Solar Air Conditioning for an Institutional Building in Subtropical Climate

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Abstract: - Air conditioning is one of the major consumers of electrical energy. The most of the ways of generating the electricity today, as well as the refrigerants being used in traditional vapour compression cooling system, produce greenhouse gas emissions which ultimately contribute to global warming. It is therefore necessary to develop process and technology to implementing renewable sources of energy for air conditioning to reduce greenhouse gas emissions and to achieve sustainable development. The use of solar energy to drive cooling cycles for space conditioning is relatively a new and attractive concept which mostly eliminates the need for CFC, HCFC or HFC refrigerants. In this presentation an overview of a hybrid solar desiccant cooling system which has been designed and installed in an institutional building of Central Queensland University, Rockhampton campus, Australia is presented. The conceptual bases of the technology, capability and limitations are outlined. The energy demand, energy consumption, and economic and environmental problem associated with the usage of fossil fuel resources in Australian commercial buildings and the issues of indoor air quality, mould growth and indoor thermal comfort are discussed. Furthermore, experimental and computational results of the performance of installed solar desiccant cooling system is presented and discussed. The results are analysed on the basis of energy savings, solar fraction (*SF*), primary energy used, coefficient of performance (*COP*) and desiccant system efficiency. Results showed that the installed solar desiccant cooling system at Central Queensland University can achieve energy savings of 19% with maximum coefficient of performance of 0.83 and desiccant efficiency of 48%.

Key-Words: Solar air conditioning, energy savings, subtropical climate, reduction of greenhouse gas emission

1 Introduction

Australia has a very sunny climate, with a very high demand for air conditioning. Relying on electricity to drive, buildings' HVAC systems will cause a significant negative impact on the environment. Relying on fossil fuel energy recourses to generate electricity is affecting global warming directly due to fossil fuel burning's high negative impact on the environment. Consequently, global warming became the most common dilemma facing the world at the present time.

Institutional buildings contain different types of functional spaces. Lecture theatres, libraries and laboratories are the most important facilities within institutional buildings, and they are usually the largest air conditioned areas which host daily students and staff activity, machinery and instruments. Institutional buildings have a very high occupational density compared to other commercial buildings [1]. This high occupancy density generates a high heat gain as well as a high emission of body odours and water vapour. It is known that the human body has a constant temperature of 36-37 °C, independent of surrounding conditions and muscle activities. As a consequence, the human body has to transmit the excess heat to the environment by means of different heat transfer mechanism. This excess heat consists of latent and sensible heat. The sensible heat is transferred by means of convection and radiation from the human body to its surroundings, while latent heat is transferred to the surrounding by diffusion of vapour through skin and exhaled air [2].

The balance between thermal comforts, indoor air quality and energy usage are building designers' main concern. Most of the research is concerns with institutional buildings are dedicated to energy savings through building construction specifications e.g. insulation and shadings and HVACs' systems performance [3]. The ordinary practice to remove contaminants and pollutant from institutional buildings is through ventilation control with active

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heating and cooling systems which causes a major energy draw. Institutional building indoor environment (sound, temperature, humidity and indoor air quality) must be fiscally and environmentally balanced. However to maintain this necessary balance between indoor air quality and energy usage will force large amount of fossil fuel burning simply wasted.

Fortunately Queensland (Australia) has one of the world's best solar resources. According to [4], Queensland has one of the highest solar energy concentrations in the world. The annual average global solar irradiance in Central Queensland region is 5.8 kWh/m^2/day [4]. Hence using solar energy to generate cooling is a very attractive concept, since in most of solar assisted air conditioning systems, solar heat is required to drive the cooling process, and this can be done by collecting solar radiation using solar collectors to convert it into thermal energy, this energy is then used to drive thermally driven cooling cycles such as desiccant, absorption and adsorption cycles.

Solar assisted air conditioning is an ideal option to achieve a high solar fraction which leads to a significant amount of energy savings and avoided greenhouse gas emission. Solar assisted air conditioning systems are environmentally friendly by being constructed in a way that minimises the need for chlorofluorocarbons CFC, Hydro chlorofluorocarbons HCFC or Chlorofluorocarbons HFC refrigerants and by using a low grade thermal renewable energy. Additionally solar assisted air conditioning can be used either as stand-alone systems or with conventional HVAC, to save energy and to improve indoor air quality. Absorption and desiccant cooling technology are the most common technology applied to solar cooling.

Solar cooling techniques have been investigated for several years under various climatic conditions and different comfort level standards. Their energy savings, desiccant effectiveness and indoor air quality have been evaluated and analysed through a number of simulation and experimental studies.

The main principle behind desiccant cooling cycle is the system's capability of removing or reducing vapours and moisture out of the treated air using a physical sorption of desiccant materials [5]. According to [6], the technology is considered as the most suitable air conditioning systems that can be used within commercial buildings, particularly institutional buildings and health care buildings in order to reduce contaminated air transmissions

Most of the researches and publications concerned with institutional buildings energy performance have considered energy savings via specific construction features such as thermal insulation, thermal mass, shading and HVAC system efficiency and performance [7]. Solar assisted air cooling techniques have been investigated recently under various climatic conditions and different comfort level standards.

However, there are limited studies and research activities available in the literatures which are concerned with Australian climates. [8] has tested a solar liquid desiccant cooling system under Brisbane climatic conditions. [9] have analysed the performance of a combined solid desiccant and indirect evaporative cooler. [10] have modelled a solar desiccant cooling system in an office building without thermal backup in three Australian cities: Sydney, Melbourne and tropical Darwin. There are no research activities available on solar cooling systems in regional Australia. Consequently, achieving the important objective of reducing the state of Queensland's greenhouse gas emissions requires more relevant research activities, especially the ones concerning solar energy as the state of Queensland's solar irradiance is reasonably abundant [11].

