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Investigation on the Feasibility of a Solar Desiccant Cooling System in Tripoli, Libya

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***Abstract*— The significant growth rate observed in Libyan commercial and residential buildings is coupled with a growth in energy consumption. This growth in energy demand needs to be met by either applying measures for saving energy or by an increase in the energy production. Solar desiccant evaporative cooling offers energy savings and promises a good sharing of renewable energy for sustainable buildings where the availability of solar radiation matches the cooling load demand. This paper presents a preliminary feasibility study for implementing desiccant systems in Libya. A mathematical model of a selected system has been developed and a simulation has been performed in order to investigate the system performance at different working conditions and an optimum design of the system structure is established. The results showed that solar desiccant cooling system is feasible under the Libyan climatic conditions with a reasonable COP at temperatures that can be obtained through the solar reactivation system. Discussion of the results and the recommendations for future work are proposed.**

***Keywords* — solar desiccant wheel cooling, system modelling and simulation, technical feasibility.**

I. INTRODUCTION

Cooling is an important aspect to provide comfort in buildings in warm climates zones such as Libya. Along with the growth of cooling loads in modern buildings (particularly in housing), the peak of the electricity demand increases especially during the day time. This causes failure to provide the required energy supply. The inertia of power plants leads to a high consumption of oil and gas by energy producer to ensure the provision of electricity and it emits harmful gases to the environment. Solar energy for cooling purposes, where energy demand matches with the solar radiation, needs low operating costs, has high durability and environmental compatibility could make a significant contribution to lowering the high grade energy demand. Some previous research found that the demand of high grade energy can be reduced by using low grade energy within the concept and principles of energy efficiency in buildings [1, 2].

A typical Desiccant cooling system includes a dehumidification system where air is dehumidified, cooled and introduced to the space. Reactivation of the desiccant material will be performed by using solar system. Water and air as working fluids in this system are friendly to the environment and providing fresh air means improving the indoor air quality and also controlling humidity. As a solar heat is used, the system should be able tolerate wide variation in solar energy input. Many countries in the world are using such system for public buildings.

In this paper, the technical feasibility of solar operated desiccant cooling system working under the Libyan climatic conditions is investigated. This is done by the estimation of the system performance for different working parameters.

II. SYSTEM DESCRIPTION

Many configurations can be found for solar operated desiccant cooling systems, some of them are using solid desiccant materials while others are using liquids. Figure (1) shows the schematic diagram of the selected system which will be considered in this study.

In this system, mixed air passes through desiccant wheel (point 1), and comes out hot and dry (point 2) due to the desiccant's heat of adsorption. Then air is cooled in the heat exchanger at point 3. Dry air then leaves the heat exchanger and passes through an evaporative cooler, which adds moisture to the air and reducing its temperature. The air then enters the space to be conditioned.

On the reactivation side, outdoor air enters the evaporative cooler (point O). This cooled air provides a heat sink for the air-to-air heat exchanger. The hottest air comes out from the heat exchanger (point 7). It is heated again by solar system before it enters the desiccant wheel (point 8) to be used in reactivation of the desiccant material then it is rejected outside. The reactivation heat source for the desiccant material is obtained from solar subsystem. The solar subsystem consists of an array of solar collectors with a water as the working fluid, which transfers energy to air via a heat exchanger.

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Water coming out of collector array stored in storage tank. An auxiliary heater is provided in series with the storage tank. The auxiliary heater compensates for energy required for reactivation process.

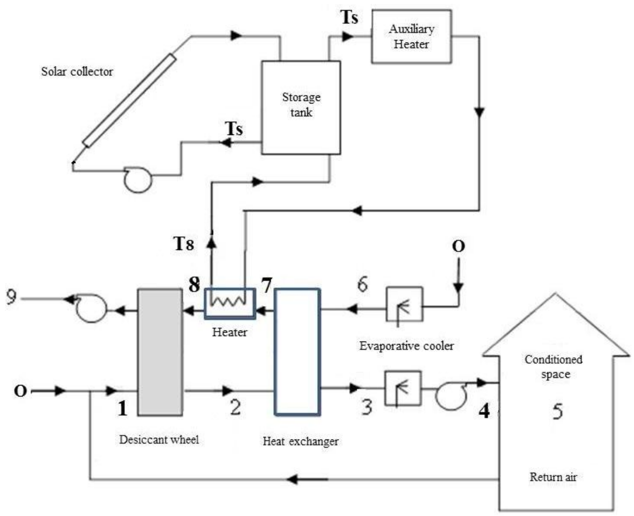

Fig.(1). Solar desiccant cooling system.

