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Site verification and modeling of desiccant-based system as an alternative to conventional air-conditioning systems for wet markets

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A B S T R A C T

Desiccant cooling system for active humidity control and to conserve energy has been in commercial applications for over two decades. However, its use in humid wet markets has never been examined. A gas-fired desiccant cooling system has been installed in a wet market in Hong Kong. In this study, the annual energy saving in conjunction with the use of desiccant cooling system was investigated by in-situ measurements, site surveys and simulations. The verified computer model was used for further simulations. It was found that for the use of a minimum ventilation rate of 10.3 L/s/person, the use of desiccant cooling system as compared to conventional system saved 4% of the energy and could achieve the desired space conditions. A parametric study under various ventilation rates indicated that use of desiccant cooling system in wet markets in hot and humid Hong Kong would lead to energy and energy cost savings, as well as CO₂ emission reduction amounting from 1% to 13%. The savings were more evident when wet markets were designed for a ventilation rate of 20 L/s/person. Furthermore, the actual occupancy profile, and lighting and small power densities determined in this study would be useful for future studies on wet market.

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1. Introduction

In most commercial and residential applications, the space relative humidity is not controlled because conventional air-conditioning systems rely on operating the cold coil at very low effective temperatures to enable the removal of moisture by condensation. As such, the desired space relative humidity can only be achieved by matching the equipment (sensible heat ratio) SHR with the building SHR, but a perfect match rarely happens. This divergence leads to excessive cooling which increases energy use, and moisture problems which accelerate microbial growth, particularly for subtropical climates like Hong Kong with very humid outdoor air.

In solving the excessive cooling and moisture problems, previous studies [1–3] advocate decoupling of dehumidification from cooling for better humidity control and energy performance. Desiccant cooling system, which can actively control the humidity level, is often used for achieving the dehumidification objective. Recent studies on desiccant cooling system have been done on optimizing the operating variables for better energy performance [4–7]; developing empirical model for investigation of the system performance [8]; improving the control and operation by experimental studies [9,10]; and widening the application to different types of buildings [11–14]. However, it is noted that they are either restricted to applications where ventilation requirements are set to remove heat generated by people, lighting and equipment, or where moisture removal is not a design intent [13,14]. As for applications where ventilation requirements are set for moisture and contaminates removal, little has been found in extant literature.

The recent occurrence of SARS (severe acute respiratory syndrome), avian flu and anthracnose in some countries makes environmental health one of the most important elements for an air-conditioning system [15]. In achieving the environmental health objective, ventilation requirement for some premises is not just for heat removal, but for moisture and contaminant removal. The wet-market in Hong Kong is one typical example.

Modernized wet markets in Hong Kong are housed in air-conditioned buildings. They traditionally sell fresh meat, fruits and vegetables, and live animals. Live animals include fish, shellfish, and poultry. Because water is used to wash the floors, keep the fruits and vegetables fresh, and keep animals alive, wet markets are an extremely damp place. For moisture and contaminant removal, wet markets often require very high ventilation rate and this, combined with the humid outdoor air, increases the latent load and thus a huge amount of energy will be consumed if conventional air conditioning system is used.
A gas-fired desiccant cooling system has been installed in a wet market in Hong Kong which may probably be the first installation of its kind in the world. The actual operating data of the pilot installation will be used to validate simulation predictions. Upon successful validation, further simulations will be conducted to evaluate the overall performance of desiccant cooling system as compared to a conventional air-conditioning system. Given designers are free to design the ventilation rates they deem proper for wet markets, the influence of ventilation rates on the energy and energy cost savings, as well as CO₂ emission reduction of desiccant cooling system will further be investigated.

2. System description

Desiccant dehumidification differs from conventional air-conditioning system in terms of the thermodynamics of the process air. Instead of cooling the air below its dew point to condense its moisture, desiccants attract moisture from the air. Desiccant has a vapor pressure lower than the air in its active state. This vapor pressure differential drives the water vapor from air onto the desiccant [16]. Figs. 1 and 2 show the psychrometric processes of the two systems.