In this research paper, a solar desiccant cooling system is investigated. The system is designed, installed, modelled and simulated at the Rockhampton campus of Central Queensland University, Queensland, Australia. The results of this study shall provide information, data, key measures and decision making tool for designers, developers and operators about solar desiccant air conditioning which can be operated under Central Queensland subtropical harsh climate. This in turn will help to develop a model for a broad range of buildings such as hospitals, health care units, institutional buildings, museums, libraries and other vital commercial buildings.

2 Solar Desiccant Cooling System

Solar desiccant cooling is effective when used in hot and humid climates since the prime feature of desiccant cooling system that they can treat sensible and latent heat load separately. Solar desiccant cooling system consists of three subsystems: solar energy system, the dehumidifier and a cheap cooling technique e.g. evaporative cooler. The desiccant dehumidifier can be controlled separately using a humidistat that measures and controls the humidity and latent load of a cooled space. The evaporative cooler is controlled using a thermostat to measure and control the sensible load. In solar desiccant cooling system, the cooling process starts in the dehumidifier as explained in Figure 1. The untreated supplied air is directed through a desiccant machine which will dry the air. Repeating the process multiple times, the desiccant material will get saturated (wet) and it will lose its sorption characteristics. Drying desiccant materials is performed to drive the moisture out of the desiccant material so it can again absorb moisture and water vapours out of the treated air in subsequent cycles. Drying the desiccant material is called regeneration. The regeneration cycle is done by heating the desiccant material until it reaches its regeneration temperature by using low grade thermal energy resources like solar energy and industrial waste heat.

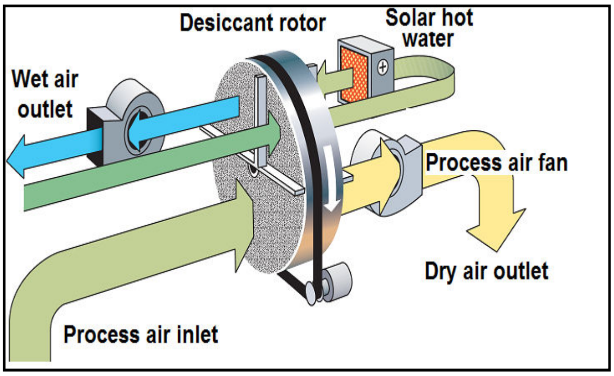

Fig.1: Desiccant cycle [12]

3 Performance parameters

The viability of this research project was assessed by means of the most commonly used performance parameters. These parameters are solar fraction, coefficient of performance, solar energy gain (energy savings), primary (parasitic) energy used, desiccant efficiency and regeneration efficiency. In this study a year round simulation and experimental results in prospective to the following indicators is used to assess the potential of solar cooling system in each climate.

3.1 Solar fraction

Solar fraction is considered as the most used technical indicator in order to evaluate the performance of solar cooling systems. Solar fraction (*SF*) measures the ratio of thermal energy produced by the solar collectors to the cooling system is total driving energy. Solar fraction depends on many factors such as load, collectors' area, hot water storage size, and availability of solar radiations. When the solar thermal energy *(Q_{solar})* is insufficient to drive the cooling process, a backup heater is used to deliver the required energy *(Q_{Aux})*. Therefore solar fraction (*SF*) can be expressed as in Equation 1 [13]

$$SF = \frac{Q_{Solar}}{Q_{Solar} + Q_{Aux}} \qquad (1)$$

3.2 Coefficient of Performance *(COP)*

Coefficient of performance (*COP*) is a general cooling system performance indicator. *COP* is defined as the ratio of cooling amount produced by cooling system to the total energy consumed by the cooling system. The cooling system with high *COP* is more efficient than the ones with lower one. The coefficient of performance (*COP*) for a conventional system (Vapour compression) is defined by Equation 2 [14]

$$COP = \frac{Q_{Ce}}{W_{el}} \qquad (2)$$

Where Q_{Ce} is refrigeration effect (kW) and W_{el} is cooling system total electric power input (kW)

Using solar desiccant cooling system, the coefficient of performance COP_{desi} is determined by Equations 3.

$$COP = \frac{\eta_h \times m_{\sup} \times \Delta h}{(m_{reg} \times \Delta h_{reg}) + Q_{eva} + Q_{el}} \times \eta_h \times \eta_s \qquad (3)$$

where Q_{ev} is the energy consumed by the evaporative cooler, , η_S is solar collectors efficiency, η_h is regeneration backup heater efficiency, m_{sup} is the mass flow of supply air, m_{reg} is the mass flow of regeneration air, Δh is the enthalpy difference between outside and supply air and Δh_{reg} is the enthalpy rise in the heater for the regeneration.

3.3 Energy Savings and Primary Energy

The system primary energy is used by the backup heaters, the hot water pump, the cold water pump, the generator pump, the dehumidifier pump, the evaporative cooler pump and the reference conventional HVAC system. The total primary energy used by the desiccant cooling system can be expressed as [15]:

$$W_e = \left(\frac{m.C_w (T_{set} - T_{in}) + U_A (\bar{T} - T_{env})}{\eta_h} \right) + \frac{Q_{ev}}{\eta_e} + Q_P \qquad (4)$$

where η_e is the evaporative cooler efficiency, Q_{evap} is the evaporative cooler capacity, m is inlet water mass flow rate, T_{set} is the set temperature of

