Theoretical study on the effect of operating parameters on the performance of adsorption refrigerator

دراسة نظرية حول تأثير العوامل التشغيلية على اداء منظومة تثليج امتزازية

Abdul Hadi N. Khalfa*   Ahmed Q. Mohammed*
ahaddi58@yahoo.com   dr.ahmed56@yahoo.com

Hassan Jawdat Fadiel*

hassan_jfsd@yahoo.com

*Engineering Technical College-Baghdad. Middle Technical University.

Abstract:

This work include a theoretical analysis of an adsorption refrigerator, the adsorption refrigerator consist of unit generator in the form of shell and tube heat exchanger, the generator can be heated either by hot water from solar collector, or from waste heat. The adsprtion pair used was active carbon-methanol. A complete thermodynamics analysis was achieved on the unit, while the thermo physical properties of methanol were calculated using Dubinin and Astakhov equation. Many key parameters were examined to find the effect off the unit starting, evaporator, condensing and peak temperatures on the cycle COP. The results show that for a certain ambient temperature there is an optimum generator temperature at which COP reaches a maximum. as the ambient temperature increases the condensing temperature increases also, thus leads to reduce cycle COP, and finally heavy materials used in building the adsorption cycle leads to increase the consumed heat that used to heat up the generator.

Key words: Adsorption, Refrigeration, active carbon, methanol.

الخلاصة:

يعنى البحث بأجراء تحليل نظري لثلاجة امتزازية، حيث تتكون الثلاجة الامتزازية من مولد البخار واذي هو على شكل مبادل حراري من نوع الغلاف والانابيب، حيث يمكن تسخين مولد البخار بعدة طرق منها الماء الساخن الناتج من المجمع الشمسي او الحرارة المطرودة من بعض العمليات الصناعية. وكان زوج الامتزاز المستعمل من نوع الميثانول - الكاربون المنشط. وتم اجراء تحليل حراري شامل للمنظومة، اما الخواص الحرارية للميثانول فقد حسبت باستخدام معادلة ديين و اساكوف. تم دراسة العديد من المتغيرات لغرض ايجاد تأثير بداية عمل المنظومة و درجة حرارة المبخر والتكثيف ودرجة الحرارة العظمى على معامل اداء المنظومة. بينت النتائج ان عند درجة حرارة محيط معينة هنالك درجة حرارة مثلى للمولد من خلالها يمكن الوصول الى اعلى معامل اداء للمنظومة. وكذلك فان زيادة درجة حرارة المحيط الخارجي تؤدي الى زيادة درجة حرارة التكثيف وبالتالي تنعكس سلبيا على نعامل اداء المنظومة. اضافة الى ان زيادة كتلة المادة التي تصنع منها المنظومة تؤدي الى استهلاك كمية حرارة اكثر لرفع درجة حرارة المولد الى درجات الحرارة العاملة.

1- Introduction:

The early scientists who worked on principles for solar adsorption system was Michael Faraday [1] in 1824. Through experiment, he succeeded in liquefying ammonia, which scientists had believed to be a "fixed" gas. He exposed the ammonia vapor to silver chloride. When the silver chloride had taken all the vapor it could adsorb, he applied heat and got a liquid. But when the heat was removed, he discovered that the liquid soon began to "boil," vaporize, and draw heat from its surrounding. F. Lemmini and F. Meunier [2] presented a numerical simulation of an adsorptive solar refrigerator after one year of operation in Rabat ,Morocco,1993. The average solar COP is 0.114. In 2005 R. A Wang and R. G. Oliveira[3] have designed a prototype adsorption chiller use

