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Research Article

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Evaluation of the Technical Feasibility of a Solar Dunkle Cycle Desiccant Cooling System in Libya

BASIM TAYIB ALMABROUK[1],*
[1] Department of Mechanical Engineering High Institute of Science and Technology,Al-Bayda, Libya
https://orcid.org/0009-0005-2573-0290
**Corresponding Email:* Boemyta@gmail.com

Abstract

This work evaluates the technical feasibility and performance of a solar-driven Dunkle cycle desiccant cooling system for a residential building under the climatic conditions of Tripoli. The study investigates the influence of several operational parameters such as reactivation temperature and the effectiveness of heat exchangers 1 and 2on the overall system performance.
A computer program was developed based on mathematical models representing the system components to calculate the coefficient of performance (COP) and the solar energy contribution during the cooling season (June to September).
The results indicate that the solar Dunkle cycle desiccant cooling system can effectively operate under the climatic conditions of Tripoli, achieving an average COP of 0.98 throughout the cooling season. The analysis also reveals that the system performance is significantly affected by both the reactivation temperature and the heat exchanger effectiveness.

Keywords: Solar energy, Dunkle cycle, Technical feasibility, Desiccant cooling system.

تقييم الجدوى الفنية لنظام تبريد ممتص للرطوبة يعمل بدورة دونكل الشمسية في ليبيا.

باسم طيب المبروك
قسم الهندسة الميكانيكية، المعهد العالي للعلوم والتقنية، البيضاء، ليبيا.

للمراسلة: Boemyta@gmail.com

الملخص.
يهدف هذا البحث إلى تقييم الجدوى الفنية وأداء نظام تبريد ممتص للرطوبة يعمل بدورة دونكل الشمسية لتكييف مبنى سكني تحت الظروف المناخية لمدينة طرابلس. كما يتناول تأثير بعض العوامل التشغيلية على أداء النظام، مثل درجة حرارة إعادة التنشيط وفعالية المبادلين الحراريين الأول والثاني.
تم تطوير برنامج حاسوبي يعتمد على نماذج رياضية تمثل مكونات النظام، بهدف حساب معامل الأداء (COP) ومساهمة الطاقة الشمسية خلال موسم التبريد (من شهر يونيو حتى سبتمبر).

أظهرت نتائج الدراسة أن نظام التبريد الممتص للرطوبة بدورة دونكل الشمسية قادر على العمل بكفاءة في ظل الظروف المناخية لمدينة طرابلس، حيث بلغ متوسط معامل الأداء 0.98 خلال موسم التبريد. كما تبين أن أداء النظام يتأثر بشكل كبير بدرجة حرارة إعادة التنشيط وفعالية المبادلات الحرارية.

الكلمات المفتاحية: طاقة شمسية، دورة دونكل، جدوى فنية، نظام تبريد بالتجفيف.

1. Introduction

Residential buildings, such as apartments and houses, require air-conditioning systems to maintain a comfortable indoor environment for occupants. Conventional air-conditioning systems are designed to control both temperature and humidity levels within buildings. However, the installation and operation costs of these traditional systems are often high, leading to the need for alternative and more cost-effective cooling technologies.

In recent years, desiccant cooling systems have emerged as a promising alternative to conventional vapor compression systems. These systems provide satisfactory thermal comfort while offering advantages such as lower operating and maintenance costs. Their performance has been found acceptable, particularly in regions characterized by high humidity levels. Furthermore, when such systems are driven by renewable energy sources especially solar energy they contribute to the sustainability and energy efficiency of residential buildings.

The most important advantages of desiccant cooling systems are as follows:

- Desiccant cooling systems use water and air as working fluids (environmentally friendly).
- Sources of energy used in the operation of a variety of Solar, Natural Gas, waste heat and thus reduce the power consumption.
- Indoor air quality is improved because of higher ventilation and fresh air rates associated with desiccant systems.
- Control of humidity and temperature are good.
- Desiccant cooling systems use a minimum number of moving parts which also reduces the operating and maintenance costs.

As a result of all in above, increased attention to this system in the past few years, especially in America, Japan, Germany and China.

Researches by government and private institutions have been proceeding on many forms and different aspects of desiccant systems. These efforts are described in the following sections.

1.1 Literature review.

A desiccant is a hygroscopic material with a great affinity for water. Desiccants can be found in either solid or liquid states. Some examples of commonly used solid desiccants are silica gel, lithium chloride (LiCl), and molecular sieves. A solid desiccant cooling system operates by absorbing water vapor from the air.

The concept of desiccant cooling was first introduced by Hausen in 1935. Based on this, many inventors like Shipman (1936), Fleisher (1939), Larriva (1941) and Altenkirch (1941, 1944) tried to develop commercial desiccant cooling systems but They were not successful in the subsequent years. However, Miller and Fonda (1933) invented the First rotary silica gel desiccant cooler, but because of lack of understanding, the potential of desiccant dehumidifiers was not realized.

Pennington (1955) patented the First ever desiccant cooling cycle which is commonly known as the ventilation cycle shone in Fig (1). A rotary heat exchanger was saturated with a solid desiccant, converting the heat exchanger into an adiabatic regenerative dehumidifier, which takes in ambient air and adsorbs the moisture in it. This air is then sensibly and again evaporatively cooled before being introduced into the conditioned space. The return air is First evaporatively cooled and allowed to pass through a sensible heat exchanger to recover the heat of adsorption from the supply air. It is then heated with a low-grade thermal energy source and used to regenerate the desiccant. Coefficient of performance (COP) values of about 0.8-1.0 are commonly predicted for this cycle (Kettleborough, 1980 and Lof et al., 1988).

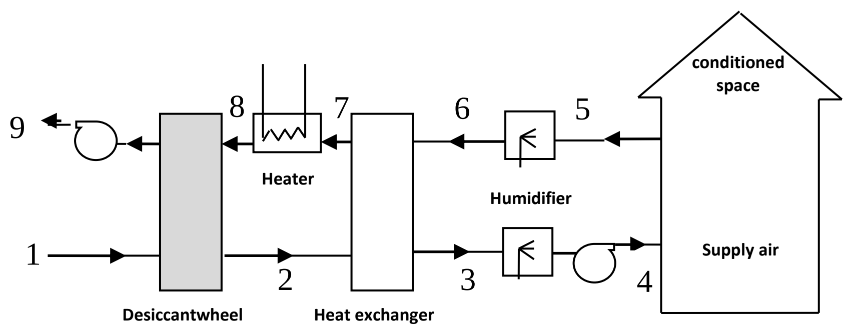


Fig. (1) Ventilation Desiccant Cooling System

The recirculation cycle, shown in Fig (2), is designed to reuse room return air as a dehumidifier process air inlet in hot and humid conditions. It elevates cooling capacity by recirculating all or some of the air.