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the heater internal thermostat in °C, T_{in} is water inlet temperature in °C, U_A is overall loss coefficient between the backup heater and its surroundings during operation, is $(T_{set} + T_{in})/2$ and T_{env} is temperature of heater surroundings for loss calculations in °C.and Q_p is the total parasitic energy used by the system main and auxiliary components. The system total energy savings to cover a certain cooling load can be expressed as [14]:

$$E_S = \frac{\left(\dfrac{W_{con}}{Q_{C,Con}}\right) - \left(\dfrac{W_d}{Q_{C,Solar}}\right)}{\left(\dfrac{W_{con}}{Q_{C,Con}}\right)} \times L_{total} \qquad (5)$$

where, E_{Con} is conventional system electric power, $Q_{C,Con}$

4 Experimental Setup

The investigated solar desiccant cooling system was designed and installed in Building No. 41 at the Rockhampton campus of Central Queensland University (as shown in Figure 2). Building 41, the Health and Safety Office consists of two identical zones, each zone had width is of 5 meter, depth of 10 meter and height of 3 meter. In addition the building had 10% glazing fraction for the north side, 25% for east side, 25% for west side and 50% for south side. Ventilation rate was defined as 1 m^3/h when the building is occupied and zero when the building is not occupied. Infiltration rate was set at 0.2 m^3/h at all times. The size of the window was 2 meter height and 10 meter width. Window was shaded by a 0.5 meter overhang which was attached within 0.1 meter above the window from the south side of the building.

In order to define internal gain, the default values provided by [16] were used with specific gain of 14 W per square meter, occupants density of 0.1 per square meter and illumination set at 2W per square meter. The building normal working hours were from eight in the morning till six in the evening, Monday to Saturday. The building structure consists of conventional room, external brick wall and a carpeted concrete floor. An air cavity of 110 mm is used to insulate the walls, sisalation foil and a plaster board as well as 90 mm of timber frame. The internal loads in the building including lighting, plug loads such as computers and occupants contribute to the overall cooling requirements. All rooms have manual switches to operate the lights. Common plug loads found in the building include desktop computers, monitors, printers, fax machines and desktop task lights.

Fig.2: Building 41 at CQUniversity

The conventional air conditioning at building 41 was a Mitsubishi package unit with 5 kWh cooling capacity. The system designed to be fully automated unattended system to serve the entire building and to maintain the room condition. The air conditioning system was sat to maintain comfort condition which is 24-26 ºC with 50-60 % of relative humidity.

The Central Queensland University's solar desiccant cooling system consisted of four connected sub-systems: solar thermal system, desiccant dehumidifier, evaporative cooler, circulation pumps and the reference conventional cooling system as shown in Figure 3 and 4.

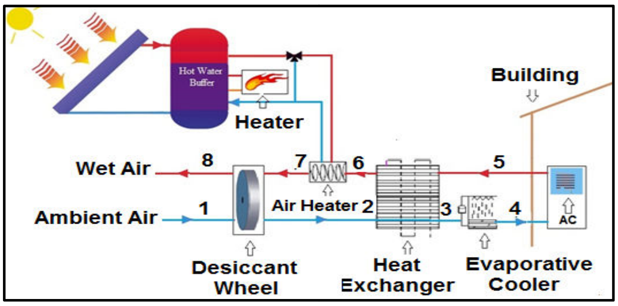

Fig. 3: Schematic of desiccant cooling system

The university solar cooling system has a design capacity of 5 kW for the reference building. The desiccant machine is manufactured by Seibu Giken DST AB, Japan and it works as a continuous process with two air streams having two different flow rates.

The air flow ratio is 3 to 1. In addition, the air stream with the higher flow rate is the processed air which is dried as it passes through the desiccant wheel, while the air stream with smaller flow rate is the regeneration air which is used to heat the wheel desiccant materials to drive away the adsorbed moisture vapour from the desiccant materials [17]. Additionally, the system is powered by 7.5 m^2 of flat plate solar collectors and is modified to fit two sets of backup heaters (4.5 kW and 9 kW). The Solar thermal flat plate collectors had the following specifications as provided by the local collectors supplier and [18]. The conversion factor η0= 0.780, the lost coefficient C_1= 4.2 (w/m^2.K), C_2 = 0.008 (w/m^2.K), the volume of the stored fluid per unit of collector area (Fluid volume/Collector area) =70 Litre /m^2, collector fluid flow rate per unit area= 0.015 kg/sm^2.

Fig. 4: Project components lay out

The data measured were: the outdoor and indoor ambient temperature, the site indoor and outdoor relative humidity, the site solar irradiance, the hot water temperature, the processed air temperature, the processed air humidity, the cooled space temperature, the cooled space relative humidity and the air flow. The experimental results and analysis of the cooling system are based on hourly measurements. The experiment was carried out for one year, November 2011 to October 2012.

4.1 Working Procedure and Control

A schematic illustration of the experimental cooling process is shown in Figure 5 which represents the cooling process in a psychometric chart taking into account two different climatic scenarios. During the cooling process in the first scenario, the assumed ambient temperature is 30 °C with 75 % relative humidity. For the second scenario, the assumed ambient temperature is 34 °C with 85 % relative humidity. In both scenarios the desiccant cooling process (as shown in Figure 4) starts with stage 1 where the treated air enters the desiccant wheel and exits after being heated and dried by the rotating desiccant wheel before reaching stage 2.

Afterwards the dried air passes through the heat exchanger to drop its risen temperature to a near ambient level, reaching stage 3. Next the treated air passes through the evaporative cooler to reduce its dry bulb temperature further and to increase its moisture content to near comfort level, reaching stage 4. This air is then passed through a conventional air conditioning unit within the reference building. Following this, the stage 5 return air is heated using the sensible heat recovery heat exchanger, thus reaching stage 6. Then the air is heated to reach stage 7 before it reaches a sufficient temperature and dryness level to regenerate the desiccant material in the desiccant wheel, being released at stage 8 as cooler air containing the moisture it has removed from the desiccant wheel.

