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An experimental study of a novel dew point evaporative cooling system

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ABSTRACT

A novel dew point evaporative cooling system for sensible cooling of the ventilation air for air conditioning application was constructed and experiments were carried out to investigate the outlet air conditions and the system effectiveness at different inlet air conditions (temperature, humidity and velocity) covering dry, temperate and humid climates. The results showed that wet bulb effectiveness ranged between 92 and 114% and the dew point effectiveness between 58 and 84%. A continuous operation of the system during a typical day of summer season in a hot and humid climate showed that wet bulb and dew point effectiveness were almost constant at about 102 and 76%, respectively. The experiment results were compared with some recent studies in literature.

1. Introduction

Evaporative cooling is one alternative to mechanical vapor compression for air conditioning applications. These systems usually require only a quarter of the electric power that mechanical vapor compression uses for air conditioning [1]. Therefore, such systems will help to reduce electricity requirements, and also contribute to reducing greenhouse gas emissions. Conventional evaporative cooling system can decrease the process air temperature theoretically approaching its wet bulb temperature, and has been used as a low energy consuming device for various cooling and air conditioning applications in industrial, agricultural and residential sectors [2–5] for providing low temperature medium fluid (i.e. air, water, etc.).

ASHRAE standard 55 for comfort conditions in summer recommends 25 °C and 50–60% relative humidity (around 10–12 g/kg humidity ratio) [6]. Since, air temperature and humidity are the two major parameters affecting thermal comfort significantly, and only sensible load can be handled by an evaporative cooling system, conventional evaporative cooling system is suitable for dry and temperate climate where the humidity is low [2,7]. However, in hot and humid climates, like Thailand, the ambient air temperature ranges from 21 to more than 35 °C, while the relative humidity varies from 45 to 95% [8].

Two common types of evaporative cooling system are direct and indirect systems. Direct evaporative cooling system has approximately 70–95% effectiveness in terms of temperature depression [9,10], and in the case of residential applications there are also concerns on hygienic issues, if maintenance is poor. Direct evaporative cooling system adds moisture to the cool air, which also makes conditions more uncomfortable for humans as (air) humidity increases. On the other hand, an indirect evaporative cooling system provides only sensible cooling to the process air without any moisture addition. Therefore, it is more attractive than direct evaporative system. However, the cooling effectiveness is generally low, around 40–60% [5,11]. This means that when the dry bulb temperature is more than 34 °C and wet bulb is more than 27 °C (in summer), the outlet air temperature leaving the system can be decreased and theoretically limited at 27 °C, i.e. its wet bulb temperature. Consequently, the supplied air temperature to a conditioning space is usually higher than 27 °C. This is a major drawback of conventional evaporative cooling.

To overcome this drawback, the enhancement of cooling effectiveness to provide much lower outlet air temperature is an interesting option for hot and humid climate conditions. For latent load reduction, the assistance of desiccant dehumidifier for moisture reduction can reduce the energy consumption significantly, especially when solar energy is used for regeneration [12,13].

A dew point evaporative cooling system can be used to supply the outlet air temperature below wet bulb temperature of ambient. In principle, the outlet air leaving the dry channel of the system has essentially lower dry bulb and wet bulb temperature than the intake air (ambient), and thus can enhance the cooling effectiveness compared to conventional systems [14,15]. In hot and humid environment, dew point evaporative cooling can be combined with a dehumidifier unit such as a cooling coil or desiccant system for

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Nomenclature

c_{pm}	specific heat of moist air (kJ/kg K)
DB	dry bulb temperature (°C)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
r	working air to intake air ratio (kg/kg)
t	temperature (°C)
WB	wet bulb temperature (°C)

Greek symbols

ε	effectiveness or efficiency
ω	humidity ratio (kg water vapor/kg dry air)

Subscripts

a	air
dew	dew point
in	inlet or intake
out	outlet
wb	wet bulb
wk	working air

provision of comfort conditions [16]. The combined system has essentially two steps: reduction of air temperature (by dew point evaporative cooling) and reduction of moisture (by desiccant method).

This paper presents the experimental results of a novel counter flow configuration of dew point evaporative cooling system. A key objective of the study was to reduce the temperature of the outlet air leaving the system without any moisture variation. The system was installed at Energy Technology Laboratory of the Asian Institute of Technology (AIT), Pathumthani, Thailand, and experimental studies were carried out for various inlet conditions representing typical tropical climate conditions. The experimental results have been evaluated to quantify the system performance in terms of temperature and humidity level of outlet air and the system effectiveness.

2. Theoretical psychometric analysis of dew point evaporative cooling

Direct evaporative cooling has a simple configuration (Fig. 1) where the air is brought into a direct contact with water and humidified as path 1–2. At adiabatic saturation conditions, the outlet air temperature can be reduced to its wet bulb temperature. Thus, the highest value of wet bulb effectiveness that can be obtained theoretically is 100% as shown in Fig. 1(a).

Indirect evaporative cooling system uses two streams of ambient air in the process (Fig. 1(b)), i.e. dry channel for intake ambient air (1), and wet channel of secondary air (2), separated by a thin-film polymer wall to prevent moisture penetration between them. Along the flow path 1–2, the intake air loses sensible heat to the wet side for water evaporation and the secondary air is cooled in direct contact with water as path 1–3. The outlet temperature can be decreased theoretically close to ambient wet bulb temperature.