waste heat or solar energy as the main heat sources. A daily ice production of between 4 and 7 kg / m^2 of solar collector with a solar COP between 0.1, and 0.15 was found. In the University of Technology in Iraq one of its students Ammar S.[4] presented a research in 2006 about solar adsorption icemaker using plate type adsorber/collector. After adsorption process the lowest temperature of evaporator was $-0.5°C$ and the COP of system was 0.39. A hybrid adsorption refrigeration unit was designed, built and tested by Nema and Fadiel [5] in 2008. They concluded that there is an optimum generator temperature at which the cycle COP reach max. The cycle COP was varied from 0.07 to 0.3. Khalifa , Hussein and Hadi,[6] have built two generators nearly continuous adsorption-desorption cycle, by allowing the generators to work alternatively either adsorbing or desorbing the medium. The adsorption chiller was driven by hot water, with a temperature range of 70 to 100ºC. It was found that using the mass recovery process increases the initial concentration of methanol in the desorption generator; hence, improving the COP and SCP of the cycle. The COP of this adsorption chiller was about 0.301, while the SCP was about 0.3532 kW/kg$_{AC}$. Hassan and Mohamad [7] have study theoretically a thermodynamic chiller based on adsorption at constant temperature, The Dubinin–Astakhov adsorption equilibrium equation is used in the analysis. The chiller is found to produce a daily mass of 2.63 kg cold water at 0 °C from water at 25 °C per kg of adsorbent. Moreover, the proposed system attains a cooling coefficient of performance of 0.66.

2- System description:

The theoretical adsorption refrigerator consists of generator (Bed), which is the main part of the field unit that acts as a compressor in the vapour compression unit. The generator was built in the form of a shell and tube heat exchanger. The difference that, the tube contains both adsorbent and adsorbate , and the water was in the shell. The unit generator consists of two identical parts; each one was built from a hemispherical shell of 240.4 mm in diameter and (100 mm) height, as shown in Figure (1). The upper end of hemispherical shell contains a valve for charging purpose, while the lower end was closed by brass disk of 1.75 mm thick. The brass disk was drilled to have 22 holes. A 22 copper tubes of 22.22 mm in diameter and 440 mm long were fixed on the brass disk. The free ends of the tubes was punched and soldered, while the other one was equipped with a screen mesh to allow refrigerant to flow through tiny holes and at the same time to keep the active carbon within the tubes. Each hemispherical shell was charged by 1.25 kg of active carbon and 87.5 gram of methanol, table 1 shows the physical properties of generator. The two parts of generator were combined together by a cylindrical shell to form the generator as shown in Figure (1). The aim of the shell is to contain the hot water coming from either solar collector or from waste heat source. A wire condenser is used to condensate the methanol and a copper container of 1liter capacity was used as unit evaporator, as shown in Figure (3).

3- Theoretical analysis:

The vapour pressure P$_s$ of methanol in a limited temperature range is described by Exell et al. [8] as a function of temperature as: -

$$ln\, P_S = 12.6973 - \frac{4024.37}{T} - \frac{87582.885}{T^2} \qquad\qquad 1$$

In the range of 250K<T<337.5K error of 2.9% max. and 337.5K<T<420K error of 0.1% max.
   While the density of saturation liquid methanol with maximum error of 0.25% is :

$$\rho_S = 937.911 - 0.058267\, T - 0.001459\, T^2 \qquad\qquad 2$$

The isotherm adsorption equation D-A equation was developed by Dubinin and Astakhov [9] as below:

$$x = x_0\, e^{-D\left[T\, ln\left(\frac{P_s}{P}\right)\right]^n} \qquad\qquad 3a$$

Where D is structural constant of the adsorbent and can be written as:

$$D = \frac{k\, R^2}{\beta^2} \qquad\qquad 3b$$

Where:

n: is usually between 1 and 3, x_0:the maximum concentration of methanol, R is gas constant,β is the affinity coefficient which characterize the polarizability of the adsorbate and is a function of adsorbate only, k: parameter depends upon the number and size distribution of micros, P_s is the saturation vapour pressure of adsorbate at temperature T, P is the equilibrium gas- phase partial pressure at the adsorbate , and x is the mass concentration percentage .