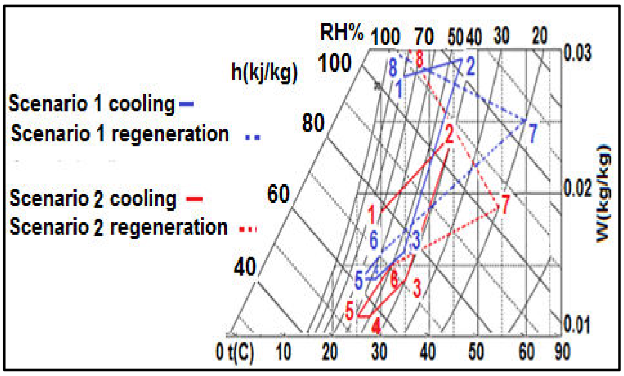

Fig. 5: psychometric chart of solar desiccant

4.2 Experimental Measurements and Results

The experimental duration was November 2011 to October 2012. The recorded outdoor temperature during summer (January to May and September to December) ranged from 22 °C to 39 °C and relative humidity varied from 45% to 98%. During winter (June-August), the outdoor temperature ranged from 6 °C to 28 °C with relative humidity of 25% to 70%.

During the cooling process in summer (November-May) it is found that the processed air temperature in stage 2, after passing the desiccant wheel, was ranging from a maximum of 49 °C with 15 % relative humidity and a minimum of 36 °C with 31 % relative humidity. In stage 3 prior to the air passing the evaporative cooler, the output air temperature was reduced to a maximum of 32 °C with 38 % relative humidity and a minimum of 23 °C with 61% relative humidity. The cooled space temperature achieved in the reference building between stages 4 and 5 stayed within appropriate comfort levels at around 24-26 °C with 45-60% relative humidity.

The desiccant wheel output air temperature is significantly affected by the ambient condition. With the increase of outdoor relative humidity, the temperature of the desiccant wheel output air becomes higher and vice versa. The air temperature supplied by the conventional HVAC unit recorded a maximum of 25 °C with 70 % relative humidity and a minimum of 23ºC with 50% relative humidity. This is because the temperature of the air supplied by the conventional HVAC unit is determined by the evaporative cooler output temperature, which depends on the temperature and relative humidity that is supplied by the desiccant wheel.

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The desiccant wheel output air temperature in stage 2 ranged from a maximum of 42 °C with 25% relative humidity and a minimum of 22 °C with 65% relative humidity. In stage 3, the evaporative cooler output air temperature was reduced to a maximum of 32 °C with 45% relative humidity and a minimum of 17 °C with 75% relative humidity. The cooled space temperature in the reference building ranged between 24-28 °C with 50-60% relative humidity. The conventional HVAC system provided air temperatures between 22 °C and 30 °C with 50% and 80% relative humidity.

The installed desiccant cooling system's measured solar fraction (*SF)* is shown in Figure 6. The Figure indicates that the system solar fraction has peaked during the cooling season in December at 0.46, followed by January, November, February, March and April at 0.44, 0.36, 0.26, 0.18 and 0.11 respectively. The minimum solar fraction was during winter in July at 0.045 followed by June, August and May at 0.05, 0.08 and 0.12 respectively. The system achieved an annual average *SF* of 0.2. As already noted the system *SF* dropped significantly during winter due to the decrease in solar irradiance and the low average relative humidity. It is clear that the *SF* has dropped significantly during winter (May to August) and this is due to solar irradiance declining intensity as well as the system reliance on the backup heater to compensate solar energy shortage needed for desiccant regeneration during this time of the year.

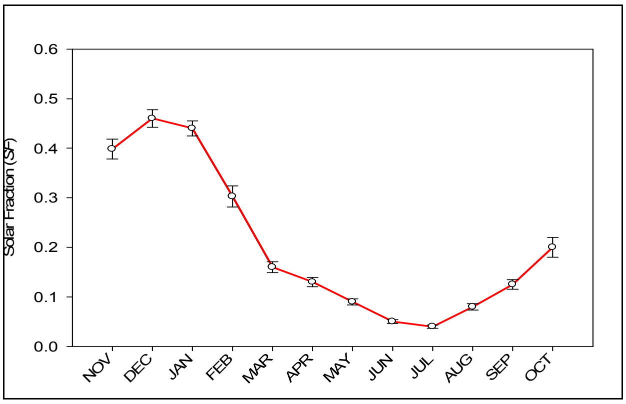

Fig. 6: Solar desiccant cooling system solar fraction

There are two main factors that affect the *COP* of desiccant cooling systems, namely: the cooling capacity and the regeneration input energy. Figure 7 presents the variation of the installed cooling system's measured *COP*. It shows that the system achieved a maximum *COP* in January of 0.73, followed by December, November, February, October, March, September and April at 0.66, 0.60, 0.53, 0.46, 0.44, 0.38 and 0.38 respectively. Moreover, during winter, the system has recorded a minimum *COP* of 0.26 in June followed by July, August and May at 0.29, 0.32 and 0.32 respectively.

The reason why the installed solar cooling system *COP* decreased sharply during winter (May to August) was the region's mild relative humidity that ranged between 25% and 75%, which mean minimum latent cooling load and maximum sensible cooling load. Accordingly most of cooling load will be dealt with using the conventional cooling system.

