# Potential of the Solar Thermal Desiccant Cooling in Asia-Pacific Region

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## *Abstract*

*The solar thermal desiccant cooling system was numerically investigated for application in the Asia-Pacific Region (East Asia and South East Asia). The system was modeled in transient system simulation (TRNSYS) program and applied in a hypothetical office building. The typical meteorological year (TMY) was used as the basis for the climatic conditions. The system was applied in the region's sixteen major cities covering the temperate, sub-temperate/sub-tropical and tropical climates. The results showed the required flat plate collector area was bigger in the tropical climate compared to temperate climate. The needed air flow rate was higher in tropical climate compared to temperate climate. However, in general, it was shown the potential and applicability of the solar thermal desiccant cooling system in the Asia- Pacific Region.*

**Keywords:** Solar energy; Desiccant system; Dehumidification; Cooling; Simulation; Asia

## Nomenclatures

| | |
|---|---|
| $C_p$ | specific heat ($kJ \cdot kg^{-1} \cdot K^{-1}$) |
| $DCOP$ | desiccant coefficient of performance |
| $E$ | electric energy (kJ) |
| $h$ | moist air enthalpy ($kJ \cdot kg^{-1}$) |
| $h_{Evap}$ | latent heat of evaporation ($kJ \cdot kg^{-1}$) |
| $m$ | mass flow ($kg \cdot s^{-1}$) |
| $OPEC$ | off-peak electric energy consumption |
| $Q$ | heat energy (kJ) |
| $OA$ | outdoor air |
| $RA$ | return air, room air |
| $SA$ | supply air |
| $SCOP$ | system coefficient of performance |
| $SF$ | solar fraction |
| $SHR$ | sensible heat ratio |
| $t$ | time (s) |
| $T$ | temperature ($^{o}C$) |
| $X$ | absolute humidity ($Kg_{H2O} \cdot Kg_{Air}^{-1}$) |

*Subscript*

| | |
|---|---|
| $AHC$ | air heating coil |
| $Aux$ | heater (daytime) |

| | |
|---|---|
| *CL* | cooling load |
| *Col* | collector |
| *Hea* | heater (nighttime) |
| *I* | inlet |
| *O* | outlet |
| *OA* | outdoor air |
| *RA* | return air |
| *SA* | supply air |
| *Sen* | sensible energy |
| *Ther* | thermal energy |
| *Tot* | total energy |
| *W* | water |
| *X* | absolute humidity |

## Introduction

Buildings consume large amount of conventional energy resources [1]. Large amount of building energy consumption is to support indoor thermal comfort conditions. In Europe, 10% of total energy consumption is for the provision of indoor thermal comfort [2]. In Middle East, 70% of the building energy consumption is for the maintenance of indoor thermal environment [3]. In Japan, 3% of total building energy consumption is for indoor cooling [4]. However, during summertime, large percentage of building energy consumption is for air dehumidification and cooling. In South East Asia, it is expected that buildings consume large percentage of electric energy to support comfortable indoor environment due to the climatic conditions (hot and humid for whole year).

There were several studies conducted on the reduction of building cooling energy consumption through the utilization of alternative energy sources, equipment, devices and methods [5,6]. Solar energy is available anywhere at different solar intensities. In most cases, the amount of cooling requirement is in phase with the amount of solar radiation [7]. Nighttime and off-peak electric energy is readily available for utilization [8]. Utilization and application of off-peak electric energy for peak demand have an impact on peak load demand reduction [9].

Thermally driven cooling technologies are potentials both for utilizing the free solar energy and the cheaper off-peak electric energy. Several studies were conducted with regard to the applicability of thermally driven cooling technologies [10]. Desiccant cooling system is a potential for application utilizing available thermal energy resources [11]. Enteria et al., (2009) [12] shows the potential of utilizing nighttime electric energy and daytime solar energy for the operation of the desiccant cooling technology.

There were several researches conducted on the potential of the solar desiccant cooling technologies with and without back-up thermal energy sources [13,14]. System potential in different climatic conditions as in the case of Europe was conducted by Mavroudaki et al., 2002 [15]. For Australian climates, White et al., 2009 [14] conducted numerical investigation comparing the Southern and Northern Australian climates. Fong et al. [16] investigated the applicability of the system in sub-tropical Hong Kong. Kodama et al., 2000 [17] made investigation for the application in damp climatic conditions. However, no investigation was conducted for application in the greater Asia-Pacific Region which is very hot and humid (East Asia during summer time and South East Asia for whole year).

The East Asian areas have temperate and sub-temperate climates while the South East Asian areas have tropical climate. Since the Asia-Pacific Region is very high in humidity and temperature compared to other regions of the world, day-long air cooling and dehumidification is needed. Katejanekarn et al. 2009 [18] showed that ordinary evaporative cooling alone was not enough to maintain the indoor thermal comfortable condition due to higher air wet bulb temperature. This paper presents the investigation of the solar desiccant cooling system potential in the Asia-Pacific Region (East Asia and South East Asia). This was done through numerical investigation in the transient system simulation (TRNSYS) program using the typical meteorological year (TMY) as the input weather conditions. The system was investigated during the hottest and humid week of the year.

