d = 1500 W.

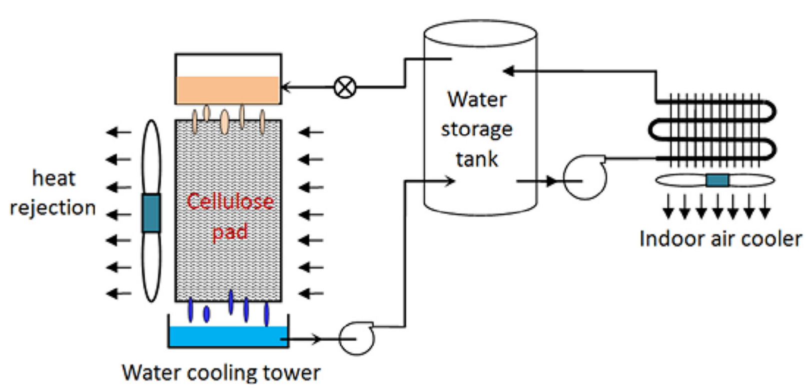

Fig. 2 – Schematic diagram of cellulose-pad water cooling system.

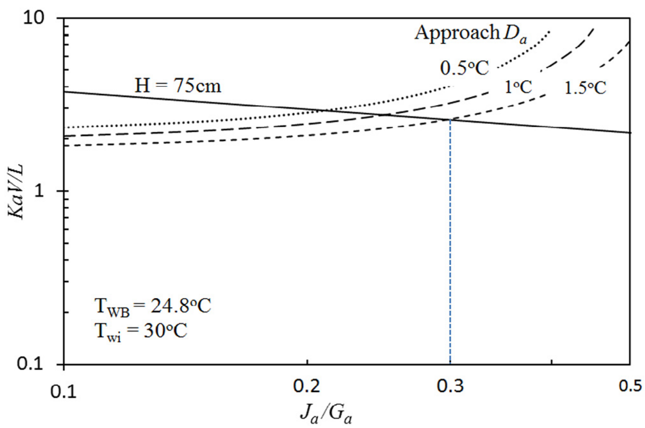

Fig. 3 – Heat and mass transfer correlation for CWCT design.

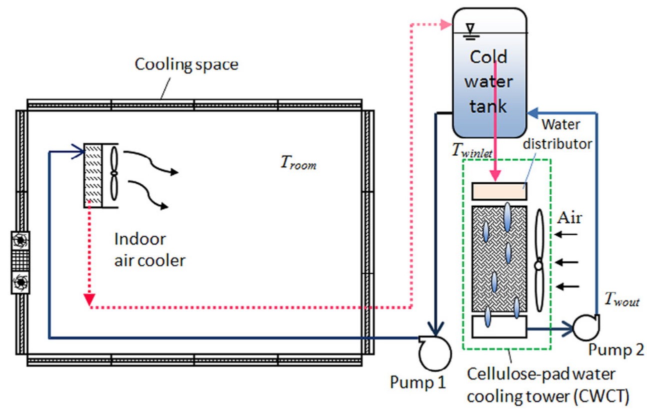

Fig. 4 – Schematic diagram of cellulose-pad water cooling system.

2.1.3. *Maximum water flowrate estimation*

According to Fig. 1, the average wet-bulb temperature is 24.8 °C in summer nights. Assuming the cooling tower temperature approach D_a is 1.5 °C, the outlet water temperature will be $T_{wout} = T_{WB} + D_a = 26.3$ °C.

The cold water stored at night will be used to cool the room at daytime and causes water temperature rise which is limited by ambient air temperature (average 32 °C) (Fig. 1). The inlet water temperature (T_{winlet}) of CWCT is assumed lower than the ambient air temperature at daytime by about 2 °C, i.e. $T_{winlet} = 30$ °C. The required water mass flowrate is $L = 0.097$ kg s^{-1} (or 5.8 kg min^{-1}), which is estimated from the energy balance relation, Eq. (4).

$$H_d = LC_p(T_{winlet} - T_{wout}) \tag{4}$$

2.1.4. *Size of cellulose-pad tower and air flowrate estimation*

For a given size of cellulose-pad cooling tower, the air flowrate affects the heat and mass transfer and is calculated from the empirical correlation, Eq. (1). The dimension of the cellulose-pad cooling tower is assumed as 60 cm width × 75 cm height × 30 cm depth. Using the design procedure of Hu and Huang (2005), we obtain the design chart for $H = 75$ cm, $T_{WB} = 24.8$ °C, $T_{winlet} = 30$ °C shown in Fig. 3. From that, for cooling tower temperature approach D_a at 1.5 °C, we obtain $J_a/G_a = 0.3$. The air mass flow rate is determined as 0.81 kg s^{-1} and the air velocity is 1.25 m s^{-1}. Table 1 summarizes the design of CWCT.

Table 1 – Design specification of CWCT.

