SOME ENGINEERING FACTORS AFFECTING THE PERFORMANCE OF SOLAR POWERED ADSORPTION REFRIGEARTION SYSTEM

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ABSTRACT

In the present study three different collector cover materials, single or, double glazing and Polyethylene-plastic cover materials three different adsorbent material masses 12, 14 and 16 kg were used respectively for the determination of the optimal collector cover material and adsorbent mass proportion to methanol quantity. Collector thermal efficiency as affected by daily incident solar radiations kW.hr/m2/day and their effect on methanol saturation which directly reflected on the coefficient of performance for the refrigeration cycle were studied. Refrigeration load in ton of refrigeration and economical evaluation were also studied. Results showed that daily average solar radiation during these experiments was ranged between 6.6 to 7.9 kW.h/m^2.day, the double glass cover material has the largest efficiency than both of single and polyethylene-plastic sheet, the thermal efficiency of single, double glazing and polyethylene plastic sheet cover were30.27%, 39.83% and 30.22% respectively. It is clear that the double glazing cover is the optimal cover material of all studied cover materials. Higher values of C.O.P % and refrigeration load were recorded for the 14 kg carbon mass than that of 12and 16 kg carbon mas.

2.INTRODUCTION

Egypt is one of the countries having solar energy in abundance. The solar energy incident on the Egyptian land has a magnitude of 12-30 MJ / m^2 /day, and the sunshine duration is between 3500 - 4500 hr per year. Solar energy can solve apart of energy demand problem however the use of solar energy in Egypt could play a useful role in satisfying energy requirements of most urban areas in appropriate circumstances. **(El-Metwally 2005)**.

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The solar energy is used as a renewable source of energy for many applications. Solar refrigeration forms one of the most important applications of solar energy. Recently, solar energy has received interest as an alternative energy source for refrigeration systems, especially in places where electricity is expensive. **Darwesh et al. (2010).** The solar adsorption refrigeration technology has attracted some researchers since last decade because it is clean, cheap and simple for use in air conditioning, ice making, food preservation and vaccine storage. **Qasem and El-Shaarawi (2013)**, Performed experiments with a solar powered ice maker that had activated carbon– methanol as working pair. In This icemaker, had a COP ranging from 0.12 to 0.14, analyzing the temperature gradient within the adsorbent bed, the authors concluded that in order to improve the performance of this system, the heat transfer properties of the adsorber must be enhanced. This could be achieved by increasing the number of fins or using consolidated adsorbent. **Li et al. (2002),** the characteristics of solar cooler are affected by many parameters. Generally speaking, we can divide these parameters into two types. One type is the parametric parameters, such as dimensions, material and characteristics of collector, which decides the physical property ofcollector. Another type of parameters is environmental parameters, such as radiation intensity, ambient temperature and wind speed, which affect the running quality of system performance. **Wang et al. (2000).**

The main objectives of the present study are:
1. Constructing and manufacturing solar powered adsorption refrigeration system,
2. Study some engineering factors affecting the coefficient of performance and refrigeration load for the system.

3. MATERIALS AND METHODS

MATERIALS

The solar powered adsorption refrigeration system was designed and constructed in the collage of Agr. Eng. Al-Azhar Univ. Nasr City Cairo. The refrigeration system was fabricated and processed in the workshop of the Faculty of Agr. Eng. and other private workshops. The experimental part of this study was carried out in Dairut – Asyut (ديروط –

اسيوط) at 27°32 N and 30°49 E since the availability of sunshine higher and clear solar insolation.

System Description

The solar adsorption refrigeration system consists of four main components: a solar collector (generator) directly in contact with adsorbent material, a condenser, an evaporator and cooling chamber as shown in **Fig. (3.1)**

Fig.(3.1) Photograph of The solar powered adsorption refrigeration system setup.

Solar Collector (Generator)

The solar collector in consisted of three parallel layers, the first one is the adsorbent plate directly in contact with seven fins of stainless steel sheet longitudinally fixed underneath the collector absorber plate 0.1 m apart and 0.001 m thick to improve the heat and mass transfer process. The second is a parallel perforated steel sheet 0.001 thick. The activated carbon is packed between the first and second layer included fins. The third layer is installed 0.01 m part of the second layer. Finally the steel

plate box is placed inside a wooden box having gross dimensions of 0.85m long, 0.85m width of $0.72m^2$ area. The walls and bottom was constructed of wooden panels 0.03m thick, the space between the bottom and the steel plate box are insulated by one layer of insulating materials is made of glass wool 0.03mm thick. **Fig. (3.2)** The adsorbent bed is made of flat plate stainless steel box, having surface area 0.80×0.80 m2, three different masses of adsorbent bed of activated carbon Maxsorb III type were packed and sealed inside the steel plate box and between fins. The absorber plate surface and the adsorbent box are made of steel sheet having gross dimensions of 0.82 m long, 0.82 m width (0.67 m2 area) and 0.001m thick painted by a blackboard paint mixed with 50% by weight of a talc powder i.e. (too fin zinc oxide, nontraditional paint) **(Ghanem. 2003).** The adsorbent bed was covered with a clear glass 3 mm thick in single or double form to reduce the reflection of radiation and heat losses by convection, or sheet of polyethylene-plastic plates. In order to guarantee better heat transfer between the front side and the adsorbent, seven fins approximately 0.10 m a part made of stainless steel are installed inside the adsorbent bed box directly in contact with the bottom side of the absorber plate and the activated carbon to improve the transfer of methanol vapor through the activated carbon layer in desorption and adsorption process, a parallel stainless steel sheet bottom 0.001m thick is uniformly perforated and installed in the adsorbent bed to permit uniform distribution of methanol in the adsorbent. For the flow of refrigerant (methanol) from the bed a hole is made at the third lower side of bed and coupling is made to connect copper tube of condenser to the bed. The schematic diagram of the adsorbent bed is shown in **Fig.(3.3)**