Fig. 2(a) and (b) shows the process of dew point evaporative cooling system. The ambient air (1) is drawn into the dry channel and loses sensible heat to the wet channel as shown in psychometric path 1–2. Consequently, the outlet air (2) leaving the system is at a lower temperature than the ambient. It would be advantageous to divert some fraction of this air to act as the working air in the wet channel. Then the working air is humidified and absorbs the heat from the dry channel as shown in path 2–3. Eventually, it is rejected to atmosphere. Thus, the outlet air temperature is reduced to below the ambient wet bulb temperature without humidity variation. In an ideal cycle, the outlet temperature can be decreased theoretically toward the dew point temperature of intake air (ambient) [14,15].

The relationship between the outlet air temperature and other factors can be derived by taking enthalpy balance of the fluids entering and leaving the system in Fig. 2(a) as:

$$\dot{m}_{in}(h_1 - h_2) = \dot{m}_{wk}(h_3 - h_2) \tag{1}$$

In the dry channel, the enthalpy change is only due to the sensible cooling to the process air. Thus, the enthalpy difference between position 1 and 2 in psychometric chart of Fig. 2(b) can be written as:

$$h_1 - h_2 = c_{pm}(t_1 - t_2) \tag{2}$$

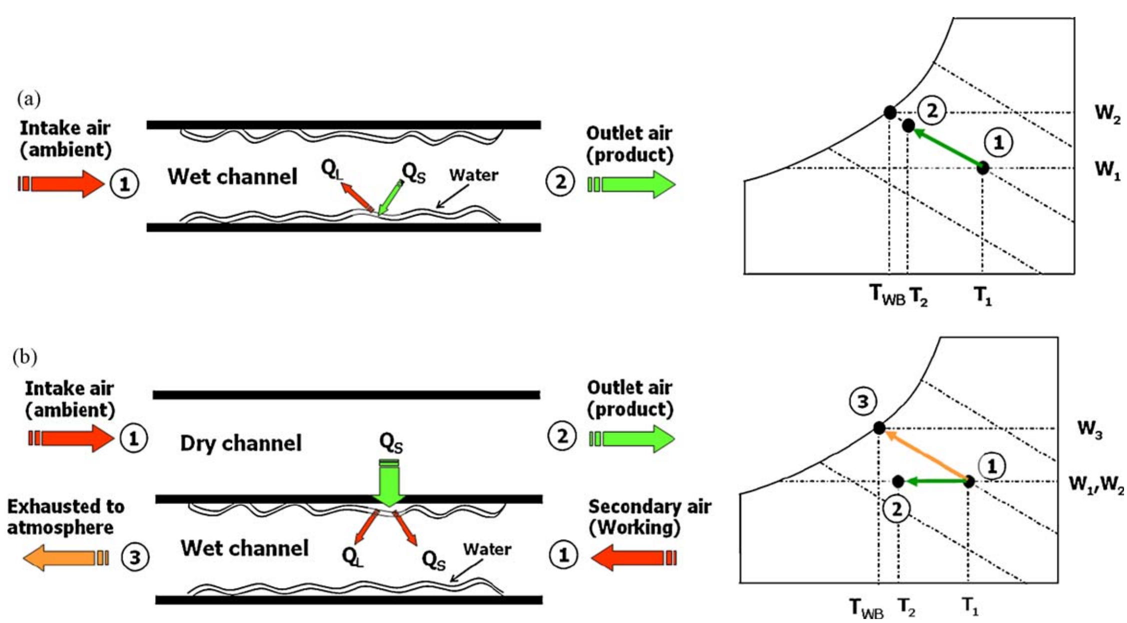

Fig. 1. (a) Direct evaporative cooling and (b) indirect evaporative cooling.

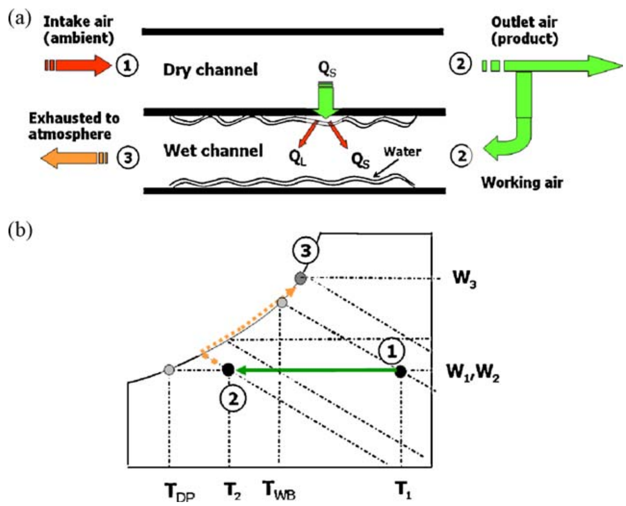

Fig. 2. (a) Dew point evaporative cooling and (b) psychometric process.

Applying the expression of enthalpy difference from Eq. (2) into Eq. (1), and rearranging gives the relationship of outlet air temperature as:

$$t_2 = t_1 - \frac{r}{c_{pm}}(h_3 - h_2) \tag{3}$$

where r is the (mass) ratio of working air to intake air and c_{pm} is the specific heat of moist air.