## Solar-desiccant cooling system

### *System description*

The solar thermal desiccant cooling system presented in Fig. 1 consists of the solar thermal subsystem and the desiccant cooling subsystem. The solar thermal subsystem composes of the flat plate collector, nighttime electric heater and thermal storage tank. The desiccant cooling subsystem composes of desiccant wheel, cross-flow heat exchangers, evaporative coolers and daytime auxiliary heater (in this case an electric heater). The system (Fig. 1) is an updated design presented by Enteria et al., 2009 [19]. In the original design, no daytime auxiliary heater was presented. In addition, one evaporative cooler was used in previous design. To make the system practical for application and at the same time enhance the cooling performance, daytime auxiliary heater and one evaporative cooler were added. The solar thermal subsystem and the desiccant cooling subsystem were connected through the air-heating coil.

### *System operation*

The original design of the solar desiccant cooling system was validated by series of numerical and experimental runs [20,21]. The optimum design and operational conditions of the original system was used as the basis for the updated solar desiccant cooling system design presented here. The system operated during nighttime (off-peak) to store thermal energy from nighttime electric heater. The stored nighttime thermal energy was used to support the early operation of the desiccant cooling system. The desiccant cooling subsystem started to operate at 8AM which solar radiation was still not available [12]. As the solar energy increases with advancing daytime, the flat plate collector started to operate to support the operation of the desiccant cooling subsystem. However, in the case of cloudy sky, the daytime auxiliary heater was used to augment the thermal energy requirement. The daytime auxiliary heater used in the study was an electric heater; however, it can be substituted with gas heater and other waste-heat sources. The desiccant cooling subsystem operated until 6PM in the afternoon.

## System and building model

### *Solar desiccant cooling system model*

The model of the solar desiccant cooling system was done in TRNSYS [22]. The model considered the heat losses in the water piping [23]. Piping heat losses were considered based on our experimental evaluation [24]. Several solar thermal and desiccant cooling system modeling did not considered these losses.

### *Building model*

The hypothetical building was designed to serve as the thermal load of the solar thermal desiccant cooling system. The building physical description and dimension was the same to Mavroudaki et al., (2002) [15]. However, the building structural components and thermal properties were different.

The building was rectangular in shape with all four walls as external. The building was cool and ventilated using the solar thermal desiccant cooling system from 8AM to 6PM during weekdays. The building had five office workers doing seated, light work such as typing. Each worker had sensible gain of 75W and latent gain of 75W. The worker had each computer with thermal gain of 230W. The office lighting had thermal gain of 17W/m$^2$. The building had an air infiltration of 0.5 ACH (air change per hour). All the thermal gains from computers, lighting and occupants started from 8AM to 6PM. The hypothetical building with solar thermal desiccant cooling system was applied in sixteen different cities in the Asia-Pacific region (See Table 1 for locations).

## Performance indices

### *Solar thermal subsystem performance*

The contribution of the solar energy in the thermal energy requirement of the system operation was expressed as the solar fraction (SF)

$$SF = \frac{Q_{Col}}{Q_{Col} + Q_{Hea} + Q_{Aux}} \tag{1}$$

Where,

$$Q_{Col} = \int_{Start}^{End} \left[ m_{Col} C_{P(W)} \left( T_{Col,O} - T_{Col,I} \right) \right] dt \tag{2}$$

$$Q_{Hea} = \int_{Start}^{End} \left[ m_{Hea} C_{P(W)} \left( T_{Hea,O} - T_{Hea,I} \right) \right] dt \tag{3}$$

$$Q_{Aux} = \int_{Start}^{End} \left[ m_{RA} \left( h_{RA(2)} - h_{RA(1)} \right) \right] dt \tag{4}$$

### *Desiccant cooling subsystem performance*

The ratio of air sensible energy load, latent energy load and total energy load reduced by the desiccant cooling subsystem was expressed as the sensible heat ratio (SHR)

$$SHR = \frac{Q_{Sen}}{Q_{Tot}} \tag{5}$$

Where,

$$Q_{Sen} = \int_{Start}^{End} \left[ m_{SA} C_{P(A)} \left( T_{OA} - T_{SA} \right) \right] dt \tag{6}$$

$$Q_{Lat} = \int_{Start}^{End} \left[ m_{SA} h_{Evap} \left( X_{OA} - X_{SA} \right) \right] dt \tag{7}$$

$$Q_{Tot} = Q_{Sen} + Q_{Lat} = \int_{Start}^{End} \left[ m_{SA} \left( h_{OA} - h_{SA} \right) \right] dt \tag{8}$$