From the Isotherm adsorption equation, the saturation pressure P_s of methanol can be approximately expressed as a function of temperature as follows[10].

$$ln\, P_s = A - \frac{B}{T} \qquad\qquad 4$$

Where A and B are constants, and can be determined by following expressions:

$$A\ =\ 12.6973\ +\ \frac{87582.885}{T_L\,.\,T_H} \qquad\qquad 5a$$

$$B = 4024.37 +\ 87582.885\,\left(\frac{1}{T_L} - \frac{1}{T_H}\right) \quad 5b$$

Where T_L and T_H are the lower and the upper temperatures in the range of saturation methanol. Similarly, the pressure-temperature relationship of an adsorbate can also be simply expressed as:

$$ln\, P = b.\left(-\frac{1}{T}\right) + a \qquad\qquad 6$$

To determine a and b (the intercept and the slope for a constant mass concentration line on the ln P vs $(-1/T)$ diagram respectively), rewrite D-A equation as:

$$ln\, P = c.\left(-\frac{1}{T}\right) + ln\, P_S \qquad\qquad 7a$$

Where

$$c = \left[\frac{ln\frac{x_0}{x}}{D}\right]^{\frac{1}{n}} \qquad\qquad 7b$$

Substitute equation 4 into equation 7a yields.

$$b = c + B = \left[\frac{ln\frac{W_0}{W}}{D}\right]^{\frac{1}{n}} + B\ \ and\ \ a = A \quad 8$$

With a series of *a* and *b*, a set of lines for a particular dsorbate/adsorbent pair can be drawn in the (ln P) vs. (-1/T) diagram. The parameters x_o, D, n and b are determined by adsorption test. Figure (2) shows such a diagram and an ideal adsorption refrigeration cycle.

Under the assumption that the specific volume of the gas is much greater than that of the liquid, the specific volume of the liquid is negligible, and if the pressure is low enough so the gas can be treated as the ideal gas, the equilibrium vapour-liquid phase transformation for a pure fluid can be expressed by Clausius-Claperyon equation[10]:

$$h_{fg} = \frac{d \ln P}{d\left(-\frac{1}{T}\right)} R \qquad\qquad 9$$

and

$$h_{ad} = \left[\frac{\partial \ln P}{\partial\left(-\frac{1}{T}\right)}\right]_X R \qquad\qquad 10$$

The subscript *x* signifies that the derivative is taken at constant concentration.

It is clear that the values of h_{fg} and h_{ad} is obtained from the slopes of the equilibrium lines for the pure refrigerant, and for a constant concentration *x* on the $(\ln P_s)$ vs (- *1/ T*) diagram. The value of h_{ad} is obtained by multiplying the slope (b) by gas constant (R) of methanol as follows.

$$h_{ad} = b.R \qquad\qquad 11$$

In Figure (2), the desorption and adsorption processes starts at temperatures T_2 and T_4 respectively, (called threshold temperatures), while the desorption process occur at condensing pressure Pc, and adsorption process occurs at evaporator pressure P_e. Since the threshold temperature cannot be measured simply, therefor it is determined using *D-A* equation.

Equation (6) can be rewritten as:

$$T_1 = \frac{b_{x_{max}}}{a_1 - \ln P_e} \quad and$$
$$\qquad\qquad\qquad\qquad 12$$
$$T_2 = \frac{b_{x_{max}}}{a_1 - \ln P_c}$$

The relation between starting temperature T_1, and threshold temperature T_2 is:

$$T_2 = \frac{a_1 - \ln P_e}{a_1 - \ln P_c}.T_1 \qquad\qquad 13$$

Similarly, the relation between condensing temperature T_3 and threshold temperature T_4, can be written as:

$$T_4 = \frac{a_3 - \ln P_c}{a_3 - \ln P_e} T_3 \qquad\qquad 14$$

Since $a_1 \approx a_3$ the relationship of the four temperatures can be written as:

$$\frac{T_2}{T_1} \approx \frac{T_3}{T_4} \qquad\qquad 15$$

3-1 Processes of adsorption cycle:

The adsorption cycle consist of four processes as follows:

Heating Process 1—2 :

In the heating process that shown in Figure (3a) the temperature of the adsorbent/adsorbate is increased from T_1 to T_2 . However, there is no methanol vapour flow from the generator, so the concentration in this process is constant at $x_{max.}$, also there is no work done. That is to say that the

heat supplied in this process is used to increase the internal energy of the active carbon, methanol and generator materials.

$$dU_{tube} + dU_{shell} + dU_{A.C.} = (m_{tube}.c_{tube} + m_{shell}.c_{shell} + m_{A.C.}.c_{A.C.})\ dT \qquad 16$$