This shows the relationship of the outlet air temperature leaving the dew point evaporative cooling system which depends on inlet air temperature, the (mass) ratio of working air to intake air, and the enthalpy difference of the working air in wet channel. The second term on the right represents the temperature drop between inlet and outlet air of dry passage. The increase of working to intake air ratio can lower the outlet air temperature. However, this may lead to a reduction of the useful air flow rate supplied to the cooling space. From psychometric path 2–3, the enthalpy difference between entering and outgoing air in the wet channel is due to the sensible and latent heat change of working air which depends on heat and mass transfer mechanism taking place, since the channel configuration (i.e. gap size, length), fluid properties and velocity dominate heat and mass transfer coefficients of flowing fluid [17]. Therefore, the optimum operating condition of this system for air conditioning application should take into account all these aspects, namely, the outlet air temperature required, outlet air flow rate, type of cooling load, inlet air conditions, channel configuration, energy used for fan and water pump, etc.

Previous work on dew point evaporative cooling used cross flow configuration of dew point evaporative cooling system which indicated wet bulb effectiveness of 110–122% and dew point effectiveness of 55–85% [14]. A numerical study for a counter flow polygonal configuration of the system for typical UK climate indicated that the air velocity in dry and wet channel, the ratio of working air to intake air, and the dimensions of channel are the major parameters influencing the system performance [15].

3. Performance indication of a dew point evaporative cooling system

The inlet and outlet conditions of the process air can be used to evaluate the performance of dew point evaporative cooling system using two indices, namely, wet bulb and dew point effectiveness. The wet bulb effectiveness is the ratio of the difference between intake and outlet air temperature to the difference between intake air temperature and its wet bulb temperature. This can be expressed as:

$$\varepsilon_{wb} = \frac{t_{a,in} - t_{a,out}}{t_{a,in} - t_{wb,in}} \tag{4}$$

Similarly, the dew point effectiveness is defined as the ratio of the difference between intake and outlet air temperature to the difference between intake air temperature and its dew point temperature. The dew point effectiveness is defined as [14,15]:

$$\varepsilon_{dew} = \frac{t_{a,in} - t_{a,out}}{t_{a,in} - t_{dew,in}} \tag{5}$$

4. Description of the novel dew point evaporative cooling system

4.1. Design considerations

A novel vertical configuration of dew point evaporative cooling system has been proposed by using countercurrent arrangement for all flowing fluids. The design considerations are summarized as follows:

- To obtain the high heat and mass transfer for all streams (i.e. air, water) in dry and wet channel of the exchanger by using counter flow arrangements for intake air–working air, working air–water and using a thin-film wall plate.
- To avoid corrosion by using water in the cooling process.
- To saturate evenly all wet surface in the wet channel continuously by using a reservoir at the top and a vertical configuration.
- To extract a certain fraction of the outlet air to act as the working air in the cooling process.

The components of the system comprise several dry and wet channels separated by polymer sheets to avoid penetration of water and the water feeding system. The water feeding should ensure to saturate all surfaces only in the wet channels by using a vertical configuration as shown in Fig. 3(a). By this method, the water could be easily supplied from the reservoir to the outlet portion of wet channels at the top side and travels vertically down to saturate the wet surfaces continuously. The supplied water can be regulated precisely using a control valve. One fan is used to transport the intake and working air for the cooling system.

Non-metallic materials, such as fiber, cellulose, ceramic, zeolite and carbon, can be used for the construction of wall surface of wet channels [18,19]. The temperature difference on dry and wet side is very small if the wall thickness is less than 0.5 mm [19,20]. For this study, a thin-film cotton sheet coated evenly by polyurethane material (PU) (total thickness 0.5 mm) was selected as the material for the wall separating dry and wet channel. Wall A and Wall B with 80 mm of width and 1200 mm of length was used for dry and wet passages having different seals on their edges as shown in Fig. 3(b). These walls are stacked with 5 mm spacing to form the rectangular configuration of a heat and mass exchanger having 4 dry channels and 5 wet channels. The arrangement ensures that two walls of each dry channel are enclosed with wet channels so that the intake air is sensibly cooled between 2 parallel surfaces along the flow path.

4.2. Experimental facility and setup

The experimental facility for dew point evaporative cooling system consists of (a) the air preconditioning unit, (b) dew point

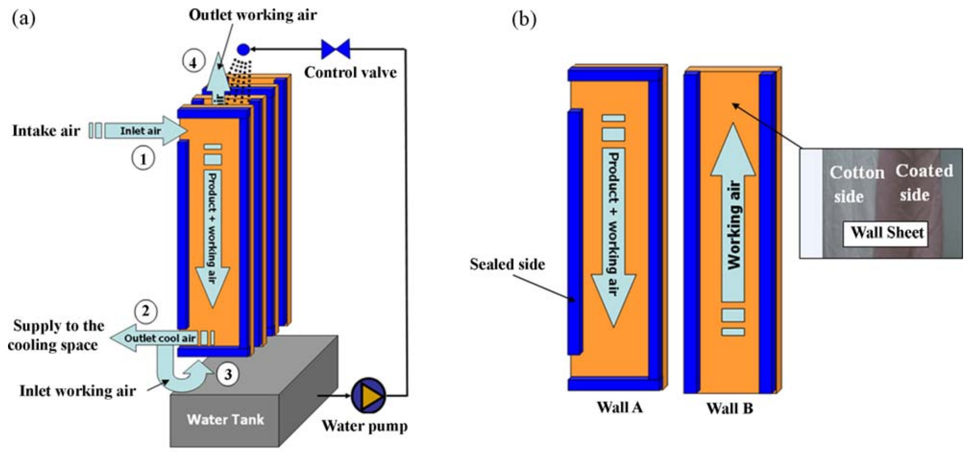

Fig. 3. (a) The dew point evaporative cooling system and (b) two types of the wall.