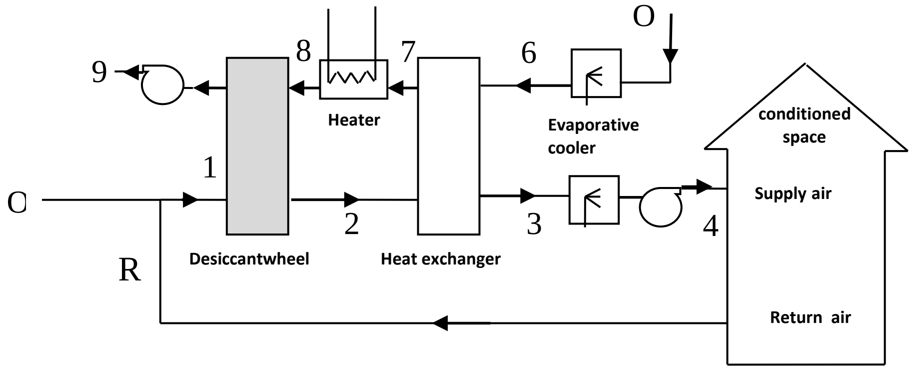


Fig. (2) Recirculation Cycle Desiccant CoolingSystem

The Dunkle cycle in Fig (3) builds on the ventilation cycle by incorporating a sensible heat exchanger. This addition improves cooling performance and efficiency by pre-cooling the process air before it reaches the evaporative cooling stage.

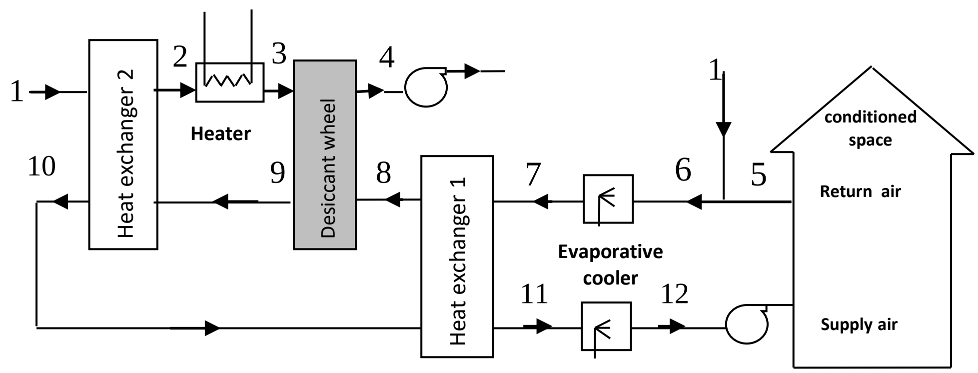


Fig.(3) Dunkle Cycle

(Davanagere et al.1999). evaluated the performance of Desiccant cooling system according to climatic conditions for four U.S. cities by using TRANSYS program, the result of his study anything to the system is able to cover the cooling load in those cities.

(Joudi et al. 2001) studied the Desiccant cooling system under the outdoor conditions of the Baghdad, the result of his study showed that the Desiccant cooling system capable of achieving thermal comfort according to these climatic conditions, and the COP has exceeded the 3 because of using the solar energy in the reactivation process.

(Fatemeh Esfandiari Niaet al.2006) presents the modeling of a desiccant wheel used for dehumidifying the ventilation air of an air-conditioning system. The simulation of the combined heat and mass transfer processes that occur in a solid desiccant wheel is carried out with MATLAB Simulink. The model is validated through comparison the simulated results with the published actual values of an experimental work, the results of simulation and the published experimental data are agreeable within an acceptable margin of error.

Studies by Kousar et al. (2021) and Ahmad et al. (2020) analyzed multi-stage configurations integrated with Maisotsenko cycle evaporative coolers, achieving coefficients of performance (COP) as high as 2.28 and energy cost savings of up to 62.9%.

Dynamic Simulation and Performance was done by Farooq et al. (2020). The Study analyzed three configuration schemes of a solar-assisted desiccant cooling system through dynamic simulation, showing that configuration 1 offered the best solar fraction and primary energy savings performance.

(Ahmad Ababneh.2025) presents the study investigates the modelling and optimization of solar-assisted desiccant cooling systems (SADCS) for industrial applications in Dubai’s extreme climate using TRNSYS 18. Four system configurations were analyzed through a parametric study of 60 simulation cases, examining airflow rates, regeneration temperatures, desiccant wheel effectiveness, and fresh air ratios to achieve thermal comfort with minimal energy use. The optimal configuration was a recirculation cycle with sensible heat recovery, operating at 50% fresh air, 50 °C regeneration temperature, and 3 ACH, providing a balanced performance between comfort and energy efficiency. This configuration achieved up to 86.5% comfort coverage while consuming only 1.39 MWh/year. Overall, SADCS showed significant reductions in electricity consumption and CO_2 emissions compared to conventional HVAC systems.

(A. Khalid, M. Hassanain, A. Mostafa, and M. Fatouh2024) presents a thermo-economic analysis of novel desiccant air-conditioning systems based on experimental data and numerical simulations. Several innovative system configurations are evaluated, including enhanced cycles such as the Dunkle cycle through the integration of heat exchangers and advanced dehumidification techniques. The results demonstrate significant improvements in energy efficiency and reductions in operating costs compared to conventional systems. The study confirms the technical and economic feasibility of advanced desiccant cooling systems for sustainable air-conditioning applications.

(K. F. Fong and C. K. Lee2020) investigates the performance of solar-assisted desiccant cooling systems applied in hot and humid climates. It presents detailed system designs, mathematical modeling, and performance evaluations of desiccant cooling cycles integrated with solar thermal energy. Although not focused exclusively on the Dunkle cycle, the paper provides valuable insights into practical desiccant system configurations and real-environment applications. The results highlight the effectiveness of solar-driven desiccant cooling in reducing energy consumption and enhancing thermal comfort in challenging climatic conditions.

1.2Objective of the Present Study.

The objective of this study is to evaluate the technical feasibility and performance of a solar operated Dunkle cycle desiccant cooling system for a residential house under the climatic conditions of Tripoli, Libya. The study also analyzes the influence of key operational parameters, including the reactivation temperature and the effectiveness of heat exchangers on the overall performance of the system.