All specific heats in the temperature range of equation (16) are shown table (1).
The change in internal energy of the methanol is:-

$$\Delta U_{meth.)_{1-2}} = m_{A.C.}.x_{\max} \int_{1}^{2} c_{v_{meth.}}\ dT \qquad 17$$

where $x_{\max}$ is the maximum concentration of methanol in the active carbon and determined by the following equation:-

$$x_{max} = x_0\ e^{-\{D[(A(T_1)-lnP_e)T_1-B(T_1)]^n\}} \qquad 18$$

Since the refrigerant methanol in all processes is adsorbed, thus it can be deal with it as in the liquid phase, therefor it can be assumed that all specific heats (namely at constant pressure, at constant volume and in the liquid phase)are the same, and are calculated by the following equation [11].

$$c_f = 3.3625 - 0.01189375T + 0.000030593T^2 \qquad\qquad 19$$

So, the average heat capacity of liquid methanol is:

$$c_{f_{1-2}}(T_2 - T_1) = 3.3625(T_2 - T_1) - 5.946875 \times 10^{-3}(T_2^2 - T_1^2) + 1.01977 \times 10^{-5}(T_2^3 - T_1^3)$$
$$20$$

Hence the change of the internal energy of the methanol is:

$$\Delta U_{meth._{1-2}} = m_{A.C.}.x_{max}.c_{f_{1-2}}(T_2 - T_1) \qquad\qquad 21$$

The total heat used to heat the generator $Q_{C.M,1-2}$ is equal to:-

$$Q_{CM_{1-2}} = (m_{tube}.c_{tube} + m_{shell}.c_{shell} + m_{A.C.}.c_{A.C.} + m_{A.C.}.x_{max}.c_{f_{1-2}})(T_2 - T_1)22$$

Desorption Process 2—3 :
When the temperature of the adsorption pair reaches the desorption temperature T_2 desorption starts as shown in Figure (3 b), and ending at maximum temperature T_3 (corresponding to the minimum concentration $x_{min.}$). During this process the refrigerant is driven off from generator continuously and the partial condensing pressure P_c of the refrigerant vapour in the system remains constant.
Taking the generator as the control volume, the energy balance of the system that does not produced or absorbed any work is [12]:

$$\delta Q = \sum m_e\ h_e - \sum m_i\ h_i + \delta\dot{Q} + dU_{CV} \qquad\qquad 23$$

where $\delta\dot{Q}$ is the heat depleted during desorption process.
The term $(\sum m_e\ h_e - \sum m_i\ h_i)$ in eq. (23) can be written as $(h_{meth.e}.dm_e)$ since there is no methanol inter the C.V. The desorbed methanol leaving the C.V. is saturated vapour and can be written as:

$$h_{meth.} = h_g = h_f + h_{fg} \qquad\qquad 24$$

The enthalpy of a saturated liquid methanol at temperature T, is[12]:-

$$h_f(T) = h_{f,0} + c_{f,T_0-T}(T - T_0) + Pv - P_0v_0 \qquad 25$$

Where $h_{f,0}$ is the enthalpy of the methanol at the reference temperature, c_{f,T_0-T} is the average specific heat of the liquid methanol from T_o to T, P and v are the saturation pressure and specific volume of liquid methanol corresponding to saturation temperature T, respectively.

Similarly, the average specific heat of liquid methanol from 273.16 K to a temperature T is:-

$$c_{f,273.15-T} = 3.3625 - 0.005946875(T + 273.15) + 0.0000101977(T^2 + 273.15T + 273.15^2)$$
$$26$$

The flow energy of methanol can be determined by combination of equations 1 and 2 as follows:

$$P_s v_f = \frac{\left[10^3 e^{\left(12.6973 - \frac{4024.37}{T} - \frac{87582.885}{T^2}\right)}\right]}{[937.911 - 0.058267T - 0.001459T^2]} \qquad 27$$

Hence, the enthalpy of liquid methanol can be written as:
$$h_f(T) = 200 + c_{f,273.15-T}(T - 273.15) + Pv - P_0v_0 \qquad 28$$

Substituting the value of R and the expression for B from equation.(5b) in equation (11) yileds the latent heat of evaporation.