A conventional system typically admits (outdoor air) OA at state 0 to mix with return air at state R to become mixed air (state 1). The mixed air will subsequently be cooled to state 2 followed by a reheating process to become supply air (state 3) for achieving the desired space temperature and relative humidity (state R). For the removal of moisture by condensation, state 2 is often conditioned to a very low temperature (10−12°C dry-bulb) and is close to saturation (90−95% relative humidity).

In a desiccant cooling system, the latent cooling is performed by desiccant dehumidification whilst the sensible cooling is treated by a cold-coil air handler. The OA at state 0 is first brought into the gas-fired desiccant dehumidifier and is dehumidified adiabatically to state 1. The dried and hot OA is then pre-cooled by the exhaust air at state R through an air-to-air heat exchanger to state 2. The pre-cooled OA is subsequently mixed with the return air to state 3 followed by a cooling process at the cold-coil air-handler (state 4) to become supply air conditions for achieving the desired space conditions (state R). The exhaust air leaving the air-to-air heat exchanger at state 5 is heated up at the gas burner to become hot air (state 6) for regeneration of the desiccant dehumidifier. Warm and humid air at state 7 will finally be exhausted to the environment.

In the regeneration process of the desiccant dehumidifier, the desiccant wheel rotates slowly into the stream of gas-fired hot air (state 6), which raises the vapor pressure of the desiccant above that of the surrounding air. With the vapor pressure differential reversed, water vapor moves from the desiccant to the regeneration air, which carries the moisture away from the system.

The schematic of the air flow of the desiccant cooling system is shown in Fig. 3.

3. Methodology

This study evaluates the overall performance of a gas desiccant cooling system as compared to a conventional air-conditioning for application in a case study wet market. The influence of ventilation requirements on the energy and energy cost savings, as well as CO₂ emission reduction of desiccant cooling system will also be evaluated. Evaluations were based on in-situ measurements, site surveys and energy simulations. In-situ measurements results were retrieved from the remote monitoring system. The retrieved results include the space air conditions and the year round energy use derived from fans (process and regeneration), chilled water circulating pumps, chillers and burners of the gas-fired dehumidifiers.

Given key parameters like occupancy, and installed lighting and small power densities are not available; site surveys have been conducted to ascertain these parameters. The in-situ measurements and site surveys results were used for simulation validation to facilitate further simulations.

A simulation program, EnergyPlus was adopted to simulate the energy use of the major equipments of the desiccant cooling system for energy analysis.

A hypothetical wet market (not the studied wet market) was designed to represent typical modernized wet markets in Hong Kong to facilitate the evaluations. It was formulated after an
extensive survey of physical characteristics of wet markets in Hong Kong.

4. The case study market and system characteristics

The case study market is housed on the ground floor of a 5-storey carport in a public housing estate. Car parking spaces are provided from 1/F to 4/F with a playground on the roof. The total floor area of the market is 2380 m² (53.0 m × 44.9 m). The headroom is 3.5 m high. It is a typical wet market in Hong Kong which sells all sorts of fresh produce and live seafood and poultry.

The desiccant cooling system installed in this market consists of two parallel systems: a dehumidification system comprises of three units of gas-fired desiccant dehumidifiers installed on the roof dedicated to dehumidify OA, and a chilled water system. The chilled water system is not different from a conventional air-conditioning dedicated to dehumidify OA, and a chilled water system. The chilled water system is not different from a conventional air-conditioning system which consists of two chillers each of 915 kW cooling capacity and the associated circulation pumps. The capacity is sufficient to cover the sensible and latent loads of the three air-handlers for treatment of mixed air with and without the operation of the desiccant dehumidifiers. This serves as a standby provision to ensure relative humidity of the wet market can always be met. To satisfy the minimum ventilation requirement for non-domestic buildings in Hong Kong [17], the market is designed for a ventilation rate of 5 air change per hour (ACH).

