

THERMODYNAMIC ANALYSIS AND SIMULATION OF COMBINED ADSORPTION HEATING AND COOLING SYSTEM

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ABSTRACT

This paper presents the description and thermodynamic analysis of a new hybrid system of solar-powered water heater and adsorption ice maker. Mathematical models were generated based on heat transfer in the collector, heat and mass transfer within the adsorbent-adsorbate pair and the thermodynamics of the adsorption process. The numerical models developed from finite difference transformation of the resulting equations were developed into a MATLAB computer code for easy implementation on a personal computer. The results reveal the ability of the hybrid system to heat 50kg water to about 96°C as well as produce ice at 7.2kg per day with a 2m² evacuated vacuum-tube-type solar collector, with a coefficient of performance of 0.62. The system is capable of reaching a specific refrigeration density of 0.48kg ice per kg-adsorbent per day. Details on the effect of condensation, evaporation and adsorption temperature on the system coefficient of performance are also reported.

Keywords: solar energy, adsorption, hybrid, adsorbent bed, simulation

NOMENCLATURE

a = Coefficient	h = Hour Angle (°)
A = Surface area (m ²) / constant of Clausius – Clapeyron equation	H = Daily total solar radiation (J/m ²) / heat of desorption (J/kg)
b = Coefficient	h _r = Radiation heat transfer coefficient (W/m ² K)
C _p = Specific heat capacity (J/kgK)	h _c = Convective heat transfer coefficient (W/m ² K)
COP = Coefficient of performance	I = Hourly solar irradiation (J/m ²)
d = Diameter (m) / day of the year / difference / change	k = Characteristic parameters of adsorption pair
g = Acceleration due to gravity (m/s ²)	K = Thermal conductivity (W/m.K) / Clearness index
Gr = Graetz number	

L = Latent heat (J/kg) / Latitude ($^{\circ}$) / Length (m)	Nu = Nusselt number
m = Mass (kg)	P = Pressure (Pa)
M = Mass of ice produced (kg)	Pr = Prandtl number
n = Characteristic parameters of adsorption pair / Time step	Q = Heat / Energy (J)
R = Geometric ratio i.e. the ratio of beam or diffuse radiation on the tilted surface to that on the horizontal surface / Gas constant (J/kgK)	r = Ratio of the hourly total or diffuse radiation to daily total or diffuse
Ra = Rayleigh number	T_F = temperature at the beginning of adsorption.
Re = Reynolds number	u = Free stream velocity (m/s)
t = Time (s)	U = Overall heat transfer coefficient ($W/m^2 K$)
T = Temperature (K)	ν = Kinematic viscosity (m^2/s)
	V = View factor
	x = Adsorption capacity (kg/kg)
Subscripts	
a = ambient	E = evaporation
ac = activated carbon/adsorbent	F = fusion
ad = adsorber	g1 = outer glass cover
A = adsorption	g2 = inner glass tube
b = beam radiation	h = horizontal surface
cool = cooling capacity	heat = heating
C = condensation	I = inclined surface
cw = cold water	max = adsorption capacity before desorption
d = diffuse radiation/diameter	min = adsorption capacity after desorption
D = desorption	o = extraterrestrial solar radiation/initial
	r = adsorbate/methanol

ref = refrigeration capacity

s = surface/sunset/saturation

T = total

w = water

Greek letters

ε = Emittance

ρ = Density (kg/m^3)

μ = Dynamic viscosity (kg/m.s)

θ = Angle of incidence ($^\circ$)

δ = Declination ($^\circ$)

w1 = initial water in the water tank

w2 = hot water

w3 = water after heat recovery

wt = water tank

$\tau\alpha$ = Transmittance-absorptance product

β = Inclination ($^\circ$) / thermal expansion coefficient (k^{-1})

σ = Stefan-Boltzmann constant

Δ = Change/ Difference

1.0 INTRODUCTION

The prevailing trends in refrigeration system have a deep engineering interest and automation with the aim of addressing two major problems. Firstly, the increase in global consumption of limited primary energy and secondly, the effect of refrigerants on the environment such as global warming and ozone depletion. The thermodynamic properties of the working fluid determine the performance of the refrigeration systems like household refrigerators, heat pumps, industrial freezers, cryogenic and air conditioning.