$$h_{fg} = 259.5 \times \left(4024.37 + \frac{2 \times 87582.885}{T}\right) \qquad 29$$

It is also reasonable to assume that methanol is adsorbed in liquid phase and desorbed in vapour phase. That is to say that the mass of the liquid methanol leaving the C.V. is:

$$dm_L = m_{A.C.} dx \qquad 30$$
While the mass of desorbed vapour methanol is:

$$m_V = m_{A.C.}(x_{max} - x) \qquad 31$$

$$dm_{meth.e} = m_{A.C.} dx \qquad 32$$

Where subscripts L and V represent the adsorbed of liquid refrigerant and the desorbed of vapour refrigerant, while the subscribe e represent the exit condition.

To simplify the calculation, taking an average value of enthalpy over the operating temperature , therefore the enthalpy of exit methanol can be written as:-

$$H_{meth.e} = m_{A.C.}. \Delta x\left(\overline{h}_f + \overline{h}_{fg}\right) \qquad 33$$

where $\Delta x = x_{max} - x_{min}$, $\overline{h}_f$ and $\overline{h}_{fg}$ are the average enthalpies of the liquid methanol and latent heat of evaporation of methanol from T_2 to T_3.

Therefore, the total enthalpy of vapour methanol that the leaving the C.V.is:

$$H_{meth.e} = m_{A.C.} \cdot \Delta x \left\{ 200 + \frac{1}{2}\left[c_{f_{0-2}}(T_2 - 273.15)\right] + c_{f_{0-3}}(T_3 - 273.15) + \left(\frac{P_2 v_2 + P_3 v_3}{2} - P_{0_c} v_{0_c}\right) + \overline{h}_{fg} \right\} \qquad 34$$

The net heat depleted in a desorption process $\delta\dot{Q}$, should be the difference between the heat of desorption and the latent heat evaporation, ie,:

$$\delta\dot{Q} = m_{A.C.} \cdot \Delta x \left(\overline{h}_{dr} - \overline{h}_{fg}\right) \qquad 35$$

Using eq. (13) and eq. (14), the average heat of adsorption $\overline{h}_{ad}$ can be determined by:

$$\overline{h}_{ad} = R \cdot \left(A_{\frac{T_2+T_3}{2}} - \ln P_c\right) \times \frac{T_2 + T_3}{2} + \frac{R(A_{T_2} - \ln P_c)T_2}{2} + \frac{R(A_{T_3} - \ln P_c)T_3}{2} \qquad 36$$

The total heat supplied to the system during process 2-3 is equal to the summation of change in internal energy of the generator tubes and active carbon and change in internal energy of the refrigerant adsorbed in active carbon, i.e.:

$$Q_{CV_{2-3}} = \left(m_{tube} \cdot c_{tube} + m_{shell} \cdot c_{shell} + m_{A.C.} c_{A.C.} + m_{A.C.} \frac{x_{max} - x_{min}}{2} \overline{c}_{f_{2-3}}\right)(T_3 - T_2) + m_{A.C.} \frac{\Delta x}{2}(P_2 v_2 + P_3 v_3) + m_{A.C.} \cdot \Delta x \cdot \overline{h}_{dr} \qquad 37$$

The total heat supplied during heating and desorption processes is the summation of $Q_{C.M.1-2}$ and $Q_{CV_{2-3}}$.

Cooling Process 3-4

From state 3 to state 4 the generator is cooled to the temperature T_4.

Refrigeration process 4-1 :

After the generator is cooled to the temperature T_4, the valve is open and the adsorption/refrigeration process that shown in Figure (3 c) take place. Take the receiver and the evaporator as the control volume for which there is no work done. Thus, net cooling effect during this process $Q_{net,c}$, is

$$Q_{net,c_{r_0-r_1}} = Q_{r_0-r_1} - \left(\delta Q_{rec_{r_0-r_1}} + \delta Q_{ev_{r_0-r_1}} + \delta Q_{leak_{r_0-r_1}}\right) \qquad 38$$

Where $\delta Q_{rec} + \delta Q_{ev} + \delta Q_{leak}$ are the sensible heat consumed to cool the receiver, evaporator and heat lost to the surroundings, respectively.