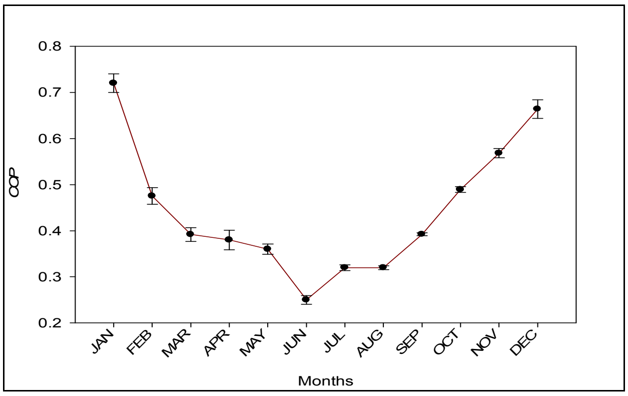

Fig. 7: System coefficient of performance COP

In this section all parasitic energy consumptions from the system components (the desiccant wheel, the heat exchanger, the pumps, the fans and the evaporative cooler) were considered. Figure 8 shows the energy usage of the hybrid desiccants cooling system is less compared to the total primary energy used by the reference stand-alone conventional cooling system. The hybrid solar desiccant cooling system reduced the annual total primary energy consumption from 6428 kWh to 5261kWh. Furthermore, the system's primary energy usage dropped to 34% and 33% in the months of December and January respectively, followed by February, November, March and April at 26%, 25%, 19% and 15% respectively. During winter, the system minimum drop in primary energy usage was in July at 9%, followed by August, May and June at 11%, 13% and 14% respectively.

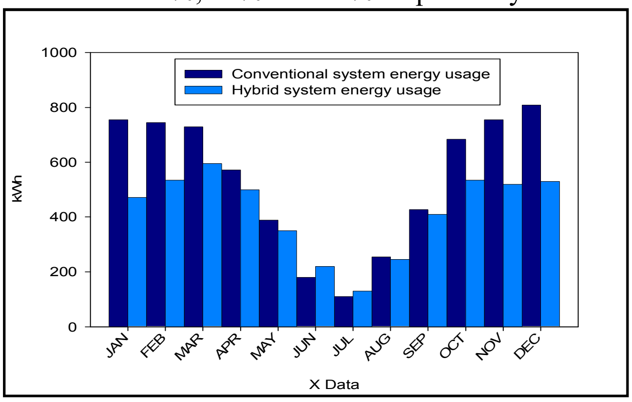

Fig. 8: Primary energy used by hybrid system and conventional cooling system

In the region of Central Queensland despite the variation of seasonal atmosphere, air conditioning is needed the whole year around. However from Figure 8 it is clear that the demand in primary energy is dropped rapidly from the month of May to the month of August and this is due to the winter season's minimum air conditioning demand

According to Figure 9, the Central Queensland University solar hybrid desiccant cooling system has a total annual energy saving of 1167 kWh which represents 19% of the total energy used by the stand-alone conventional cooling system. The maximum energy saving during the cooling season was in December at 43%, followed by January, November, February, October, March, September and April where it reached 41%, 36%, 29%, 19%, 15%, 14% and 13% respectively. The system recorded a minimum saving in the month of June and July at 8% followed by August and May at 9% and 9% respectively. The energy savings by the desiccant cooling system can be boosted further by increasing the number of solar panels and when the latent load is higher than sensible load.

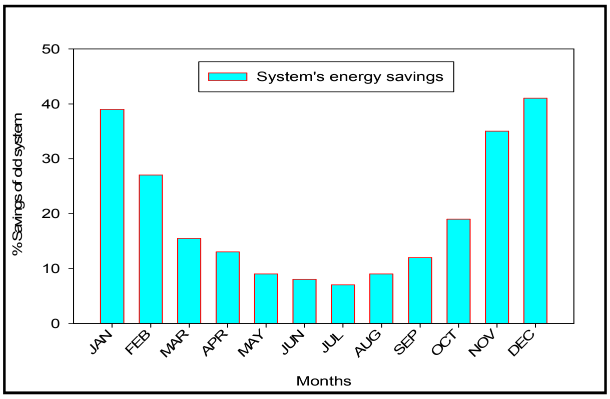

Fig. 9: Solar desiccant cooling system energy savings

5 System Simulation

The system is simulated using TRNSYS 16 software. TRNSYS is a FORTRAN-based transient systems simulation program developed at the Solar Energy Lab (SEL) of University of Wisconsin, to assess the performance of thermal and electrical energy systems. The software has been available since 1975 [19]. TRNSYS software has been extensively used to simulate solar energy applications and buildings' energy performance. It is also considered to be software with flexible open source architecture by facilitating the addition of mathematical models, the available add-on components, and the ability to interface with other simulation programs. TRNSYS structure is modular in nature and has wide range of energy systems' components which is called Types or modules.

The Types are configured and assembled using a graphical front end which is a visual interface known as TRNSYS Simulation Studio and building input data which can be entered through another visual interface called TRNBuild. Then the software simulation engine solves and analyses energy systems' algebraic and differential equations associated with the energy system. The software library contains components for multi zone building model, weather data readers, solar electric photovoltaic and solar thermal systems, low energy buildings and HVAC systems, renewable energy systems; cogeneration including fuel cells and other systems requires dynamic simulations.

5.1 Building Simulation

The main objective of building simulation was to assess the actual thermal performance of the reference building in order to provide guidance for energy consumption, performance and assessments. The study is conducted on Building 41, the Health and Safety Office at the Rockhampton campus of CQUniversity, Queensland, Australia. A base model of the reference building is modelled and developed using the Google sketch simulation tool and then added to TRNSYS software to evaluate the building cooling load and different cooling systems energy performance.

The building simulation model is shown in Figure 10. In TRNSYS software simulation studio the equivalent Type 56 which represents multi-zone building is used to model the building's thermal behaviour. The same building including physical structure and parameters is proposed in two of Central Queensland Subtropical regions.

Fig. 10: Building 41, at the Rockhampton campus of CQUniversity model

Using energy rate control, the models calculated cooling loads based upon the net heat gains or losses from the building. During modelling the building, cooling loads are calculated independently of the cooling equipment's operation. Building temperature and relative humidity for cooling seasons are set according to [20]. Then the software determines the energy necessary to keep the room at these set points. The system analysis is conducted based on the modelled building heat loss or gain, along with any additional gains due to sun radiation, lights, people, machineries, cooking, etc.

