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Research Article

SIMULATION OF A HYBRID DEW-POINT EVAPORATIVE COOLING SYSTEM

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ABSTRACT:
This article aims to investigate the factors influencing the performance of a hybrid dew-point evaporative cooling system (HDPEC) in a 7-Eleven store in Thailand. Those factors are the channel length, channel gap, air flow rate, and operational strategy of the system. The simulations were conducted using the TRNSYS program. The hybrid system consists of a dew-point evaporative cooler and a vapor-compression air-conditioning system (VAC). Three operational strategies were examined. The first is that both evaporative cooler and VAC operate 24 hours simultaneously. The second is that the cooler and VAC are on when the room temperature is higher than 25˚C and they are off when the room temperature is less than 23˚C. The third is that only the VAC operates and it is on when the room temperature is higher than 25˚C and is off when the room temperature is less than 23˚C. It was revealed that the second strategy consumes the least electricity compared to the others. In addition, the performances of HDPEC in the provinces of Bangkok, Nakhon Ratchasima, Sakon Nakhon, and Chiang Mai were compared. It was found that the annual power consumption of a 7-Eleven store with HDPEC in Bangkok is 39.62% less than that of the store with VAC alone. This is the largest reduction of power consumption among the provinces studied. The wet-bulb effectiveness and the payback period of the system in Bangkok are 1.37 and 3.5 months, respectively.

Keywords: *HDPEC, evaporative cooling, TRNSYS, 7-Eleven, air-conditioning, hybrid*

1. INTRODUCTION

The rate of electricity consumption has been continuously increasing worldwide. It was found that 50.75% of electricity demand in the residential sector was consumed by air conditioning systems [1]. Based on 2019 data, Thailand had approximately 10,268 branches [2] of 7-Eleven stores and they use vapor-compression air-conditioning systems (VAC). As a result, if energy consumption of the air-conditioning systems of 7-Eleven stores could be reduced, the electricity consumption of Thailand could be reduced.

The VAC is a common choice to serve the space cooling and/or space heating needs for residential and commercial buildings. However, synthetic refrigerants used in VACs have adverse effects on the global environment. Although researchers have proposed several alternative refrigerants, which have lower adverse effects, they usually come with a trade-off between the lower effects and toxicity, flammability, lower efficiency, and increasing costs when using these alternatives. For that reason, alternative solutions to VAC have been widely developed and investigated [3].

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Direct evaporative cooler (DEC), an environmentally friendly alternative to VAC, reduces the air temperature by flowing air through water sprayed on to cooling pads made of cellulose material [4]. The lowest temperature of cooled air from DEC is theoretically equal to the wet-bulb temperature of the inlet air. However, all wet-bulb effectiveness (ε_{wb}) of DECs in Table 1 are less than 1.0. This implies that the typical DEC cannot reduce the air temperature to reach its wet-bulb temperature.

Table 1: Comparison between the present study and of relevant research.

Reference	Type	Method	Flow type	$T_{db,inlet}$ °C	RH (%)	ε_{wb}
Bishoyi and Sudhakar [5]	DEC	experiment	Cross-flow	32.0	83.9	0.8
Sohani and Sayyaadi [6]	DEC	simulation	Cross-flow	28.9-40.7	18.3-44.2	0.61-0.95
Kim et al. [7]	IEC	experiment	Cross-flow	27.5±0.5	35.7	0.4-0.81
Riangvilaikul and Kumar [8]	DPEC	experiment	Counter-flow	45.0	42.8	1.1
Duan et al. [9]	DPEC	experiment	counter-flow	40.0	23.6	0.96-1.07
Zhan et al. [10]	DPEC	simulation	Counter & Cross-flow	28.0	47.8	1.14-1.36
Xu et al. [11]	DPEC	simulation	Counter-flow	28.0	83.1	1.3
Delfani et al. [13]	HDPEC	experiment	Cross-flow	31.4	13.9	N/A
Cui et al. [14]	HDPEC	simulation	Counter-flow	30.0	70.0	0.7
Present study	HDPEC	simulation	Cross-flow	23.72	64.44	1.37

Bishoyi and Sudhakar [5] experimentally investigated the cooling performance of DECs made of Aspen and Honeycomb. The results show that the cooling capacity and wet-bulb effectiveness of Honeycomb cooling pads are 2.34 kW and 0.8, respectively, as shown in Table 1, and they are higher than those of Aspen cooling pads.

Sohani and Sayyaadi [6] also studied the employment of DEC as the cooling system for a residential building with a floor area of 97.1 m^2. Simulations for representative cities with various climate types were conducted. Multi-objective optimization was used to minimize the life-cycle cost (LCC) and the annual water consumption (AWC) as well as to maximize the annual average coefficient of performance (AACOP) of the cooling system. It was found that the LCC and AWC decrease by 52.8% and 74.3%, respectively, and AACOP increases by 146.9% compared with the reference system in the design optimization process scenario. Meanwhile, the LCC and AWC decrease by 20.9% and 64.6%, respectively, and AACOP increases by 70.1% compared with the reference system in the retrofit optimization process scenario. However, as the outlet air from DEC is exposed directly to water, the outlet air humidity is relatively high and this could lead to the growth of microorganisms. Furthermore, the wet-bulb effectiveness is 0.61-0.95 as shown in Table 1, depending on the local weather conditions and the scenario selected.

Kim et al. [7] investigated the cooling performance of two different types of cross-flow indirect evaporative cooler (IEC), which indirectly cool the supply air without increasing the air humidity. Each IEC was tested in two operation modes. The first mode used the return air from the testing room as the working air, while the second model used a portion of the supply air from the cooler as its working air. It was revealed that both ICEs perform better in the first mode. The wet-bulb effectiveness for the first mode is approximately 0.7 while it is approximately 0.5 for the second mode.

In order to enhance the cooling performance of the IEC, a dew-point evaporative cooler (DPEC) has been proposed as an alternative evaporative cooling technology [7]. Theoretically, the outlet air temperature from DPEC can approach the dew-point temperature of the inlet air. In other words, ε_{wb} of DPEC can be greater than 1.0 as indicated in Table 1. Generally, DPEC consists of a series of parallel plates. The space between two adjacent plates forms the channel through which the air flows. There are two types of channel: the wet channel and the dry channel. As shown in Fig. 1, there are also two streams of air. The ambient air is induced into the dry channel and the working air is induced into the wet channel. The water evaporation in the wet channel absorbed heat from the air in both channels. As a result, the air in the dry channel is cooled and can be used as the supply air for a conditioned space. On the other hand, the humid air in the wet channel passes as exhaust to the ambient surroundings.

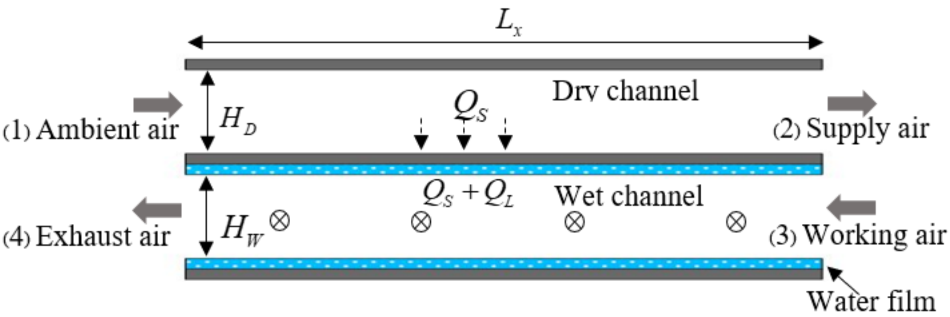

Fig. 1. Schematic diagram of DPEC.

Riangvilaikul and Kumar [8] studied a counter-flow DPEC. The supply air was divided into two streams. One stream was used as the working air and the other stream was supplied to the testing room. It was found that the wet-bulb effectiveness was 1.1 when the ambient air was 45˚C and 42.8% RH. The temperature difference between the ambient and supply air was 12.5˚C.