evaporative cooling system, and (c) measurement devices as shown in Fig. 4. The desired inlet air conditions are provided by an air preconditioning unit (Hilton air conditioning laboratory apparatus made from P.A. Hilton Ltd., England). The unit is equipped with a 1.5 kW pre-heater and re-heater and a 5 kW water boiler system for humidification. A cooling coil of a vapor compression cycle controls air temperature and humidity ratio of preconditioning air. The intake fan of dew point evaporative cooling unit connected to the delivery portion of the pre air conditioning unit regulates precisely the desirable amount of air flow by a variable speed drive controller. The exchanger body is insulated to avoid heat losses. The actual experimental setup is shown in Fig. 5.

4.3. Measurement setup

To evaluate the performance of the dew point evaporative cooling system, the following data were measured:

- Temperature (DB, WB) and velocity of the intake air entering and leaving the dry channel.
- Temperature (DB, WB) and velocity of the working air entering and leaving the wet channel.

The measurement locations are shown in Fig. 4. Thermocouples were pre-calibrated and verified before the tests in an isothermal bath using a standard precision thermometer. A hot-wire anemometer measures the air velocity. The specifications of measuring devices used during the experiments are listed in Table 1.

5. Experimental studies description

Two types of experiments were conducted to investigate the performance of dew point evaporative cooling system:

- *Type* 1: Static studies aimed to know the performance of the proposed system at specific (constant) inlet conditions:

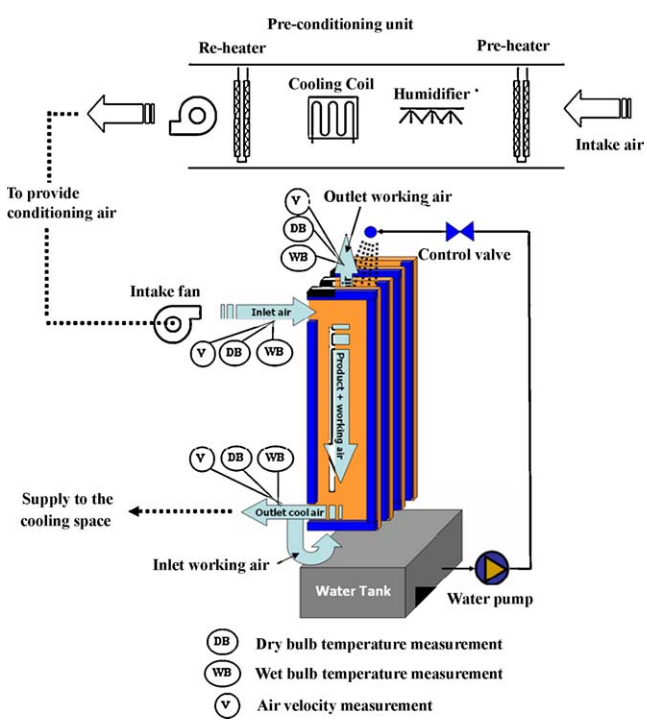

Fig. 4. Schematic drawing of experimental set up.

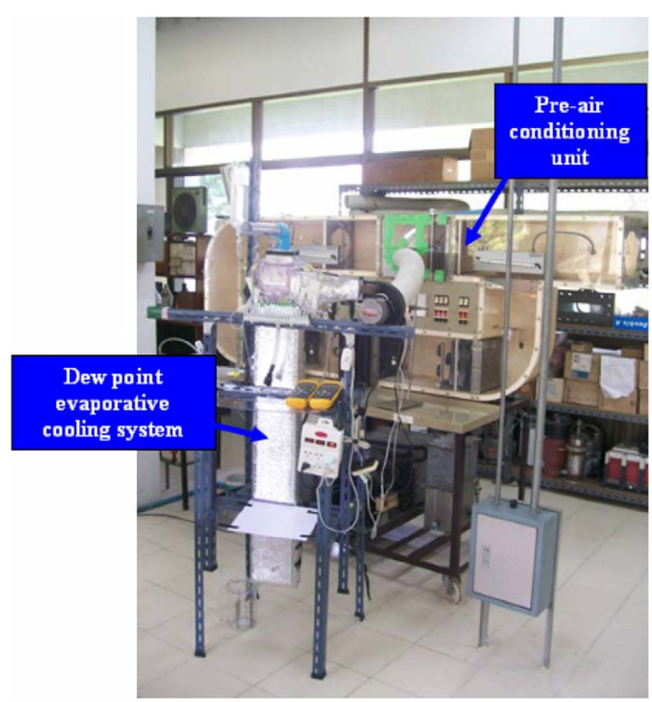

Fig. 5. Experimental setup of dew point evaporative cooling system.

Table 1
Specification of measuring devices.

Parameters	Instruments	Accuracy	Range
Temperature	Thermocouple Type K	±0.2 °C	0–100 °C
Air flow	Hot-wire anemometer	±5%	0.2–20.0 m/s

(a) Influence of inlet air temperature and humidity on outlet air temperature and effectiveness.
(b) Influence of air velocity on outlet air temperature and effectiveness.