III. Background & Mathematical Modeling

A. *Desiccant wheel*

The dehumidifying capacity of the wheel depends on the rotational speed of the wheel, the air flow rate and the temperature of the regeneration air stream. It also depends on the supply of air stream conditions, i.e., temperature, absolute humidity and absolute humidity of regeneration stream [3].

In the field of desiccant wheel modeling, considerable work has been developed. Maclaine-Cross and Banks [4] have developed an analogy method, with reference to the rotating sensible heat exchangers, for predicting the coupled heat and mass transfer process in desiccant dehumidifier wheel (method of characteristics).

In this study, the model of Jurinak [5] has been adopted, as it is considered flexible in the application and reliable with reference to the scope of this analysis. This model has also been used in the TRNSYS software.

In this model, the dehumidifier assumed to be of infinite capacity (the matrix capacity rate is the matrix mass rate times the specific heat of the solid) [5]. Thus, the rotational speed of the wheel and the mass flow rate of the process or regenerated air streams do not influence the calculations. The model considers equal capacity rates for the two streams, leading to equal mass flow rate for the two air streams.

The model consists of the following set of equations (1 to 4).

$$F_{1,n} = \frac{-2865}{T_n^{1.49}} + 4.344\, w_n^{0.8624} \qquad (1)$$

$$F_{2,n} = \frac{T_n^{1.49}}{6360} - 1.127\, w_n^{0.07969} \qquad (2)$$

$$\eta_{F1} = \frac{F_{1,2}-F_{1,1}}{F_{1,8}-F_{1,1}} \qquad (3)$$

$$\eta_{F2} = \frac{F_{2,2}-F_{2,1}}{F_{2,8}-F_{2,1}} \qquad (4)$$

Where, F1 and F2 are Maclaine-Cross characteristic potentials, entering the desiccant wheel model of Jurinak, η_{F1} and η_{F2} are Maclaine-Cross characteristic potential effectiveness and set to 0.08 and 0.95, respectively. Those values correspond to a high efficiency wheel, as the η_{F1} and $\eta_{F2} \leq 1$. Tn is the air temperature (K), w_n absolute humidity (kg/kg), and n is ports of Desiccant wheel.

B. *Heat exchanger*

As deriving from the theory [7, 8], for equal capacity rates of the hot and cold side, the heat exchanger effectiveness (η_{hx}) can be described by Eq. (5). Analysis of the η_{hx} values for various operation conditions of a heat exchanger permits the assumption of a steady typical value for η_{hx}, while there is no humidity exchange through the heat exchanger:

$$t_7 - t_6 = \eta_{hx}(t_2 - t_6) \qquad (5)$$

For equal the energy loss by the hot side and the energy gained by the cold side; we can obtain temperature at point 3 as follows [6].

$$t_3 = t_2 - (t_7 - t_6) \qquad (6)$$

C. *Evaporative cooler*

The effectiveness of an evaporative cooler is given by the following relation:

$$\eta_{ev} = \frac{t_3 - t_4}{t_3 - t_{wb-3}} = \frac{t_5 - t_6}{t_5 - t_{wb-5}} \qquad (7)$$

Where η_{ev} is the supply and exhaust air stream evaporative cooler efficiency, t_{wb-3} and t_{wb-5} are wet-bulb temperatures at points 3 and 5, respectively. The evaporative cooling process is considered adiabatic and constant wet-bulb temperature line. Therefore:

$$t_{wb-3} = t_{wb-4}$$

$$t_{wb-5} = t_{wb-6}$$

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D. Mixing box

The return air from the building is mixed with outside air in the mixing box. A common process in air conditioning systems is the adiabatic mixing of two moist air streams. Adiabatic mixing is governed by three equations [8]:

$$m_o h_o + m_5 h_5 = m_1 h_1$$
$$m_1 = m_o + m_5 \qquad (8)$$
$$m_o w_o + m_5 w_5 = m_1 w_1$$

Where: m : mass flow rate (kg/sec).
h : enthalpy of the air (kJ/kg).
w : humidity ratio (kg/kg).