There are various methods of refrigeration and they can be classified as follows; non-cyclic, cyclic, thermoelectric and magnetic. The cyclic method is important for this study, and it could be either the vapour cycle or gas cycle. The vapour cycle can further be classified as vapour compression and vapour absorption refrigeration.

The investigation of using either solar or kerosene/gas to power either vapour compression or vapour absorption refrigerators has attracted the attention of scientific

community since the experimental reference works of Pons and Guillemot (1986) and Headley et al (1994). Solar energy can be transformed to electricity or to heat to power refrigeration cycle and more interest have been paid to solar thermal driven refrigeration technologies especially solar adsorption systems (Sumathy and Li, 1999). New solar-powered continuous solid adsorption refrigeration and heating hybrid system was proposed by Zhang et al, (2002); the COP and the cooling capacity were obtained.

Yeung and Sumathy (2003) presented a solar hybrid adsorption heating and cooling system employing an evacuated vacuum tube collector with an exposed area of $2m^2$ as the heat source, and the COP was obtained. Wang et al, (2000) tested a new hybrid system capable of both heating and cooling effect and the results shown that this simple design was effective. In an attempt to overcome the intermittent character of a single bed solar adsorption cycle, a novel model of the combined cycle of a solar powdered

adsorption-ejection refrigeration system was established by Li et al (2002).

Khattab (2006) performed a simulation and optimization analysis of a novel solar-powered adsorption refrigeration module. A computer simulation program was developed in visual basic by Alghoul et al (2007) in order to guide in the design of a multipurpose solar continuous adsorption system and predicted its applicability under various operating conditions.

The field of study is still vast. A lot of research works still need to be done for enhancing the heat and mass transfer to improve performance of solar adsorption systems. More modern solar energy collecting and transferring technologies, and more advanced optimization and simulation models are also being anticipated.

2.0 MATHEMATICAL ANALYSIS

The mathematical models describing the operation and the performance of the proposed hybrid system are developed in this section. The necessary parameters for the simulation are listed in table 1. The schematic design of the hybrid system is represented in figure 1.

Solar collector

The hourly total solar radiation intensity (I_β) on an inclined (sloped) surface, corresponding to the metrological conditions in Ibadan Nigeria (latitude $7.433^\circ N$) is assumed to vary sinusoidally and can be obtained by

$$I_\beta = \left[(I - I_d) \left(\frac{\cos \theta_i}{\cos \theta_h} \right) \right] + I_d (1 + \cos \beta) / 2$$

Where,

$$\cos \theta_i = \cos(L - \beta) \cos \delta \cos h + \sin(L - \beta) \sin \delta \quad 2$$

and

$$\cos \theta_h = \cos L \cos \delta \cos h + \sin L \sin \delta \quad 3$$

Note that $\beta = L + 10$ as suggested by Fagbenle (1991). More so Fagbenle (1991), the extraterrestrial solar radiation, H_0 , for April 11 in Ibadan Nigeria, is $37.5824 MJ/m^2$ while a value of $24 MJ/m^2$ for the average total solar radiation, H (based on previous experimental data), is employed for the same day.

$$K_T = H/H_0 \quad 4$$

The value of K_T obtained is employed to find the diffuse fraction of the total solar radiation H_d , from the Collares-Pereira and Rabl's recommended correlation chart of daily diffuse fraction with K_T (Duffie and Beckam, 1991). In converting the daily total solar radiation, H , to hourly equivalent, the following equation as suggested by Collares-Pereira and Rabl (Duffie and Beckam, 1991) is used:

$$r_i = \frac{I}{H} = \frac{\pi}{24} \left(a + b \cos h \right) \frac{\cosh - \cosh_s}{\sin h - \left(\frac{2\pi h_s}{360} \right) \cosh_s} \quad 5$$

Where,

$$a = 0.409 + 0.5016 \sin(h_s - 60) \quad 6$$

$$b = 0.6609 - 0.4767 \sin(h_s - 60) \quad 7$$

As recommended by Liu and Jordan (Duffie and Beckam, 1991) I_d is estimated from equation 8 below:

$$r_d = \frac{I_d}{H_d} = \frac{\pi}{24} \frac{\cosh - \cosh_s}{\sin h_s - \left(\frac{2\pi h_s}{360} \right) \cosh_s} \quad 8$$