Size of CWCT	0.6 m (width) × 0.75 m (height) × 0.3 m (depth)
Maximum cold water storage (M_t)	1000 kg
Maximum heat dissipating rate (H_d)	1500 W
Wet-bulb temperature at night (T_{WBn})	24.8 °C
Outlet water temperature (T_{wout})	26.3 °C
Inlet water temperature (T_{winlet})	30 °C
Maximum water mass flow rate (L)	0.097 $kg\ s^{-1}$(5.8 $kg\ min^{-1}$)
Maximum air mass flow rate (m_{air})	0.81 $kg\ s^{-1}$
Maximum air flow velocity (U_{air})	1.25 $m\ s^{-1}$
Cooling tower temperature approach D_a	1.5 K
Evaporation temperature glide ($T_{winlet} - T_{WB}$)	5.2 K

2.2. *Overall design of cellulose-pad cooling system*

The CWCT designed in Section 2.1 is used to cool a room with interior dimension 8 m (length) × 3.5 m (width) × 2.8 m (height) for the experiment. The room is a zero-energy house with overall U-value 0.24 W m^{-2} K^{-1}. The whole cellulose-pad cooling system is shown in Fig. 4.

The cold energy stored at night will be used for cooling purpose at daytime. An indoor air cooler was installed inside the cooling room as shown in Fig. 5. The cooler is an air heat exchanger (fan-coil unit) made from aluminum finned copper tubes, with overall dimension 39 cm (length) × 35 cm (width) × 4 cm (depth). Cold water flows inside the tubes to absorb the heat of the room air. Nine DC fans (12 V/3 W/24 m^3 h^{-1}) were used to circulate the room air through the air cooler at maximum flowrate 127 m^3 h^{-1} with 27 W total fan

Fig. 5 – Installation of indoor air cooler.

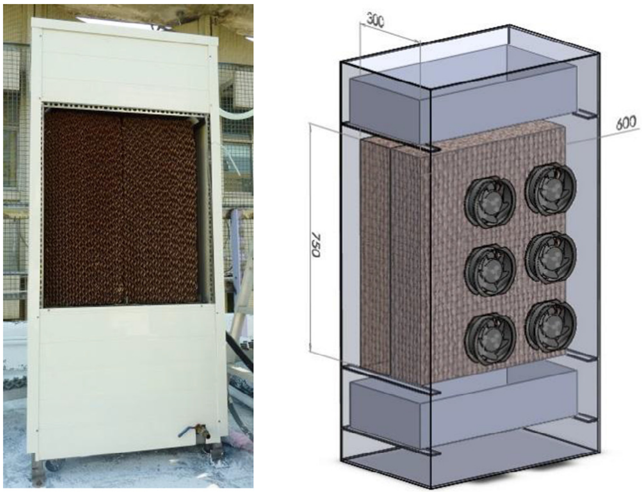

Fig. 6 – Installation of CWCT.

power consumption. Two 12VDC water pumps were installed in series in the water line to pump cold water from storage tank through a pipe about 10 meters. Pump 1A consumes about 24 W at maximum flowrate 20 liter min^{-1} and its maximum head is 3.1 mH_2O. Pump 1B consumes about 18 W at maximum flowrate 13.3 liter min^{-1} and its maximum head is 4 mH_2O. The maximum power consumption for fans and pumps at daytime for room cooling is 69 W.

In CWCT, the water flow is supplied by a 12VDC pump with maximum power consumption of 18 W at maximum flowrate of 13.3 liter min^{-1} and maximum head of 4 mH_2O. Six 12VDC fans (15 cm × 15 cm) with head 16.8 mm H_2O were installed for driving air through the CWCT. The air velocity is in the range 1.5~2.4 m s^{-1}. The total maximum power consumption of CWCT is 95 W. Fig. 6 shows the installation of CWCT.

2.3. *Design of control and measurement systems*

The cellulose-pad water cooling system operates in two modes: nighttime mode and daytime mode. In nighttime mode, water is cooled and stored in the storage tank. In daytime mode, cold water is used for cooling of a room. A microprocessor-based control system was designed in the present study to control the operations. Fig. 7 shows the control scheme of nighttime mode. The CWCT is turned on when the wet-bulb temperature of ambient air is lower than dry-bulb temperature by 1 °C (i.e. $T_a - T_{WB} > 1$ °C) at night. The turn-on time can be changed for any desired time interval for studying the specific performance.

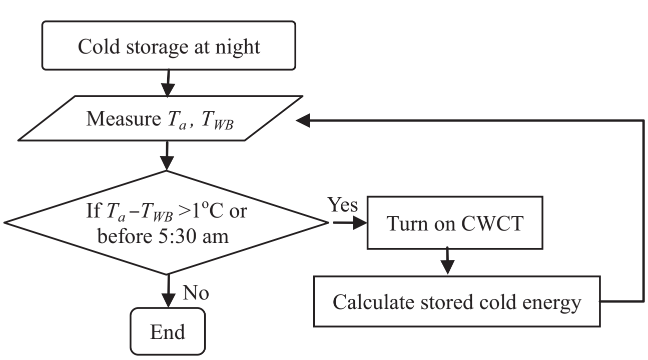

Fig. 7 – Control scheme of nighttime mode.