The desiccant cooling subsystem performance in reducing air temperature and humidity content based on thermal energy requirement was expressed as the desiccant coefficient of performance (DCOP)

$$DCOP = \frac{Q_{CL}}{Q_{Ther}} \tag{9}$$

Where,

$$Q_{CL} = \int_{Start}^{End} \left[ m_{SA} \left( h_{OA} - h_{SA} \right) \right] dt \tag{10}$$

$$Q_{Ther} = \int_{Start}^{End} \left[ m_{RA} \left( h_{RA(1)} - h_{HA} \right) \right] dt + \int_{Start}^{End} \left[ m_{RA} \left( h_{RA(2)} - h_{RA(1)} \right) \right] dt \tag{11}$$

### *System performance*

The amount of electric energy consumption by the system operation was expressed as the ratio of off-peak electric energy consumption to total electric energy consumption called off-peak electric energy consumption (OPEC)

$$OPEC = E_{Nighttime} / \left( E_{Nighttime} + E_{Daytime} \right) \tag{12}$$

Where,

$$E_{Nighttime} = \int_{Start}^{End} \left( E_{Hea} \right) dt + \int_{Start}^{End} \left( E_{Hea,Pump} \right) dt \tag{13}$$

$$E_{Daytime} = \int_{Start}^{End} \left( E_{Aux} \right) dt + \int_{Start}^{End} \left( E_{AHC,Pump} \right) dt + \int_{Start}^{End} \left( E_{Col,Pump} \right) dt + \int_{Start}^{End} \left( E_{Fans} \right) dt \tag{14}$$

The performance of the system was based on the reduction of the air temperature and humidity content to the system total electric energy consumption expressed as system coefficient of performance (SCOP)

$$SCOP = Q_{CL} / \left(E_{Daytime} + E_{Nighttime}\right) \qquad (15)$$

## Simulation and Evaluation

### *System Specification*

For different climatic conditions, the system had different specifications to maintain the indoor thermal comfort conditions (See Table 2). In this table, it was shown that in Tokyo, Japan, the required flat plate collector area was 8m$^2$. The volumetric air flow rate was 330m$^3$/h for outdoor air with 165m$^3$/h supply air.

In the case of other locations and conditions, Table 2 shows the different sizes of the solar thermal system flat plate collector requirements and air flow rates to support room cooling, dehumidification and ventilation. This system size supported the required comfortable condition of the room presented. As shown in the table, northern part of the Asia-Pacific Region (East Asia and upper part of South East Asia) needed 8m$^2$ of the flat plate collector except for Korea. In the lower portion of the South East Asia, the required flat plate collector was 12m$^2$ except for Timor.

The required air flow rates for the East Asia was 330m$^3$/h compared to the middle of the East Asia and South East Asia which had 200m$^3$/h air flow rate. The southern part of the South East Asia required 330m$^3$/h air flow rate to support the required indoor environmental comfort.

The smaller requirement of flat plate collector area in the East Asia even at higher volumetric air flow rate is due to the lower outdoor air absolute humidity content compared to the South East Asia. Thus, the required thermal energy for dehumidification is lower compared to the South East Asia. In South East Asia, it needs large area of solar collector to support the dehumidification of high volumetric flow rate of air since the absolute humidity content is higher compared to East Asia.

In the case of the lower East Asia and upper South East Asia, smaller size of flat plate collector is needed with lower air flow rate due to the high collection of solar energy compared to north East Asia and not as high absolute humidity content compared to lower South East Asia. Thus, different sizes of the flat plate collectors are needed for different air flow rates to support the needed indoor environmental conditions.

### *Building Performance*

In the case of Jakarta and Port Moresby, the desiccant cooling had an air flow rate of 330m$^3$/h compared to 200m$^3$/h in Dili with 12 m$^2$ flat plate collector (See Table 2). This was due to the higher dehumidification requirement (higher outdoor air absolute humidity content). The average air temperatures (outdoor air, supply air and room air) were presented in Fig. 2.

As presented, the outdoor air temperature was higher in the location of Vientiane and Yagon ($35^oC$ and above). The outdoor air temperature was decreasing from north to south of the two cities. The supply air temperature was between $17^oC$ and $24^oC$. With this condition, the room temperature was maintained within $22^oC$ and $25^oC$ with system specifications presented in Table 5. In the case of air absolute humidity contents (outdoor air, supply air and room air), Fig. 3 showed the results.

As presented in the figure, the region outdoor air average absolute humidity was within 15g/kg and 21g/kg. This makes the region one of the most humid climates. The supply air absolute humidity content was maintained within 3g/kg and 7g/kg. With these conditions, the room absolute humidity was maintained within 8g/kg and 11g/kg. This makes the building indoor environment thermally comfortable.

The air equivalent relative humidity (outdoor air, supply air and room air) was presented in Fig. 4. It shows the region average humidity between 45% and 74%.

### *System Performance*

The performance of the solar desiccant cooling system is presented in Fig. 5. It shows the sensible heat ratio (SHR), off-peak electric energy consumption (OPEC) and solar fraction (SF). The solar desiccant cooling system shows reduction of a large amount of air latent energy content. Based on the sensible heat ratio, it was within 20% to 30%. It means almost three-fourth of the air thermal load controlled by the solar desiccant cooling system was an air latent energy (moisture).