The energy absorbed by the outer glass cover of the double-glazed simple solar collector is given by

$$m_{g1}Cp_g \frac{dT_{g1}}{dt} = I\beta(\tau\alpha)_{g1} \frac{A_{g1}}{2} + hr_{g12}(T_{g2} - T_{g1}) \frac{A_{g2}}{2} - U_{g1}(T_{g1} - T_a) \frac{A_{g1}}{2} \quad 9$$

where,

$$h_{g12} = \frac{\sigma(T_{g12}^2 + T_{g21}^2)(T_{g12} + T_{g21})}{\left(\frac{1}{\epsilon_{g1}}\right) + \left(\frac{d_1}{d_2}\right)\left[\left(\frac{1}{\epsilon_{g2}}\right) - 1\right]} \quad 10$$

and

$$U_{g1} = hc_{g1a} + hr_{g1a}$$

Based on available correlation Holman, 2001,

$$Nu_{d1} = 0.386 Re^{0.592} Pr_a^{1/3} \quad 11$$

where, $Re = \frac{\rho_a u d_1}{\mu_a}$

and

$$hr_{g1a} = \sigma\epsilon_{g1}(T_{g1}^2 + T_a^2)(T_{g1} + T_a) \quad 12$$

The rate at which the collector inner glass absorbs energy is obtained by the following energy balance equation from which the state equation of the inner glass tube is determined:

$$m_{g2}Cp_g \frac{dT_{g2}}{dt} = I\beta(\tau\alpha)_{g2} \frac{A_{g2}}{2} - hr_{g12}(T_{g2} - T_{g1}) \frac{A_{g2}}{2} - hc_{g2w}(T_{g2} - T_{w1})A_{g2} \quad 13$$

Since the heat transfer from the glass tube to the water occurs by natural convection, the heat transfer coefficient, hc_{g2w1} , can be obtained by the following correlation (Peary and Chilton, 1973)

$$hc_{g2w1} = \frac{NuK_{w1}}{d_2} \quad 14$$

which is valid for natural convection in a horizontal tube with the wall temperature increasing at a uniform rate where

$$Nu = 1.81(Gr Pr_{w1})^{0.214} \quad \text{for}$$

$$Gr = \frac{g\beta_{g2-w1}\Delta T d_2^3}{\nu_{w1}^2}$$

Energy analysis of the water tank along with adsorbent bed

(A) The useful energy collected by the double-glazed simple solar collector will contribute to the heating of the water in the water tank inside which the adsorbent bed is immersed. Thus, the rise in the water temperature along with the metallic tank is said to be equal to the increase in the internal energy of water and the heat loss to the adsorber inside the water tank, which is as assumed to be well insulated. Hence neglecting the heat losses, the rate of increase in water temperature can be evaluated by performing an energy balance which gives the following energy equation from which the final water temperature (hot water temperature), T_{w2} is obtained:

$$(m_w Cp_w + m_{ad}) \frac{dT_w}{dt} = hc_{g2-w1}(T_{g2} - T_{w1})A_{g2} - hc_{w2ad}(T_{w2} - T_{ad-1})A_{ad} \quad 15$$

The heat transfer coefficient (hc_{w2ad}) corresponding for the water to the adsorber (which is horizontal and cylindrical in shape) can be calculated by the following correlation (Peary and Chilton, 1973)

$$hc_{w2ad} = \frac{Nu_d K_{w2}}{d_{ad}} \quad 16$$

$Nu_d = 0.48 Ra_d^{0.25}$ for $10^4 < Ra_d < 10^7$

$$Ra_d = Gr_d Pr_{w2} \quad \text{where}$$

$$Gr_d = \frac{g\beta_{w2-ad}(T_{w2} - T_{ad-1})d_{ad}^3}{\nu_{w2}^2}$$

(B) Performing an energy balance on the hot water and the adsorber gives the following:

$$(m_{ad}Cp_{ad} + m_{ac}Cp_{ac} + m_{ac}Cp_r) \frac{dT_{ad}}{dt} = hc_{w2ad}(T_{w2} - T_{ad-1})A_{ad} - H_D m_{ad} \frac{dx}{dt} \quad 17$$