5.2 Solar Desiccant Cooling System Simulation

A model of desiccant cooling system under Central Queensland subtropical climate regions is developed in order to evaluate the system's performance parameters taking into account using different solar collectors' area. The solar desiccant cooling system modelling procedure started with selecting and connecting the system's component (Types) of the hybrid desiccant cooling system in TRNSYS simulation studio. The simulation software generates the building model automatically, couples the solar system installation to the building model and uses the system's components (Types) and its control strategy to generate the effect of dehumidifiers, the heat exchanger and evaporative cooler efficiencies on the overall cooling system performance.

5.3 Simulation Results

Changing collector's area has a big influence on solar desiccant cooling system solar fraction. Figure 11, shows an average of 0.6 of solar fraction during the cooling seasons was achieved. This was achieved by installing 50 m^2 of solar collector's area. The system solar fraction peaked in the months of December and January, reaching 0.82 and 0.81 respectively followed by November, February, March and April at 0.79, 0.65, 0.43 and 0.2. The system's minimum performance for the same area of collectors was in the month of May, June, July and August.

Additionally, installing 20 m^2 of solar collectors' area, the systems best performance was in the months of December and January, reaching 0.65. When installing 10 m^2 and 5 m^2 of solar collectors, the system's best performance was recorded in the months of December at 0.50 and 0.34 respectively. Finally, the results showed that the annual average solar fraction of the university installed system which consists of 10 m^2 of solar collector's area and 0.4 m^3 of hot water storage is 0.22.

The efficiency of the solar desiccant cooling system can be quantified by using the system coefficient of performance *COP*. Figure 12 shows the effect of the collector's area on the coefficient of performance for the proposed cooling system. It shows that the system achieved a maximum *COP* of 1.24 in the month of December by installing 50 m^2 of solar collectors' area followed by the months of January, February, October, March and April at 1.22, 0.95, 0.86, 0.64 and 0.27 respectively.

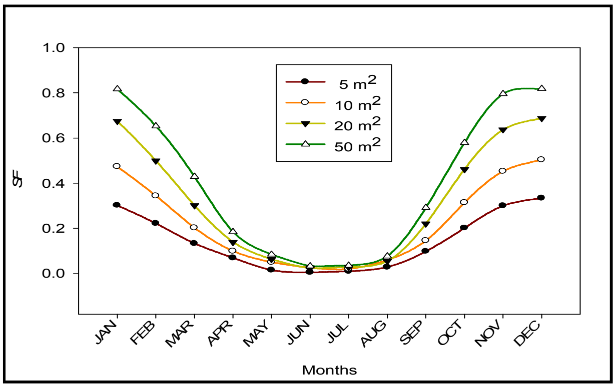

Fig.11: Desiccant cooling system solar fraction

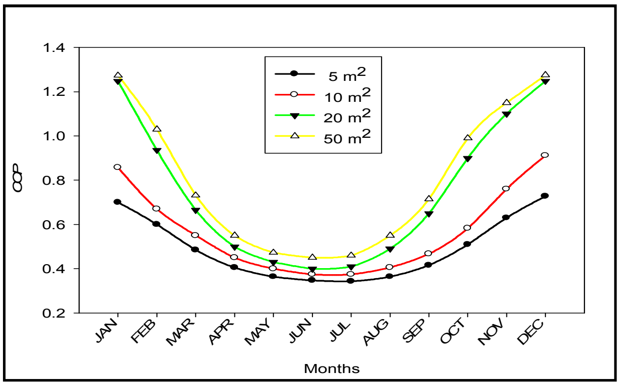

Fig. 12: Desiccant cooling system COP

Furthermore installing 20 m^2 of solar collectors' area, the system performance shows a similarity to the performance of 50 m^2 of solar collectors' area by delivering *COP* of 1.23 in the month of January, December and November. Installing 10 m^2 and 5 m^2 of solar collectors' area the system's best COP was found to be 0.89 and 0.63 respectively. Moreover, in winter, the system has recorded a minimum *COP* in the month of June at 0.42, 0.39, 0.32 and 0.28 for 50 m^2, 20 m^2, 10 m^2 and 5 m^2 of collectors' area respectively. As already noted, the system *COP* barely changed after installing more than 20 m^2 of solar collectors. Therefore, the recommended solar collector's area to be installed is 20 m^2.

The variation of the proposed desiccant cooling system using primary energy is shown in Figure 13. Results showed that the maximum primary energy required by the system was 650 kWh when installing 5 m^2 of solar collectors' area during March while the minimum primary energy required was 110 kWh in June for the same collectors' area. By increasing the collectors' area to 10 m ^{2} and 20 m ^{2}, the maximum primary energy required was 597 kWh and 523 kWh in February respectively and the minimum was in June at 110 kWh 99 kWh respectively. The minimum required primary energy recorded during the cooling season (January to May and September to December) was in the month July by installing 50 m^2 of solar collector's area at 95 kWh.