In addition, Duan et al. [9] used a finite element model to design and develop a regenerative evaporative cooler (REC), which is a closed-loop IEC which can theoretically also provide the outlet air temperature below the dew-point temperature of the inlet air. It was found that the wet-bulb effectiveness obtained was 0.96-1.07. It was also found that the cooling capacity per unit volume provided by the proposed cooler is increased by 62-108% of the other RECs in related studies.

Furthermore, Zhan et al. [10] studied a Maisotesenko-cycle (M-cycle) counter-flow and cross-flow DPEC. Both systems had the same physical size and were tested with the same operating conditions. It was reported that the cooling capacity of the counter-flow system was 20% higher than that of the cross-flow system. In addition, the wet-bulb and dew-point effectiveness of the counter-flow system was 15-23% higher than that of the cross-flow system. However, the COP of the counter-flow system was 10% lower than that of the cross-flow system.

Xu et al. [11] proposed a heat and mass transfer with a novel shape of the dry and wet channels. Then several simulations were conducted and the results compared with a conventional flat-plate heat and mass transfer with the triangular supporting guides between the plants. It was found that the cooling capacity of the proposed system is 32.9-37% higher than that of a conventional system that has the same size and the same operating conditions. It was also found that the dew-point and wet-bulb effectiveness of the proposed system were 29.7-33.3% higher than those of the conventional system. Additionally, the pressure drop of the proposed system is 55.8-56.2% lower than that of the conventional system.

Chauhan et al. [12] conducted an experiment by using DEC with VAC and found that this system was able to lower energy consumption with the highest level at 23.8% at an ambient air temperature of 43˚C and relative humidity of 18.1%. In addition, they also found that this system had a payback period of 8.1 years.

Delfani et al. [13] conducted an experiment by using DPEC with VAC that was called in this article “hybrid dew-point evaporative cooling” (HDPEC). They found that HDPEC was able to lower electric power consumption by 55% compared to the electric power consumption of stand-alone VAC. Cui et al. [14] conducted an experiment to compare conventional and regenerative DPEC systems. Based on the model, it was found that regenerative DPEC had wet-bulb effectiveness higher than that of conventional DPEC with the approximate value of 0.7. In this article, wet-bulb effectiveness was less than 1 because there were different definitions for the calculation.

Coca-Ortegon et al. [15] used TRNSYS software in modeling hybrid liquid desiccant systems (HLDS) using a dehumidifier to lower the humidity of the air and mutually operating with vapor compression chillers that were modeled under the climate of Kuala Lumpur which is a city with high temperature and humidity. In addition, they also found that the best seasonal coefficient of performance in the air conditioning process without renewable energy was 2.19, while the solar collector surface was 1,323 m^2. Ahmed et al. [16] presented the modeling of a new membrane liquid desiccant air-conditioning (LDAC) system by using TRNSYS software to analyze the characteristics of a membrane LDAC system. They found that the system coefficient of performance (COP) was 0.68 when operating on design conditions. It could be seen that TRNSYS software has potential in modeling air-conditioning systems and analyzing the energy usage.

This article aims to numerically investigate the factors influencing the operation of a HDPEC in order to find suitable dimensions and operation strategies for 7-Eleven stores. In addition, the operations of HDPEC were compared between the provinces of Bangkok, Nakhon Ratchasima, Sakon Nakhon, and Chiang Mai to evaluate the feasibility of using the system in these provinces. Also, an economic assessment was also presented in this article.

2. DESCRIPTION OF HDPEC

The HDPEC in this study consists of a DPEC and a VAC and it provides the conditioned air for a 7-Eleven store. Figure 2 shows the schematic diagram of the system. The store temperature was regulated in a range of 23-25˚C. The return air (1) is induced to VAC for cooling and supplied to the store as the supply air (2). Also, the return air (1) is induced to the dry and wet channels (4) of DPEC. In the DPEC, the air in the dry channels is cooled by transferring its heat to the wet channels, where the water is evaporated into the air. The output air from the dry channels is supplied to the store as supply air (3), whereas the humidified air from the wet channels is the exhaust at (5). Additionally, the ambient air is drawn into the store at (6) via a ventilation fan(s) and the door openings. The specifications of the 7-Eleven store are shown in Table 2.

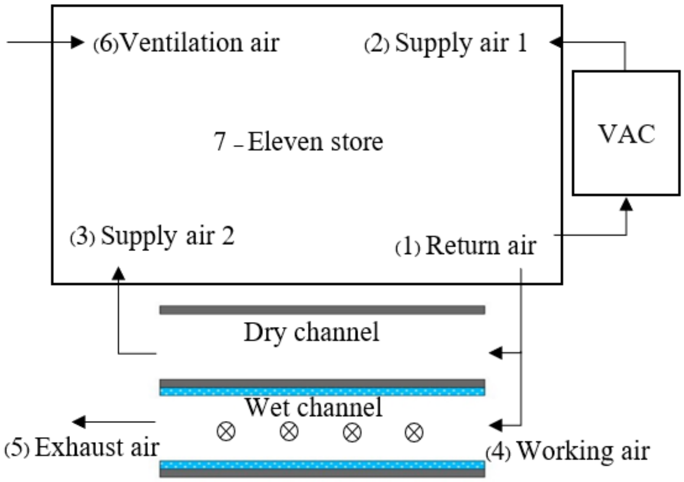

Fig. 2. Schematic diagram of HDPEC.

Table 2: Specifications of 7-Eleven store.

Parameter	Value
Ceiling height	4 m
Width	6.5 m
Length	18 m
Lighting heat gain	31.6 W/m^2
Electric appliance heat gain	8.61 W/m^2
Office occupancy	54 persons/h
The ventilation air	1.8 ACH (602.64 m^3/h)
Internal heat gain	10 W/m^2
Room set point temperature	25℃

3. TRNSYS SIMULATION

This article was conducted to study HDPEC by using TRNSYS software that could model thermal and electrical energy systems. This program has the ability to model operations throughout the year and for selected different climates. This program contains basic modules of thermal and electrical energy systems. The TRNSYS software model of HDPEC of the 7-Eleven store is shown at Fig. 3.