The refrigeration process is occurring in two steps. Firstly, a portion of the liquid refrigerant in the receiver/evaporator cools the receiver, evaporator materials, and the rest of the liquid refrigerant from condensing temperature T_C to the designed evaporator temperature T_e. Then the remaining liquid refrigerant evaporates at the evaporating pressure P_e (corresponding to the saturation temperature T_e) until evaporates completely and absorbs heating load.

In the first step, the temperature of the refrigerant temperature decreases from T_c to T_e (Denoting the process as r_0-r_1), the flow of heat across the control volume is:

$$Q_{r_0-r_1} = (m_{r_0} - m_{r_1})(h_{g_e} - u_{r_1}) - m_{r_0}(u_{r_0} - u_{r_1}) \qquad 39$$

Since the first step happens very quickly, it is reasonable to assume that there is no heat exchange between the system and its surroundings, ie, $Q_{r_0-r_1} = 0$, so that:

$$m_{r_1} = m_{r_0} \frac{h_{g_e} - u_{r_0}}{h_{g_e} - u_{r_1}} = m_{r_0} \frac{h_{fg} + P_e v_e - c_f(T_c - T_e)}{h_{fg_e} + P_e v_e} \qquad 40$$

In the second step, the evaporation process occurs at constant temperature (Denoting the process as r$_1$-r$_2$)

$$Q_{r_1-r_2} = m_{r_1}\left(h_{g_e} - m_{r_1}\right) \qquad 41$$

and:

$$Q_{net,c_{r_1-r_2}} = m_{r_1}\left(h_{g_e} - u_{r_1}\right) - \left(Q_{rec_{r_1-r_2}} + Q_{ev_{r_1-r_2}} + Q_{leak_{r_1-r_2}}\right) \qquad 42$$

Thus, the net cooling of load of the cycle is:

$$Q_{net,c_{r_0-r_2}} = m_{r_0}\left(h_{g_e} - u_{r_0}\right) - \left(Q_{rec_{r_0-r_2}} + Q_{ev_{r_0-r_2}} + Q_{leak_{r_0-r_2}}\right) \qquad 43$$

While the net change in internal energy is:

$$Q_{rec_{r_0-r_2}} + Q_{ev_{r_0-r_2}} = (m_{rec}c_{rec} + m_{ev}c_{ev})(T_c - T_e) \qquad 44$$

The mass of the refrigerant at the beginning of the evaporation is:

$$m_{r_0} = m_{A.C.}(x_{max} - x_{min}) = m_{A.C.}.\Delta x \qquad 45$$

Assuming that the methanol in evaporator entrance as saturated liquid, therefore it can be say that:

$$u_{r_0} = u_{f_{r_0}} = h_{f_{r_0}} - P_{r_0}v_{f_{r_0}} \qquad 46$$

Where P$_{ro}$, V$_{fro}$ and h$_{fro}$, are the pressure, specific volume, and the specific enthalpy of the saturated liquid corresponds to the temperature T_c
So $\left(h_{g_e} - u_{r_0}\right)$ may be expressed as:

$$h_{g_e} - u_{r_0} = h_{fg_e} - c_{r_f}(T_c - T_e) + P_e v_{f_e} \qquad 47$$

The latent heat of evaporation at evaporator temperature h_{fg_e} can be determined as in equation (31). And the net cooling effect is:

$$Q_{net_c} = m_{A.C.}.\Delta x\left[h_{fg_e} - c_f(T_c - T_e) + P_e v_{f_e}\right] - (m_{rec}c_{rec} + m_{ev}c_{ev})(T_c - T_e) - Q_{leak} \qquad 48$$