2. 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. Fig. (4) Shows the schematic diagram of the Dunkle cycle desiccant cooling system which will be considered in this study, and Fig. (5) Shows the psychometric processes.

On the process side, mix the return air from conditioned space with outdoor air at point (6),then passed through a evaporative cooler, which adds moisture to the air, reducing its temperature through procedure (6-7), then heated in the heat exchanger (sensible heating) through procedure (7-8),through procedure (8-9) pass the air through desiccant wheel, this air exits is hot and dry, then cooled in the heat exchanger(2) (sensible cooling) through procedure (9-10), then cooling again in the heat exchanger(1) through procedure (10-11),finally passed through a evaporative cooler to add moisture and reducing the temperature through procedure (11-12), before it enters the conditioned space.

On the regeneration said, heat the outdoor air in heat exchanger (2) through procedure (1-2), then heated again through procedure (2-3), 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. Before it enters the desiccant wheel and remove moisture from desiccant through procedure (3-4), then exits to the outdoor air.

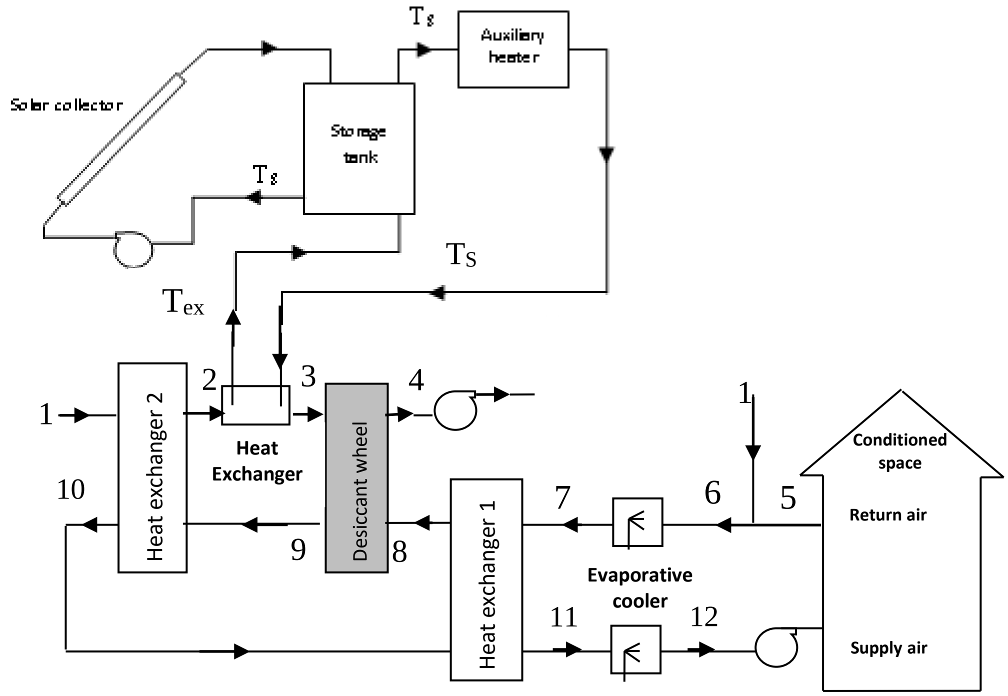


Fig.(4) Dunkle Cycle Desiccant Cooling System

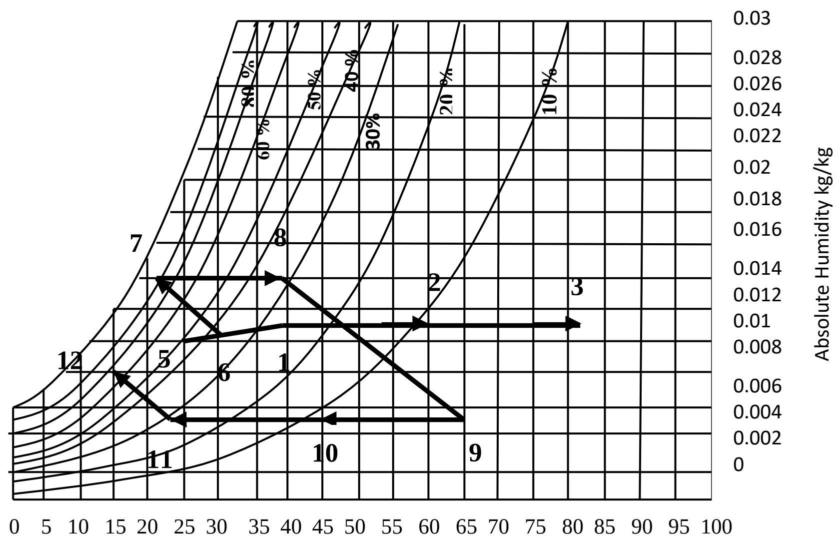


Fig.(5) psychometric processes

3. Modelling of the system.

Modelling of the desiccant air-conditioning system performance for space air conditioning depends on a group of parameters and/or conditions, implication the system's operation parameters, environmental conditions and space requirements. The development of a model is a necessary tool for the study of the relation of the above-mentioned parameters, on a design, control strategies, as well as performance analysis basis. The modelling of the desiccant system, for the layout presented in Fig. (4), refers actually to the integration of models of individual components of the system (desiccant wheel, heat exchanger, heater, evaporative cooler, mixing box and solar collector storage subsystem). Such a model has been developed for the purposes of this work, and is described as follows (the indices 1, 2, . . ., 12 impose different positions of the system as these are defined in Fig. (4)).

Several assumptions will be considered for simplify this study; the following are the most important:

1. The dehumidifier of infinite capacity (rotational speed of the wheel and the air mass flow rate do not influence the calculations).
2. Equal capacity rates for the two streams in the heat exchanger1 (mass flow rate and specific heat equal for both streams).
3. The heat exchanger and the evaporative cooler effectiveness are 90%.
4. The energy input for the water pump and air fans is neglected in the cycle analysis.
5. Indoor conditions recommended standard design conditions for a residential
6. Air conditioner are T_5= 24°C, RH_5=50% [30].
7. Influence of the air fans on rise air temperature is neglected.

4- Mathematical Modeling

4.1 Desiccant wheel

As deriving from the operation principle of the desiccant wheel, the dehumidifying capacity of the wheel depends to the rotational speed of the wheel, the flow rate (actually the velocity of air in the face of the wheel) and the temperature of the regeneration air stream. It also depends to the supply air stream conditions (temperature, absolute humidity), as well to the absolute humidity of the regeneration stream [9].