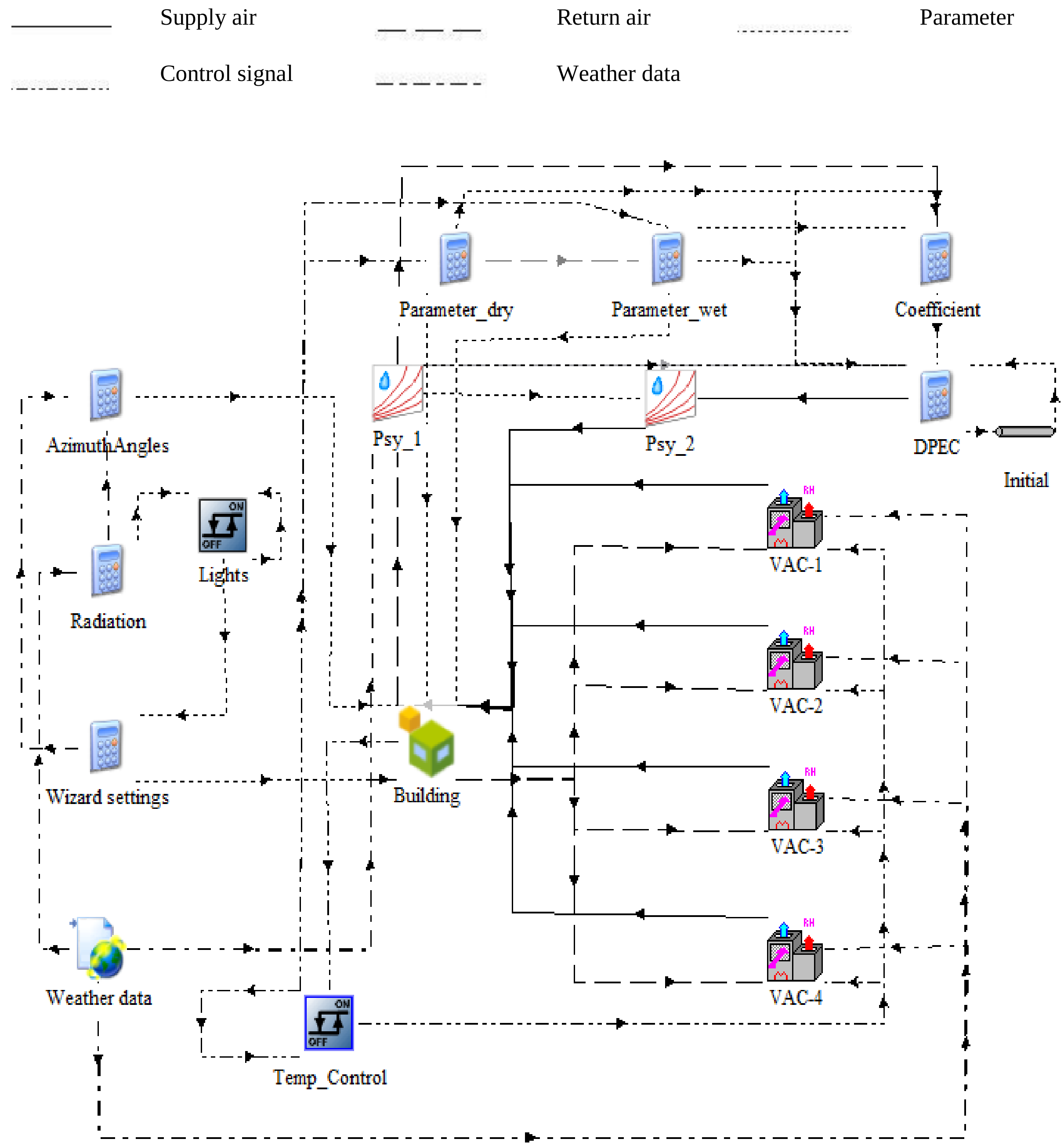

Fig. 3. Diagram of HDPEC system simulation using TRNSYS software.

Figure 3 shows that HDPEC had important modules for the 7-Eleven store (building). There are 4 units of VAC, DPEC, temp control, weather data, azimuth angles, radiation, and wizard settings. An explanation of the functions of each type of device is shown in Table 3.

Table 3: Description of TRNSYS modules.

Module Name	Picture	Type	Description
Building	 Building	56	7-Eleven store
Air to air heat pump	 VAC	954c	Receiving inlet condition and supply of cooled air to the building
Equation	 DPEC	-	Receiving inlet air condition to calculate the outlet air temperature of DPEC (Eq. (3), (10) and (12))
Equation	 Coefficient	-	Calculating heat and mass transfer of DPEC from defined parameters by user (Eq. (4)-(6))
Equation	 Parameter dry	-	Defining parameters for the dry channel of DPEC by user (such as gap, length of dry channel)
Equation	 Parameter wet	-	Defining parameters for the wet channel of DPEC by user (such as gap, length of wet channel)
Equation	 Azimuth angles	-	Receiving weather data to calculate azimuth angles for the building (see detail in TRNSYS mathematical reference chapter 5.9)
Equation	 Radiation	-	Receiving weather data to calculate radiation heat load for the building (see detail in TRNSYS mathematical reference chapter 5.9)
Equation	 Wizard settings	-	Calculating shading heat load for the building (see detail in TRNSYS mathematical reference chapter 5.9)
Controller	 Temp_control	2	Signal control for ON/OFF HDPEC to control the room temperature
Psychometric	 Psy	33e	Calculating air properties from dry bulb and relative humidity
Pipe/duct	 Initial	31	Initial air temperature of dry and wet channels for calculation of the water temperature
Weather data	 Weather data	15	Sending ambient condition to the HDPEC system

4. MATHEMATICAL MODELING

This section explains the governing equations of HDPEC used by TRNSYS software modelling.

4.1 DPEC

DPEC generated cooling by evaporating water. In this modeling process, wet-bulb effectiveness was used for measuring the cooling capability of DPEC by calculation based on Eq. (1) [8]

$$\varepsilon_{wb} = \frac{T_{db,amb} - T_{db,\mathrm{supply}}}{T_{db,amb} - T_{\mathrm{wb,amb}}} \tag{1}$$

DPEC consisted of dry and wet channels that were switched with one another in large quantities. When flowing was determined as turbulent flow, the equation was as follows:

4.1.1 Dry channels

Air temperature changing of the dry channel could be calculated by using Eq. (2) [17]

$$\frac{\partial T_p}{\partial x} = -\frac{L_x}{C_{p,m} m_p}\left[U_{pw}\left(T_p - T_w\right)\sigma + U_{ps}\left(T_p - T_s\right)(1-\sigma)\right] \tag{2}$$

Assuming the surface of the wet channel to be wet thoroughly, $\sigma = 0$, the equation (2) becomes

$$\frac{\partial T_p}{\partial x} = -\frac{L_y}{c_{p,m} m_p}\left[U_{pw}\left(T_p - T_w\right)\right] \tag{3}$$

To determine the heat transfer coefficient of the dry channel, the Nusselt number can be calculated by

$$Nu_p = \frac{h_p D_h}{k} \tag{4}$$

$$Nu_p = 0.023 \mathrm{Re}_p^{0.8} \mathrm{Pr}^{1/3} \tag{5}$$

The Re_p is the Reynold number of the air in the dry channel.

$$D_h = 4\frac{A_c}{p} \tag{6}$$

4.1.2 Wet channels

Changing of temperature and humidity of the air in the wet channel can be calculated by using Eq. (7) and (8)

$$\frac{\partial T_s}{\partial y} = \frac{L_x h_s}{m_s c_{p,m}}\left[1 + \frac{c_{p,v}}{c_{p,a}}\left(\omega_{\mathrm{int}} - \omega_s\right)\right]\left(T_w - T_s\right)\sigma + \frac{L_x}{m_s}\left[U_{ps}\left(T_p - T_s\right)\right](1-\sigma) \tag{7}$$

$$\frac{\partial \omega_s}{\partial y} = \frac{L_x}{m_s} h_d\left(\omega_{\mathrm{int}} - \omega_s\right) \tag{8}$$

When i was determined as specific enthalpy which can be calculated by Eq. (9)

$$i_v = i_0 + c_{p,v} T_w \tag{9}$$

Assuming that $\sigma = 0$, the equation is shown in Eq. (7) reduced to Eq. (10)

$$\frac{\partial T_s}{\partial y} = \frac{L_x h_s}{m_s c_{p,m}} \left[1 + \frac{c_{p,v}}{c_{p,a}} \left(\omega_{\text{int}} - \omega_s \right) \right] \left(T_w - T_s \right) \tag{10}$$

Subsequently, the heat transfer coefficient of the wet channel is calculated by using Eq. (11)

$$h_s = 36.31 \left(\rho u_s \right)^{0.68} \left(L_y / D_{eq} \right)^{-0.08} \tag{11}$$

Considering the temperature changing of water on the wall of the wet channel, calculation is defined as follows:

$$\frac{\partial T_w}{\partial y} = \frac{L_x \sigma}{m_w c_{p,w}} \left[h_d \left(\omega_{\text{int}} - \omega_s \right) \left(i_v - T_w c_{p,w} \right) + h_s \left(T_s - T_w \right) + U_{p,w} \left(T_p - T_w \right) \right] \tag{12}$$

Heat transfer coefficient of the water film in the wet channel is determined as shown in Eq. (13)

$$h_w = Nu_w k_w \left(\frac{\rho_w^2 g}{\mu_w^2} \right) \tag{13}$$

$$Nu_w = \frac{h_w \delta_w}{k_w} = 1.88 \tag{14}$$

$$\delta_w = \left(\frac{3 \Gamma \mu_w}{g \rho_w^2} \right)^{1/3} \tag{15}$$

$$\Gamma = \frac{m_w}{2 \left(N - 1 \right) L_x} \tag{16}$$

Subsequently, the overall heat transfer coefficient is obtained as shown in Eq. (17)

$$\frac{1}{U_{pw}} = \frac{1}{h_p} + \frac{\delta_{plate}}{k_{plate}} + \frac{1}{h_w} \tag{17}$$