Condenser

The condenser function in the system is to condense vaporized methanol flows from bed to condenser through copper tube and cooling it by air applied on fins of condenser, thus vaporized methanol is converted into liquid methanol. This liquefied methanol now goes to evaporator, It has the following specification:

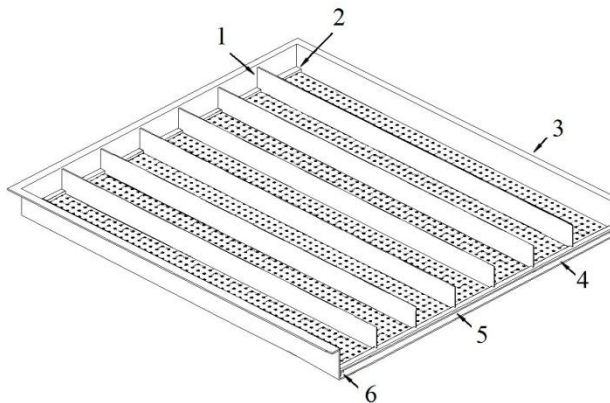- Tube material : copper material
- Tube diameter(condenser) : 9.525 mm

- Number of tubes used : 1
- Fin material : Aluminum
- Total number of fins : 14
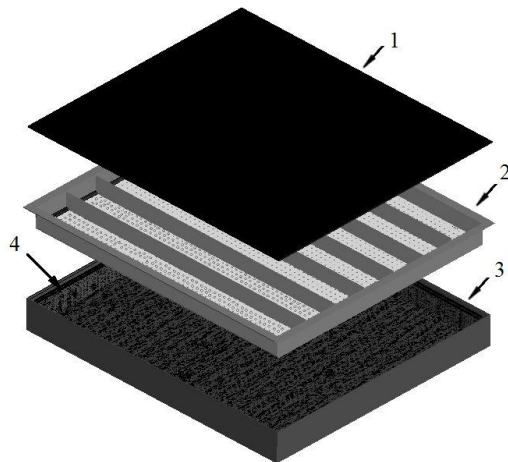
1 Front plate   2 Adsorbent bed
3 Wooden box
                                       4 Insulating materials

Fig. (3.2) Schematic Of The Different Parts Of The Collector.

1. Fin.   2 Supporting plate.
3 Flat plate stainless steel box.   4 False bottom
5 Perforated steel plate.   6 Angle (L) install perforated plate.

Fig.(3.3) Schematic Of Adsorbent Bed

Evaporator

The evaporator must have sufficient volume to collect the entire condensed methanol. In order to enhance the heat transfer effect, the heat exchange surface is designed as a series of four trapezoidal cells with dimension of 0.3m × 0.3 m × 0.1 m. Evaporator is made up of stainless steel sheet 0.001m thick. The evaporator is placed inside a cooling chamber, the lower portion of evaporator is made up of trapezoidal shape with dimensions of 0.03m × 0.04m, to increase surface area and ensures better heat transfer, hole is made on upper side of evaporator to be fitted to the end of the condenser pipe.

Cooling Chamber

The cooling chamber was constructed of stainless steel sheet 0.0005 m thick, with dimensions of 0.5 m long, 0.5 m wide and 0.5m high, the side walls and bottom of cooling chamber were insulated by two insulating materials, the first one is a foam layer 0.05m thick, the second is glass wool 0.03m thick, all of them were located in a wooden panel box which is made of wooden panels with dimensions of 0.58m long and 0.58m wide, and 0.58m height, The chamber door was made of wooden panels with gross dimensions of 0.58m long and 0.58 m wide, the door was tightly sealed by a rubber gasket.

Methanol

Methanol also known as methyl alcohol, which is a chemical with the formula CH_3OH, methanol acquired the name "wood alcohol". Methanol used in the present work is made in India purity of 99.8%

Activated Carbon

The activated carbon Maxsorb III type, and methanol is best suitable pair for the adsorption refrigeration process and also refrigerant methanol has zero ozone depletion potential and very low global worming potential, thus the refrigerator is eco-friendly. The activated carbon is a high density adsorbent and provides maximum volume activity, high hardness and mechanical strength ensures negligible losses during backwashing, air scouring and multiple reactivations.

Measuring Instruments

Solar intensity device

A black and white pyranometer designed, constructed and tested by **Ghanem (1989)** and calibrated in the Solar Energy Department, National Research Center (NRC), Giza, Egypt. It was used to measure the instantaneous total solar radiation incident on the horizontal plain as shown in **Fig. (3.4).** The output voltage of the pyranometer is converted to solar intensity as one mV. equal to 182.6 W/ m^2.

Temperature Sensors

Temperatures were measured using LM 35 sensors. The LM35 series are precision integrated-circuit temperature sensors, whose output voltage is linearly proportional to the Celsius (Centigrade) temperature. The LM35 does not require any external calibration their accuracies are ±1⁄4°C at room temperature and ±3⁄4°C over a full −55 to +150°C temperature range, **Fig. (3.5)** Temperatures were measured in ten points: front and rear face of the collector, on condenser, on evaporator and the cold chamber.

The LM 35 temperature sensors are connecting to the 10 way selector switch as shown in **Fig. (3.6)** in order to facilitate the transition between the measuring points. Their specifications are:

- 10 way selections to 1 output
- Plastic screw spacers
- Up to 5A current rating for each input
- Can be used for selection data or power signals
- Size: 75×65 mm

The outlet of selector switch and the positive terminal are connecting to digital multimeter as shown in **Fig. (3.7)** to read the system temperatures.