In the field of desiccant wheel modeling, there has been considerable work developed. Maclaine-Cross and Banks [10] have developed an analogy method, with reference to the rotating sensible heat exchangers (or regenerators), for predicting the coupled heat and mass transfer process in desiccant dehumidifier wheel (method of characteristics). On the basis of the evolution of this approach, i.e. the non-linear analogy method [11,12], the model of Jurinak and Banks [13] is referred. On the same field, this of analytical approach, Van Den Bulck et al. [14] introduced a wave analysis, proposing after all an e-NTU method. On a different basis of approach, Beccali et al. [15] propose a model based on the interpolation of experimental data obtained from the industry.

In this study, the model of Jurinak has been adopted, 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. The model consists of the following set of equations [16]:

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

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

(n=9,8,3, ports of Desiccant wheel)

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

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

Where, F1,F2 is Maclaine-Cross characteristic potentials, entering the desiccant wheel model of Jurinak, Tn air temperature (K),wn absolute humidity (kg/kg), (ηF1,ηF2) is F1, F2 characteristic potentials effectiveness, entering the desiccant wheel model of Jurinak.
The model assumes dehumidifier of infinite capacity (the matrix capacity rate is the matrix mass rate times the specific heat of the solid [17]), thus the rotational speed of the wheel and the mass flow rate of the process or regenerated air streams do not influence the calculations. (One has to note that the model considers equal capacity rates for the two streams, leading to equal mass flow rate for the two air streams). Required values of [η_{F1}, η_{F2}] = [0.08, 0.95] are set, which correspond to a high efficiency wheel, as the condition η_{F1}, $\eta_{F2} \leq 1$ is always valid and the ideal wheel refers to $\eta_{F1} = 0, \eta_{F2} = 1$.

4.2 Heat exchangers:

As deriving from theory [18], for equal capacity rates of the hot and cold side (which is our case), the heat exchanger effectiveness (η_{hx}) can be described by Eq. (5), for heat exchanger 1 and Eq. (6), for heat exchanger2 Analysis of theη_{hx} values for various operation conditions of a heat exchanger [10], permits the assumption of a steady typical value forη_{hx}, while there is no humidity exchange through the heat exchanger:

$$t_8 - t_7 = \eta_{hx1}(t_{10} - t_7) \quad \text{.......(5)}$$
$$t_9 - t_{10} = \eta_{hx2}(t_9 - t_1) \quad \text{... ....(6)}$$

And for equal the energy lost by the hot side and the energy gained by the cold side, we can obtain [18].

$$(t_8 - t_7) = (t_{10} - t_{11}) \ldots\ldots\ldots\ldots(7)$$
$$(t_2 - t_1) = (t_9 - t_{10}) \ldots\ldots\ldots\ldots(8)$$

4.3 Evaporative Coolers:

The effectiveness of an evaporative cooler is given by the following relation [19]:

$$\eta_{ev1} = \frac{t_{11} - t_{12}}{t_{11} - t_{wb-11}} \ldots\ldots\ldots\ldots\ldots(9)$$
$$\eta_{ev2} = \frac{t_6 - t_7}{t_6 - t_{wb-6}} \ldots\ldots\ldots\ldots(10)$$

Where η_{ev1},$\eta_{ev2}$is efficiency of evaporative coolers 1,2 respectively and t_{wb-11} and t_{wb-6} is wet-bulb temperature at point 11, 6 respectively, see fig.(1))
The evaporative cooling process is considered adiabatic, on a constant wet-bulb temperature line [20].

$$t_{wb-11} = t_{wb-12}$$
$$t_{wb-7} = t_{wb-6}$$

4.4 Mixing Box

In this box, the return air from the building is mixed with the outside air. A common process in air conditioning systems is the adiabatic mixing of the two moist air streams. The adiabatic mixing is governed by three equations [20]:

$$m_1 h_1 + m_5 h_5 = m_6 h_6 \ldots\ldots\ldots\ldots (11)$$
$$m_{6=} m_1 + m_5 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(12)$$
$$m_1 w_1 + m_5 w_5 = m_6 w_6 \ldots\ldots\ldots\ldots (13)$$

Where: -
m : Mass flow rate (kg/sec).
h : Enthalpy of the air (kJ/kg).
w : Humidity ratio (kg/kg).

4.5 Solar collector storage subsystem

This subsystem provides the energy required for the reactivation of the desiccant. Fig.(1) shows a schematic of the system component configuration chosen. The system consists of an array of solar collectors with a 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 aids in supplying the hot water at the required present regeneration temperature.

4.5.1. Collector:

The collector liquid with a water as the working fluid. Type of collector is flat plate. Area of collector array is chosen to be 75 m^2. Recommended flow rates of the working fluid and water loops are (0.01 - 0.02) kg/s m^2 of collector area [14]. Hence an average value of 0.015 kg/s m^2 is chosen. This gives a flow rate of 1.125 kg/s. Also, conventional design of a storage tank volume requires about 50 - 100 liters/m^2 of collector area [21]. Hence a mean value of 75 liters (0.075 m^3/m^2 of collector area) is chosen. This suggests a storage tank volume of 5.6 m^3.

Hottle-Whillier's equation gives a general expression for collector efficiency [22].

$$\eta_c = \frac{Q_u}{A_c\, I_t} = F_R(\tau\alpha)_n - F_R U_C \frac{(T_{ex}-T_1)}{I_t} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

where, ηc is the collector efficiency, Qu rate of useful energy gain by the collector, Ac collector surface area, It total incident radiation, FR The heat removal factor, (τα)n transmissivity-absorptivity product, T_1 ambient temperature, Tex inlet temperature of fluid of collector, and Uc collector overall heat loss coefficient.

The value of factors FR(τα)n and FRUc depend on the type of the collectors, layer of the cover glass and selective material. Typical values of these factors are shown in Table (1).