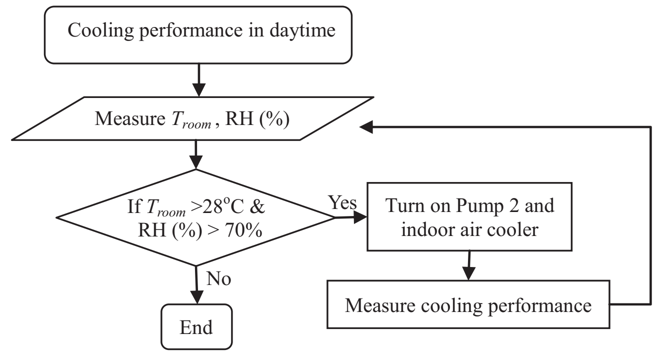

Fig. 8 – Control scheme of daytime mode.

Fig. 8 shows the control scheme of daytime mode. The indoor air cooler is turned on when $T_{room} > 28$ °C and relative humidity RH > 70% in daytime. The daytime operation can also be changed for any desired time interval for studying the performance. The system always operated between 9:00 ~ 17:00 (8 hrs) in daytime mode and 19:30 ~ 5:30 (10 hrs) in nighttime mode.

A PC-based monitoring system was designed to measure the system performance. Fig. 9 shows the measuring positions and instruments used. All the measured data are transmitted to a PC via 7018Z and 7017A5 transmitters.

3. Experimental results

The experiment of the whole cellulose-pad water cooling system in nighttime and daytime modes is run at highest flowrate of air and water. Data are collected to analyze the performance characteristics in nighttime and daytime modes.

3.1. *Performance of CWCT in nighttime mode (cold storage)*

Fig. 10 shows the instantaneous performance of cellulose-pad water cooling system. The heat dissipation and cold storage in nighttime mode are run between 19:30 (August 2) and 5:30 (August 3), for 10 hours. The average heat dissipation rate of CWCT was 564 W. Fan and pumps are run in full capacity. The fan consumed 82 W, and the water pump consumed 13 W, with total power consumption 95 W in nighttime mode. The COP in cold storage (heat dissipation) performance, COP_{nt}, is 5.94. It can be seen from Fig. 10 that the initial temperature difference between wet-bulb temperature T_{WBi} and water inflow T_{wi}, i.e. the initial evaporation temperature glide D_G, is around 2 °C. Fig. 11 shows the daily performance of the cellulose-pad water cooling system. The operating time is 10 hours for nighttime mode (heat dissipation) and 9 hours for daytime mode (room cooling).

We run the cold storage performance in nighttime continuously in summer and obtain the results of Table 2. It is seen that the COP of cold storage (COP_{nt}) in nighttime mode

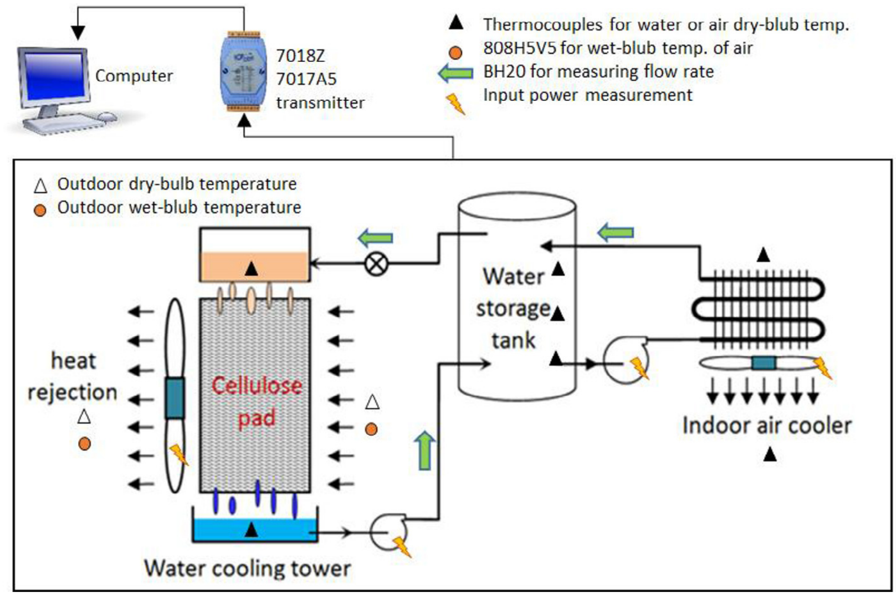

Fig. 9 – Schematic of monitoring system.

increases with evaporation temperature glide D_G which is defined as:

$$D_G = T_{wi} - T_{WBi} \quad (5)$$

where T_{wi} is the initial water temperature in tank in nighttime mode, T_{WBi} is the initial wet-bulb temperature of outdoor air. D_G is an important parameter affecting the heat and mass transfer of CWCT which ranges from 1.9 to 6.5 °C in summer. The measured COP_{nt} ranges from 3.8 to 11 for the present CWCT. The average heat dissipation rate (H_d) ranges from 360 to 1100 W.