E. Solar collector storage subsystem

Solar collector storage subsystem illustrated in Figure 1 shows a schematic of the system component configuration chosen. The system consists of an array of solar collectors with water as the working fluid, which transfers energy to air via a heat exchanger. This hot water is stored in a storage tank. An auxiliary heater is provided in series with the storage tank. The auxiliary heater is essential to supply compensated energy required to heat to the desired regeneration temperature. The model of this subsystem was found as standard model [9, 10]. The collector efficiency is given by the following equation [10]:

$$\eta_c = \frac{Q_u}{A_c I_t} = F_R(\tau\alpha)_n - F_R U_C \frac{(T_{ex} - T_1)}{I_t} \qquad (9)$$

Where: Q_u is the useful energy gain in the collector, A_C is the collector surface area, I_t is the solar radiation, F_R is the collector heat removal factor, $(\tau\alpha)_n$ is the collector transitivity- absorptivity product at normal incidence, and U_c is the collector overall heat loss coefficient.

F. System coefficient of performance

The overall system coefficient of performance (COP) is defined as the ratio between heat recovered by the system from the space to energy input to the system i.e.,

$$COP = \frac{\text{heat removed by the system from the space}}{\text{Energy input to the system}}$$
$$= \frac{(h_5 - h_4)}{(h_8 - h_7)} \qquad (9)$$

IV. SIMULATION PROCESS

Using the mathematical model, a simulation Code on Fortran PowerStation basis was built to simulate the calculations for a system parametric study and finding out the proper parameters for maximum COP and the solar contribution factor for optimum conditions. This factor indicates how much energy can be used from a solar subsystem. In order to simplify the process of simulation steady state condition is assumed within this work. Figure 2 shows a simplified information flow diagram, where climatic data for the location is introduced together with building information to building subroutine to calculate building cooling load and also to the solar energy collection subroutine to calculate solar useful energy collected. The outputs are introduced then to the main program to alternate the calculations through a different DO loops searching for the optimum results and working conditions.

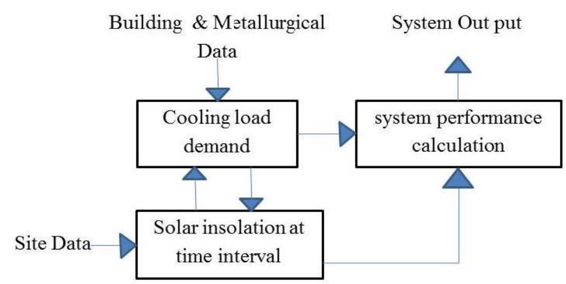

Fig (2). DC-SOLAR information flow diagram

The program was executed using Tripoli climatic conditions. The influence of such conditions on the reactivation temperature, heat exchanger efficiency and the evaporative cooling efficiency as well as the cooling system performance was studied. The results obtained for a selected dwelling in Tripoli-Libya, for a specific day as an example, recommended indoor conditions, to be as T_5= 26°C, RH_5=50%. This program can be run through the cooling season.

V. RESULTS AN DISCUSSION

The solar insolation in W/m^2 throughout the day is shown in Figure 3. The peak insolation occurs at about solar noon while low insolation values occur in the morning up to 10 a.m. and after noon, from 5 p.m.

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The energy required by the cooling system in the weak collection time has to be provided by auxiliary heater or by introducing sensible heat storage within the system.

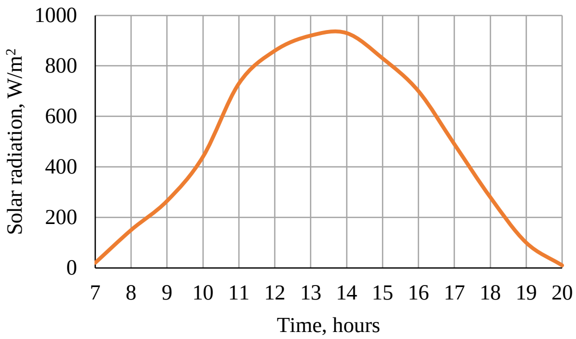

Fig (3). Solar insolation throughout the day (21 of Jun , Tripoli)

Table 1 shows the contribution of solar energy as a function of collector area based on the selected building cooling load. Also, energy obtained from the solar system in 24 hours is shown in the table.

Table 1
Contribution of solar energy in the system

Collector area (m^2)	Solar energy (kWh)	Contribution of solar energy (%)
50	137.8	40.3
75	209.7	61.4
100	281.6	82.9
120	341.3	100

Figure 4, shows the effect of reactivation temperature on the output temperature from the dehumidifier (t_2). As the reactivation temperature increases (increasing the energy input), the output air temperature is increased, these temperatures related to the desired system size and optimum working conditions.