Table (1) The Value of $F_R(\tau\alpha)_n$ and F_RUc for Some Type of Solar Collector

Solar Collector Type	**$F_R(\tau\alpha)_n$**	**F_RU_c**
Flat-Plate, Selective-Surface, Single-Glass Cover	0.80[a]	5.00[a]
Flat-Plate, Selective-Surface, Double-Glass Cover	0.80[b]	3.50[b]
Evacuated Tubular Collectors	0.80[a]	Range 1-2[a]
Parabolic-Through Concentrating solar collector (PTC)	0.70[c]	2.5[c]

(a) from Fléchon, Lazzarin et al., 1999
(b) from Huang, Chang et al., 1998
(c) average data from Table 4.1 in Henning, 2004

Q_u can be expressed considering the collector energy balance [15].

$$Q_u = F_R A_c[I_t(\tau\alpha)_n - U_C(T_{ex} - T_1)] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (15)$$

4.5.2 Heat exchanger:

The heat exchanger used is of the parallel flow mode constant effectiveness type. For this, the maximum possible heat transfer is calculated based on the minimum capacity rate fluid and the cold and hot side fluid inlet temperatures [18], minimum capacity rate fluid in our case is the air.

$$Effectiveness = \epsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} \ldots\ldots \quad (16)$$

$$\epsilon = \frac{\dot{m}_a c_{pa}\,(t_8 - t_7)}{\dot{m}_a c_{pa}\,(t_s - t_7)} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. \quad (17)$$

$$\epsilon = \frac{t_8 - t_7}{t_s - t_7} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (18)$$

Heat balance in the heat exchanger can be obtained from the following equation:

$$\dot{m}_a c_{pa}(t_8 - t_7) = \dot{m}_W c_{pw}(t_s - t_{ex}) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. \quad (19)$$

Where, $\dot{m}_a$ is mass flow rate of the air, c_{pa} is specific heat of the air,t_8 is the air temperatures at point 8, t_7 is the air temperatures at point 7,$\dot{m}_W$ is mass flow rate of the water, c_{pw} is specific heat of the water, t_s is inlet temperatures of the water,$t_{ex}$is exit temperatures of the water.

4.5.3Storagetank:

The storage tank used is of the fully mixed type. The load flow enters at the bottom of the tank and the hot source stream enters at the top of the tank. The tank is well insulated with storage tank loss coefficient and area product (UA)s value of 11.1W/°C.

The water in the storage tank is assumed to be unratified at a uniform time dependent temperature Ts, resulting from the well mixing of hot water from the collector and cool water back from the heat exchanger. An energy balance on the storage tank gives [22]:

$$(MC_p)_s \frac{dT_s}{dt} = Q_u - Q_l - Q_{Tl} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (20)$$

Where:

$(MC_p)_s$ = mass and specific heat product of water in the tank.

t = time.

Q_u = rate of collected solar energy delivered to the storage tank.

Q_l = heating load supplied by solar energy via the load heat exchanger.

Q_{TL}= rate of energy loss from the storage tank.

Above equation can be rewritten in finite –difference form as follows:

$$(MC_p)_s \frac{\acute{T}_s - T_s}{\Delta t} = Q_u - Q_l - Q_{Tl}$$

Or

$$\acute{T}_s = T_s + \frac{\Delta t}{(MC_p)_s}(Q_u - Q_l - Q_{Tl}) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (21)$$

Where T_s and $\acute{T}_s$ are respectively, the old and the new temperatures of the tank at the beginning and the end of the time period (Δt).

Heating load supplied by solar energy via the load heat exchanger can be expressed by:

$$Q_l = (\dot{m}C_p)_s(T_S - T_{ex}) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (22)$$

Where:
$(\dot{m}C_p)_s$ = mass flow rate and specific heat product of water circulating through the loadheat exchanger.
$(T_S - T_{ex})$ = temperature drop of water through the load heat exchanger.

The rate of storage tank energy loss can be expressed by:

$$Q_{TL} = (UA)_s(T_S - T_1) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (23)$$

Where:T_1 = temperature of environment.

4.6 Solar fraction (SF):

The solar fraction is the amount of energy provided by the solar technology divided by the total energy required, the solar fraction thus is zero for no solar energy utilization, to 1.0 for all energy provided by solar.

$$\mathrm{SF} = \frac{\text{energy by solar}}{\text{total energy required}}$$

$$\mathrm{SF} = \frac{(\dot{m}C_p)_w(T_S - T_{ex})}{\dot{m}_a c_{pa}(T_8 - T_7)} \dots\dots\dots\dots\dots\dots\dots\dots (24)$$

4.7 Coefficient of performance (COP):

The term coefficient of performance has been devised to measure the effectiveness of refrigerating machines and is usually defined as the heat removed from the space to be cooled divided by energy input for the cooling system. Usually, the COP is used to analysis and compare cooling systems for their thermal performance.

$$\mathrm{cOP} = \frac{\text{heat removed by the system from the space}}{\text{Energy input to the system}}$$

$$\mathrm{COP} = \frac{(\mathrm{h}_5 - \mathrm{h}_{12})}{(\mathrm{h}_3 - \mathrm{h}_2)} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (25)$$

Where: h is the enthalpy of air KJ/Kg.
(The energy input for the water pump and air fans is neglected in the cycle analysis).

4.8 System Simulation.

To elaborate the technical feasibility of the system we have to use the system simulation in order to study the technical performance of the selected system at various working conditions. This tool is a very helpful one where input data for the location and system is provided to the code and output results is foreseen out of which we can decide whether the system is fit or not. This procedure is very common one in research and development activities, and we are following this way otherwise we have to go to laboratories work which is not in hands.

5. Description of The Used Program (code).

Microsoft FORTRAN Power Station 4.0 provides complete FORTRAN development environment for Windows NT (Intel) 32-bit platforms and integrates FORTRAN compiler technology with Microsoft's Developer Studio. It is an implementation of the FORTRAN programming language that supports the Fortran 66, Fortran 77, and Fortran 90 standards.

In the current study Fortran Power Station 4.0 is used to write two programs, one calculates the cooling load of the house, by use the Cooling Load Temperature Difference (CLTD) method. And other calculates the thermal conditions at the entrance and exit of each component of the system on the basis of equations (1 to 25).

6. Validation of the Jurinak model.

To validate the developed numerical model based on the Jurinak approach, the simulation results were compared with previously published studies that employed the same modeling framework for desiccant cooling systems. This validation strategy is commonly adopted when experimental data are unavailable and aims to demonstrate the credibility and reliability of the numerical predictions.

Naimeh and Faraa (2005) evaluated several desiccant evaporative cooling cycles, including the ventilation cycle, recirculation cycle, Dunkle cycle, and modified ventilation cycle, under the climatic conditions of Damascus. Their study reported coefficient of performance (COP) values for the Dunkle cycle ranging between 0.4 and 0.9, depending primarily on heat exchanger effectiveness [23]. In comparison, the present numerical model predicts COP values in the range of 0.6 to 1.0 under comparable operating conditions. This close agreement indicates that the developed model successfully reproduces the thermal performance trends reported in the literature and confirms the capability of the Jurinak model to realistically capture the behavior of Dunkle cycle desiccant cooling systems.