Table 3 is the test results of nighttime-mode performance (cold storage) in winter which shows that D_G ranges from 2.1 to 4.2 °C, lower than in summer. By combining the performance in summer and winter, Fig. 12 shows that COP_{nt} of CWCT in nighttime mode (cold storage) increases linearly with D_G.

The experiments were carried out in Taipei, located in subtropical area with high humidity. This causes a lower COP_{nt}. It is shown from Fig. 12 that COP_{nt} can be higher than 12 for the location and operating condition with $D_G > 8$ °C, mostly happens in winter.

Table 4 shows that the measured cooling tower temperature approach D_a is mostly close to the design point (1.5 °C).

3.2. *Performance of room cooling in daytime mode*

The daytime-mode performance is actually the room cooling using the indoor air cooler in the zero-energy house. Table 5

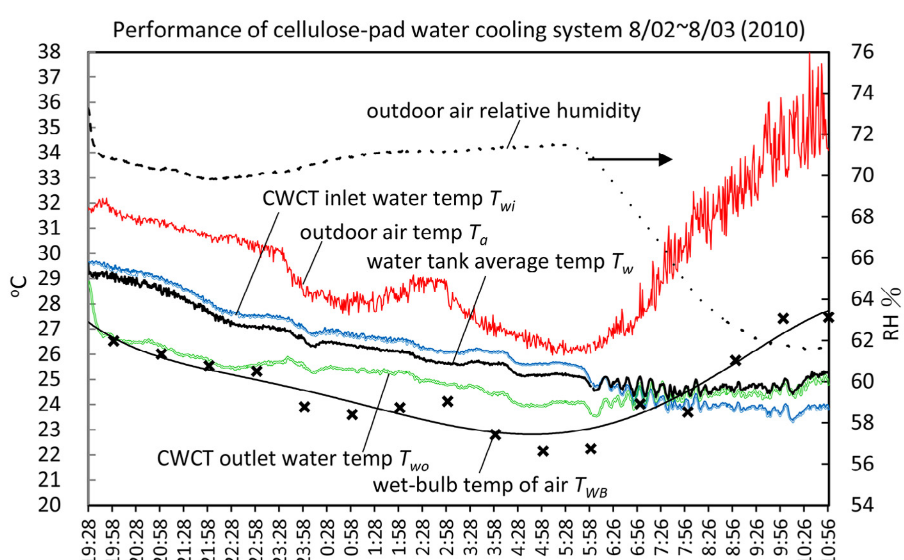

Fig. 10 – Performance in nighttime mode.

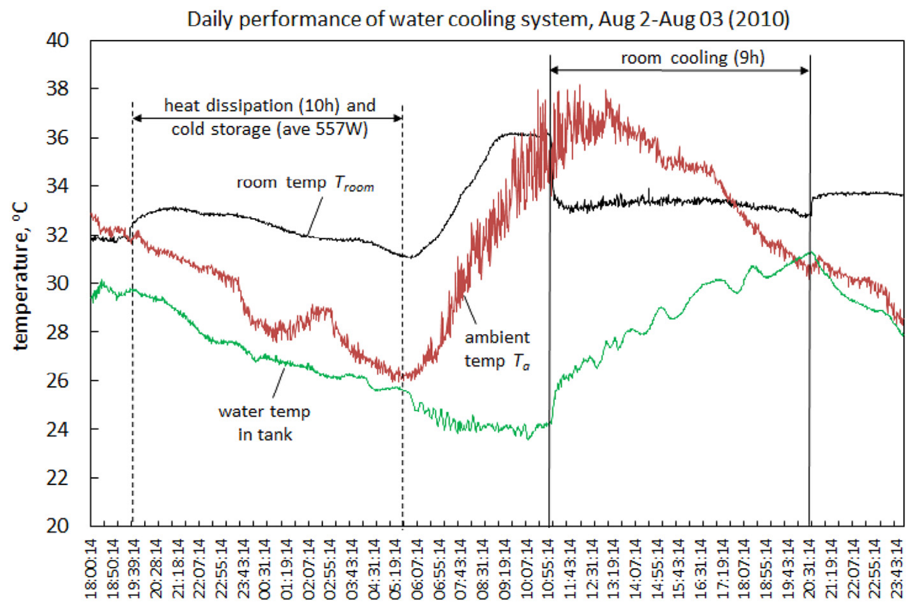

Fig. 11 – Performance in nighttime and daytime modes.

shows the measured results in summer. The COP is between 8.8 and 14.5 with average cooling rate 300~800 W. Fig. 13 shows that COP_{day} varies linearly with the initial temperature difference between room air (T_{ri}) and cold water in tank (T_{wi}).

4. Performance analysis of CWCT in cyclic operation

4.1. *Overall COP of CWCT in cyclic operation*

The cold water stored in nighttime is to be used for room cooling at daytime and absorbs the heat of the room. For cyclic operation day by day, it requires that heat removed from the room at daytime (room cooling) equals to the heat dissipated to the ambient at night. The energy absorbed by the cold water at daytime is the energy removed from the room E_{room}. The COP in daytime mode is defined as

$$COP_{day} = \frac{E_{room}}{E_{day}} = \frac{Q_c}{P_{day}} \tag{6}$$

where P_{day} is the average power consumption in daytime mode, and E_{day} is the total energy consumption ($=P_{day} \times t_{day}$), t_{day} is the operating time of daytime mode.