The operation of the nighttime electric heater made possible the early day operation of the solar desiccant cooling system. As shown in Fig. 5, the off-peak electric energy consumption reached more than 55% of the entire energy consumed in the operation of the solar desiccant cooling system (For cities of Yagon, Manila and Dili). These cities had high solar radiation with outdoor air temperature and humidity. Seoul has lower OPEC value due to lower solar radiation during the operation and it means daytime electric heater operates to compensate the system thermal requirement.

The evaluation of the thermal energy requirement was based on the solar energy contribution – solar fraction. It showed the high solar fraction was possible in the southern region (South East Asia) which reached 90% compared to the East Asia (less than 80%).

The system thermodynamic performances were presented in Fig. 6 – desiccant coefficient of performance (DCOP) and the system coefficient of performance (SCOP). The desiccant cooling subsystem performance – the desiccant coefficient of performance (DCOP) showed it had value of between 1 and 0.5 (Fig. 6). It means it needed greater amount of thermal energy to produce cooling load. However, as shown in Fig. 5, the solar energy contributed sizable amount of thermal energy requirement of the system. This is due to the contribution of solar fraction (between 70% and 90%).

As presented in the results, the system coefficient of performance was high in the South East Asia compared to the East Asia. The SCOP reached above 3 in Brunei. As shown in Fig. 3, Brunei has highest outdoor air absolute humidity content. The solar fraction in Brunei reached 90% (See Fig. 5). The solar thermal desiccant cooling system has low SCOP in Seoul (SCOP <1.5) due to low available solar radiation. This resulted to high consumption of thermal energy from electric heater (daytime).

## Conclusions

The paper showed the potential and applicability of the solar thermal desiccant cooling system in the hot and humid Asia-Pacific Region (East Asia and South East Asia). The studies focused on the application of the developed solar thermal desiccant cooling system based on the numerical investigation in transient system simulation (TRNSYS) program.

The results showed the needed area of the flat plate collector varied depending on locations with available solar radiation, required air flow rates, and dehumidification rates. In addition, the needed air flow rate to maintain the indoor thermal comfort depends on the available solar radiation, outdoor air temperature, and humidity content.

In general, the presented solar thermal desiccant cooling system maintained the required indoor thermal comfortable conditions in the Asia-Pacific Region (East Asia and South East Asia) with specifications dependent to the region’s different climatic conditions.

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## Figures:

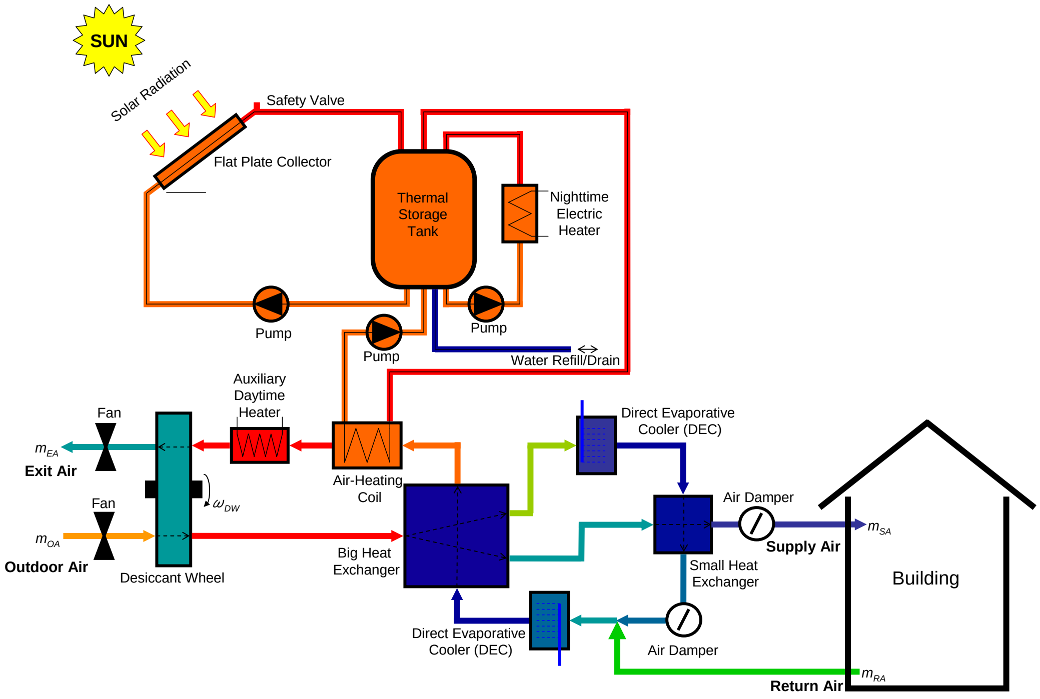

**Figure 1** Solar thermal desiccant cooling system used in the study.

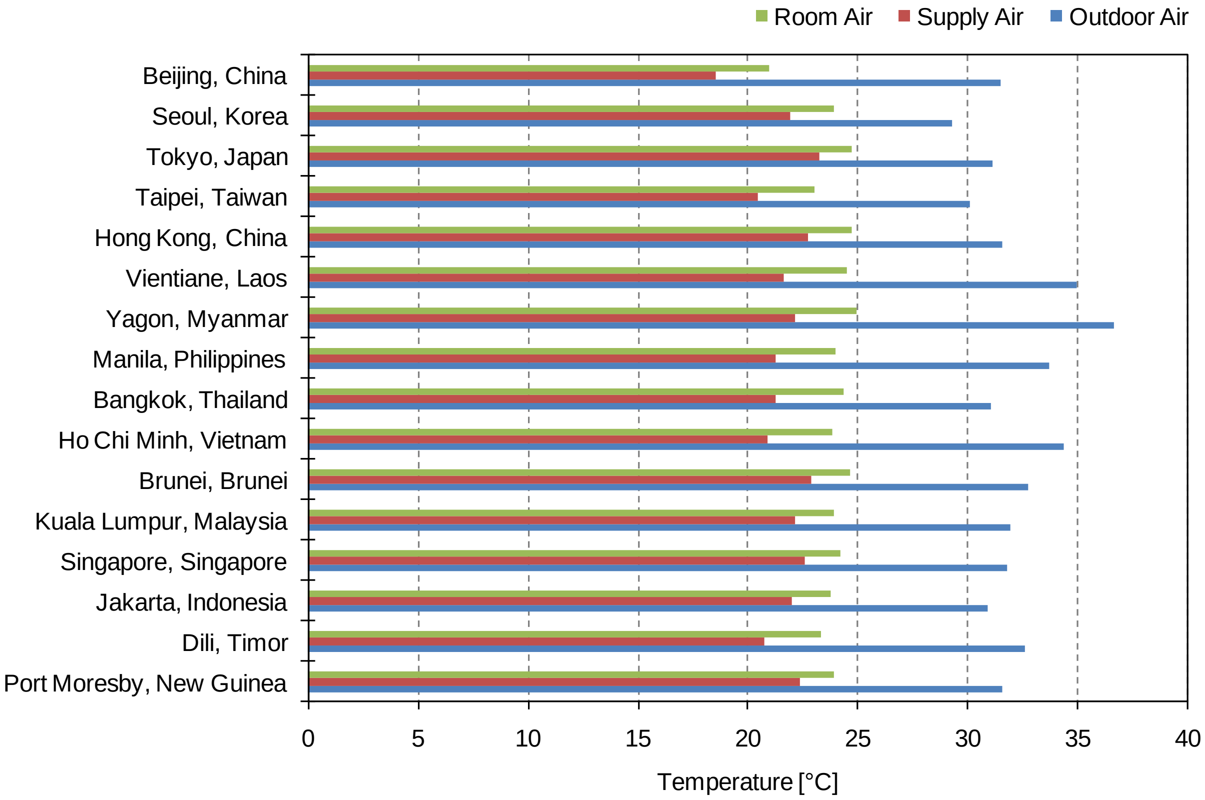

**Figure 2** Air temperatures at different locations and weather conditions in the Asia-Pacific Region (East Asia and South East Asia).