VANE Meter

The ambient temperature in °C, relative humidity (%) and air velocity were measured using VANE type anemometer, 3 in one, professional,

Pressure Gauge

The system pressure can be measured between the collector and the condenser during desorption time and between the collector and the

evaporator at adsorption process by glycerin Pressure gauge type, this device was made in china.
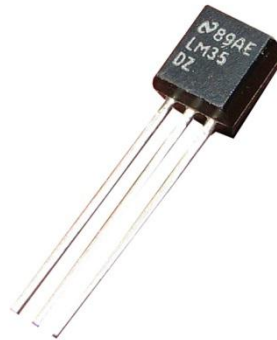

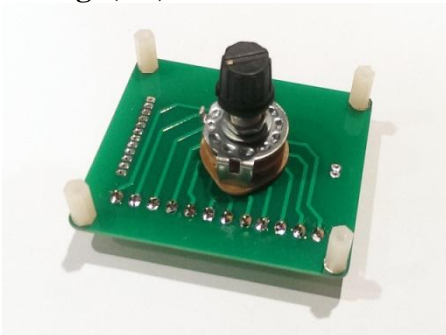**Fig. (3.4): The solar meter**

Fig (3.5): The LM 35 Sensors

Fig (3.6): Selector Switch

Fig (3.7): The Digital Multimeter

METHODS

In the present work some engineering factors affecting the performance of solar adsorption refrigerator were studied follows:

Collector efficiency calculations

The amount of solar radiation received by the collector is:

$$\mathbf{Q_i = I\ \tau \propto A} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (\mathbf{3.1})$$

Where:

I = the intensity of solar radiation, in W/m2,

A = a collector surface area, in m2

τ = is the transmittance of collector cover,

$\propto$ = is the absorption of plate collector.

The rate of heat loss (QL) depends on the collector overall heat transfer coefficient (UL) and the collector temperature.

$$Q_L = U_L A \left(T_p - T_a \right) \ldots \ldots \ldots \ldots \ldots \ldots (3.2)$$

Where:

T_p = average Flat-plate collector temperature, °c,

T_a = ambient temperature, °c, and

U_L = the collector overall heat transfer coefficient, W/m2.°c

The overall heat transfer coefficient of the collector UL is expressed by

$$U_L = U_T + U_B + U_S \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3.3)$$

Where U_T, U_B and U_S are the heat losses coefficients of the top, bottom and sides of the collector (generator/absorber), respectively. Since U_S is too small it can be neglected **(Qasem and El-Shaarawi, 2013)** U_T is calculated according to **Duffie and Beckman, 2013.**

$$U_T = \left[\frac{Ng}{\frac{C}{T_p}\left[\frac{T_p - T_a}{Ng + f}\right]^e} + \frac{1}{h_w} \right]^{-1} + \left[\frac{\delta(T_p + T_a)(T_p^2 + T_a^2)}{\left[(\varepsilon_p + 0.0.0059Ngh_w)^{-1} + \frac{2Ng + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - Ng\right]} \right] \ldots \ldots (3.4)$$

Where:

N_g = number of glass covers

f = (1 + 0.089hw − 0.1166hwεp) (1 + 0.07866Ng)

C = 520(1 − 0.000051β2) for 0°< β < 70°; for 70°< β < 90°, use β= 70°

e = 0.430(1 − 100/Tp)

$β$ = collector tilt (27.32 o)

$ε_g$ = emittence of glass and plastic sheet are 0.88 ,0.90 respectively.

$ε_p$ = emittence of plate

T_a = ambient temperature (°c)

T_p = mean plate temperature (°c)

$δ$ = Stefan-Boltzman constant (5.6704×10-8 W/m2 °c 4)

h_w = wind heat transfer coefficient (W/m2°c). And equal h_w = 5.7 + 3.8V when the wind speed equal to 5 m/s and when it was more than 5 m/s the following equation following can be used $h_w = 7.44(v^{0.8})$

The back losses coefficients UB depend on the insulation material and its *t*hickness which can be evaluated by:

$$U_B = \frac{K_s}{L_s} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots . . \ldots \ldots \ldots . (3.5)$$

Where:

Ks= Thermal conductivity of the insulation material. W/m.°c, and

Ls = The thickness of the insulation material, mm

Thus, the rate of useful energy extracted by the collector Qu expressed as a rate of extraction under steady state conditions is equal amount lost by the collector to its surroundings sub stracted form the rate of useful energy absorbed by the collector, less the. This is expressed as follows:

$$Q_u = Q_i - Q_L \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (3.6)$$
$$= I \tau \propto A - U_L A (T_p - T_a) \dots\dots\dots\dots\dots\dots\dots. (3.7)$$

Useful heat of the collector can also be measured by the amount of heat carried away by the fluid i.e methanol desorbed by activated carbon and passed through the condenser vice versa as,

$$Q_u = m\ Cp\ (T_p - T_i) \dots\dots\dots\dots\dots\dots\dots\dots\dots. (3.8)$$

Where:

T_i inlet fluid temperature, °C

Using equation (3.9) heat removal factor of the system FR can be found as follows:

$$F_R = \frac{m\ Cp\ (T_p - T_i)}{A[I \tau \propto -U_L\ (T_p - T_a)]} \dots\dots\dots\dots\dots\dots\dots\dots. (3.9)$$

The maximum possible useful energy gain in a solar collector occurs when the whole collector is at the inlet fluid temperature. The actual useful energy gain Q_u, is found by multiplying the collector heat removal factor FR by the maximum possible useful energy gain. This allowed by rewriting of equation (3.8) as:

$$Q_u = F_R A[I \tau \propto -U_L\ (T_i - T_a)] \dots\dots\dots\dots\dots\dots. (3.10)$$

Equation (9) is a widely used for measuring collector energy gain and is generally known as the "Hottel Whillier-Bliss equation" **(Smith., Weiss.1977)**

The collector efficiency η defined as the ratio of the useful energy gain Qu to the incident solar energy over a particular time period:

$$\eta = \frac{\int Q_u\ dt}{A \int I\ dt} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (3.11)$$

The instantaneous thermal efficiency of the collector is:

$$\eta = \frac{Q_u}{IA} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (3.12)$$

$$\eta = \frac{F_R A[I \tau \propto -U_L\ (T_i - T_a)]}{IA} \dots\dots\dots\dots\dots. (3.13)$$

$$\eta = F_R \, \tau \propto -F_R \, U_L \left(\frac{T_i - T_a}{I}\right) \dots \dots \dots \dots \dots \dots \dots (3.14)$$