- *Type* 2: Dynamic studies aimed to know the performance when the inlet conditions continuously vary during its operating period.

A description of the experimental conditions studied is summarized in Table 2. The specifications and parameters used for all experimental are summarized in Table 3. For each experiment, the data were recorded once steady state condition was reached, and each experiment was repeated to ensure consistency (and repeatability) of the measured data.

6. Results and discussion

The experimental observations, results obtained and the interpretations of these results are discussed in this section. Some comparative analyses with data/results from literature are also presented.

6.1. Static studies (type 1a: performance under specific inlet air conditions)

Two performance indices, namely, wet bulb and dew point effectiveness were used to evaluate the system performance. The higher values of these indices indicate how close the outlet air temperature approaches the theoretical wet bulb and dew point temperature of the intake air. The humidity ratio between inlet and outlet air is not different due to the separation between dry and wet passages by wall (thin-film polymer sheet).

Fig. 6 shows the outlet air temperatures obtained from the experimental study for different inlet air conditions. For inlet air temperatures ranging from 25 to 45 °C and inlet humidity ratio varying from 6.9 to 26.4 g/kg, lower outlet air temperature was obtained at lower inlet air temperature and lower humidity levels. In addition, the outlet air temperature was found to be almost linearly dependent on the inlet air temperature at constant humidity ratio. The slope of the outlet temperature at constant humidity ratio is positively small, ranging between 0.15 and 0.17. This implies that if the inlet air temperature is increased by 10 °C, the outlet air temperature increases by about 1.5–1.7 °C. On the other hand, the outlet air temperature is considerably influenced by inlet humidity ratio at constant inlet temperature. This means that if the inlet humidity ratio is decreased by 10 g/kg, the outlet air temperature decreases by about 6.8 °C for the same inlet temperature. The results indicate that the system could supply outlet air temperatures below 25 °C when inlet humidity condition is below 11.2 g/kg. This shows the potential for air conditioning application in hot and dry climate conditions. The performance

Table 3
Basic specifications of the parameters used in the experiment.

Parameters	Specification/value
Wall material	Cotton sheet coated with polyurethane
Wall thickness	0.5 mm
Channel length	1200 mm
Channel width	80 mm
Channel gap	5 mm
Working air to intake air ratio	0.33 kg/kg
Water supplied	60 g/h

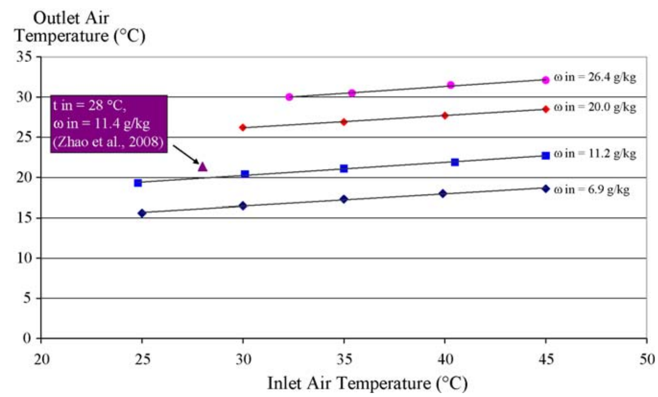

Fig. 6. Outlet air temperature for different inlet air conditions.

Table 2
Details of experimental conditions.

Type of experiment	Experimental details
1. Static studies	
(1a) Influence of inlet air temperature and humidity	
Inlet air temperature	25, 30, 35, 40 and 45 °C (5 temperatures)
Inlet air humidity	7, 11, 20 and 26 g/kg (4 humidity conditions)
Intake air velocity	2.4 m/s
Measured data	Temperature, humidity and velocity at inlet and outlet of system (at steady state)
Calculated results	Wet bulb and dew point effectiveness
(1b) Influence of inlet air velocities	
Inlet air temperature	34 °C
Inlet air humidity	19.0 and 11.2 g/kg (2 humidity levels)
Intake air velocity	1.5, 2.4, 3.3, 4.2, 5.1 and 6.0 m/s (6 levels)
Measured data	Temperature, humidity and velocity at inlet and outlet of system (at steady state)
Calculated results	Wet bulb and dew point effectiveness
2. Dynamic studies	
(2) Experiment under dynamic ambient conditions of inlet air	
Inlet air temperature	Ambient condition (8 a.m. to 8 p.m.)—varies continuously
Inlet air humidity	Ambient condition (8 a.m. to 8 p.m.)—varies continuously
Intake air velocity	2.4 m/s
Measured data	Temperature, humidity and velocity at inlet and outlet of system (at steady state)
Calculated results	Wet bulb and dew point effectiveness

characteristic of the developed system can be correlated by the following equation:

$$t_{outlet}\ (^{\circ}\mathrm{C}) = 7.65 + 0.152 t_{in,ambient}(^{\circ}\mathrm{C}) + 681\omega_{in,ambient}(\mathrm{kg/kg}) \quad (6)$$

Eq. (6) is specific for this experimental setup and valid for the experimental conditions (inlet air temperature between 25 and 45 °C, humidity ratio about 6.9–26.4 g/kg dry air, intake air velocity of 2.4 m/s and working to intake air ratio of 0.33). The equation predicts outlet air temperatures within about ±5%.