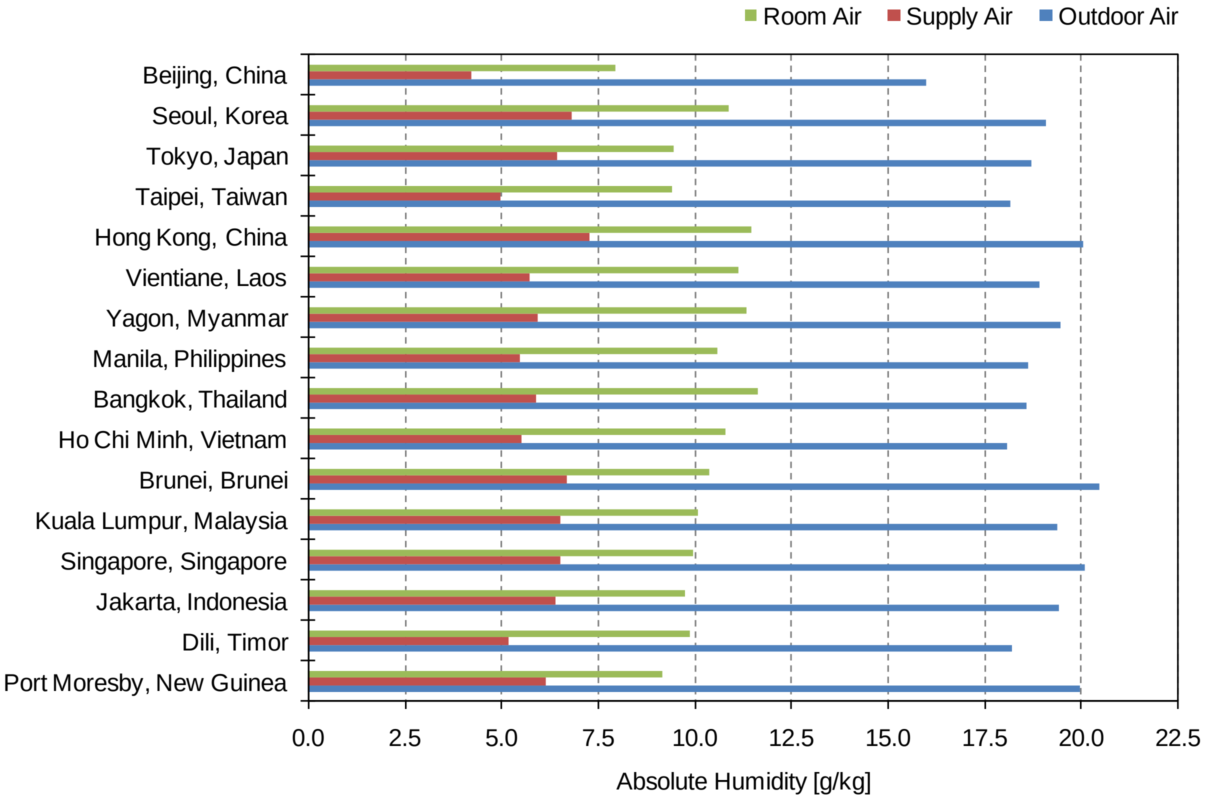

**Figure 3** Air absolute humidity at different locations and weather conditions in the Asia Pacific Region (East Asia and South East Asia).

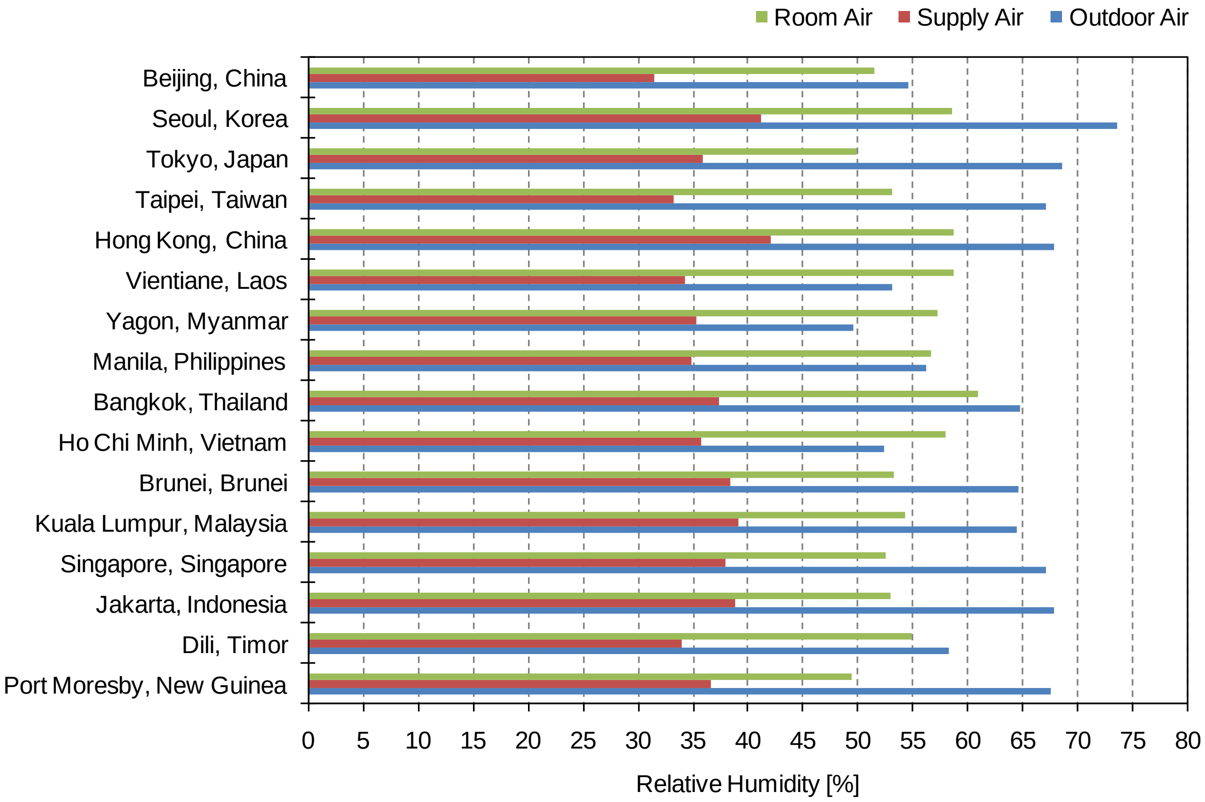

**Figure 4** Air relative humidity at different locations and weather conditions in the Asia-Pacific Region (East Asia and South East Asia).