The building, design and system characteristics of the case study wet market are summarized in Table 1.

### Table 1
Summary of building, design and system characteristics.

| Description                      | Details                      |
|----------------------------------|------------------------------|
| **Building characteristics**     |                              |
| Air-conditioned area (m²)        | 2380                         |
| Window-to-wall ratio             | 0.4                          |
| Glass characteristics            | 10 mm clear glass            |
| Wall characteristics             | 600 mm thickness             |
| **Design characteristics**       |                              |
| Temperature (°C)                 | 25                           |
| Relative humidity (%)            | 65                           |
| Ventilation rate (ACH)           | 5                            |
| Infiltration rate (ACH)          | 0.1                          |
| **System characteristics**       |                              |
| Desiccant system                 |                              |
| Number of dehumidifiers          | 3                            |
| Desiccant materials              | Silica gel                   |
| Regeneration source              | Town gas burner              |
| Rated moisture removal rate (kg/hr) | 238                  |
| Rated process air flow (kg/s)    | 5.64                         |
| Rated gas input (KW)             | 100                          |
| Regeneration fan efficiency (%)  | 65                           |
| **Heat wheel system**            |                              |
| Flow arrangement                 | Counter flow                 |
| Power input (KW)                 | 0.035                        |
| **Chilled water system**         |                              |
| Chillers                         |                              |
| Number of chillers               | 2                            |
| Heat rejection method            | Air-cooled                   |
| Rated cooling capacity (KW)      | 915                          |
| Measured COP                     | 1.9                          |
| **Pumps**                        |                              |
| Number of units                  | 3 (one Standby)              |
| Pumping system                   | Single-loop                  |
| Rated flow rate (L/s)            | 45                           |
| Rated pump head (kPa)            | 225                          |
| Rated pump power (KW)            | 14.5                         |
| Efficiency (%)                   | 70                           |
| **Air handling units**           |                              |
| Number of units                  | 3                            |
| Rated cooling capacity (KW)      | 600                          |
| Fan efficiency (%)               | 65                           |

5. Simulation validation

In-situ measurements and site surveys, with details as discussed below, were adopted to collect data for simulation validation.

5.1. In-situ measurements

A remote monitoring system is deployed to monitor the performance of the desiccant cooling system. It can relay commands signals to carry out remote control of the dehumidifiers. The operation of the system can be viewed either locally on site or in the remote office through the internet. The records (air states in hypothesis are with reference to Fig. 2) can be retrieved by this system which includes spaces air temperature and relative humidity (state R), dried and hot outdoor air temperature and relative humidity (state 1), hourly operation schedule and energy use of major equipments. The monitoring system also enables the set points of relative humidity to be adjusted actively in order to achieve individual needs at a different zone. The system configuration is shown in Fig. 4.

The indoor and outdoor air conditions and the energy use of major equipments were retrieved from the remote monitoring system for comparison with simulation results. Exemplary results in one summer period are shown in Fig. 5. The average indoor temperature and RH were 25.2 °C and 65.8% respectively.

5.2. Site surveys

In the site surveys, the wet market was divided into three zones. Each zone was assigned with a responsible person to record to the number of customers as well as stall retailers in every 60 min intervals, and to ascertain the installed lighting and small power densities.

Occupancy flow measurements were taken on three consecutive days to cover weekdays and weekends during which the occupancy levels were recorded from 7:00 to 19:00. The hourly average values are summarized in Table 2.

The (installed lighting power density) ILPD was ascertained by counting the numbers and types of luminaires installed in the market. There were altogether four types of luminaries including 60 W fluorescent lamps, 48 W fluorescent lamps, 400 W spot lights and 200 W light bulbs. The surveyed ILPD is summarized in Table 3.

The (installed small power density) ISPD was determined by a walk-through survey on the power ratings of the fixed and portable appliances including refrigerators, boilers, cooking devices, and ventilation fans. The surveyed ISPD is also summarized in Table 3.