The mass transfer coefficient in the wet channel can be calculated by using Eq. (18) [18]

$$Le = \frac{h_s}{c_{ps} h_d} \tag{18}$$

4.2 System performance

Air flows through the dry and wet channels of DPEC which are small channels causing a pressure drop that could be calculated from Eq. (19) [19] as follows:

$$\Delta P = f\,\frac{L}{D_h}\frac{\rho v^2}{2} + \zeta\,\frac{\rho v^2}{2} \tag{19}$$

When f is the friction factor and ζ is the minor loss coefficient, it is found that the pressure drop in the wet channel would be three times greater than the pressure drop in the dry channel. [20]

$$\Delta P_W = 3 \cdot \Delta P_D \tag{20}$$

As a result, energy consumption rates of the fans of dry and wet channels are shown in Eq. (21) and (22)

$$P_{f,\mathrm{d}} = \frac{\dot{\forall}_D \cdot \Delta P_D}{3600 \cdot \eta_0 \cdot \eta_1} \cdot \eta_2 \tag{21}$$

$$P_{f,\mathrm{w}} = \frac{\dot{\forall}_W \cdot \Delta P_W}{3600 \cdot \eta_0 \cdot \eta_1} \cdot \eta_2 \tag{22}$$

Where η_0 ranges from 0.7-0.8, η_1 ranges from 0.85-0.95 and η_2 ranges from 1.05-1.1 [20]. As a result, the total energy consumption rate of DPEC is calculated from Eq. (23) as follows:

$$P_{DPEC} = P_{f,d} + P_{f,w} + P_{pump} \tag{23}$$

The energy consumption rate of VAC can be calculated from Eq. (24) as follows:

$$P_{VAC} = P_{f,indoor} + P_{f,outdoor} + P_{compressor} \tag{24}$$

When calculating the COP of the system is shown in Eq. (25)

$$COP = \frac{\dot{Q}_{VAC} + \dot{Q}_{DPEC}}{P_{VAC} + P_{DPEC}} \tag{25}$$

Total electric power consumption of HDPEC could be calculated from Eq. (26) as follows:

$$EEC = EEC_{VAC} + EEC_{DPEC} \tag{26}$$

$\dot{Q}_{VAC}$ and $\dot{Q}_{DPEC}$ are heat transfer rates for transferring heat from air-conditioned room by VAC and DPEC, respectively. In this study, the 4th order Runge-Kutta method was used for solving thermal problems. From the experiment, it was found that the output temperature of the wet channel was higher than the wet-bulb temperature of the input of the wet channel by 2˚C approximately. As a result, the temperature of the output of the wet channel was determined to be higher than the wet-bulb temperature by 2˚C. These equations allow thermal problems to be solved more easily.

According to the flowchart as shown in Fig. 4, it could be seen that after starting modeling, users must determine the length and width of the dry and wet channels as well as the flow rate of the air and specify the running time required, for example, from January 1st to December 31st, etc. Subsequently, the program would calculate the heat and mass transfer of DPEC according to the defined conditions. If the temperature of temp control of the room was over 25˚C, VAC and DPEC would start to be operated to supply cool air to the room until the room temperature was lower than 23˚C. Temp control would command the system to stop operating and the cooling room would be heated by sunlight and room heat generated by occupants and electrical devices. When the modeling period was complete upon the due date, i.e., simulation time equal to the specified running time, the simulation would be terminated. However, if the due date was not completed, HDPEC would be operated continuously according to specified conditions in order to control the temperature of the air-conditioned room.

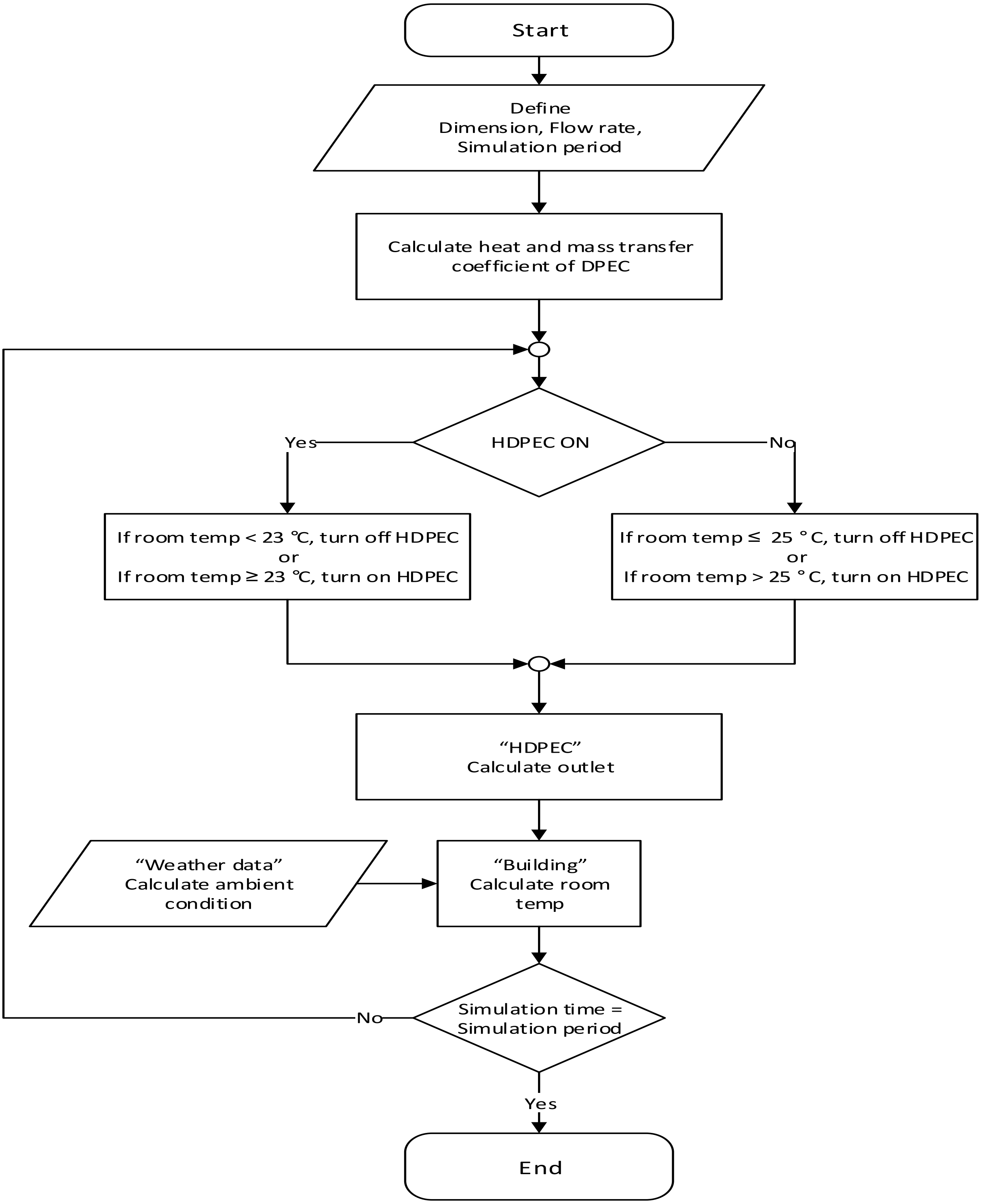

Fig. 4. Flowchart of the HDPEC system's operation.

5. MODEL VALIDATION

In this article, the researcher modeled DPEC using TRNSYS software. The obtained results were compared with experimental results conducted at Suranaree University of Technology in Nakhon Ratchasima province by installing the experimental set with the experimental room as shown in Fig. 5.

Figure 5 is the installation of DPEC with the room for experimentation while details of the experimental set are shown in Table 4.

Fig. 5. Installation of the DPEC experiment set.

Table 4: Information of the DPEC Experimental Set.