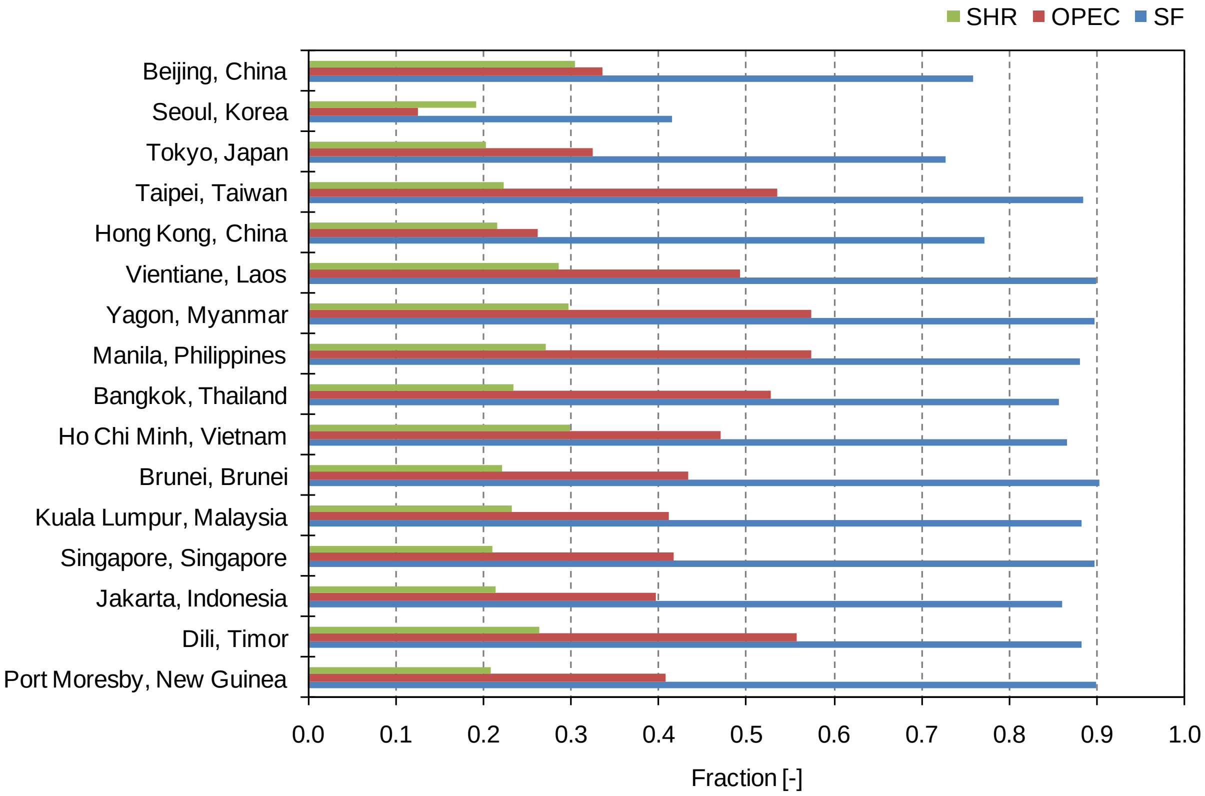

**Figure 5** System sensible heat ratio, off-peak electric energy consumption and solar fraction at different locations and weather conditions in the Asia-Pacific Region (East Asia and South East Asia).