5.3. Simulation

Energy simulation based on the site survey results (including occupancy, lighting and small power densities and occupation patterns), average space air-conditions as indoor set-point conditions (25.2 °C dry-bulb and 65.8% RH), building envelop details (Table 1), and physical geometry of the wet-market were input to EnergyPlus for simulating the annual energy use of the major equipments.

The power requirement of the chiller was calculated based on the DOE-2 model. The DOE-2 model is an empirical model and its mathematical expressions are shown in Equations (1)–(3), in which CAPFT is the cooling capacity factor; EIRFT is the energy input to cooling output factor; and EIRPLR is the energy input to cooling output factor at part load condition [18–20].
\[
\text{CAPFT} = a_1 + b_1 (T_{cw,s}) + c_1 (T_{cw,s})^2 + d_1 (T_{a,e}) + e_1 (T_{a,e})^2 + f_1 (T_{cw,s})(T_{a,e})
\]
\[
\text{EIRFT} = a_2 + b_2 (T_{cw,s}) + c_2 (T_{cw,s})^2 + d_2 (T_{a,e}) + e_2 (T_{a,e})^2 + f_2 (T_{cw,s})(T_{a,e})
\]
\[
\text{EIRFPLR} = a_3 + b_3 (\text{PLR}) + c_3 (\text{PLR})^2
\]

where $T_{cw,s}$ is the supply chilled water temperature; $T_{a,e}$ is the condenser air entering temperature, which is the hourly weather conditions of Hong Kong in 1995 (the typical meteorological year.

**Table 2**
Occupancy flow in the wet market.

| Hour | Weekday | Weekend |
|------|---------|---------|
| 7:00 | 216     | 204     |
| 8:00 | 275     | 254     |
| 9:00 | 330     | 291     |
| 10:00| 304     | 276     |
| 11:00| 257     | 253     |
| 12:00| 208     | 195     |
| 13:00| 212     | 201     |
| 14:00| 269     | 248     |
| 15:00| 260     | 249     |
| 16:00| 299     | 285     |
| 17:00| 326     | 296     |
| 18:00| 289     | 267     |
| 19:00| 215     | 208     |
for Hong Kong [21]; PLR refers to the chiller's Part Load Ratio; \( a_1 \) to \( f_2 \), \( a_2 \) to \( f_2 \) and \( a_3 \) to \( c_3 \) are the default coefficients in EnergyPlus. CAPFT, EIRFT and EIRFPLR equal to 1 (dimensionless) at reference condition. Chiller capacity control is to maintain the designed \( C_{\text{eqw}} (= 7 \degree \text{C}) \) at constant flow condition.

The energy consumption of the desiccant dehumidifier was simulated based on a default performance model in EnergyPlus. It is a generic model for determining the leaving dry bulb temperature (\( T_l, \degree \text{C} \)) and humidity ratio (\( w, \text{kg/kg dry air} \)) of the process air and the specific regeneration energy (\( E_r, \text{J/kg H}_2\text{O} \)) and regeneration velocity (\( v_e, \text{m/s} \)) of the desiccant wheel shown mathematically in Equation (4).

\[
X = C_1 + C_2(T_e) + C_3(T_e)^2 + C_4(w_e) + C_5(w_e)^2 + C_6(v) + C_7(v)^2 + C_8(T_e)(w_e) + C_9(T_e)^2(w_e)^2 + C_{10}(T_e)(v) + C_{11}(T_e)^2(v)^2 + C_{12}(w_e)(v) + C_{13}(w_e)^2(v)^2 + C_{14} \ln(T_e) + C_{15} \ln(w_e) + C_{16} \ln(v)
\]

where \( X \) is the modeled parameter (i.e. \( T_l, w_e, E_r \) and \( v_e \)); \( T_e \) and \( w_e \) are the process air entering dry bulb temperature (\( \degree \text{C} \)) and humidity ratio (\( \text{kg/kg dry air} \), which follow the hourly weather conditions of Hong Kong in 1995 (the typical meteorological year for Hong Kong [21]); \( v \) is the process air velocity (\( \text{m/s} \)); and \( C_1 \) to \( C_{16} \) are the default coefficients in EnergyPlus which correspond to the parameter modeled.