Where $dx = \Delta x = x_{max} - x_{min}$.

$$H_D = RA \left(\frac{T}{T_s} \right) \quad \text{and can be evaluated from}$$

Clausius Clapeyron equation where $T_s = T_c$

In this study, activated carbon-methanol is employed as the working pair. To determine the initial of the adsorbate (methanol) being

adsorbed by the adsorbent (activated carbon), Durbinin–Radushkeritch equation is used:

$$x = x_o \exp \left[-k \left(\frac{T}{T_s} - 1 \right)^n \right] \quad 18$$

Energy analysis between filled water and adsorber

The heat recovered by the filled cold water can be estimated using the following heat balance equation:

$$m_w C_{pw} (T_{w3} - T_{cw}) = m_{bc} C_{pac} (T_{ad,2} - T_{cw}) + m_{ad} C_{pad} (T_{ad,2} - T_{cw}) + x_{max} m_{bc} C_{pr} (T_f - T_{cw}) \quad 19$$

The temperature of cold water (T_{w3}) after heat recovery can then be estimated.

Refrigeration capacity and system efficiency

The refrigeration quantity is estimated as

$$Q_{ref} = \Delta x m_{ac} L_E \quad 20$$

The energy used in cooling the evaporated adsorbate from condensing temperature (T_c) to evaporation temperature (T_E) is estimated by the expression,

$$Q_{cool} = m_{ac} \Delta x C_{pr} (T_c - T_E) \quad 21$$

The total energy input to the system is calculated as:

$$Q_{heat} = (m_{bc} C_{pac} + m_{bc} C_{pr} x_{max}) (T_B - T_A) [m_{bc} C_{pac} + m_{bc} C_{pr} \left(\frac{x_{max} + x_{min}}{2} \right)] (T_D - T_E) + m_{bc} H_D \Delta x \quad 22$$

The system

$$COP = \frac{Q_{ref} - Q_{cool}}{Q_{heat}} \quad 23$$

3.0 RESULTS

From the above equations, the initial simulation results required for further analysis are presented in table 2.

Figure 4 shows the effects of Condensing Temperature and Evaporation temperature on system COP. The COP decreases with the increase in the condensing temperature, T_c for a given desorption temperature, T_D , evaporating

temperature, T_E , and adsorption temperature, T_A . This is due to an increase in condensing temperature which increases the system pressure with the amount of adsorbed adsorbate decreasing under the same desorption temperature, T_D . This reduces the system COP. However, the condensing temperature cannot be too low as it is being limited by the ambient temperature.

As shown in Figure 4, the system COP varies directly with the evaporating temperature, T_E , if $T_A = T_c = 28^\circ C$. When T_E is decreased, the implication is that more energy would be needed to cool the refrigerant liquid from condensing temperature to evaporating temperature. In order to obtain the required cooling effect, more refrigerant vapour would be desorbed from the adsorbent which in a way demands more heat supply to the system, which ultimately reduces the system COP.

For ice making purposes, it is recommended to limit T_E to $-5^\circ C$ and $-10^\circ C$, and for air-conditioning, T_E can be set to $5^\circ C$. For solar ice maker, T_E is determined by the initial amount of adsorbed adsorbate and the adsorption temperature.

The influence of the adsorption temperature on the hybrid system COP is observed as shown in figure 5. The COP appreciably increases at lower adsorption temperature, T_A but drops as the adsorption temperature increases. For solar ice making, the adsorption temperature should be kept at the lowest possible value during the adsorption stage. T_A is also limited by the ambient temperature, that is, cannot be lower than the ambient temperature. This is justified that, at a higher adsorption temperature, the system pressure increases and this will hinder not only the flow of adsorbate vapour from the evaporator to the adsorber tube-bed (adsorption process), but also the regeneration of activated carbon. In regular conventional adsorption ice makers, the heat from the adsorbent bed is

released by natural convection to the surrounding, but in this work, in order to enhance the adsorption process and the regeneration of the adsorbent, water is used as a medium of cooling.