Numerical integration

Numerical integration used in the present work is derived using Simpson's rule as reported by Toledo (2007), this procedure assumes a parabolic curve between area increments. The curve is divided into an even number of increments, therefore, including the value of the function at the lower and upper limits, (there will be i values of the height of the function and $(i - 1)$ area increments. The number of increments $(i - 1)$ must be an even integer. Thickness of area increments must be uniform, therefore, for the limits x = a to x = b, $\Delta x = (b - a)/(i - 1)$. Simpson's rule is as follows:

$$\int_a^b f(x)dx = \frac{x}{3}\,[F(a) + 4F(1) + 2F(2) + 4F(3) + 2F(4) + \cdots 2F(i - 2)$$
$$+ 4F(i - 1) + F(b)] \dots \dots \dots \dots \dots \dots \dots \dots \dots (3.15)$$

Adsorption and desorption calculations
Adsorption equilibrium

The adsorption and desorption concentrations (x) are usually determined by Dubinin–Astakhov equation (**Critoph , 1991**):

$$X(T, P) = x_o \, \exp\left[-D\left(T\,Ln\left(\frac{p_{sat}}{p}\right)\right)^n\right] \dots \dots \dots \dots \dots \dots \dots (3.16)$$

Or

$$X(T, T_{sat}) = x_o \, \exp\left[-K\left(\frac{T}{T_{sat}} - 1\right)^n\right] \dots \dots \dots \dots \dots \dots \dots (3.17)$$

Where:
x concentration ratio of adsorbate inside adsorbent (kgm / kgac)
K adsorbent constant
x_o limited adsorption capacity,
D Dubinin–Astakhov constant (°c -1)
n Dubinin–Astakhov constant
P system pressure (kPa)
T temperature (°c)
Sat saturated
The parameters xo, D, n Special adsorbent material and given especially Table (1) **(Qasem and El-Shaarawi 2013)**

Table (3.1): Parameters xo, D , n, ρ and CP in Activated Carbon

Activated carbon	x_o (kg kg^1)	D ($°c^{-1}$)	n	ρ (Kgm^{-3})	C_P (kJ kg^{-1}°c^{-1})	k
Maxsorb III	1.24	4.022×10^{-6}	2.0	281	0.93	13.38

Thermodynamic analysis of the refrigeration cycle

Setp: 1   Isosteric heating process

In this process the heat absorbed by the solar collector is utilized to heat up the activated carbon **Q_{A-B}** as shown by a line A–B, in Clapeyron diagram **Fig. (3.8).** Thus, the isosteric sensible heat addition is expressed as:

$$Q_{A-B} = M_{ac} (cp_m + x\, cp_{ac}) (T_B - T_A) \dots\dots\dots\dots\dots\dots (3.18)$$

Where:

M_{ac} mass of activated carbon (kg).,

Cp_{ac} , cpm specific heat at constant pressure (kJ kg-1 °c-1) that given Table (1)

T_A Temperature at point A. This temperature is taken at 8:00 AM session at the start of isosteric heating at adsorbent bed.

T_B Temperature at point B, This temperature is taken at 10:00 AM of the day and at the end of isosteric heating and start of desorption process at adsorbent bed. **(Nilesh et al. 2014b)**

Setp: 2   Isobaric desorption process

The desorption process is represented by the line (B–D). in Fig. (3.8) the heat provided to desorb methanol during this process Q_{B-D} can be written as:

$$Q_{B-D} = M_{ac} (cp_m + x\, cp_{ac}) (T_B - T_A) + M_{ac}\, \Delta H (x_B - x_D) \dots..(3.19)$$

Where:

x_B adsorption capacity at point B, which calculated according to Dubinin–Astakhov equation

x_D adsorption capacity at point D. which calculated according to Dubinin–Astakhov equation

ΔH heat of desorption or adsorption per unit mass of methanol (kJ kg-1), it can also be estimated as a function of latent heat (Le) of methanol **(Critoph , 1991)**

$$H = L_e \left(\frac{T}{T_{sat}} \right) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (3.20)$$

The summation of the desorption heat together with the isosteric heat is called the generation or regeneration energy and is given as:
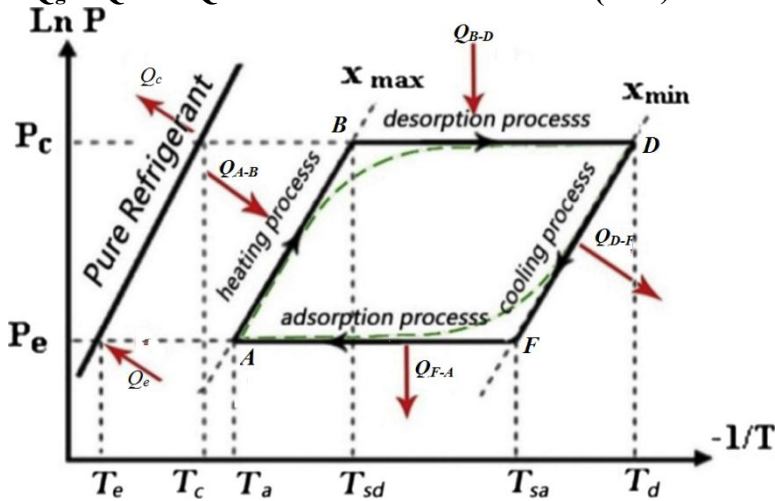
$$Q_g = Q_{A-B} + Q_{B-D} \ \dots\dots\dots\dots\dots\dots\dots\dots.(3.21)$$

Fig.(3.8). Schematic view of the adsorption process on Clapeyron diagram

Performance calculations for system

Thermal analysis of evaporator

The quantity of heat transferred in the evaporator during refrigeration process is:

$$Q_{eva} = Le \ Mac \ \Delta x \ \dots\dots\dots\dots\dots\dots\dots..\dots\dots\dots.(3.21)$$

Where:

Δx the difference between of the minimum and maximum concentration value $(x_{max}-x_{min})$ which calculated according to Dubinin–Astakhov equation.