Fig. 6 also compares the outlet air temperature predicted by simulation for typical UK summer climate by Zhao et al. [15] when inlet air temperature and humidity ratio are equal to 28 °C and 11.4 g/kg respectively with the present experimental study. The experimental results obtained in the present study gives slightly lower values as compared to the simulation results by Zhao et al. [15]. The difference is probably due to the difference in system configuration and operating parameters.

Fig. 7 displays the performance of dew point evaporative cooling device in terms of wet bulb effectiveness. Knowing the outlet and inlet condition, the relationship expressed by Eq. (4) can be used to calculate the wet bulb effectiveness. It could be observed that the experimental setup can lower the outlet air temperature below its inlet wet bulb condition (wet bulb effectiveness could be greater than 100%). For inlet temperatures ranging from 30 to 45 °C, the wet bulb effectiveness varied between 100 and 115%. The experimental observations show that higher inlet temperature leads to greater wet bulb effectiveness owing to the larger temperature depression.

When the inlet air humidity ratio is low, higher wet bulb effectiveness was obtained. This can be explained by heat and mass transfer mechanism between dry and wet side of this exchanger. When the humidity ratio of inlet air is small, air has more capacity to absorb the moisture from water evaporation owing to the larger driving force of vapor pressure gradient. As a result, more latent heat is needed to evaporate the water on the wet surface. Thus, more sensible heat of process air can be transferred from dry side to wet side. Thus, lower temperature of outlet air can be obtained from the exchanger.

Fig. 8 shows the dew point effectiveness for different inlet air temperatures and humidity. Dew point effectiveness can be obtained by using the relationship given by Eq. (5). When inlet temperature ranged from 30 to 45 °C, the dew point effectiveness varied between 63 and 85%. Higher inlet air temperature gives higher dew point effectiveness. Another interesting observation from the experimental results is that higher dew point effectiveness could be obtained by increasing the inlet humidity ratio at constant inlet temperature (Fig. 8). This leads to an increased outlet air temperature shown in Fig. 6. It can be graphically explained by psychometric process as shown in Fig. 9. For example, consider two inlet air conditions shown in the chart (at 30 °C). They have different levels of humidity ratios (7 and 20 g/kg, respectively). In psychometric chart, as the slope of saturation curve is not constant, the differences between the inlet and dew point temperature, or the dew point depression, is greater at low humidity ratio condition. Therefore, the dew point effectiveness derived by Eq. (5) may be decreased in a lower humidity ratio condition. This implies that the level of dew point cooling effectiveness indicates how close the outlet air temperature approaches the theoretical dew point temperature for a specific inlet condition. To summarize the outcome of the influence of inlet conditions, lower outlet air temperature could be obtained when the inlet temperature and humidity are maintained at the lowest level.

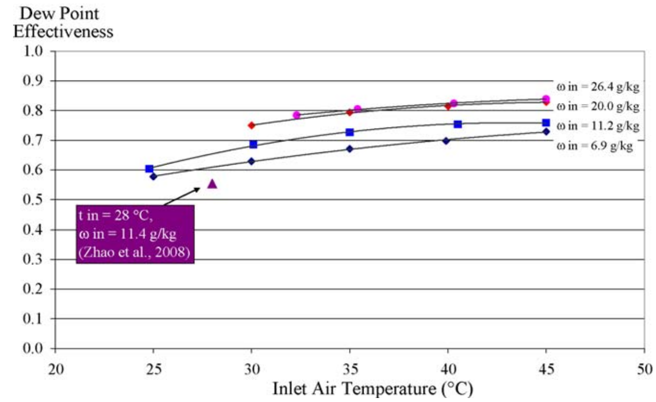

Fig. 8. Dew point effectiveness for different inlet air conditions.

The wet bulb and dew point effectiveness values obtained from this experimental study are slightly greater than the results predicted by Zhao et al. [15] for the same inlet air condition.

6.2. Static studies (type 1b: performance under influence of inlet air velocities)

Fig. 10 shows the outlet air conditions when the intake air velocity was varied. Lower humidity ratio at inlet condition gives larger decrease in the outlet air temperature. When the velocity is increased, the outlet temperature increases. However, for velocities below 2.5 m/s, the outlet temperatures for both inlet conditions are below their wet bulb temperatures which equals to 26.6 and 21.6 °C, respectively. For very low velocity, the trends of both outlet temperatures approach the inlet dew point temperatures at 24 and 15.7 °C, respectively.

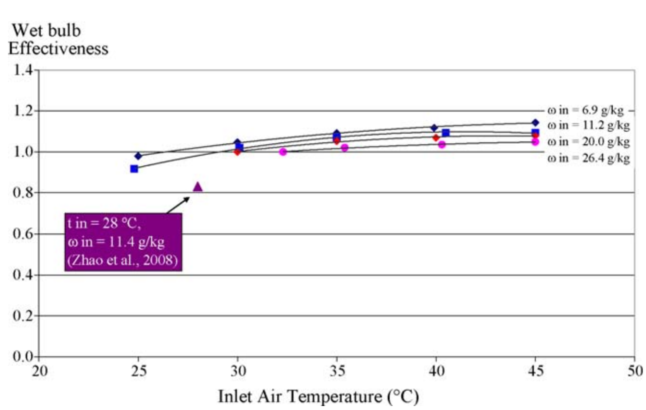

Fig. 7. Wet bulb effectiveness for different inlet air conditions.