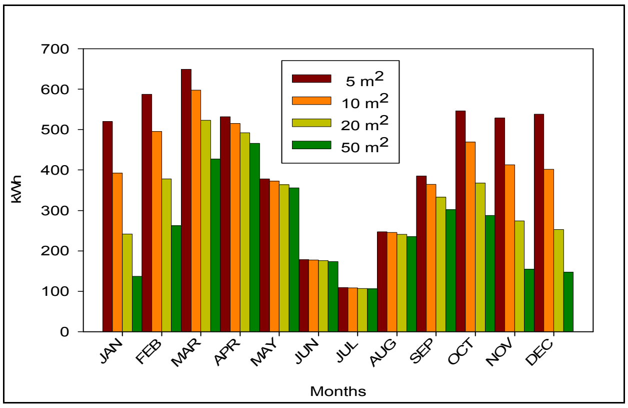

Fig. 13: Primary energy used by the desiccant cooling system

Energy saving achieved by Rockhampton's solar desiccant cooling system is illustrated in Figure 14. Results showed that, installing 50 m^2 of solar collectors' area will achieve 2023 kWh of annual energy savings which represents 30% of total annual energy used by the conventional cooling system. Thus the maximum energy savings of 395 kWh was achieved in December at 395 kWh for the same collectors' area followed by the months January, November, February, October, March, September, April, May, August and June.

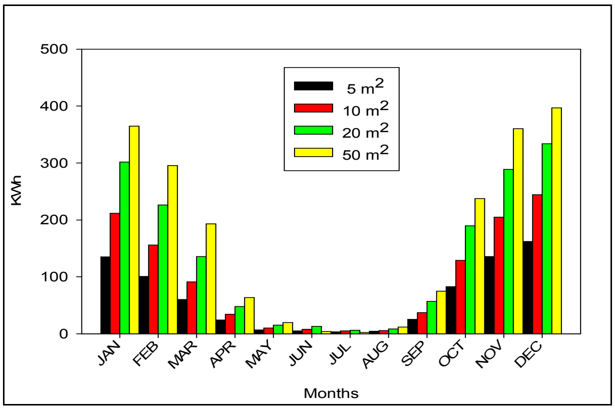

Fig. 14: Energy savings by the desiccant cooling system

The Rockhampton's desiccant cooling system achieved 1621 kWh by installing 20 m^2 of solar collectors' area which represents 24 % of total annual energy used by the conventional cooling system. The maximum amount of energy savings of 333 kWh was achieved in the month of December while the minimum was in the month of July at 6 kWh. The system estimated energy savings for 10 m^2 and 5m^2 of solar collectors' area was 1136 kWh and 743 kWh respectively which is accounted for by 17% and 12% of the total annual energy used by the stand alone cooling system.

5.4 Comparison between Experimental and Simulated Results

The installed cooling system's *SF* measured and simulated results are compared in Figure 15. It can be seen from the figure that the system achieved an annual average *SF* of 0.20 actual and 0.22 simulated of solar fraction. The system solar fraction has peaked during the cooling season in the month of December reaching 0.46 measured and 0.44 simulated, followed by the month January, November, February, March and April reaching 0.44, 0.36, 0.26, 0.18 and 0.11 measured respectively while the simulated results is 0.41, 0.40, 0.30, 0.16 and 0.16 for the months of January, November, February, March and April respectively.

The maximum deviation between measured and simulated *SF* was found to be a maximum in the month of June at 13.2%, followed by March, August, May, July, October, February, November, April, January, December and September at 12.5%, 12.4%, 11.11%, 10%, 9.5%, 9%, 8.8%, 7.7%, 4.5%, 4.3% and 4% respectively.

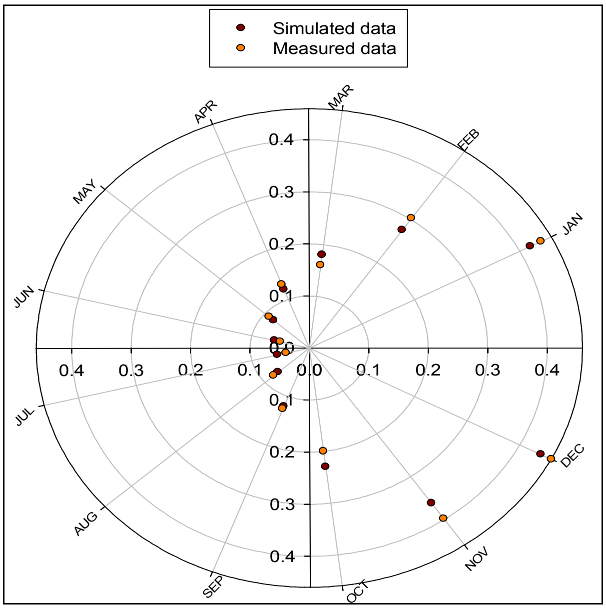

Fig. 15: Desiccant cooling system's simulated Vs measured *SF*

Figure 16 presents the difference between the installed cooling system's simulated and measured *COP*. The figure shows that the system achieved a maximum *COP* in the month of January at 0.74 measured and 0.72 simulated, followed by the

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months of December, November, February, October, March, September and April reaching 0.66, 0.60, 0.53, 0.46, 0.44, 0.38 and 0.38 measured respectively, while the recorded simulated results were 0.70, 0.56, 0.49,0.48, 0.39,0.39 and 0.30 respectively. During winter, the system has recorded a minimum *COP* of 0.26 measured and 0.25 simulated in the month of June followed by the months of July, August and May at 0.29, 0.32 and 0.32 respectively, while the simulated results were 0.32, 0.36, 0.36, 0.35 and 0.36 respectively.

As already noted there is a variation between the used primary energy by the hybrid desiccant cooling (measured and simulated) and the conventional stand alone cooling system as illustrated in Figure 17. The figure shows the hybrid solar desiccant cooling system reduced the annual total primary energy consumption from 6428 kWh to 5261 kWh measured and 5150 kWh simulated.

In December the system primary energy usage dropped from 809 kWh to 510 kWh measured and 540 kWh simulated. In March, the system used primary energy dropped from 749 kWh to 629 kWh measured and 600 kWh simulated. Additionally in February the system used primary energy also dropped from 755 kWh to 535 kWh measured and 550 kWh simulated. The minimum used primary energy drop was in the month of July from 110 kWh to 90 kWh measured and 85 kWh simulated.