Parameters	Description
Test room area	16 m^2
Test room height	4 m
DPEC core	90x40x30 cm.
Pump power	100 W
Fan power	1.49 kW
Data logger	Wisco DL2200
Thermocouple	Type T ±0.9˚C

After installing the experimental set as shown in Fig. 5 and conducting the experiment under the climate of Nakhon Ratchasima province, and comparing experimental results with simulation results obtained by TRNSYS software, the obtained results were as shown in Fig. 6.

Figure 6 showed a comparison between the simulation results and actual experimental results under the same operational conditions and climate. It could be seen that the temperature of the supply of cooled air was consistent with the percentage of average error at 2.37% that may be caused by the uncertainty of measurement with the deviations for type T thermocouples at ±0.9˚C.

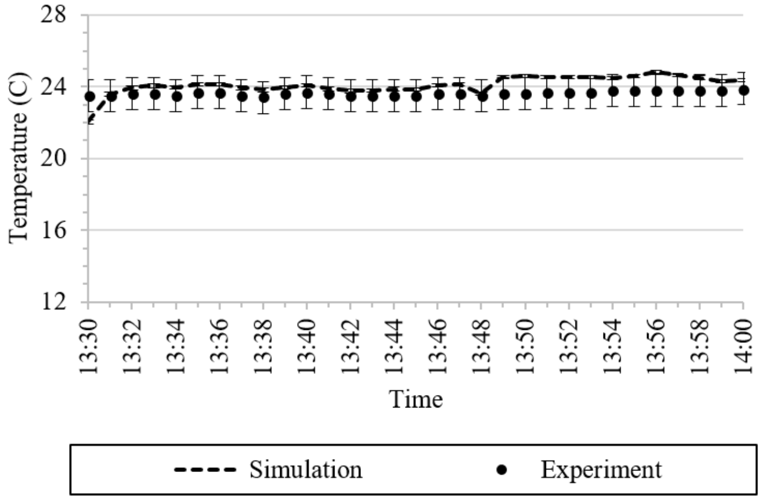

Fig. 6. Temperature of the supply cooled air during the experiment and simulation using TRNSYS software.

6. RESULTS AND DISCUSSION

This article aims to model HDPEC and study the factors influencing the operation of the system by adjusting the variables of DPEC, whereas 4 units of VAC had the same operational conditions. The results shown in this section are the results of simulation in the event that HDPEC was able to control room temperature in the range of 23-25º C. In the simulation, the variables of DPEC were adjusted as shown in Table 5.

Table 5: Case study of HDPEC.

Parameters	Value			
	Case 6.1	**Case 6.2**	**Case 6.3**	**Case 6.4**
Dry channel length	10 - 100 cm	25 cm	25 cm	25 cm
Wet channel length	30 cm	30 cm	30 cm	30 cm
DPEC width	100 cm	100 cm	100 cm	100 cm
Dry channel gap	6 mm	3 – 9 mm	6 mm	6 mm
Wet channel gap	10 mm	5 – 15 mm	5 mm	5 mm
Dry channel air flow rate	600 - 6,000 m^3/h	4,000 m^3/h	4,000 m^3/h	4,000 m^3/h
Wet channel air flow rate	602.64 m^3/h	602.64 m^3/h	602.64 m^3/h	602.64 m^3/h
Simulation time	1 day	1 day	1 year	1 year
Location	Nakhon Ratchasima	Nakhon Ratchasima	Nakhon Ratchasima	-Nakhon Ratchasima -Sakon Nakhon -Chiang Mai -Bangkok

In the simulation, the length of the wet channel was fixed at 30 cm (the length of DPEC) because the area of the 7-Eleven store was limited. As a result, if DPEC was large, it was inconvenient for actual installation. The simulation was divided into 5 sections as follows:

6.1 Impact of dry channel length

DPEC contained dry and wet channels where the length of these channels is an important factor influencing the system's operation. As a result, this section was considered as a simulation by adjusting the length of the dry channels under the climate of May, which is considered to be summer. The pressure drop which occurred in the dry channel of DPEC is shown in Fig. 7.

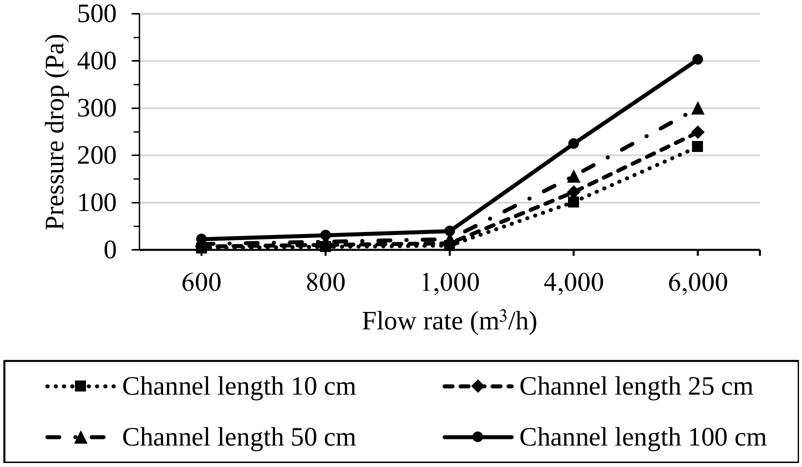

Fig. 7. Pressure drop with variation of air flow rate and length of dry channel.

From the results of the simulation in Fig. 7, it is shown that by increasing the length of the dry channel from 10 to 100 cm, the pressure drop caused by air flow would be increased in accordance with Eq. (19). In addition, when the air flow rate was increased from 600 to 6,000 m^3/h, the pressure drop would be approximately increased by 100 times, caused by increasing the flow rate, whereby the pressure drop also increased according to Eq. (19). As a result, to design DPEC, an appropriate length and flow rate in the dry channel should be selected in order to hinder the pressure drop from increasing, which may affect the selection of the size of the ventilating fan of DPEC. When considering electric power consumption of HDPEC, it is shown in Fig. 8.

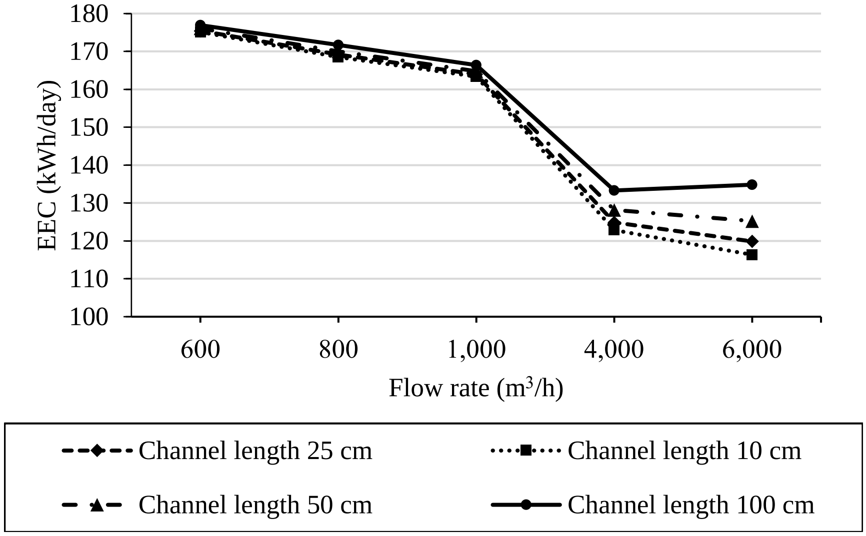

Fig. 8. EEC of the system with varying air flow rate and length of dry channel.

In Fig. 8 which air flow rate was equal, it is found that when the length of the dry channel increased, EEC also increased because of the pressure drop increment causing the ventilating fan to consume more electric power. When considering the same length of dry channel, it is found that increasing the flow rate may reduce EEC because of an increase in heat transfer of DPEC, and separate heat from the air-conditioned room at a higher rate. Although the ventilating fan of DPEC consumed more electric power, the EEC of the system may be lower due to less operation of VAC. Subsequently, EEC would decrease when the flow rate was higher than 4,000 m^3/h because of the rapidly increasing pressure drop. Therefore, DPEC consumed more electric power. When considering the COP of the system, it can be seen in Fig. 9.