The heated water is then cooled by CWCT and stored at night. The total energy dissipated is E_d. The COP in nighttime mode is defined as

$$COP_{nt} = \frac{E_d}{E_{nt}} = \frac{H_d}{P_{nt}} \tag{7}$$

where P_{nt} is the average power consumption in nighttime mode, and E_{nt} is the total energy consumption ($=P_{nt} \times t_{nt}$), t_{nt} is the operating time of nighttime mode.

For cyclic operation of the water cooling system in a whole day, it requires $E_{room} = E_d$. Therefore:

$$COP_{day}E_{day} = COP_{nt}E_{nt} \tag{8}$$

The overall system COP for a day cycle operation, COP_o, is defined as:

$$COP_o = \frac{E_d}{E_{nt} + E_{day}} = \frac{E_{nt}}{E_{nt} + E_{day}} COP_{nt} = \frac{E_{day}}{E_{nt} + E_{day}} COP_{day} \tag{9}$$

The test on August 3 satisfies approximately the requirement of cyclic operation $E_{room} = E_d$ and the data are analyzed. Table 6 shows that the overall COP_o is 5.1, which is not very high since all the fans and pumps are run at full capacity with a very large design margin.

4.2. *Analysis of cyclic performance of CWCT at various operating conditions*

The experimental data obtained in the present study can be used to calculate the overall cyclic performance at various operating conditions. COP_o is affected by operating conditions. The test result of Table 6 is for initial room temperature at 35.7 °C in daytime mode and initial water temperature 31.1 °C in nighttime mode. COP_o can be analyzed for different operating conditions using the test results.

In nighttime mode, Fig. 12 shows that measured COP_{nt} of CWCT in nighttime mode (cold storage) increases linearly with D_G. The empirical correlation can be derived as:

$$COP_{nt} = 1.4153(T_{wiN} - T_{WBi}) + 1.1641 \tag{10}$$

Assume that the room temperature setting (T_{room}) and the ambient wet-bulb temperature (T_{WB}) are not changed all day long. The initial water temperature in nighttime mode (T_{wiN}) actually approximates the final room temperature in daytime mode (room cooling) by 3 °C, i.e. $T_{wiN} = T_{room} - 3$ °C, due to heat transfer limit of the air cooler. The evaporation temperature

Table 2 – Nighttime mode (cold storage) performance in summer.

	Date (2010)							
	8/02	8/03	8/05	8/05	8/10	8/11	8/12	8/13
Time of operation	19:28 ~05:30	20:35 ~05:30	20:41 ~ 05:49	20:41 ~ 02:12	22:04 ~ 02:54	00:02 ~ 05:17	00:00 ~ 05:25	00:06 ~ 05:32
Operating time, hr	10.1	8.9	9.2	5.5	4.9	5.3	5.4	5.4
Initial ambient temperature, T_{ai} (°C)	32.2	31.2	29.7	29.7	27.9	28.0	26.3	32.2
Average wet-bulb temperature of outdoor air, $T_{WB,av}$ (°C)	24.7	23.9	23.8	23.7	25.2	21.9	24.8	23.9
Initial wet-bulb temperature of outdoor air T_{WBi} (°C)	27.5	25.8	23.7	23.7	25.7	22.2	25.0	24.4
Initial water temperature in tank, T_{wi} (°C)	29.4	31.1	27.1	27.1	28.3	28.3	27.7	28.2
Final water temperature in tank, T_{wf} (°C)	25.6	24.9	25.0	25.3	25.9	24.5	25.7	24.4
Total cold energy storage, E_d (kWh)	5.70	6.68	3.33	2.38	2.67	4.36	2.53	3.81
Average heat dissipation rate, H_d (W)	564	750	362	432	545	822	468	705
Average power consumption, P_{nt} (W)	95	95	95	95	105	95	103	103
Total energy consumption in nighttime mode, E_{nt} (kWh)	0.96	1.07	1.11	0.66	0.79	0.64	0.69	0.69
Initial evaporation temperature glide in nighttime mode, D_G ($T_{wi} - T_{WBi}$) (K)	1.9	5.3	3.4	3.4	2.6	6.1	2.7	3.8
COP of cold storage, COP_{nt} (=E_d/E_{nt})	5.94	7.90	3.81	4.54	4.50	8.65	4.53	6.82
	8/13	8/14	8/18	8/19	8/19	8/23	9/13	9/14
Time of operation	22:00 ~ 06:37	00:00 ~ 05:28	00:00 ~ 05:30	00:41 ~ 05:30	00:41 ~ 06:18	23:01 ~ 05:29	21:58 ~ 06:49	21:41 ~ 06:00
Operating time, hr	8.6	5.4	5.5	4.8	5.5	6.2	8.9	8.3
Initial ambient temperature, T_{ai} (°C)	28.8	28.1	28.0	28.9	28.9	29.3	28.8	25.8
Average wet-bulb temperature of outdoor air, $T_{WB,av}$ (°C)	23.4	24.6	24.1	23.4	23.4	24.5	23.4	22.4
Initial wet-bulb temperature of outdoor air T_{WBi} (°C)	24.4	25.0	24.3	22.5	22.5	25.1	24.5	23.2
Initial water temperature in tank, T_{wi} (°C)	30.9	28.5	28.1	27.8	27.8	26.8	31.0	26.9
Final water temperature in tank, T_{wf} (°C)	23.7	25.0	25.0	24.3	24.0	24.6	23.8	23.0
Total cold energy storage, E_d (kWh)	9.64	4.01	3.68	4.35	4.65	2.82	9.86	6.21
Average heat dissipation rate, H_d (W)	1,122	742	668	907	845	455	1,108	748
Average power consumption, P_{nt} (W)	105	104	104	104	105	103	100	98
Total energy consumption in nighttime mode, E_{nt} (kWh)	1.30	0.70	0.71	0.62	0.58	0.80	1.11	1.02
Initial evaporation temperature glide in nighttime mode, D_G (=$T_{wi} - T_{WBi}$) (K)	6.5	3.5	3.8	5.3	5.3	1.7	6.5	3.7
COP of cold storage, COP_{nt} (=E_d/E_{nt})	9.52	7.15	6.44	8.74	8.05	4.40	11.06	7.66