Figure 6 shows the influence of desorption temperature, T_D , on the mass of adsorbate desorbed. It is observed that with the temperature lower than 305K, no methanol was desorbed but beyond this value, desorption starts and a further increase in the desorption temperature makes more methanol to be desorbed. However, it should be noted that, for a given T_C, T_A and T_E , the cycle requires a minimum desorption temperature as the additional heat added to the system is only used in sensible heating of the adsorbent and the adsorbate and there is no additional energy for the desorption of methanol. Although a higher desorption temperature could result in a more amount of cooling effect, the COP may not increase or may even decrease as there is no methanol desorbed. It is often recommended that for adsorption systems with methanol and activated carbon as the working pair, the desorption temperature should range from 70 to 90°C in order to prevent phase change and more so that water is used as the heat transfer fluid.

It is observed from figure 7 that an increase in the solar collector area increases the system COP. This is due to the fact that, the amount of solar radiation collected increases with an increase in the collector area, which enhances a higher desorption temperature of the adsorption bed. For the given size of adsorber tube-bed and the water tank, the system performance increases with the collector area, up to 2- m^2 but, the increase rate is rather slow with further increase in the collector area. The reason is because, despite that the desorption temperature increases with the collector area, it is often not advisable to maintain the adsorption bed at higher temperatures, since most of the input energy is utilized only to raise the sensible heat of the

bed, which correspondingly hinders the adsorption process during cooling period. This is reflected on the slow increase of the system COP.

Figure 8 shows the effect of the adsorbent mass on the system COP. The effects of the solar collector area, the amount of water in the water tank and the mass of the adsorbent tube-bed are considered simultaneously as we have in any solar-powered adsorption refrigeration system, that their influences are closely related due to the fact that, the heat energy gained by the solar collector is initially used to heat the water in the water tank, which in turn heats up the adsorbent tube-bed to effect desorption. It is revealed from the figure that, if the mass of water, m_w , for any combination of collector size and the adsorbent mass, m_{ac} , is too high, the system would not achieve any cooling (ice making) effect which automatically makes the system COP zero. This is owing to the reason that, the heat gained by the water is utilized only for the sensible heating of the adsorbent tube-bed, and does not contribute to desorption of methanol required. In the same manner, if the mass of the adsorbent, m_{ac} , is too low for any mass of water, the system COP turns out to be zero. This is because, less mass of adsorbent makes initial the intake of methanol insufficient to effect any cooling (ice making).

4.0 CONCLUSION

A thermodynamic analysis and simulation of combined adsorption heating and cooling system has been carried out to study the performance of a hybrid system of solar-powered water heater and ice-maker using solid adsorption with activated carbon and methanol in the ambient condition of Ibadan, Nigeria. It is obvious from the simulation results that the amount of water used in the water tank plays an important role in the performance of the system compared to the collector size and the adsorbent mass. In this work, efforts are made by sensitizing the combinations of these key

parameters so as to improve the system performance and for further improvement in the future. In order to further improve the performance of the system, evacuated-vacuum-tube-type solar collector is employed instead of the conventionally used simple type solar collector. Therefore, a 2- m^2 solar collector installed in Ibadan Nigeria is capable of heating 50kg water to about $96^\circ C$ and also producing

7.2kg ice per day with the COP of 0.62, thereby making hot water and ice available for domestic uses. This concept of energy utilization provides an effective way for Nigeria energy conservation and sustainable development of her environment.