Variation of absolute humidity of supply air leaving the desiccant wheel and introduced to the heat exchanger with reactivation temperature is shown in Figure 5. Whereas the reactivation temperature increased with increasing reactivation temperature, the absolute humidity decreased. The limit of decreasing in W_2 is a matter of optimum working condition of the system.

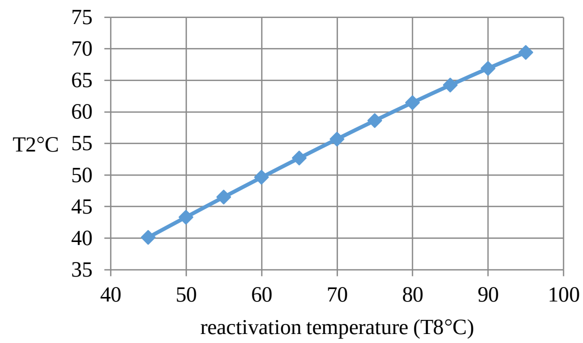

Fig.(4). The effect of reactivation temperature on of supply air temperature leaving the desiccant wheel

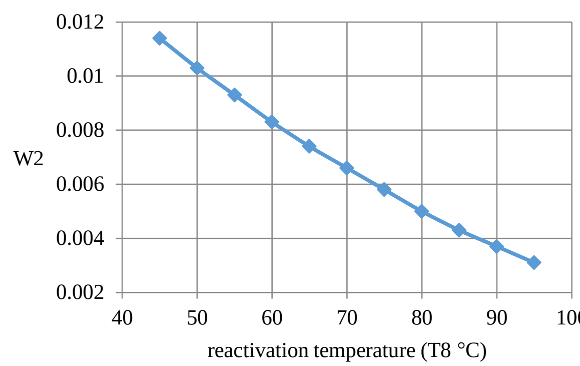

Fig.(5). The effect of reactivation temperature on the absolute humidity of the supply air leaving the evaporative cooler.

Increasing of reactivation temperature leads to a decrease in absolute humidity at point 2 (W2) and then decreasing in wet bulb temperature.

Figure 6 shows that the increase in reactivation temperature tends to a decrease in the temperature of supply air leaving the evaporative cooler and to a decrease in its absolute humidity, as shown in Figure 7. This is because the increasing in reactivation temperature leads to a decrease in absolute humidity at point 2 (W2) and then a decrease in wet bulb temperature.

Figure 8 shows the effect of variation of reactivation temperature on the system performance coefficient. In general we note that the coefficient of performance increases with increasing reactivation temperature until reaching the maximum value, (COP=0.7) after that it starts to decrease. This result meets the ones shown in many mediterranean region researches.

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Note that the COP were negative at the reactivation temperature Less than 46 °C, implying that the enthalpy of indoor was lower than the enthalpy of supply.

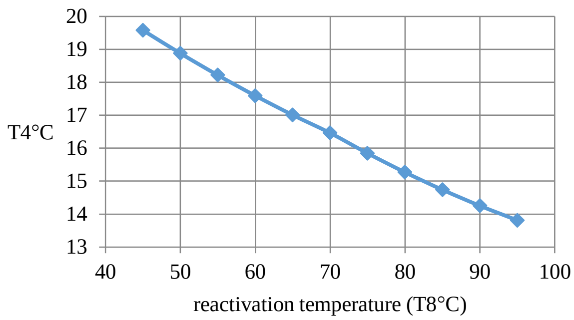

Fig.(6).Variation of the reactivation temperature with air temperature leaving the evaporative cooler