Equation (4) is valid for the following range of process inlet conditions: dry-bulb temperatures of 1.67–48.9 \( \degree \text{C} \) and humidity ratios of 0.002857 kg/kg to 0.02857 kg/kg. The output of desiccant dehumidifier is to maintain the leaving process air staying at the designed outlet conditions, which in this analysis is moisture content below 0.004 kg/kg dry air.

In the simulations, a perfect control and feedback system was assumed such that the output of the equipments would match perfectly with the system cooling demand. Table 4 compares the actual and simulated energy use results, illustrating that the difference between the two sets of results is between -3.3% and +5.8%. The difference is considered acceptable considering that the best achievable uncertainty in the measured energy use as estimated based on the Kline and McClintock's equation [22,23] is ±0.6% (assuming all instruments conform to ASHRAE Standard 114-1986). The comparison of energy breakdown of major equipments in Table 5 also shows that their difference is ±2%. To check consistency of the two sets of energy breakdown, bivariate correlation was conducted. It was found that the correlation coefficient was 0.996 to indicate correlation was significant at the 0.05 level (2-tailed).

To compare the actual and simulated indoor conditions, their standard errors with the set-point conditions in one summer week period (1st to 7th July) are compared in Table 6. Their standard errors are shown to be comparable, illustrating the simulated results agree well with the actual conditions.

Given the acceptable range of the difference and the fundamental consistency in the two sets of data, validations are regarded as acceptable to conclude that it is reasonable to adopt simulations for further investigations.

6. Energy use comparison

Further simulations were conducted to compare the energy performance of desiccant cooling system with the conventional system. In this comparison study, the actual occupancy (Table 2), the measured lighting and small power densities (Table 3), and the installed equipment characteristics of the studied wet market were used in the simulation, but not the space air conditions and ventilation rate for which the ASHRAE recommendations could not be fulfilled [24].

According to ASHRAE [25], the indoor air condition in summer should have a wet bulb temperature below 20 \( \degree \text{C} \) and dry bulb temperature between 22.5 \( \degree \text{C} \) and 27 \( \degree \text{C} \). Thus for the actual space air temperature of 25.2 \( \degree \text{C} \), the corresponding relative humidity should be set at 50%. They were therefore set as indoor set-point conditions in the simulation.

Whilst for ventilation rate, although there has been no guideline for wet markets, it is noted that the actual rate of 5 ACH (corresponds to approximately 3.3 L/s/person) is far lower than the ASHRAE recommended rate for supermarkets of 7.6 L/s/person [24]. Given wet markets are much more humid than supermarkets, to avoid the possible microbial growth, ASHRAE's highest recommended rate for the same air class, i.e. 10.3 L/s/person, has been adopted in the simulation.

With regards to the equipment characteristics, the desiccant cooling system was assumed provided with the actual installed equipments (Table 1), only that our pilot simulation indicated that

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**Table 4**

| Equipment    | Fuel       | Actual data | Simulated data | Difference (%) |
|--------------|------------|-------------|----------------|---------------|
| Fan          | Electricity| 1495.7      | 1582.6         | 5.8           |
| Pump         | Electricity| 175.8       | 176.8          | 0.6           |
| Chiller      | Electricity| 2632.5      | 2677.6         | 1.7           |
| Burner       | Gas        | 2488.2      | 2407           | -3.3          |
| Total energy use |          | 6792.2      | 6844           | 0.8           |

**Table 6**

| Data            | Space air temperature error | Space relative humidity error |
|-----------------|-----------------------------|-------------------------------|
| Actual          | 0.62                        | 2.46                          | 2.47                        | 3.75                        |
| Simulated       | 0.53                        | 2.05                          | 1.25                        | 1.90                        |
with lowering the space relative humidity, the moisture removal rate of the desiccant dehumidifier was too small and thus an electric heater was added to increase the temperature of the regeneration air for supplementing the dehumidification requirements. Electric heater was chosen for their common use in buildings in Hong Kong [26]. The electric heater capacity was 173 kW determined using EnergyPlus auto size option.