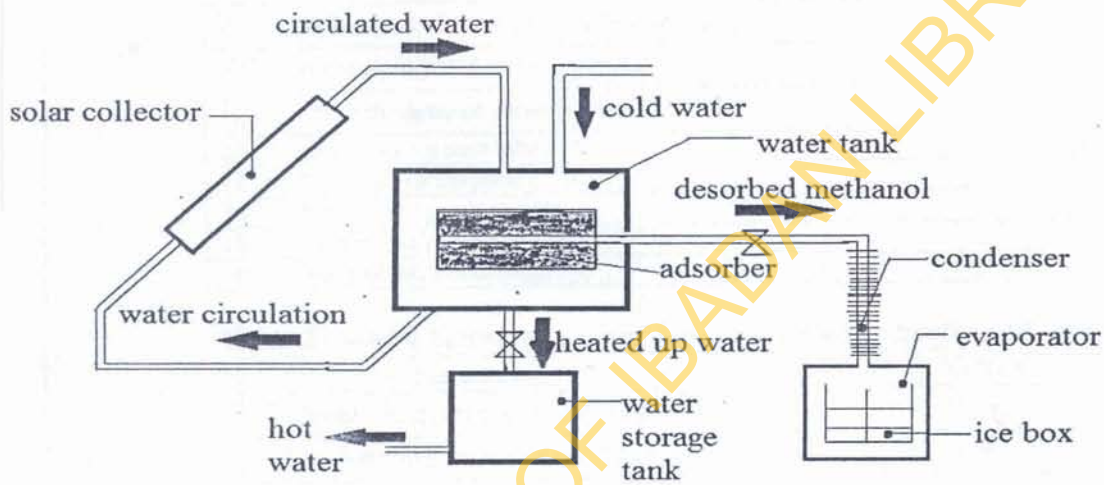


Figure 1: Schematic representation of the solar hybrid system.

Table 1: System and simulation parameters

No	Parameters	Values
1	Specific heat of adsorbent (Cp_{ac})	900J/kg K
2	Specific heat of adsorbate (Cp_r)	750 J/kg K
3	Specific heat of glass (Cp_g)	700 J/kg K
4	Specific heat of adsorber tube (Cp_{ad})	902 J/kg K
5	Density of adsorbent (rho_ac)	500kg/ m^3
6	Density of glass (rho_g)	2,500kg/ m^3
7	Ambient temperature (T_a)	26.5 °C
8	Cold water temperature (T_{CW})	24.5 °C
9	Latent heat of vaporization adsorbate (L_E)	1102 × 10 ³ J/kg
10	Inner diameter of vacuum tube (d_2)	0.05m
11	Outer diameter of vacuum tube (d_1)	0.06m
12	Length of vacuum tube (L)	1m
13	Maximum adsorption capacity(x_0)	0.284
14	Adsorption parameter (k)	10.21
15	Adsorption parameter (n)	1.39
16	Mass of adsorbent (m_{ac})	15kg

Table 2: Simulation results of the hybrid solar water heater and ice maker

Parameters	Values
Condensing temperature(T_C)	28 °C
Evaporating temperature (T_E)	-5 °C
Desorption temperature (T_D)	70 °C
Adsorption temperature (T_A)	28.2 °C
Refilled water temperature (T_{w3})	28.2 °C
Hot water temperature (T_{w2})	95.5 °C
COP	0.62
Ice produced per day (M)	7.2kg

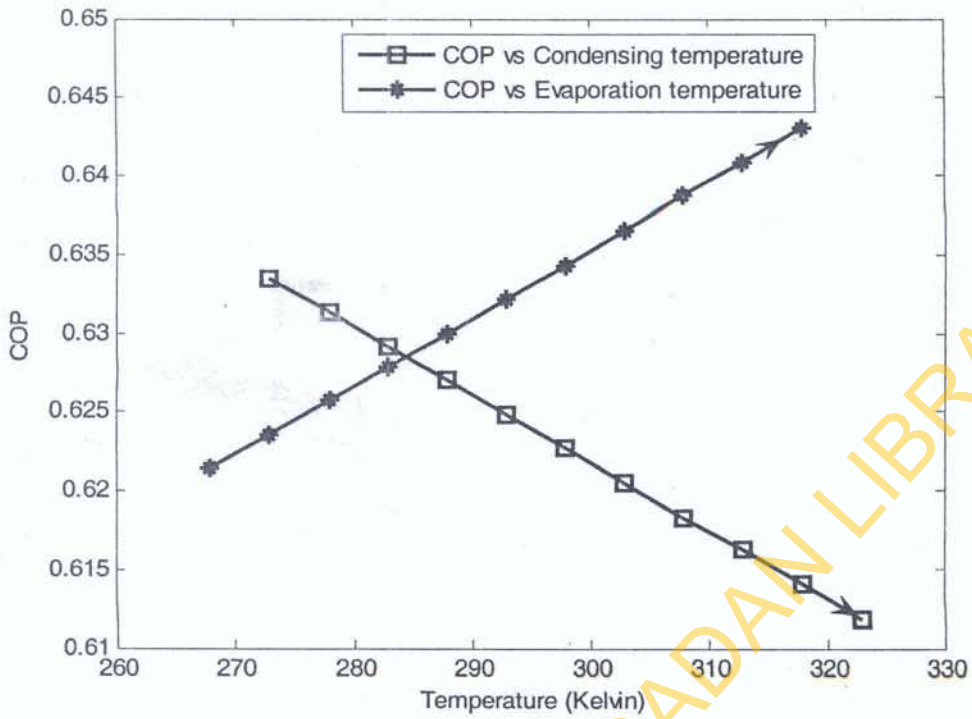


Figure 4: Effects of Condensing Temperature and Evaporation temperature on system COP