The coefficient of performance

The performance of the refrigeration system alone is described by the coefficient of performance of its cycle (**COP**) without including the solar collector performance. (**Qasem and El-Shaarawi (2013)**).

$$COP = \frac{Qeva}{Q_g} = \frac{Le \ Mac \ \Delta x}{Q_{A-B}+Q_{B-D}} \ \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.(3.22)$$

Refrigeration load

The refrigeration load is the rate of heat energy removal from a given space (or object) in order to lower the temperature of space to a desired

level (**Paul Singh and Dennis 2014)** The refrigeration load of any product   store is made up of the basic components as follow:

- **Heat removed form cold room.**

$$Q_r = \frac{m_a c_{p\,(T_o - T_i)}}{t} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.23)$$

Where:

  Q_r heat removed form cold room kW,

  m_a mass of air inside cold room,

  Cp specific heat of air kJ/ kg. °c. ,

  T_o temperature of air outside the refrigerator °c, and

  T_i temperature of air inside the refrigerator °c

- **Heat leakage (conduction through)**

  QL = U A (To − Ti)………………….……….……….(3.24)

  Q$_{Ref.\ load}$ = Q$_r$+Q$_L$………………………........……(3.25)

Where:

  Q_L rate of heat transmitted through wall, kW.,

  U overall head coefficient kW/ m2.°c.,

  A surface area of refrigeration unit m2.,

  T_o temperature of air outside the refrigerator °c, and

  T_i temperature of air inside the refrigerator °c.

<u>4.RESULTS AND DISSCUSIONS</u>

Effect of collector cover material on collection efficiency as affected by ambient conditions

1.Solar intensity and ambient conditions

For evaluating the effect of cover material on collection efficiency of the solar collector or generator ambient conductions were measured: hourly solar intensity, ambient temperatures and wind speeds respectively. Fig.(4.1),(4.2) showed solar intensity and ambient air temperature distribution as affected by local time during refrigerator testing. It is clear that solar intensity has a trend of approximately normal distribution curve before noon and afternoon, due to the physical phenomena of the nature of the solar radiation and the measuring instrument as it is electromagnetic flux. Maximum values of the solar intensity were at the noon time ranged between $1.0 - 1.24$ kW/m^2 and minimum values at morning and afternoon which are ranged between 0.51-0.73 kW/m^2 and

0.39- 0.65 kW/m^2 before and after noon respectively. It is also clear that ambient air temperature do not obey the normal distribution trend curve as solar intensity which may be interpreted by the interaction of various effects between ambient air humidity and wind speeds.
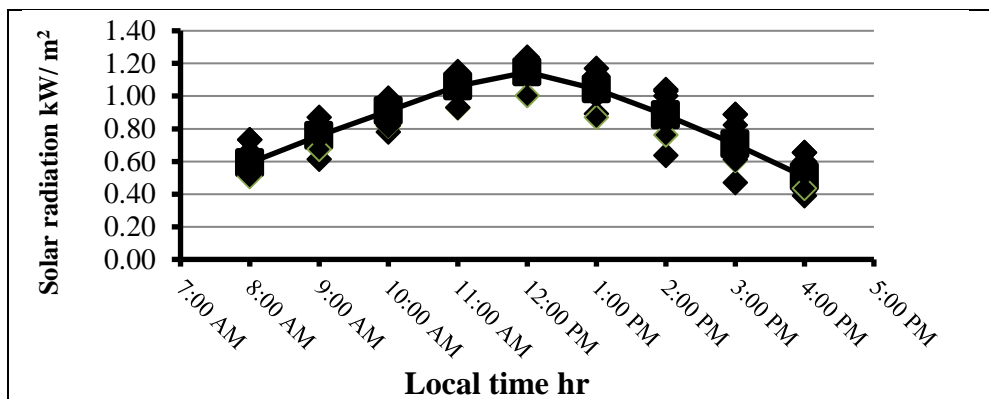

Fig (4.1): Solar intensity distribution as affected by local time during experiments.
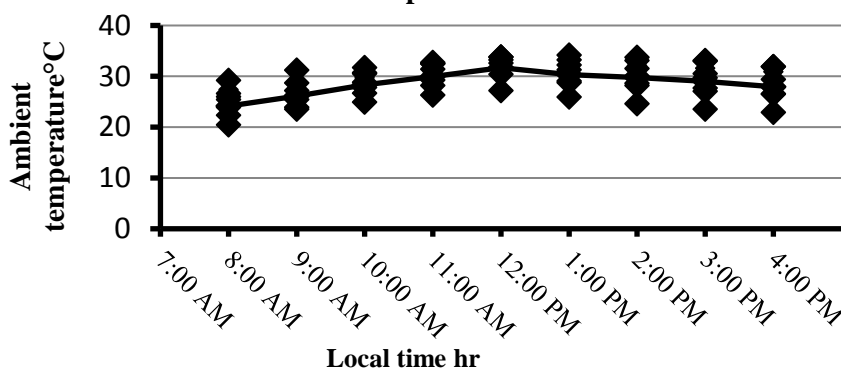

Fig (4.2): Ambient air temperature distribution as affected by local time during experiments.

2.Solar collector efficiency as affected by cover material type

In present study three different cover materials namely: single glazing, double glazing and polyethylene-plastic sheet were tested. Fig (4.3) showed the effect of cover material on thermal efficiency of the collector or generator as affected by local time. It is clear that double glass cover material has the highest average efficiency 39.83% compared to 30.27% and. 30.22% for both single and polyethylene-plastic sheet respectively.

That may be interpreted by lower total daily heat losses and higher total daily useful heat 0.922 kW/day and 2.671 kW/day of double glazing compared to 1.116, 1.165 kW/day and 1.522, 1.961 kW/day for both single glass and polyethylene cover respectively. Previously results discussed may be interpreted by stagnant insulation of air gap between the two consecutive glass covers than that of one glass cover or polyethylene-plastic sheet as depicted in Figs.(4.4) and (4.5) respectively.