Furthermore, the influence of regeneration air temperature on system performance shows strong qualitative and quantitative agreement with the findings of Naimeh and Faraa (2005). Both studies demonstrate that increasing the regeneration temperature enhances the moisture removal capacity of the desiccant material, resulting in an improvement in COP up to an optimal temperature range. Beyond this range, further increases in regeneration temperature lead to diminishing performance gains due to higher thermal energy requirements. This characteristic behavior is well captured by the present model, further supporting its validity.

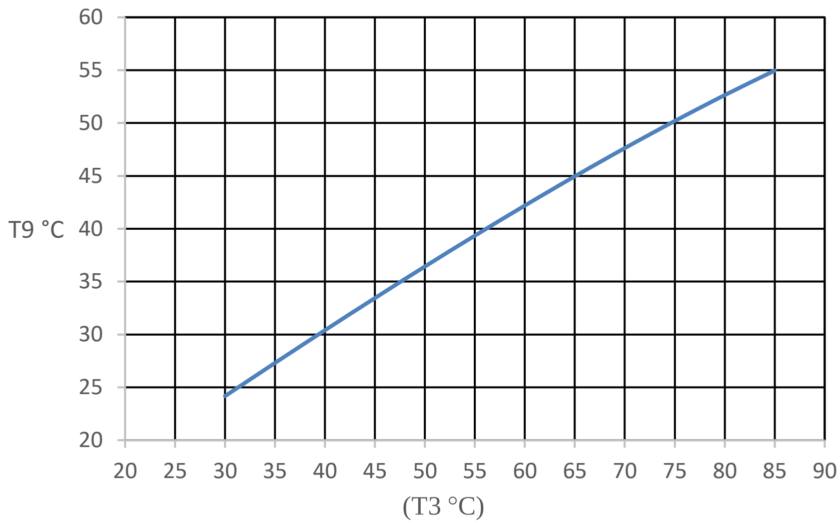
Additional support for the applicability of the Jurinak-based modeling approach is provided by Martínez et al. (2017), who developed and experimentally validated a dedicated outdoor air system (DOAS) incorporating a desiccant wheel and a vapor-compression refrigeration system. Their model, implemented in the TRNSYS simulation environment, employed the Type 1716 component from the TESS library, which is based on the Jurinak F1 and F2 potential equations. Validation was achieved through laboratory experiments, using the root mean square error (RMSE) to assess the agreement between measured and simulated air and refrigerant properties.

The results obtained with the developed model showed that the DOAS was able to maintain an indoor humidity ratio depending on outdoor conditions. Laboratory tests were also used to investigate the effect of changes in the regeneration air temperature and the process air flowrate on the process air humidity ratio at the outlet of the wheel. The results are consistent with the technical literature [24].

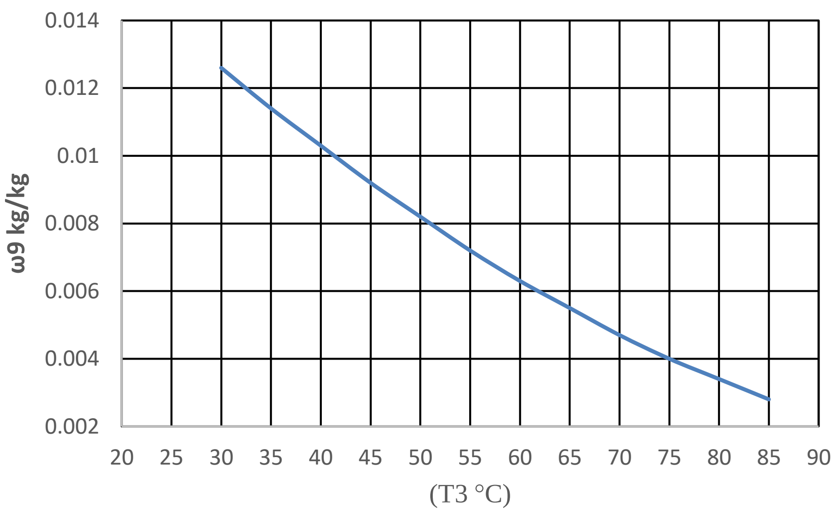
Given that the present work adopts the same Jurinak-based desiccant wheel model and simulation framework, the close agreement between numerical predictions and both experimental and numerical results reported in the literature confirms the reliability of the developed model. Consequently, the proposed model is considered suitable for evaluating the performance and technical feasibility of solar-driven Dunkle cycle desiccant cooling systems under Libyan climatic conditions.

7. RESULTS.

Fig. (6) and Fig. (7) shows the influence of the reactivation temperature (T3) on the temperature and absolute humidity of the supply air leaving the Desiccant wheel, (point 9, Fig. (1)).

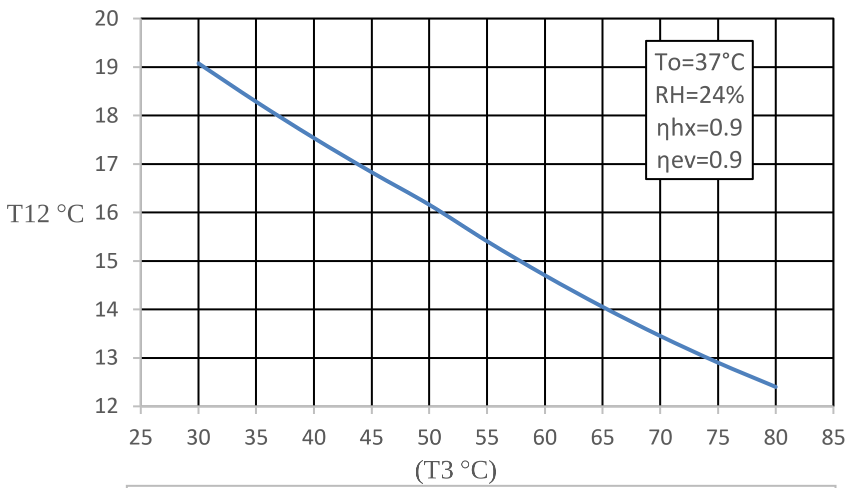


Fig(6).Influence of the reactivation temperature (T3) on the temperature of the supply air leaving the Desiccant wheel (T9)