COP of the Ideal Refrigeration Cycle:
The coefficient of performance (COP) of a cycle is defined as the ratio of cooling effect to the energy transferred to the cycle to accomplish this effect. so the net COP of the cycle is:

$$COP_{net} = \frac{Q_{net_c}}{Q_h}$$

$$= \frac{m_{A.C.}.\Delta x\left[h_{fg_e} - c_f(T_c - T_e) + P_e v_{f_e}\right] - (m_{rec}c_{rec} + m_{ev}c_{ev})(T_c - T_e) - Q_{leak}}{\left(\begin{array}{c}(m_{tube}.c_{tube} + m_{shell}.c_{shell} + m_{A.C.}c_{A.C.} + m_{A.C.}x_{max}.\overline{c}_{f_{1-3}})(T_3 - T_1) \\ + m_{A.C.}\frac{\Delta x}{2}\left(P_{s_2}v_{f_2} + P_{s_3}v_{f_3}\right) + m_{A.C.}.\Delta x.\overline{h}_{dr}\end{array}\right)} \qquad 49$$

The Gross COP of the cycle is:

$$COP_{gross} = \frac{Q_{gross_c}}{Q_h}$$

$$= \frac{m_{A.C.}.\Delta x\left[h_{fg_e} - c_f(T_C - T_e) + P_e v_{f_e}\right]}{\left(m_{tube}.c_{tube} + m_{shell}.c_{shell} + m_{A.C.}x_{max}.\overline{c}_{f_{1-3}}\right)(T_3 - T_1) + m_{A.C.}\frac{\Delta x}{2}\left(P_{s_2}v_{f_2} + P_{s_3}v_{f_3}\right) + m_{A.C.}.\Delta x.\overline{h}_{dr}} \qquad 50$$

4- Results and discussions:

The effect of many key variables were examined to find the effect of operating temperatures on the cycle COP, and the software called Engineering Equation Solver software (EES) was used to solve the equation that mentioned in theoretical analysis. Referred to Figure (2) that show a complete adsorption cycle, the unit generator submitted to many temperatures, namely; evaporator temperature T_e, condensing temperature T_c, starting temperature T_1, peak temperature T_3. Figure (4) shows the effect of peak temperature on the cycle COP at different starting temperature, it can be seen from the figure that, starting temperature affecting COP directly, while peak temperature affecting COP inversely, this is due to as the peak temperature increases the amount of heat supplied to the generator increases. The higher condensing temperature reduced COP, as shown in Figure (5), since condensing temperature affect the condensation process and the degree of sub-cooling. Figure (6) shows the effect of starting temperature on the cycle COP , at different condensing temperatures. It can be seen from the figure that as the starting temperature increases the COP increases too. This is due to that less energy is consumed in rising generator temperature. Figure (7) shows the effect of starting temperature on the COP at different values of certain peak temperatures. It can be seen from the figure that maximum COP is at a lower generator temperature. Figure (8) shows the effect of evaporator temperature on the cycle COP, it can be seen from the figure that as the evaporator temperature increases the cycle COP increases also. Figures (9) and (10) show the effect of evaporator temperature on the cycle COP, for different values of starting temperature and generator temperature respectively. It can be seen from all figures that the COP is affected strongly by the condenser temperature, while the generator temperature has a little effect on the cycle COP.

5- Conclusions:

1- For a certain ambient temperature there is an optimum generator temperature at which the reaches a maximum .Therefore, there is no need to increase the generator temperature more than necessary.
2- The adsorption refrigeration system is suitable when the ambient temperature is low, as the ambient temperature increases the condensing temperature increases also, thus leads to reduce cycle COP
3- Heavy materials used in building the adsorption cycle leads to increase the consumed heat that used to heat up the generator.

Table (1): Specifications and specific heat of adsorption system components.