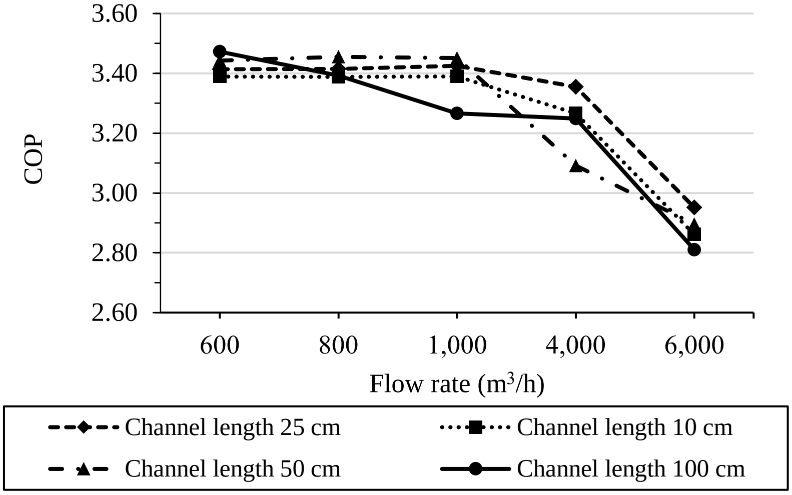

Fig. 9. COP of the system with variation of air flow rate and dry channel length.

In Fig. 9 it could be seen that the COP of the system is insignificantly changed in the range of 600-1,000 m^3/h with the highest level of change at 6.33%. However, when the flow rate was increased, ranging from 4,000-6,000 m^3/h, COP decreased rapidly because this range had a high pressure drop as shown in Fig. 7. As a result, DPEC consumed more electric power and lowered the COP of the system. In addition, it is also found that at the lengths of 50 and 100 cm for the dry channel, COP significantly decreases when the flow rate increases rapidly due to an increasing flow rate causing more turbulence in the flow and also increased the pressure drop. Therefore, COP decreased according to Eq. (25)

6.2 Impact of channel gap

Since this section is related to the study of the influence of the width of the dry and wet channels in DPEC from section 6.1, the length of the dry channel and flow rate is specified as 25 and 4,000 m^3/h as shown in Table 5. DPEC with the size of 25x30x200 cm is obtained because it made the system have suitable COP and EEC. The simulation is shown in Fig. 10.

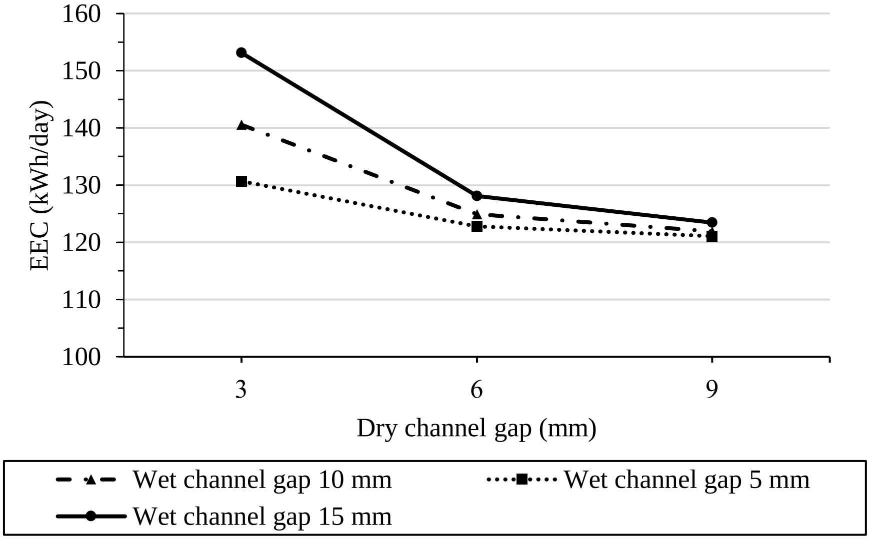

Fig. 10. EEC of the system as varying wet and dry channel gap functions.

Figure 10 reveals that when the wet channel gap is smaller, EEC would decrease because this simulation specifies that DPEC had the same size. As a result, when the size of the wet channel gap is reduced, the number of wet channels could be increased making DPEC have a greater area for heat transfer. Although reducing the size of the channel may cause a higher pressure drop, the flow rate in the wet channel would be low at 602.64 m^3/h, lowering the pressure drop of this channel. However, when the size of the dry channel gap was smaller, it could be seen that EEC was increased due to the high flow rate of this channel at 4,000 m^3/h. As a result, when the channel is smaller, the pressure drop will be higher and EEC increases. When considering COP, it was as shown in Fig. 11.

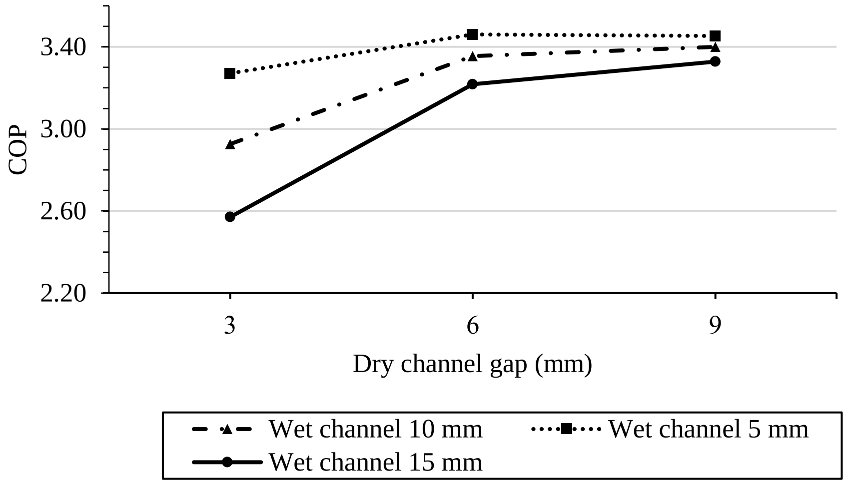

Fig. 11. COP of the system with varying wet length and dry channel gap.

In Fig. 11, the small wet channel gap had a large area of heat transfer due to the higher number of dry and wet channels increasing COP. Although the smaller wet channel had a higher pressure drop, the air flow rate in the wet channel was quite low compared to the dry channel. As a result, the pressure drop was low. When considering the larger dry channel gap, it could be seen that the COP was higher due to the lower pressure drop reducing EEC of the system.

From the simulation in 6.1 and 6.2, an appropriate DPEC size was obtained. In this article, the dry channel gap was selected to be 6 mm because it was the size of material that was generally available, for example, polycarbonate sheets that were cheap because the simulation in 6.1 and 6.2 was determining DPEC to be always operated. However, the next section was the study of the appropriate operation strategy of DPEC.

6.3 Operational strategy of DPEC

In modeling, VAC was determined to be operated during 24 hours and temperature was controlled to be in the range of 23-25˚C, i.e., VAC would be operated when the room temperature was higher than 25˚C and it would be stopped when the room temperature was lower than 23˚C. For DPEC, the operational strategy could be determined into three types as follows:

- Always-on DPEC: DPEC was determined to be operated 24 hours with VAC.
- Syn-with VAC: DPEC was determined to be operated with VAC with the same operational condition, i.e., to be operated when the room temperature was higher than 25˚C and stopped when the room temperature was lower than 23˚C.
- VAC-alone: VAC was determined to be operated without DPEC for generating cooling .

In Fig. 12, it could be seen that VAC-alone consumed the maximum electric power at 6,216 kWh/month in May because it was considered as the hottest month in summer, while October-February was winter, and therefore the operation of the cooling system decreased. When considering the EEC of the system that was mutually operated with DPEC, it was found that Always-on DPEC and Syn-with VAC had similar EEC levels helping to lower the total electric power consumption of the system, especially in May, for which Syn-with VAC could reduce EEC by 41.54% or 2,581.88 kWh approximately when compared to VAC-alone.