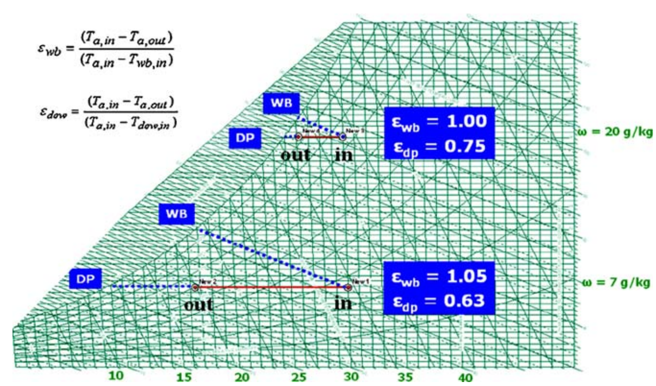

Fig. 9. Psychometric process of two different inlet air conditions (adapted from ASHRAE psychometric chart).

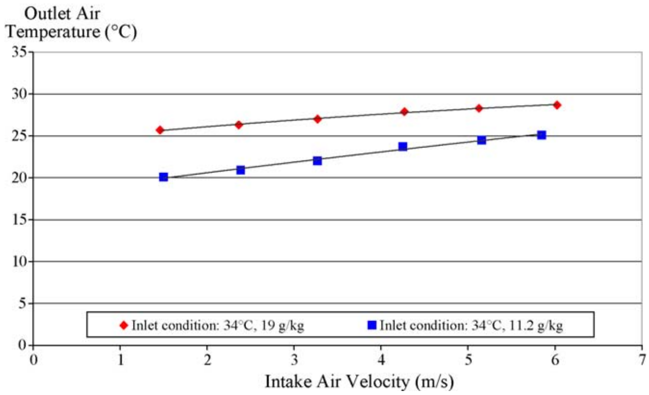

Fig. 10. Outlet air temperature for different intake air velocities.

Fig. 11 shows the wet bulb and dew point effectiveness for different intake air velocities. When the velocity is increased, both the effectiveness values decrease. For velocities below 2.5 m/s, the wet bulb effectiveness values are greater than 100% and dew point effectiveness values are higher than 70%.

6.3. Dynamic studies (type 2: performance under continuously varying conditions of inlet air)

The purpose of these experiments was to observe the performance of dew point evaporative cooling device under real ambient conditions on a typical day in summer season of Thailand. The difference between type 1 and type 2 was the provision of the supply condition of inlet air. In type 1, the inlet condition of process air was controlled and supplied by the air preconditioning unit, where as in type 2, the air preconditioning unit was not used (ambient condition was maintained). Furthermore, in case 2, the experiment was conducted continuously (8 a.m. to 8 p.m.) to observe the dynamic performance of the system.

Fig. 12 shows the hourly experimental data of dew point evaporative cooling device under ambient conditions of inlet air. The inlet ambient temperature varies slightly from 30.5 °C in the morning, reached its maximum value of 34 °C in the afternoon, and then continuously decreased to 33.2 °C in the nighttime. The inlet air humidity ratios slightly decreased from 20.5 g/kg in the morning to the minimum level of 18 g/kg in the nighttime. The outlet air temperature, on the other hand, did not vary much, ranging from 25.7 to 27 °C, and very close to the ambient wet bulb temperature. However, it slightly decreased in the evening when the inlet ambient air humidity was lower (at that time).

The performance of this system in terms of wet bulb and dew point effectiveness is shown in Fig. 13. Both effectiveness values

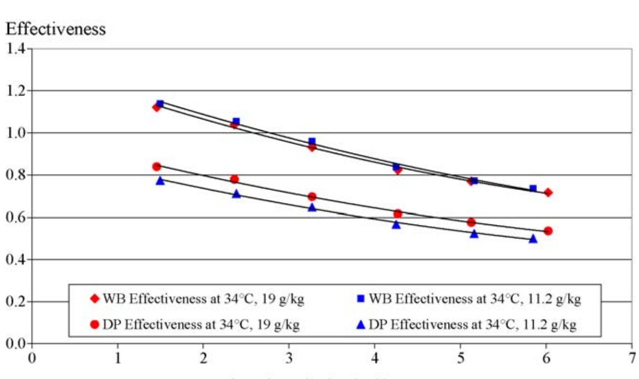

Fig. 11. Wet bulb effectiveness for different intake air velocities.

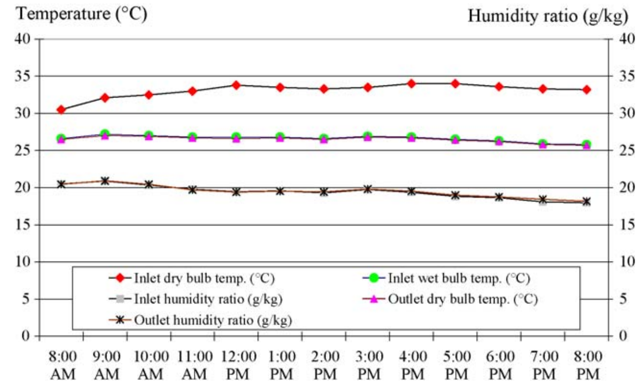

Fig. 12. Performance of experimental setup under continuously varying conditions.