Thermal efficiency of single, double glazing and polyethylene plastic sheet cover were decreased form morning to afternoon. Thermal efficiencies of both single and polyethylene-plastic cover were approximately the same for all same local time form morning to afternoon. The Maximum and minimum average efficiency values for both single and polyethylene plastic covers ranged between 33.41 to 27.98 % for all local day times with standard deviation ranged between 1.13 to 0.01 and too small coefficient of variation ranged between 0.0337 to 0.0003.Comparing double glazing with average values of both single and polyethylene plastic covers based on thermal efficiency increment percent as:

$$\eta_{\text{Increment}} = \frac{\eta_{\text{Double}} - \eta_{\text{avg.}}}{\eta_{\text{avg.}}} \dots \dots \dots \dots \dots \dots \dots . \dots \dots . (4.1)$$

Data showed that the maximum and minimum efficiencies of the double glass cover collector were increased from 42.83 to 35.89 % than that of both single and polyethylene cover respectively. From previous discussion, it is clear that the double glazing cover is the optimal cover material of all studied cover materials in present work.

Effect of adsorbent mass load on the refrigeration performance
Evaluating the effect of carbon mass on the coefficient of performance of the refrigerator

Table (4.1), Fig. (4.6) showed that the coefficient of the performance C.O.P % of the double glazing collector for all studied carbon mass materials tested were increased linearly by increasing the average daily solar radiation calculated using the Simpson rule for numerical integration and satisfied only with the range of 6.6 to 7.9 kW.hr/m2/day. It is also clear that there is a direct proportion between the COP % and the average daily solar radiation. Higher values of COP % were recorded

for the 14 kg carbon mass than that of 12and 16 kg carbon mass respectively, these results agreed with that reported by **Qasem and El-Shaarawi (2013),**
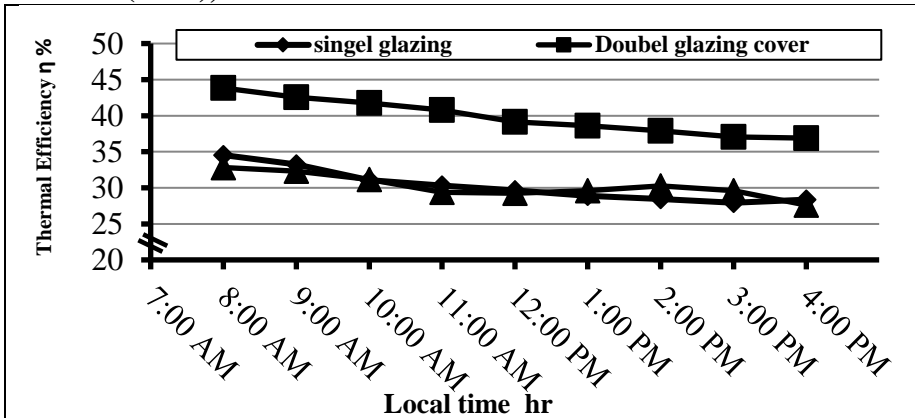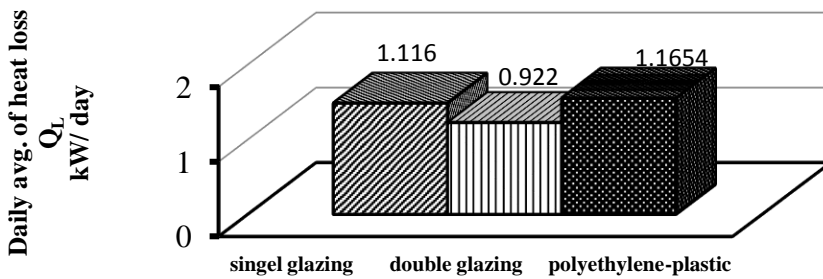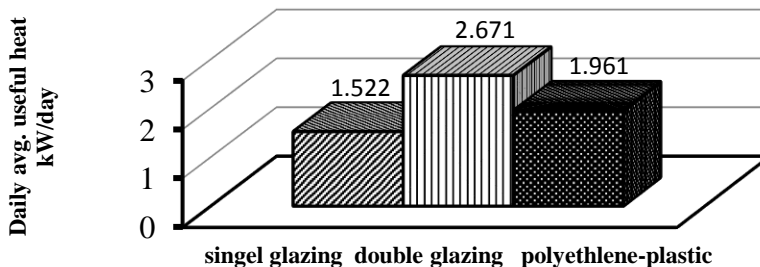

Fig (4.3): Solar collection efficiency as affected by caver material type.

Fig(4.4):Comparing heat losses of single, double glazing and polyethylene-plastic sheet cover.

Fig(4.5):Comparing useful heat, of single, double glazing and polyethylene-plastic sheet

that increasing desorption material mass has a negative effect on the COP % and there is an optimal mass of the adsorbent material required for a defined methanol quantity and the quantity for the best C.O.P % is recorded as 14 kg mass of the adsorbent bed material which is an optimal quantity compatible to 3.4 kg of methanol.Satisfied equations for the effect of carbon mass and daily solar radiation in kW. h/ m^2. day on coefficient of performance are follows:

$$COP_{12kg\ car.} = 0.1089I_d - 0.411 \qquad R^2 = 0.994…………...(4.2)$$

With total daily solar radiations ranged 6.6 to 7.6 kW.h/m^2.day.

$$COP_{14kgcar.} = 0.0486\ I_d + 0.135 \qquad R^2 = 0.954……………..(4.3)$$

With total daily solar radiations ranged 6.6 to 7.7 kW.h/ m^2.day.

$$COP_{16\ kg\ car.} = 0.031\ I_d + 0.0467 \qquad R^2 = 0.911.……………(4.4)$$

With total daily solar radiations ranged 6.8 to 7.9 kW.h/ m^2.day.

Table (4.4):Effect the total daily solar radiation, and adsorbent material mass on coefficient of performance.