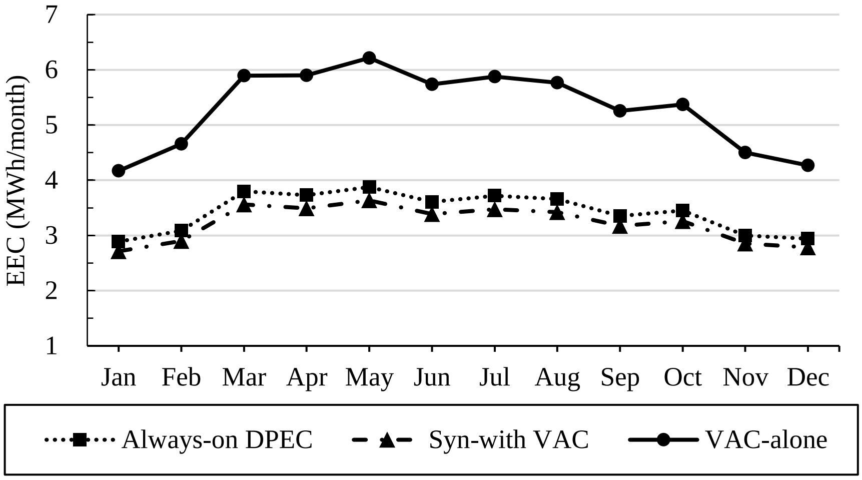

Fig. 12. EEC of the system with different strategies of DPEC operation.

For Fig. 13, when considering EEC throughout the year, it could be seen that VAC-alone was the system consuming high electric power compared to Always-on DPEC and Syn-with VAC, whereas EEC was higher than Syn-with VAC by 64.54% approximately. When considering Always-on DPEC and Syn-with VAC, it was found that the electric power consumption of both types of VAC (EEC-VAC) was not significantly different. However, the electric power consumption of DPEC (EEC-DPEC) was different by 5.96% approximately because Always-on DPEC determined DPEC to be always operated causing DPEC to consume excessive electric power even in the periods with low heat load such as night time. As a result, the study in 6.4 would use the operational conditions of Syn-with VAC in the simulation.

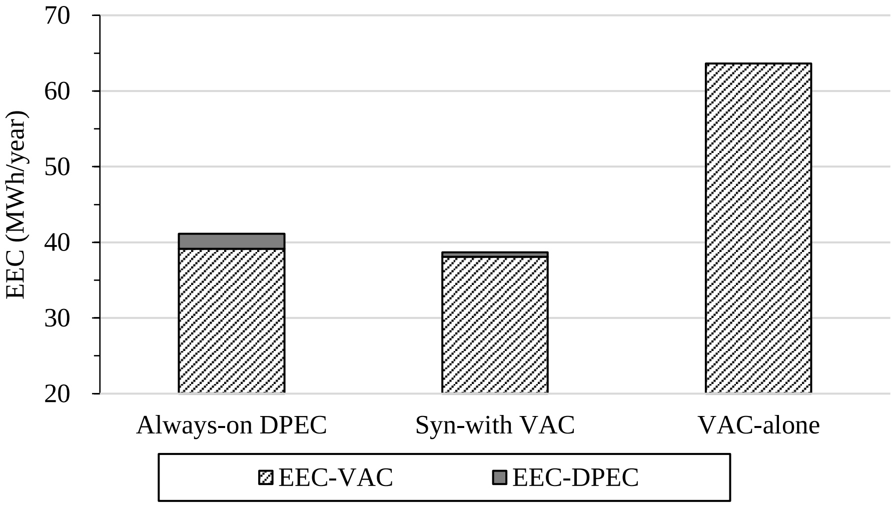

Fig. 13. EEC of the system with different strategies of DPEC operation.

6.4 Impact of geometry

This section compares the operation of HDPEC in 4 provinces of Thailand including Chiang Mai, Sakon Nakhon, Nakhon Ratchasima, and Bangkok, covering different climates. The results of the simulation are shown in Fig. 14.

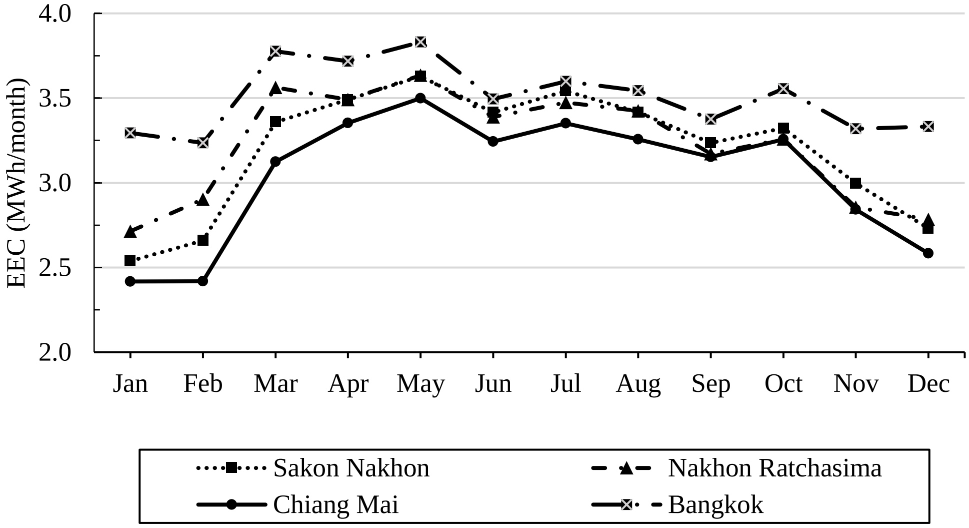

Fig. 14. EEC of the system operated in different climates.

When comparing the EEC of the system operated in provinces with different climates as shown in Fig. 14, it could be seen that the province with the highest level of EEC was Bangkok because its climate was tropical and therefore HDPEC would have a high heat load. The province with the lowest level of EEC was Chiang Mai because it was located in the northern part of Thailand with average temperatures lower than those of the other three provinces. As a result, HDPEC of Chiang Mai had the lowest heat load and the lowest level of EEC throughout the year and for VAC-alone as shown in Fig. 15.

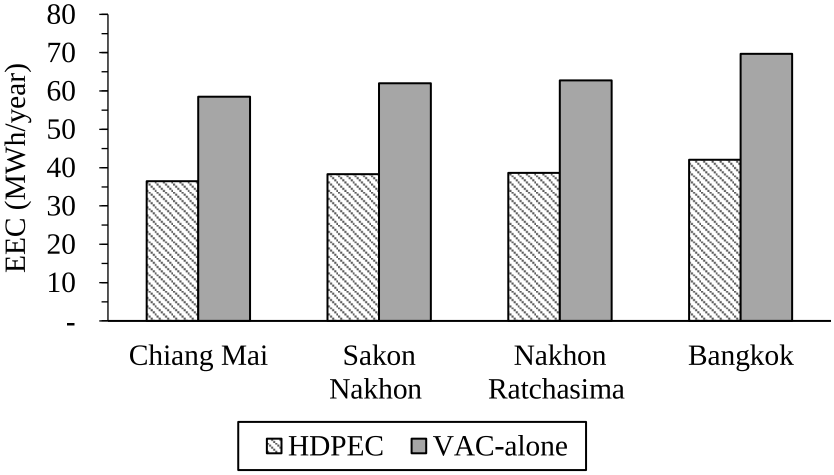

Fig. 15. EECs of the system operated in different climates compared to its VAC systems.

In Fig. 15, when comparing HDPEC with the system using VAC-alone, it was found that the province using HDPEC that could lower electric power consumption at the highest level was Bangkok, which reduced EEC by 39.62%. Chiang Mai was the province with HDPEC that could lower electric power consumption at the lowest level by lowering EEC by 37.61% approximately because it was the province with the lowest EEC. DPEC had high wet-bulb effectiveness when the temperature and humidity of the air was high. Since Chiang Mai was the province with lowest average temperature and humidity, its HDPEC had wet-bulb effectiveness as shown in Fig. 16.