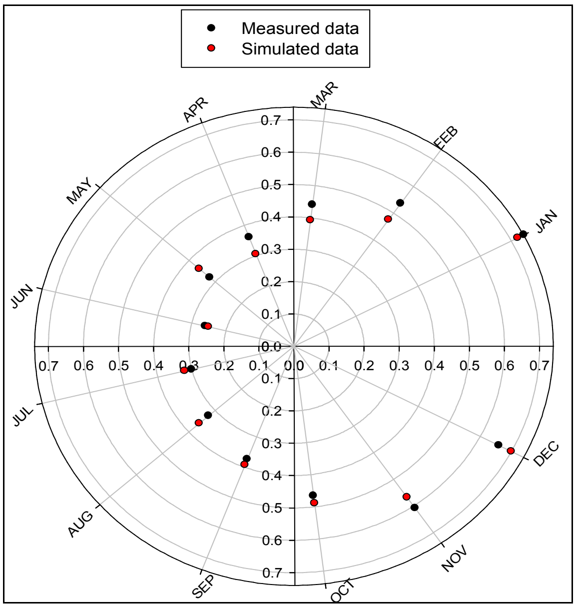

Fig. 16: Desiccant cooling system's simulated Vs measured *COP*

Figure 18, shows the CQUniversity solar hybrid desiccant cooling annual energy savings. As shown in Figure 18 the cooling system achieved total annual savings of 1167 kWh actual and 1350 kWh simulated which represents 19% actual and 21% simulated of the total energy used by the stand-alone conventional cooling system.

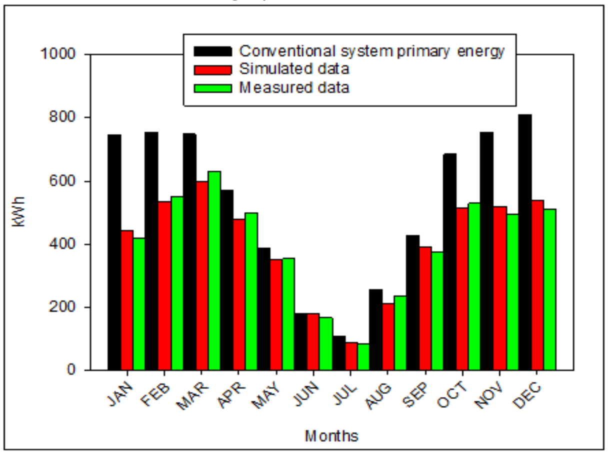

Fig. 17: System's used primary energy

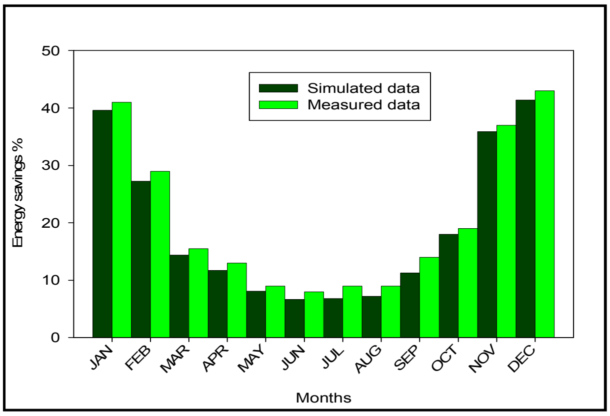

Fig. 18: System's Energy savings

The cooling system's maximum energy saving is achieved at 43% measured and 39% simulated during the cooling season in the month of December followed by January, November, February, October, March, September and April at 41%, 36%, 29%, 19%, 15%, 14% and 13% measured respectively and 39%, 37%, 26%, 17%, 13%, 11% and 11.7% simulated respectively .

5.5 Result Validation and Uncertainty

The TRNSYS simulation model was validated using the experimental data obtained from the Rockhampton installed solar desiccant cooling system. In this research all system measured and analysed data are based on hourly values. To validate the system performance results, a comparison analysis has been carried out between the experimental values and the simulated results. Measured data was used as input parameters for the TRNSYS simulation model of Rockhampton's desiccant cooling system in order to evaluate the performance of the system using the system's performance parameters: solar fraction, coefficient

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of performance, primary energy and savings energy. The average relative error between simulation and experimental data is calculated as in equation 6 [21] [22].

$$E = \frac{1}{N}\sum \frac{|C_{value} - M_{value}|}{M_{value}} \times 100 \qquad (6)$$

In the above equation E is the error percentage, C_{value} is calculated value, M_{value} is measured value and N is the number of samples.

It is clear that there are variations between simulated results and measured values, and that is due to some external causes and other resources such as discrepancies between actual and simulation input in weather data, building operational data and physical properties which were beyond the control of the author. The difference between outdoor measured ambient temperature and simulated temperature varies between ±2% to ±4%, while the difference between measured and simulated outdoor relative humidity is ±3%. In addition, the difference between simulated and measured system parameters namely: system *COP*, primary energy used, energy savings and solar fraction was ±8%, ±10%, ±9% and ±7% respectively.

6. Conclusion

A solar desiccant cooling systems has been designed and installed in the health and safety unit office building (building 41) at Rockhampton campus of CQUniversity, Australia. The performance of the installed cooling system has been experimentally investigated and analysed. Numerical simulation of solar cooling technologies also has been carried out using TRNSYS software taking into account different solar collectors' area. The numerical results have been validated using the experimental measurement of the installed solar desiccant cooling system. Through the extensive experimental investigations and numerical modelling, more comprehensive understanding of solar assisted air conditioning characteristics has been achieved. The study presented a new and comprehensive assessment, facts, results, limitations and strategy concerning installation of solar assisted air conditioning in the region.

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