The conventional air-conditioning system has been assumed identical to the installed chilled water system (Table 1). However, to enable a side-by-side comparison of the energy performance of the two systems, despite the fact that conventional air-conditioning systems control only the space air temperature, it has been assumed in the simulation that chilled water flow rate of the conventional system can be varied to cool and dehumidify the supply air to a level for achieving the desired space air temperature as well as relative humidity; whilst overcooling is assumed offset by electric reheating to maintain the space air temperature. The electric heater capacity was again determined using EnergyPlus auto size option.

Through hour-by-hour simulations, the year-round space air temperatures and relative humidities achieved by the use of the desiccant cooling and conventional systems were found. The standard error between the simulated results and the set-point conditions for the two systems is summarized in Table 7. It can be seen that their standard errors are comparable to conventional systems, accounting for more than 40% of the total energy use. The evaluations confirm that the simulation results are reasonable, and small energy saving can be achieved by the use of desiccant cooling system.

7. The parametric study

The parametric study is to enable the study of the influence of ventilation requirements on the energy performance, CO2 emission level, and energy cost saving of desiccant cooling system.

7.1. The hypothetical wet market

To provide a basis for the parametric study, a hypothetical wet market representing typical wet markets in Hong Kong was formulated. A survey of physical characteristics of 19 modernized wet markets in Hong Kong was conducted. It was found that their areas were ranging from 800 m² to 18,650 m², with a mean of 4,140 m²; the aspect ratios (ratio of width to length) were between 0.4 and 0.8, with a mean of 0.625; and the headrooms were between 3 m and 4.5 m, with a mean of 4 m. Considering the collected data are of equal importance and should be given equal weight, mean values were used to set physical dimensions (W × L × H) of the hypothetical wet market which is 30 m × 48 m × 4 m.

7.2. Range of variations

Given it is common to see humid environment is provided with 100% outdoor air for control of airborne infectious diseases [27], the range of variations has been set between 10 L/s/person at 10 L/s intervals and complete outdoor air flow rate corresponding to 58 L/s/person. In the parametric study, the equipment capacities were determined using EnergyPlus auto size option; only that the rated performance of major equipments including chillers, fans, and with lowering the space relative humidity, the moisture removal rate of the desiccant dehumidifier was too small and thus an electric heater was added to increase the temperature of the regeneration air for supplementing the dehumidification requirements. Electric heater was chosen for their common use in buildings in Hong Kong [26]. The electric heater capacity was 173 kW determined using EnergyPlus auto size option.

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pumps, burners and desiccant dehumidifiers were assumed identical to those installed at the studied wet market.

7.3. Energy saving

Fig. 7 indicates the energy use comparison between the two systems under various ventilation rate conditions. It is noted that the energy use of both systems is comparable when the ventilation rate is at 10 L/s/person which is consistent with the comparison study result in the earlier section. This again confirms the reliability of the parametric study results. Parametric study results revealed that further increase in ventilation rate lead to a steady increase in energy use. However, the rate of rise for the conventional system was higher than that of the desiccant cooling system when the ventilation rate was increased from 10 L/s/person to 20 L/s/person. The results indicate that at a ventilation rate of 20 L/s per person, energy saving of 8% can be achieved by the use of desiccant cooling system. The significant saving can be explained by the fact that the increase in ventilation rate can offset part of the space cooling load in mild season, but further increase in ventilation rate leads to a sharp increase in space cooling load as shown in Fig. 8.