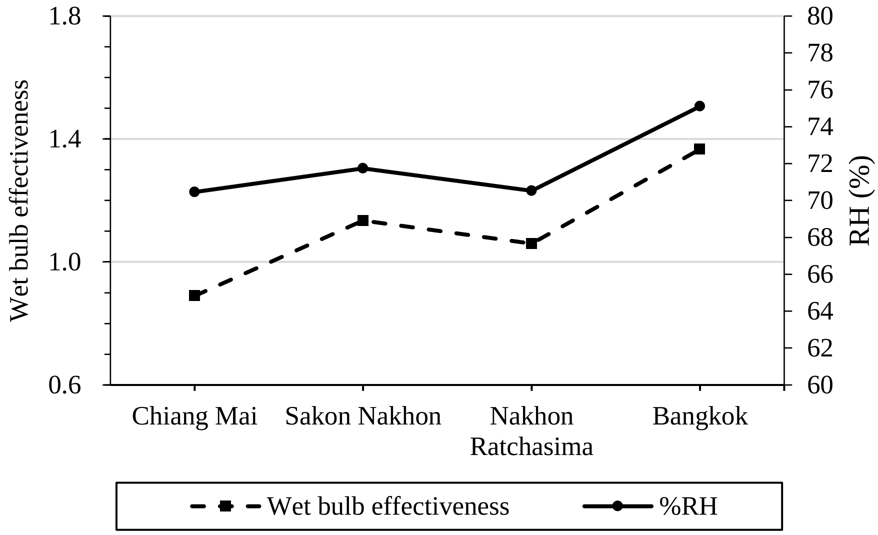

Fig. 16. Wet-bulb effectiveness and relative humidity when the system operated in different climates.

In Fig. 16, it was found that Bangkok had the highest level of wet-bulb effectiveness due to high relative humidity (RH). From Eq. (1), the denominator was reduced while the numerator was slightly changed, therefore wet-bulb effectiveness was at the highest level at 1.37 approximately. DPEC used the principle of water evaporation to generate cooling, therefore wet-bulb effectiveness was low in the humid climate. Since this article covers the simulation of DPEC mutually operating with VAC, whereas DPEC would use air with low humidity obtained from the air-conditioned room using VAC in controlling temperature and humidity, HDPEC had high wet-bulb effectiveness although Bangkok had high humidity.

6.5 Economy

This section is an analysis of the economic impact of HDPEC in order to find the payback period for the operation in Thailand. The initial cost of devices in DPEC is as shown in Table 6.

Table 6: Initial cost of DPEC.

Lists	Price (USD)
DPEC core	52.74
Pump	30.48
Blower	524.34
DPEC housing	60.97
Electrical system	30.48
Water pipe and distributor	15.24
Total	714.26

From Table 6, the initial cost for creating DPEC was approximately 714.26 USD. When the electrical energy cost for 1 unit (1 kWh) was determined as 0.09 USD/unit, the payback period was as shown in Fig. 17.

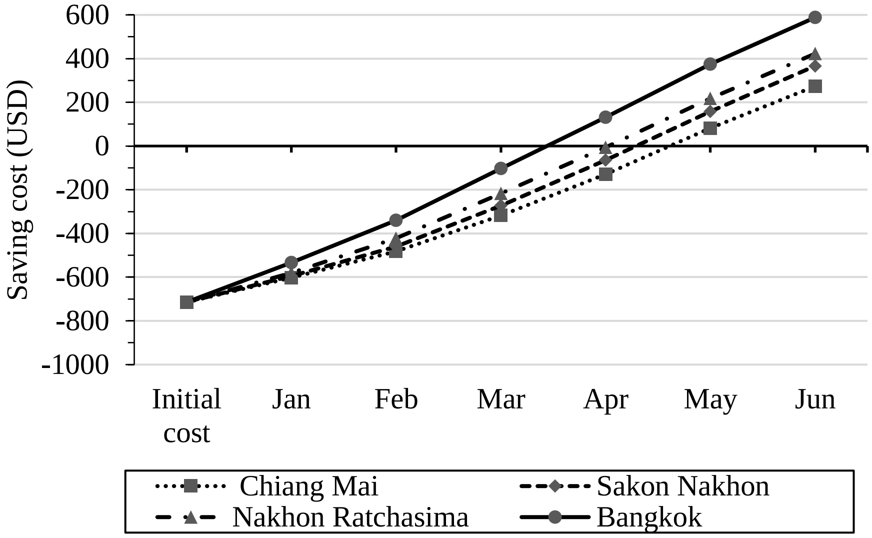

Fig. 17. Comparison of payback periods of the HDPEC system.

Fig. 17 shows that the initial cost was -714.26 USD and it was determined to be a negative value because it had to be paid to install the DPEC in order to lower the electric power consumption of cooling systems in 7-Eleven stores. When subtracting the initial cost from the electrical energy cost that could be saved in each month, it was found that Bangkok has the shortest payback period at around 3.5 months, because it could lower the power consumption of VAC at the highest level as explained in Fig. 15. Chiang Mai has the longest payback period at around 4.5 months. As a result, the province with the greatest potential for applying HDPEC according to this study is Bangkok because it had the shortest payback period.

7. Conclusions

This article aims to study the factors influencing the operation of HDPEC for a 7-Eleven store by numerical simulations using TRNSYS software. It was found that the obtained results are consistent with the experimental results with an average error of 2.37%. It was also found that the pressure drop depends on the channel length and flow rate. The maximum COP of 3.47 is achieved when the channel length is 100 cm and the flow rate is 600 m^3/h. The optimal dry and wet channel gaps are 6 and 5 mm, respectively. According to the simulations, the operational strategy, in which the cooler and VAC are both on when the room temperature is higher than 25˚C and they are both off when the room temperature is less than 23˚C, consumes the least electricity compared to others. Its annual power consumption is 41.54% less than that of the system with VAC alone.

Nomenclatures

A	Area (m^2)
C_p	Specific heat capacity at constant pressure (J/kg K)
D_h	Hydraulic diameter (m)
f	Friction factor
h	Convective heat transfer coefficient (W/m^2K)
H	Channel gap (mm)
h_d	Convective mass transfer coefficient (g/m^2s)
i	Specific enthalpy (J/kg)
i_0	Specific enthalpy of vapor at 0 ̊C (J/kg)
k	Thermal conductivity (W/m K)
Le	Lewis number
L_x	System length (m)
L_y	System height (m)

$\dot{m}$	Mass flow rate (kg/s)
M	Total mass flow rate (kg/s)
M_v	Mass transfer rate (kg/s)
N	Number of plates
Nu	Nusselt number
P	Electric power (W)
Pr	Prandtl number
Q_L	Latent heat (kJ)
Q_S	Sensible heat (kJ)
Re	Reynolds number
T	Temperature (ºC)
U	Overall heat transfer coefficient (W/m^{2}K)
v	Velocity (m/s)

Subscripts

c	*Cross section*
D	Dry channel
W	Wet channel
Int	Water and air interface
p	Primary air or dry channel air
s	Secondary air or wet channel air
m	Moist air
v	Vapor
pw	Dry channel air to water
ps	Dry channel air to wet channel air
w	Water film, wet channel
wb	Wet-bulb

Greek symbols

ΔP	Pressure loss (Pa)
ε	Effectiveness
Γ	Water linear flow rate (kg/m s)
σ	Surface wetting factor
ω	Humidity ratio (kg/kg)
∀	Volumetric flow rate (m^3/h)
ζ	Minor loss coefficient

Abbreviations

COP	Coefficient of performance
DEC	Direct evaporative cooling
DPEC	Dew-point evaporative cooling
EC	Evaporative cooling
EEC	Electrical energy consumption
HDPEC	Hybrid dew-point evaporative cooling
VAC	Vapor-compression air-conditioning

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