Table 3 – Nighttime mode (cold storage) performance in winter.

	Date (2010)			
	11/29	12/01	12/02	12/06
Operating time (20:00–5:30), hr	9.5	9.5	9.5	9.5
Average humidity, RH (%)	80.7	80.7	78	69
Average outdoor temperature T_a (°C)	20.3	20.5	19	18.3
Average wet-bulb temperature T_{WB} (°C)	18.1	18.5	16.5	14.8
Initial tank temperature T_{wi} (°C)	20.6	20.9	21	20.2
Final water temperature in tank, T_{wf} (°C)	18	18.2	16.5	14
Initial wet-bulb temperature of outdoor air T_{WBi} (°C)	18.2	18.8	17.4	16
Average heat dissipation rate, H_d (W)	344	350	552	716
Total cold energy storage, E_d (kWh)	3.27	3.32	5.24	6.80
Average power input, P_{nt} (W)	98	98	98	98
Initial evaporation temperature glide D_G (=$T_{wi} - T_{WBi}$) (K)	2.4	2.1	3.6	4.2
COP of cold storage, COP_{nt} (=H_d/P_{nt})	3.52	3.57	5.65	7.29

glide D_G (=$T_{wiN} - T_{WB}$) and COP_{nt} can then be calculated by Eq. (10), for a given T_{room} and T_{WB}.

For a given power consumption in nighttime mode $P_{nt} = 95$ W and the total dissipated energy in nighttime mode E_d (=E_{room}), we can calculate H_d, t_{nt} and E_{nt} using the following relations:

$$H_d = COP_{nt}P_{nt}; \quad t_{nt} = E_d/P_{nt}; \quad E_{nt} = P_{nt}t_{nt}$$

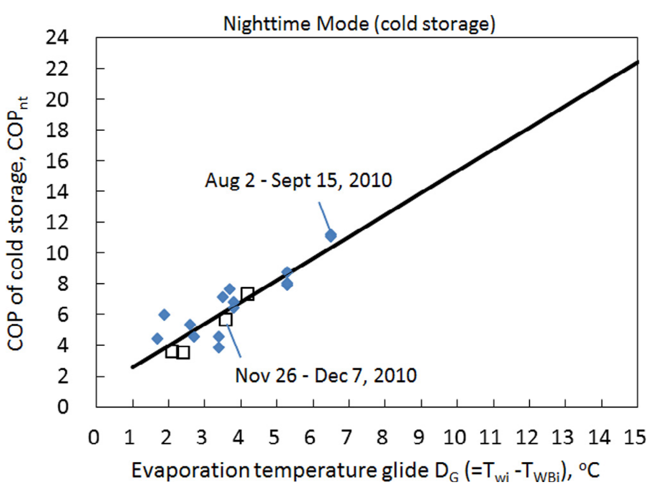

Fig. 12 – Measured COP of nighttime-mode performance.

The final water temperature at the end of nighttime mode (T_{wfN}) is calculated from the energy balance of the tank: $T_{wfN} = T_{wiN} - E_d/M_t\ C_p$, which will be the initial water temperature of daytime mode. The cooling tower temperature approach at the end of nighttime mode is calculated by $D_a = T_{wfN} - T_{WB}$ which should be greater than 1 K as the operation limit.