8. Prediction of other savings

Based on the energy consumptions of the conventional ($E_{CR}$, kWh) and desiccant cooling ($E_{DC}$, kWh) systems; and the energy split between gas ($E_{DC,gas}$) and electricity ($E_{DC,elc}$) of desiccant cooling system, the percentage energy cost saving and emission reduction ($P$) can be predicted based on the use of Equation (5) derived as follows:

$$P = \frac{C}{a \times E_{CR}} \times 100\%$$  \hspace{1cm} (5)

and:

$$C = a \times E_{CR} - (a \times (1 - w) \times E_{DC} + b \times w \times E_{DC}) \quad (6)$$

where $C$ is the energy cost saving/CO2 emission reduction between the use of desiccant cooling system and conventional system.

Using Equations (5) and (6) becomes:

$$P = \frac{1 \times (1 - w + \mu \times w) \times E_{DC}}{E_{CR}} \times 100\%$$  \hspace{1cm} (7)

where:

$$w = \text{the ratio of gas consumption to total energy consumption of desiccant cooling system, } (E_{DC,gas}/E_{DC})$$

$$1 - w = E_{DC,elc}/E_{DC}$$

$$a = \text{the tariff/EEF per kWh of electricity}$$

$$b = \text{the tariff/EEF per kWh of town gas}$$

$$\mu = \text{the ratio of } b \text{ to } a$$

8.1. Energy cost saving

In Hong Kong, two private sector power companies, the (China Light and Power Co. Ltd.) CLP and the (Hong Kong Electric Co. Ltd.) HEC have the monopoly right to generate, distribute and sell electricity, each to a different part of Hong Kong, whilst the Hong Kong and China Gas Company Limited (Towngas) is the only utility supplying gas to Hong Kong.

According to CLP and Towngas, the electricity and town gas tariffs are $0.86/kWh and $0.6/kWh respectively, and thus ($\mu$) in Equation (3) is 0.698 ($=0.6/0.86$).

Fig. 9 compares the annual energy costs of the two systems. It is noted that there is larger energy cost saving as compared to energy saving for the desiccant cooling system because gas tariff is lower. The saving ranged between 4% and 10%, and the optimal saving, as expected, occurred when the ventilation rate was at 20 L/s/person.
The environmental impact is assessed based on (carbon dioxide) CO2 saving because the amount of CO2 in the atmosphere accounts for about 50% of the global warming effect [28]. The (effective emission factor) EEF for electricity generation in Hong Kong based upon the use of different fuels is estimated to be 0.615 kg/kWh, and that for gas is 0.238 kg/kWh [29]. To estimate the amount of CO2 saving, (μ) in Equation (3) is 0.387 (=0.238/0.615).

Fig. 10 compares the annual CO2 emission level of the two systems. It is noted that similar to energy cost saving, there is larger emission reduction for the desiccant cooling system as compared to energy saving because the effective emission factor for gas is much lower. The reduction ranged between 6% and 13%, and again, the optimal reduction occurred when the ventilation rate was at 20 L/s/person.

9. Conclusion

A desiccant cooling system was installed in a wet market in Hong Kong. In this study, the actual energy performance of the desiccant cooling system was evaluated. Evaluations indicated that use of gas desiccant cooling system was an effective and flexible mean for providing active humidity control. Further evaluations based on in-situ measurements, site survey and simulation studies were conducted. In-situ measurements provided the actual energy use data for comparison with simulated results and for validation simulation. Further simulations based on actual system, equipment and operating characteristics of the studied wet market were conducted for prediction of energy benefits of desiccant cooling system as compared to conventional air-conditioning system, which was found to be 4%.

To evaluate the influence of ventilation rates on the associated energy and energy cost savings as well as CO2 emission reduction of desiccant cooling system, a parametric study under various ventilation rates was performed. The parametric study was conducted based on a hypothetical wet market formulated to represent typical physical characteristics of wet markets in Hong Kong. The study results revealed that use of desiccant cooling system in wet markets in hot and humid Hong Kong would lead to energy, energy cost and CO2 emission reduction up to 13% when the ventilation rate is 20 L/s/person. The results of this study would be useful for establishing essential criteria for the use of supplementary energy source with respect to air-conditioning system design.

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