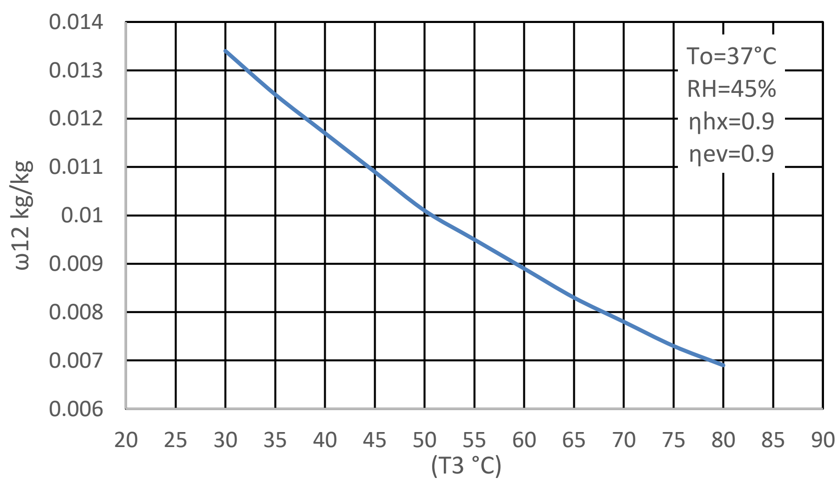


Fig(7).Influence of the reactivation temperature (T3) on the absolute humidity of the supply air leaving the Desiccant wheel

Fig. (8) and Fig. (9) shows the influence of the reactivation temperature on the temperature and absolute humidity of the supply air leaving the evaporative cooler 2 (point12, Fig. (4)).



Fig(8).Influence of the reactivation temperature (T3) on the temperature of the supply air leaving the evaporative cooler 2



Fig(9).Influence of the reactivation temperature (T3) on the absolute humidity of the supply air leaving the evaporative cooler 2

Fig.(10) Shows the Influence of reactivation temperature on the performance coefficient of the system.

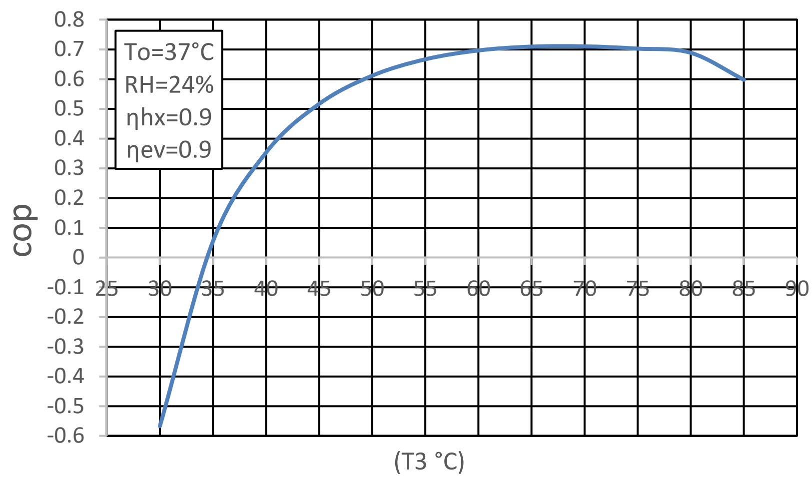


Fig. (10). Influence of the reactivation temperature (T3) on the COP of the system.

Fig. (11) shows average of the COP in each month, and average of the COP through the cooling season is 0.98.

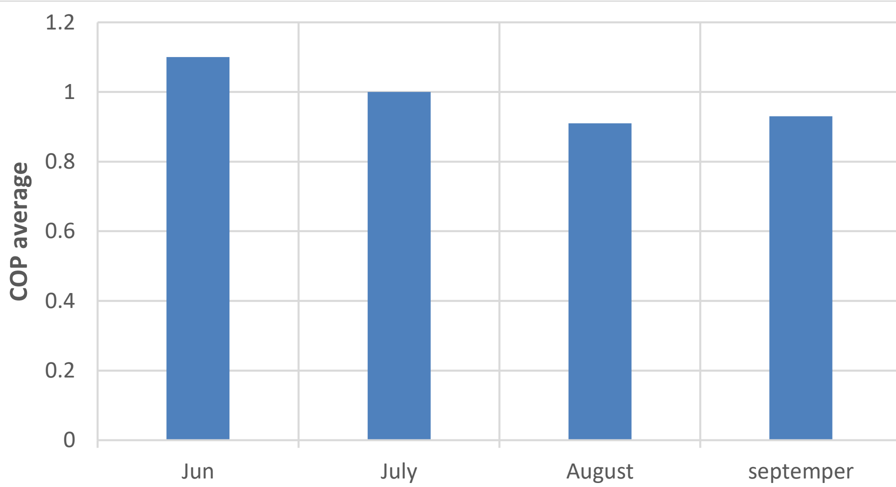


Fig.(11) Average of COP in the cooling season

Fig. (12) Shows the variations of the cooling load and cooling effect with time on the Sixth of Jun 2023.

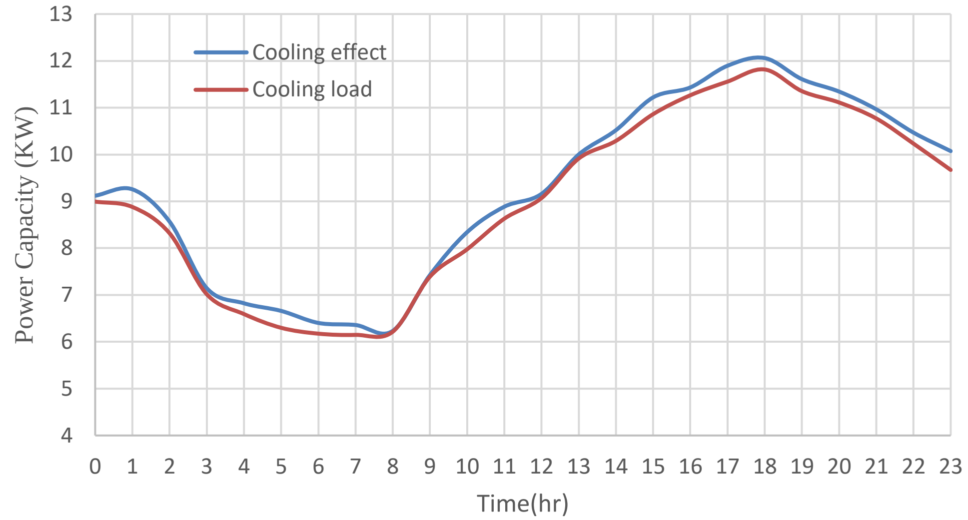


Fig.(12) variations of the cooling load and cooling effect with time

Fig. (13) Shows the contribution of solar energy in provide the required energy to operation the air conditioning system, in each month through the cooling season.