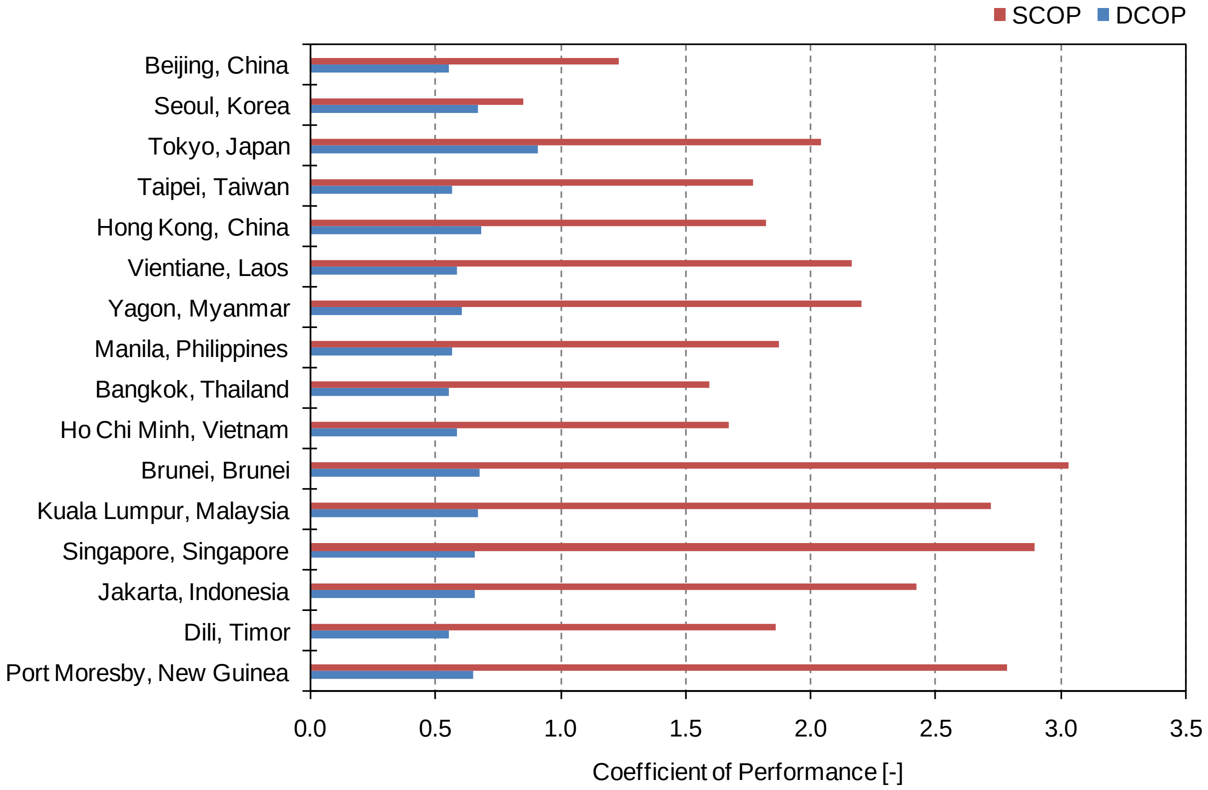

**Figure 6** System coefficient of performance at different locations and weather conditions in the Asia-Pacific Region (East Asia and South East Asia).

**Table 1** Asia-Pacific Region (East Asia and South East Asia) major countries and cities

| Country | City | Latitude | Longitude |
|---|---|---|---|
| China | Beijing | 39° 54′ 50″ N | 116° 23′ 30″ E |
| Korea | Seoul | 37° 34′ 8″ N | 126° 58′ 36″ E |
| Japan | Tokyo | 35° 42′ 2″ N | 139° 42′ 54″ E |
| Taiwan/China | Taipei | 25° 2′ 0″ N | 121° 38′ 0″ E |
| China (SAR) | Hong Kong | 22° 15′ 0″ N | 114° 10′ 0″ E |
| Laos | Vientiane | 17° 58′ 0″ N | 102° 36′ 0″ E |
| Myanmar | Yagon | 16° 48′ 0″ N | 96° 9′ 0″ E |
| Philippines | Manila | 14° 35′ 0″ N | 120° 58′ 0″ E |
| Thailand | Bangkok | 13° 45′ 8″ N | 100° 29′ 38″ E |
| Vietnam | Ho Chi Minh | 10° 46′ 10″ N | 106° 40′ 55″ E |
| Brunei | Bandar Seri Begawan | 4° 53′ 25″ N | 114° 56′ 32″ E |
| Malaysia | Kuala Lumpur | 3° 8′ 0″ N | 101° 42′ 0″ E |
| Singapore | Singapore | 1° 17′ 0″ N | 103° 50′ 0″ E |
| Indonesia | Jakarta | 6° 12′ 0″ S | 106° 48′ 0″ E |
| East Timor | Dili | 8° 34′ 0″ S | 125° 34′ 0″ E |
| Papua New Guinea | Port Moresby | 9° 25′ 0″ S | 147° 17′ 0″ E |

**Table 2** Specification of the solar thermal desiccant cooling system at different locations and weather conditions

| | Flate Plate Collector | | Air Flow Rate | |
|---|---|---|---|---|
| | Inclination [°]* | Area [m²] | Outdoor Air [m³/h] | Supply Air [m³/h] |
| Beijing, China | 39 | 8 | 330 | 165 |
| Seoul, Korea | 37 | 10 | 330 | 165 |
| Tokyo, Japan | 35 | 8 | 330 | 165 |
| Taipei, Taiwan | 25 | 8 | 200 | 100 |
| Hong Kong, China | 22 | 8 | 330 | 165 |
| Vientiane, Laos | 17 | 8 | 200 | 100 |
| Yagon, Myanmar | 16 | 8 | 200 | 100 |
| Manila, Philippines | 14 | 8 | 200 | 100 |
| Bangkok, Thailand | 13 | 8 | 200 | 100 |
| Ho Chi Minh, Vietnam | 10 | 8 | 200 | 100 |
| Brunei, Brunei | 4 | 12 | 330 | 165 |
| Kuala Lumpur, Malaysia | 3 | 12 | 330 | 165 |
| Singapore, Singapore | 1 | 12 | 330 | 165 |
| Jakarta, Indonesia | 6 | 12 | 330 | 165 |
| Dili, Timor | 8 | 8 | 200 | 100 |
| Port Moresby, New Guinea | 9 | 12 | 330 | 165 |

**Facing Equator*