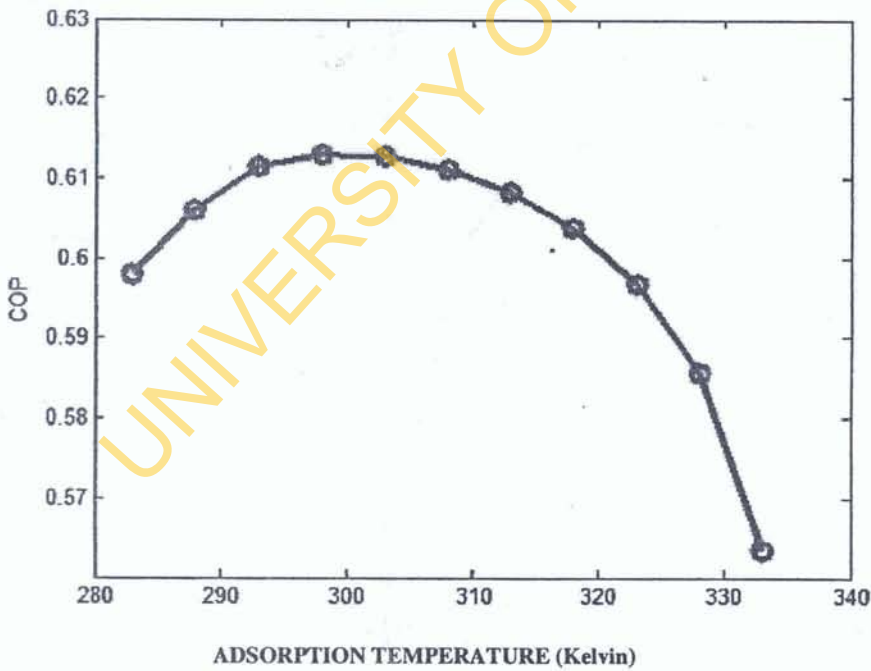


Figure 5: Effect of adsorption temperature on the system COP

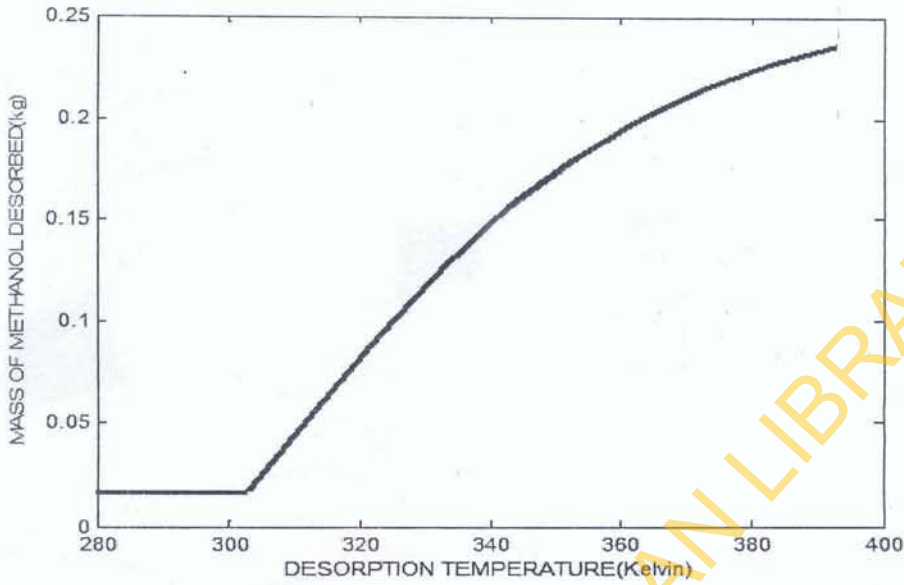


Figure 6: Effect of desorption temperature on mass of desorbed adsorbate

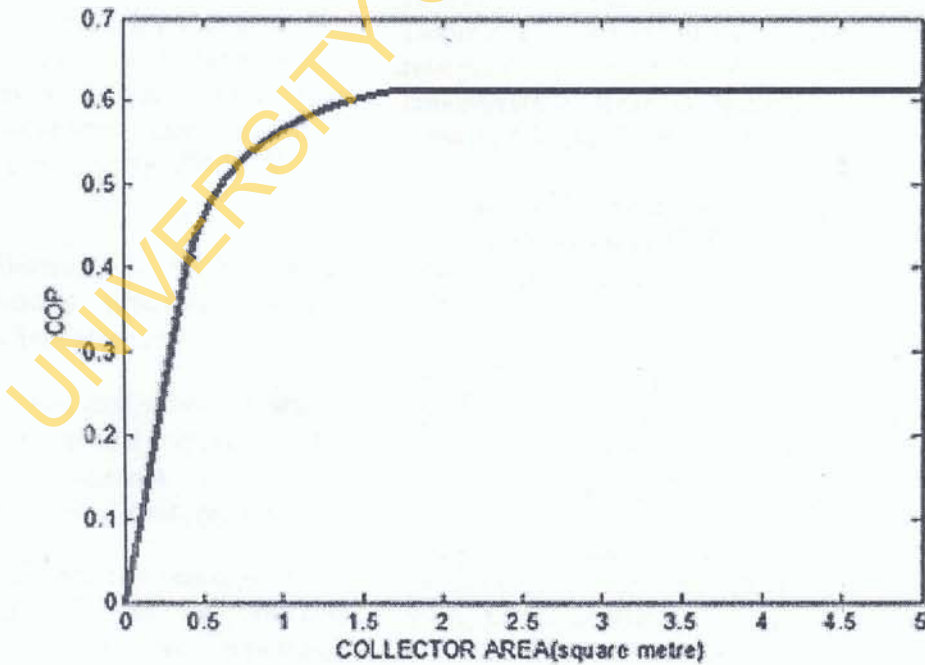