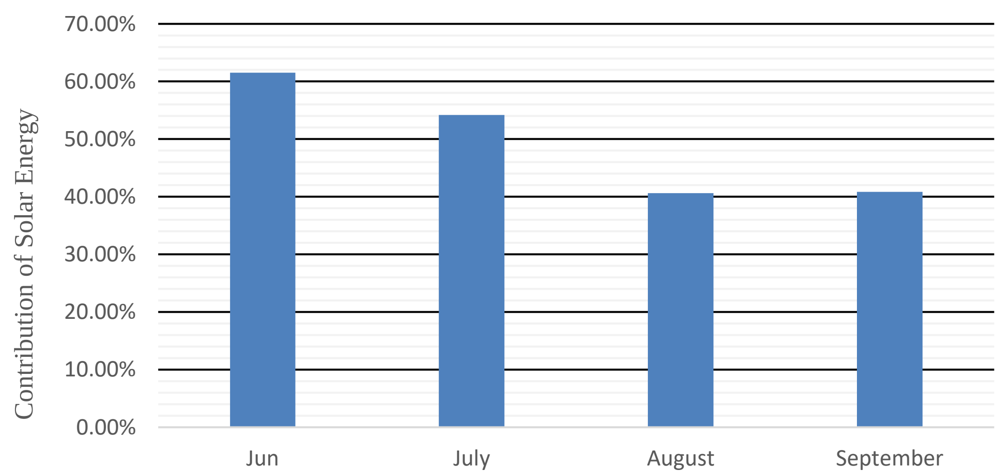
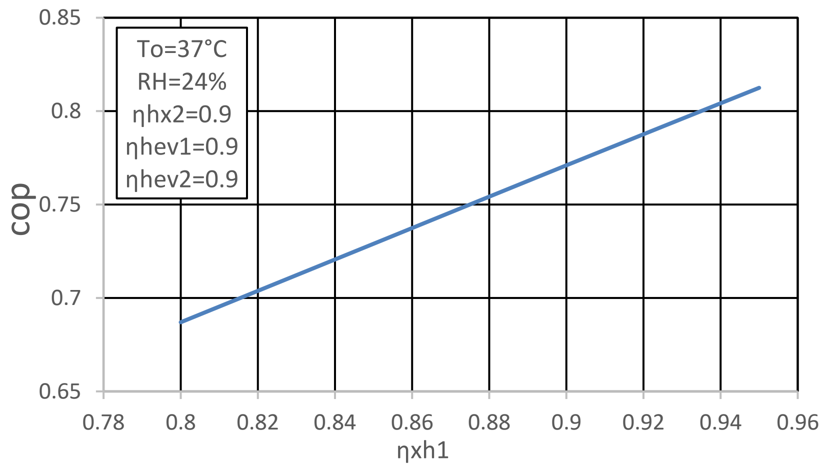


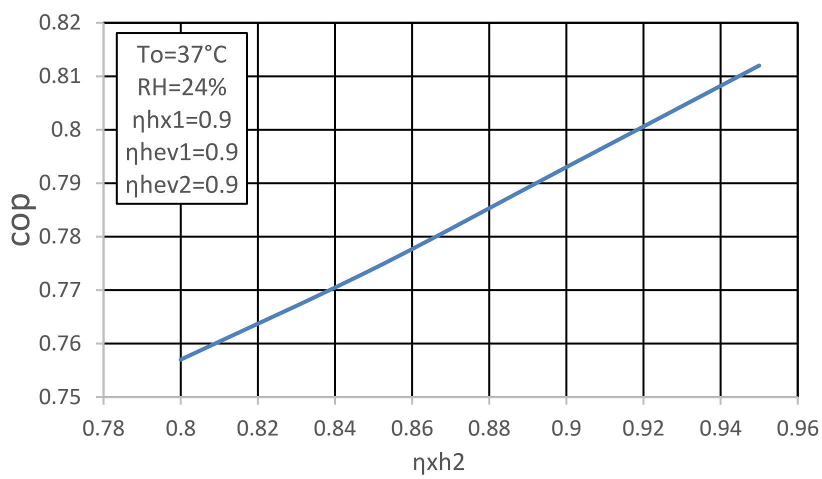
Fig (13). Contrbution Of Solar Energy

Fig. (14) shows the influence of the heat exchanger1 effectiveness on the COP.



Fig(14).Influence of the heat exchanger1 effectiveness on the COP.

Fig. (15) shows the influence of the heat exchanger2 effectiveness on the COP.



Fig(15).Influence of the heat exchanger2 effectiveness on the COP.

8. DISCUSSION

Fig. (6) shows an increasing in the reactivation temperature led to an increasing in the temperature of supply air leaving the Desiccant wheel. And to a decreasing in absolute humidity Fig. (7). That is due to the heat and mass transfer processes that occur in a desiccant wheel.

Fig. (8) shows an increasing in this temperature takes to a decreasing in the temperature of supply air leaving the evaporative cooler and to a decreasing in absolute humidity Fig. (9). This is because the increase reactivation temperature lead to a decreasing in absolute humidity at point 9 (ω9) and then decreasing in wet bulb temperature. As shown in evaporative cooler equation$T_{12} = T_{11} - \eta_{ev2}(T_{11} - T_{W11})$a decreasing in wet bulb temperature lead to a decreasing in T_{12} .

Fig. (10) Shows the Influence of reactivation temperature on the performance coefficient of the system, In general note that coefficient of performance increases with increasing reactivation temperature until reaching the maximum value, after it starts decreasing. Because it initially increases cooling effect with increasing reactivation temperature at greater rate than the increase in input energy up access to great value to the coefficient of performance, and then cooling effect increases with increasing temperature reactivation at a lower rate of increase in input energy which leads to a decrease in the coefficient of performance.
Note that the COP were negative at the reactivation temperature Less than 35°C, implying that enthalpy of indoor (point 5, Fig. (4)) was lower than enthalpy of supply (point 12, Fig. (4)) from Eq.(25). This shows that the room condition was not comfortable enough in most of the time.

Fig. (12) Shows the variations of the Cooling load and Cooling effect with time on the Sixth of Jun 2023, it shows that the system is able to provide thermal comfort inside the building.

Fig. (13) It shows that in the Jun month was the contribution of solar energy 61%, while in September month it decreases to about 40%, due to increased temperature and humidity during the cooling season, which leads to an increase in the power required to operate the system.

Fig.(14) and Fig.(15)shows an increasing in the heat exchanger1and 2 effectiveness lead to an increasing in COP, note that the performance coefficient is a strong function of heat exchangers effectiveness.

9. CONCLUSIONS AND FUTURE WORK.

This study presents a solar Dunkle cycle 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 0.98. 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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