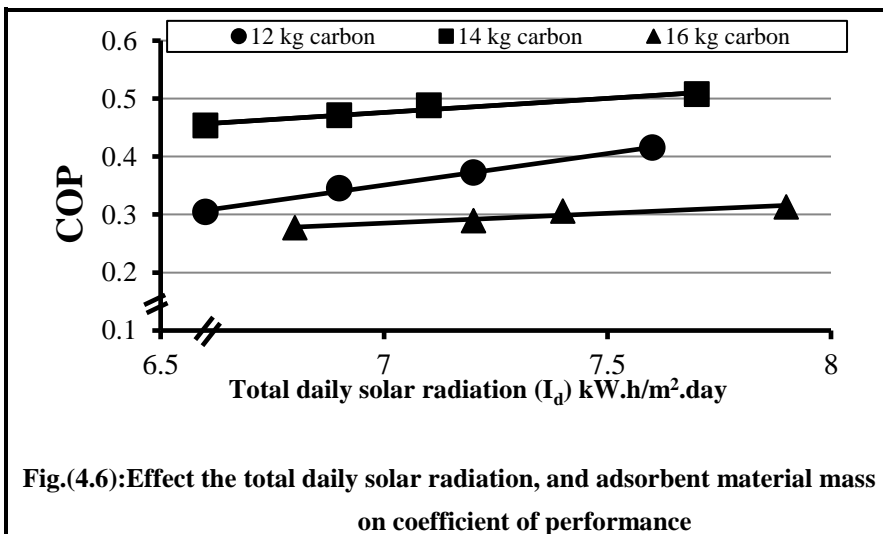| Double glazing cover					
12 kg carbon		14 kg carbon		16 kg carbon	
I_d kW.h/m^2.day	COP	I_d kW.h/m^2.day	COP	I_d kW.h/m^2.day	COP
6.6	0.304	6.6	0.453	6.8	0.277
6.9	0.345	6.9	0.471	7.2	0.289
7.2	0.372	7.1	0.488	7.4	0.306
7.6	0.415	7.7	0.507	7.9	0.313

Fig.(4.6):Effect the total daily solar radiation, and adsorbent material mass on coefficient of performance

Effect of the adsorbent bed mass on the refrigeration load

The effect of the three different adsorbent bed masses 12, 14, and 16 kg activated carbon were evaluated on the refrigeration load. R_L in ton ref. Table (4.2) and Fig (4.7) showed that the refrigeration load of the double glazing collector for all studied carbon mass materials tested were increased linear by increasing the average daily solar radiation calculated using the Simpson rule for numerical integration and satisfied only with the ranged 6.6 to 7.9 kW.hr/m2/day. It is also clear that there is a direct proportion between the refrigeration load and the average daily solar radiation. Higher values of refrigeration load were recorded for the 14 kg carbon mass than that of 12and 16 kg carbon mass respectively, so it can be concluded that 14 kg mass of the adsorbent bed material is an optimal quantity compatible to 3.4 kg of methanol.

Table (4.2): Effect the daily solar radiation, and adsorbent material mass on refrigeration load.

Double glazing cover					
12 kg carbon		14 kg carbon		16 kg carbon	
I_d kW.h/m²/day	Ref. load ton ref.	I_d kW.h/m²/day	Ref. load ton ref.	I_d kW.h/m²/day	Ref. load ton ref.
6.6	0.03488	6.6	0.044916	6.8	0.040593
6.9	0.036724	6.9	0.046116	7.2	0.042484
7.2	0.037777	7.1	0.046793	7.4	0.044113
7.6	0.039136	7.7	0.048108	7.9	0.044782
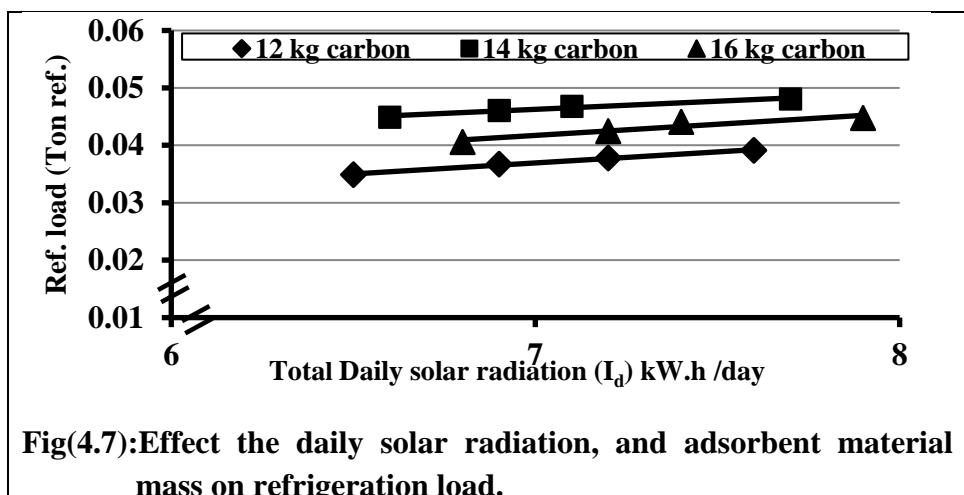
Fig(4.7):Effect the daily solar radiation, and adsorbent material mass on refrigeration load.

Satisfied equations for the effect of carbon mass and daily solar radiation in kW. h/ m^2/ day on refrigeration load. are follows:

$$RL_{12kg\ car.} = 0.0038Id + 0.01 \qquad R^2 = 0.993…………..(4.5)$$

With the range between 6.6 to 7.6 kW. h/ m2/ day.

$$RL_{14kgcar} = 0.0028Id + 0.0265 \qquad R^2 = 0.973…………....(4.6)$$

With the range between 6.6 to 7.7 kW. h/ m2/ day.

$$RL_{16\ kg\ car.} = 0.0039Id + 0.0146 \qquad R^2 = 0.9031………..…(4.7)$$

With the range between 6.8 to 7.4 kW. h/ m2/ day.