In daytime mode, Fig. 13 shows that the measured COP_{day} in daytime mode (indoor air cooling) increases linearly with the difference between initial room air temperature (T_{ri}) and wet-bulb temperature of outdoor air (T_{WBi}). The empirical correlation can be derived as:

$$COP_{day} = 1.6803(T_{ri} - T_{wiD}) + 2.7445 \tag{11}$$

Assume that the tank is thermally insulated very well. The initial water temperature in daytime mode T_{wiD} is actually the final water temperature at the end of nighttime mode, T_{wfN}, i.e. $T_{wiD} = T_{wfN}$. Since $T_{ri} = T_{room}$, we can calculate COP_{day} using Eq. (11), for a given T_{room} and further calculate Q_c, t_{day} and E_{day} using the following relations:

$$Q_c = P_{day}COP_{day}; \quad t_{day} = E_{room}/Q_c; \quad E_{day} = t_{day}P_{day} \tag{12}$$

The measured average power consumptions $P_{day} = 56$ W and $P_{nt} = 95$ W in daytime-mode and nighttime-mode, respectively, are used in the analysis. The overall COP for daytime and nighttime mode operations in a day cycle is calculated by the relation:

Table 4 – Cooling tower temperature approach in nighttime mode (cold storage) performance in summer.

	Date (2010)					
	8/2	8/11	8/13	8/16	8/18	9/15
Initial dry-bulb temperature of outdoor air T_{DBi} (°C)	31.8	28.7	26.3	25.5	27.4	27.8
Initial wet-bulb temperature of outdoor air T_{WBi} (°C)	27.3	25.6	24.1	23	24.3	24.1
Initial inlet water temperature of CWCT, T_{winlet} (°C)	29.6	27.5	28.5	26.9	28.2	29.6
Initial outlet water temperature of CWCT T_{wout} (°C)	28.8	26.2	25.3	24.5	26.0	27.1
Average heat dissipation rate H_d (W)	564	822	705	362	668	748
Cooling tower temperature approach D_a (=$T_{wout} - T_{WBi}$) (K)	1.5	0.6	1.2	1.5	1.7	2.0

Table 5 – Daytime mode (room cooling performance).

Daytime mode (room cooling performance)	8/02	8/03	8/06	8/11	8/12	8/13	8/18	9/15
Time of operation	11:00~19:00	11:00~19:00	10:30~15:00	14:30~17:30	10:30~18:00	10:30~18:00	10:30~18:00	9:30~18:00
Operating time, hr	8	8	4.5	3	7.5	7.5	7.5	8.5
Initial ambient temperature, T_{ai} (°C)	34.2	35.5	29.6	31.7	33.3	32.9	30.5	30.1
Initial room temperature, T_{ri} (°C)	34.9	35.7	31.5	33.8	33.2	33.7	31.8	31.4
Final room temperature, T_{rf} (°C)	31.5	30.1	30.1	31.0	29.5	30.8	30.1	29.6
Initial water temperature in tank, T_{wi} (°C)	24.2	24.0	24.5	25.6	25.6	24.5	24.7	23.0
Final water temperature in tank, T_{wf} (°C)	29.6	30.2	26.9	27.5	27.9	28.5	28.2	27.3
Average power consumption P_{day} (W)	56	56	56	55	55	54	55	55
Total energy consumption, E_{day} (kWh)	0.45	0.45	0.25	0.16	0.41	0.41	0.41	0.47
Average cooling rate Q_c (W)	707	811	558	673	322	558	483	530
Total energy removed from room, E_{room} (kWh)	5.65	6.49	2.51	2.02	2.42	4.19	3.62	4.50
Initial room air–water temperature difference, $T_{ri} - T_{wi}$ (K)	5.30	5.50	4.60	6.28	5.30	5.20	3.64	4.10
COP of daytime mode, COP_{day}	12.6	14.5	10.0	12.3	10.9	10.3	8.8	9.7

$$COP_o = \frac{E_d}{E_{nt} + E_{day}} \tag{13}$$

Fig. 14 shows the variation of overall COP_o with room temperature for $E_d = E_{room} = 4$ kWh and $T_{WB} = 23$ °C. It is seen that COP_o increases with room temperature T_{room} and is higher than 8.0 for $T_{room} > 40$ °C. COP_o reaches 9.4 at $T_{room} = 45$ °C. At this case, the cooling tower temperature approach D_a is around 15 °C. This means that in some applications with higher room temperature, such as data center (Fakhim et al., 2011; Zhang et al., 2015), the present cellulose-pad water cooling system is feasible.

5. Discussions

There are many defects in the design of the present experimental system which can be improved to obtain much better results. First, the fan and pumps are run at full capacity and consume a lot of power. The water and air flowrates can be optimized through an intelligent control. This research will be published in another paper.

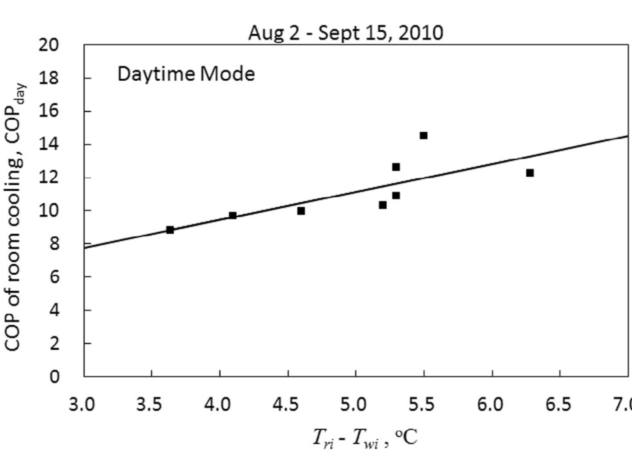

Fig. 13 – Measured COP_{day} in daytime mode.