Figure 7: Effect of Solar Collector area on the system COP

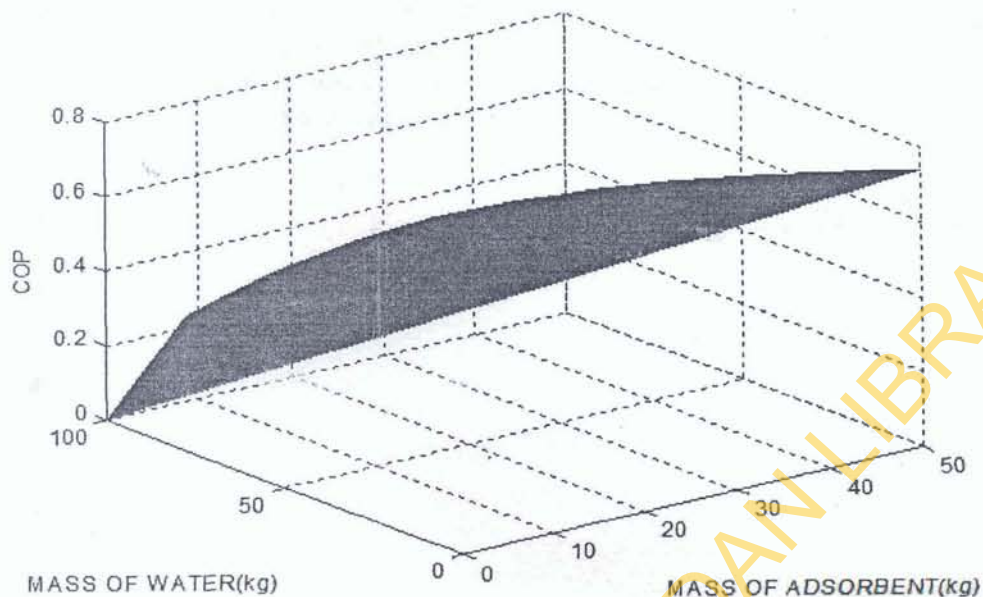


Figure 8: Effect of mass of water and adsorbent mass on the system COP

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