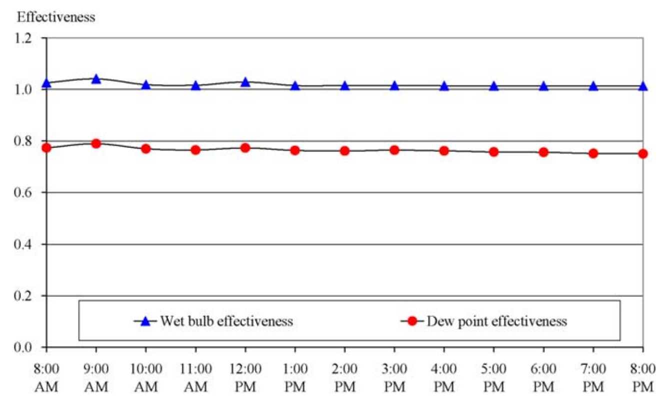

Fig. 13. Experimental results of wet bulb and dew point effectiveness.

did not vary much during the continuous operation. The wet bulb effectiveness ranged from 101 to 104%, whereas the dew point effectiveness varied between 75 and 79%.

7. Comparison between present study and previous studies

Table 4 compares the performance of dew point evaporative cooling systems of the present and previous studies. The present study investigated the outlet air condition and system effectiveness for various inlet air conditions. Compared to Zhao's simulated results, the present experimental study's results provide similar values of outlet performance indicator. One notable difference is that Zhao's study used the intake air velocity of 1 m/s, channel gap of 10 mm and working to intake air ratio of 0.5 kg/kg for the preset operating condition, whereas in the present study these parameters were 2.4 m/s, 5 mm and 0.33 kg/kg, respectively.

Compared to the conventional system, the dew point evaporative cooling can decrease the air temperature without moisture change. The direct evaporative cooling experimented by Wu et al. [9] showed that the humidity ratio of process air increased during the cooling process. Indirect evaporative cooling studied by Alonso et al. [21] also showed a similar feature as dew point evaporative cooling, i.e. no moisture variation for the outlet air. However, lower effectiveness values were observed as compared to the results of present study. The higher effectiveness greater than 100% was observed by the experimental results of two-stage indirect/direct evaporative cooling system studied by Heidarinejad et al. [7] for various conditions of inlet air. However, increasing humidity was observed.

Other studies for dew point evaporative cooling employ different configurations. Idalex uses a cross flow arrangement for heat transfer between dry and wet channels. Zhao's method uses counter

Table 4
Comparison of operating parameters and performances of the present and previous studies.

Observed information	Present study	Zhao et al. [15]	Wu et al. [9]	Alonso et al. [21]	Heidarinejad et al. [7]
System descriptions					
Method	Experiment	Simulation	Simulation and experiment	Experiment	Experiment
Type of cooling device	Dew point evaporative cooling	Dew point evaporative cooling	Direct evaporative cooling	Indirect evaporative cooling	Two stage indirect/direct evaporative cooling
Flow arrangement	Counter flow	Counter flow	Cross flow	Cross flow	Cross flow
Parameters					
Inlet air temperature	25.0–45.0 °C	28 °C	27.2–37.1 °C	25–35 °C	33.0–46.6 °C
Inlet air humidity	7.0–26.0 g/kg	11.4 g/kg	7.3–17.6 g/kg	10.6–12.4 g/kg	8.2–17.1 g/kg
Intake air velocity	2.4 m/s	1 m/s	2 m/s	0.022 m^3/s	0.472 m^3/s
Channel length	1.0 m	1.0 m	0.138 m	0.3 m	0.4 m
Channel gap	5 mm	10 mm	N/A	3 mm	7 mm
Outlet air temperature	15.6–32.1 °C	21.4 °C	22.4–30.4 °C	20.8–24.8 °C	16.1–24.8 °C
Outlet air humidity	7.0–26.0 g/kg (↔)	11.4 g/kg (↔)	10.9–20.3 g/kg (↑)	10.6–12.4 g/kg (↔)	12.0–21.0 g/kg (↑)
Wet bulb effectiveness	0.92–1.14	0.83	0.70–1.00	0.77–0.93	1.08–1.11
Dew point effectiveness	0.58–0.84	0.55	N/A	N/A	N/A

flow arrangement for heat transfer between dry and wet channel in a polygonal structure and the water travels vertically as cross flow humidification. In the present study, a vertical counter flow configuration of dew point evaporative cooling system has been considered. By using countercurrent humidification in wet channel, the rate of heat removed from the water on the wet surface could be high and the lowest temperature water collected at the bottom side (which can be recirculated by a pump). This design is more compact and is easy to construct and operate.

8. Conclusion

A novel configuration for dew point evaporative cooling has been experimentally shown to provide good performance at various operating conditions, covering dry, moderate and humid climate. The wet bulb effectiveness ranged between 92 and 114%, whereas the dew point effectiveness varied between 58 and 84% for various inlet conditions. At inlet air temperature more than 30 °C, the velocity of intake air should be kept below 2.5 m/s to obtain wet bulb effectiveness greater than 100%. The dry and wet bulb effectiveness did not vary much during continuous operation under real ambient condition. This indicates the potential of this system for air conditioning applications.

For hot and dry climate application, this system alone can provide comfort condition for a living at inlet temperature and humidity ratio less than 45 °C and 11.2 g/kg, respectively. To make this system commercially viable to humid climates, the assistance of solar regenerated desiccant dehumidifier should be studied, which could lead to alternate solutions for mechanical vapor compression systems.

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