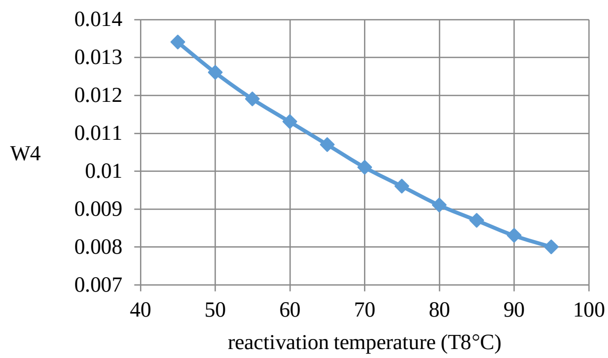

Fig.(7). The effect of the reactivation temperature on absolute humidity

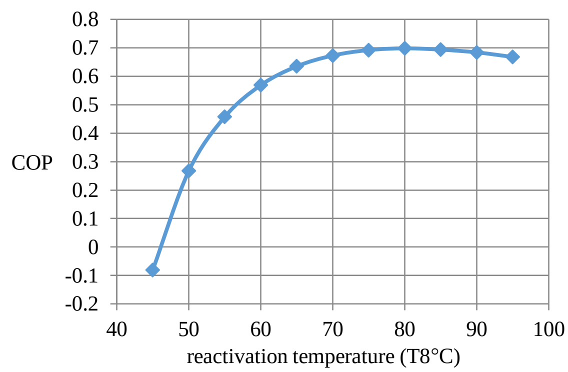

Fig.(8). Variation of COP with reactivation temperature

Reactivation temperature of 80°C can be achived by using a flat plate collector of a selective surface. That means the solar operated system is technically feasible.

Note that the performance coefficient is a strong function of heat exchanger efficiency, so a serious attention has to be taken in designing the heat exchanger.

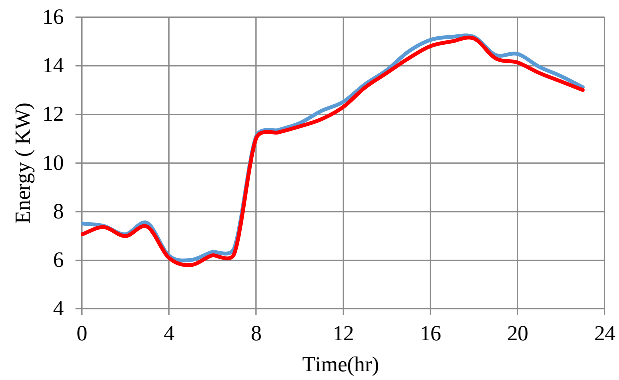

Fig.(9). Variation of cooling Load and cooling effect with time.

Figure 9 shows the variations of cooling load (red curve) and the cooling effect (blue curve) during the day. From this figure we see that the system can provide the space with the desired cooling load. Therefore, under the selected conditions, the Solar Desiccant Cooling system, selected in this study, is technically feasibile to be used in the city of Tripoli.

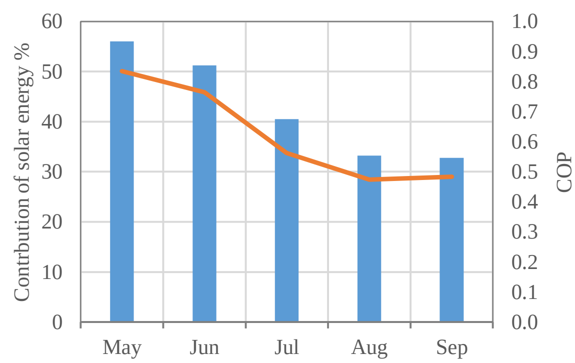

Fig.(10). Contrbution of solar energy and COP during cooling season

Figure 10 shows both the contribution of solar energy for the supply of the AC system and the average COP values during the cooling season. The COP presented for each month with the average COP equal to 0.63 through the cooling season.

The contribution of solar energy in each month through the cooling season shows that in May the contribution of solar energy is 56%, while in September it decreases to about 32%, due to the increase in both temperature and humidity during the cooling season. This leads to an increase in the power required to operate the system.

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However, providing between half and one third of the energy demand required for air conditioning can be considered a high feasibility of using the desiccant cooling system.

VI. Conclusions And Future Work

This study presents a solar operated desiccant cooling system working under Tripoli – Libya climatic conditions. Influence of some working parameters on the performance and to each other were investigated. Some observation was drawn from this study that shows the solar desiccant cooling system have the ability to work under the climatic conditions of Tripoli with acceptable coefficient of performance of 0.7. The Coefficient of performance increases with increasing the heat exchanger efficiency at a rate greater than the increase evaporation cooler efficiency, in other words COP is a strong function of heat exchanger efficiency. The loss of matching in time where the solar radiation is low can be covered with auxiliary system unless a sensible heat storage considered to be used. System simulation carried out at static condition while dynamic simulation will be an issue for future work to find out the optimum design for the solar operated desiccant cooling system for Libyan climate.

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