5.CONCLUSION

Solar adsorption refrigeration devices are of significance to meet the needs for cooling requirements such as air-conditioning, ice-making and medical or food preservation in remote areas. They are also noiseless, non-corrosive and environmentally friendly. Various solar powered cooling systems have been tested extensively; however, these systems are not yet ready to compete with the well-known vapor compression system. For these reasons, research activities in this sector are still increasing to solve the technical, economic and environmental problems.

The main results of this research work can be summarized as follows:

1. The solar intensity and ambient air temperature distribution as affected by local time during refrigerator testing. It is clear that solar intensity has a trend of approximately normal distribution curve before noon and after noon, due to the physical phenomena of the nature of the solar radiation and the measuring instrument as it is electromagnetic flux.

2. Three different collector cover materials, single or, double glazing and Polyethylene-plastic cover materials. the double glass cover material has the largest efficiency than both of single and polyethylene-plastic sheet that may be interpreted by lower heat losses and higher useful

heat due to stagnant insulation of air gap between the two consecutive glass covers. it is clear that the double glazing cover is the optimal cover material of all studied cover materials in present work.

3. The effect of the three different adsorbent bed masses 12, 14, and 16 kg activated carbon were evaluated on the coefficient of performance and the refrigeration load. RL in ton ref. Higher values of C.O.P % and the refrigeration load. RL in ton ref. were recorded for the 14 kg carbon mass than that of 12and 16 kg carbon mass respectively, there is an optimal mass of the adsorbent material required for a defined methanol quantity and the quantity for the best C.O.P % and the refrigeration load. are recorded as 14 kg mass of the adsorbent bed material   which is an optimal quantity compatible to 3.4 kg of methanol.

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الملخص العربى

بعض العوامل الهندسة المؤثرة على أداء دورة تبريد بالطاقة الشمسية بالادمصاص

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م / أحمد صلاح عيسى ****

تعتبر الطاقة الشمسية من اهم مصادر الطاقات الجديدة والمتجددة والبديلة للوقودالاحفورى. وتبوأت الطاقة الشمسية مكانًا مهما ضمن البدائل المتعلقة بالطاقات المتجددة، وتأتي أهميتها بأنها مصدراً نظيفاً للطاقة من حيث تأثيرها على البيئة وهي طاقة مباشرة وسهلة الاستخدام متجددة ومستمرة الحفاظ على نفس الجودة ولا تخضع لاحتكار الشركات أو المستثمرين بل يمكن أن يستفيد منها جميع سكان الأرض. وفى السنوات الاخيرة لوُحظ ان هناك اهتمام كبير لتطبيقات الطاقة الشمسية فى تكنولوجيا التبريد حيث ان التبريد اصبح ضروريا لحفظ المنتجات الزرعية والمواد الغذائية والادوية الطبية واللقاحات والتحصينات البيطرية وتبريد المنشأت الزراعية. ونظام التبريد بالطاقة الشمسية بالادمصاص واحد من أهم الطرق الواعدة فى التبريد بأستخدام الطاقة الشمسية والتى يمكن ان يلبى الاحتياجات الخاصة بالتبريد مثل تكييف الهواء وصناعة الثلج وحفظ المواد الغذائية والمواد الطبية واللقاحات والتحصينات البيطرية وخصوصا فى المناطق النائية والمستصلحة حديثا ونظرا لما له من مزايا أهمها أنه أمان تماماً على البيئة إنخفاض تكلفة الصيانة والاصلاحات غير قابلة للتأكل ولا يحدث منها ضوضاء.

ويهدف هذا البحث الى :

١. أنشاء وتصنيع وحدة تبريد بالطاقة الشمسية بالادمصاص باستخدام زوج العمل كربون نشط ـ ميثانول.

٢. دراسة بعض العوامل الهندسية التى تؤثر على أداء الدورة للوصول لافضل أداء .وحمولة التبريد

٣. دراسة أقتصادية

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وقد أظهرت النتائج ما يلى :

١. تأثير العوامل البيئية على أداء دورة تبريد بالطاقة الشمسية بالادمصاص وكفاءة المجمع الشمسى (المولد) مثل شدة الاشعاع الشمسي ك وات / م² ودرجة حرارة الجو م° وسرعة الرياح م/ ث وأظهرت النتائج ان شدة الاشعاع الشمسى متأثرة بالتوقيت المحلى يتبع تقريبا منحنى التوزيع الطبيعى قبل وبعد الظهر، وكانت القيم العظمى لشدة الاشعاع الشمسى عند الظهر وتتراوح ما بين ١،٠ – ١،٢٤ ك وات / م ² ، والقيم الدنيا في الصباح وبعد الظهر والتي تراوحت بين ٠،٥١-٠،٧٣ كيلووات / م ٢ و ٠،٣٩- ٠،٦٥ كيلووات / م ٢ قبل وبعد الظهر على التوالي. ومن الواضح ان درجة الحرارة المحيطة لاتتبع منحنى التوزيع الطبيعى نظرا لتأثرها برطوبة الهواء وسرعته.

٢. اختباروتقييم ثلاثة أغطية على كفاءة المجمع الشمسى (المولد) غطاء زجاج – غطائين من الزجاج – غطاء من البلاسيتك (البولى إيثلين) ، وقد أوضحت النتائج ان أعلى قيم للكفاءة الحرارية مع الغطائين من الزجاج وذلك للاستفادة العظمى من الطاقة الحرارية وانخفاض الفاقد منها عنها فى الغطاء الواحد من الزجاج والبلاسيتك (البولى اثيلين)

٣. اختبار وتقييم ثلاث كتل من مادة الادمصاص (الكربون النشط) ١٢ ،١٤، ١٦ كجم للوصل الى افضل معاما اداء % وافضل حمولة تبريد طن تبريد وقد اوضحت النتائج ان اعلى قيم لمعامل الاداء وحمولة التبريد كانت عند ١٤ كجم من الكربون النشط .