Second, the design match between CWCT and air cooler is needed in order to make the cyclic operation more balance between day and night. The present heat exchange rate of air cooler is near 800 W (Table 6) and heat dissipation rate H_d of CWCT is near 750 W.

Table 6 – System overall performance (2010/8/03).

Daytime mode (room cooling by indoor air cooler)	
Time of operation	11:00~19:00
Operating time, hr	8
Initial ambient temperature, T_{ai} (°C)	35.5
Initial room temperature, T_{ri} (°C)	35.7
Final room temperature, T_{rf} (°C)	30.1
Initial water temperature in tank, T_{wiD} (°C)	24.0
Final water temperature in tank, T_{wfD} (°C)	31.1
Average power consumption, P_{day} (W)	56
Total energy consumption, E_{day} (kWh)	0.45
Average room cooling rate, Q_c (W)	811
Total energy removed from the room, E_{room} (kWh)	6.49
Initial temperature difference between room air and water, $T_{ri} - T_{wiD}$ (K)	5.50
COP of daytime mode, COP_{day} (=E_{room}/E_{day})	14.5
Nighttime mode (cold storage by CWCT)	
Time of operation	20:35~05:30
Operating time, hr	8.9
Initial ambient temperature, T_{aiN} (°C)	31.2
Average wet-bulb temperature of outdoor air, T_{WB} (°C)	23.9
Initial wet-bulb temperature of outdoor air, T_{WBi} (°C)	25.8
Initial water temperature in tank, T_{wiN} (°C)	31.1
Final water temperature in tank, T_{wfN} (°C)	24.9
Total cold energy storage, E_d (kWh)	6.68
Average heat dissipation rate, H_d (W)	750
Average power consumption, P_{nt} (W)	95
Total energy consumption in nighttime mode, E_{nt} (kWh)	1.07
Initial evaporation temperature glide in nighttime mode, D_G ($T_{wiN} - T_{WBi}$) (K)	5.3
COP of cold storage, COP_{nt} (=E_d/E_{nt})	7.9
Overall COP, $COP_o = E_d/(E_{day} + E_{nt})$	5.1

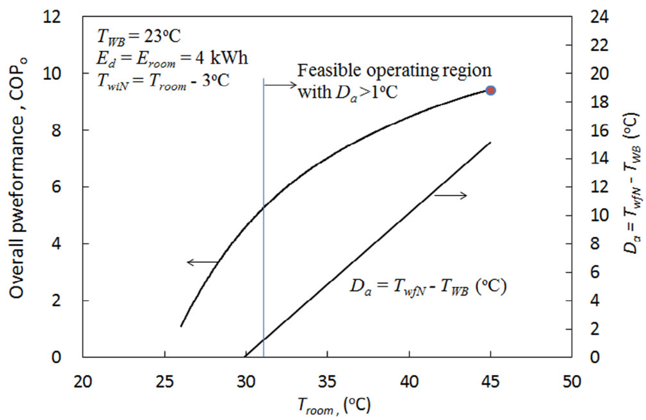

Fig. 14 – Overall performance analysis.

The evaporative cooling in cellulose-pad is more efficient if it is run at dry weather with lower wet-bulb temperature. This means that CWCT is recommended to be installed in dry region to effectively utilize the day/night natural energy. But, the water consumption is another issue. The water consumption of CWCT is about 1.5 kg h^{-1} per kW of cooling capacity from the test results.

6. Conclusion

The present study develops a cellulose-pad water cooling system utilizing the natural energy drawn from diurnal ambient air temperature difference. Water is used as the working medium which was cooled at night by cellulose-pad cooling tower (CWCT) and stored for cooling application at daytime.

A CWCT was designed and built to study the performance of CWCT experimentally. The test results show that the coefficient of performance of CWCT for heat dissipation of water at night, COP_{nt}, is between 3.8 and 11 and varies linearly with the evaporation temperature glide D_G. The measured D_G in Taipei area ranges from 1.9 to 6.5 K in summer and 2.1 to 4.2 K in winter. Extrapolation of experimental results in Fig. 12 shows that COP_{nt} will be higher if CWCT is used in dry areas with lower wet-bulb temperature, i.e. higher D_G.

The COP for room cooling at daytime run in use with an air cooler in a room, COP_{day}, is between 8.8 and 12.6. For day cycle operation (daytime room cooling and nighttime heat dissipation of water), the measured overall cooling COP_o is 5.1.

CWCT should be run in cyclic operation. That is, the heat removed from the room at daytime (room cooling) should be equal to the heat dissipated to the ambient at night by CWCT. The test results are used to analyze the cyclic performance of CWCT at various operating conditions. It is estimated that COP_o will be larger than 8.0 for room temperature higher than 40 °C and reaches 9.4 at room temperature 45 °C. The present water cooling system may be suitable for the cooling application with higher room temperature such as in data center. The performance of the present CWCT can be further improved through an intelligent control for optimal mass flowrate of water and air.

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