materials	Number	Length (mm)	Diameter (mm)	Weight (kg)	C kJ/kg K
copper tubes	44	440	22.22	0.1189	0.39
brass plate	2	—	240	0.467	0.4
generator shell	1	450	240.4	2.634	$0.5_{\text{at }100^{o}C}$
Active carbon	—	—	—	2.5	1
Methanol	—	—	—	0.175	$2.47_{\text{at }25^{o}C}$
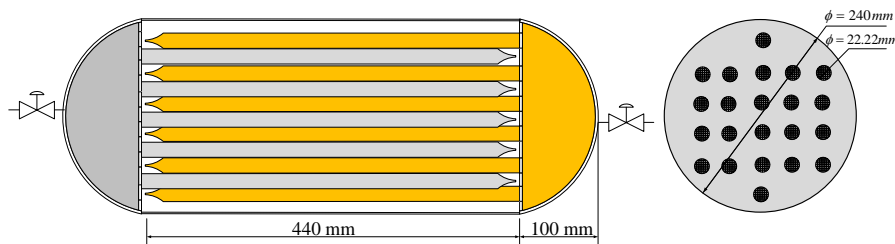
Figure (1) Unit generator of adsorption cycle
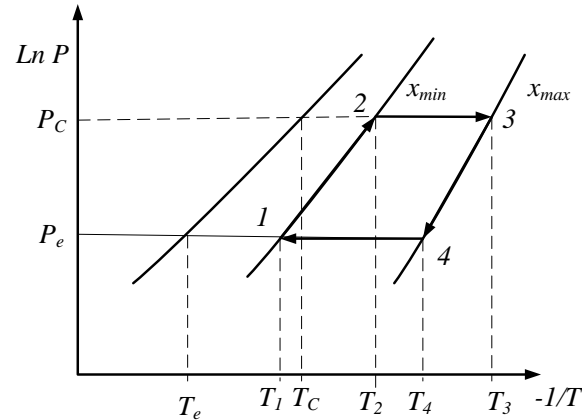
Figure (2): The *P-T-x* diagram of the ideal basic adsorption cycle

a-   Heating process 1-2   b-   Heat and desorption 2-3

c-   Cooling and adsorption 4-1
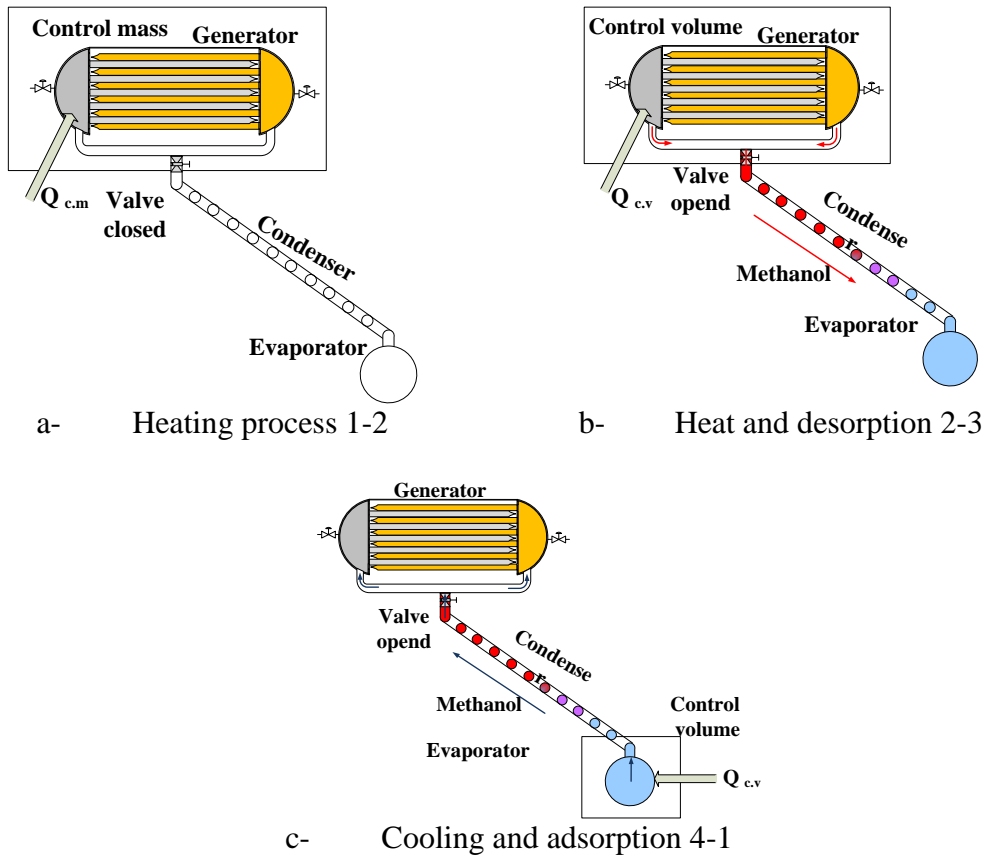
Figure (3) adsorption refrigeration cycle
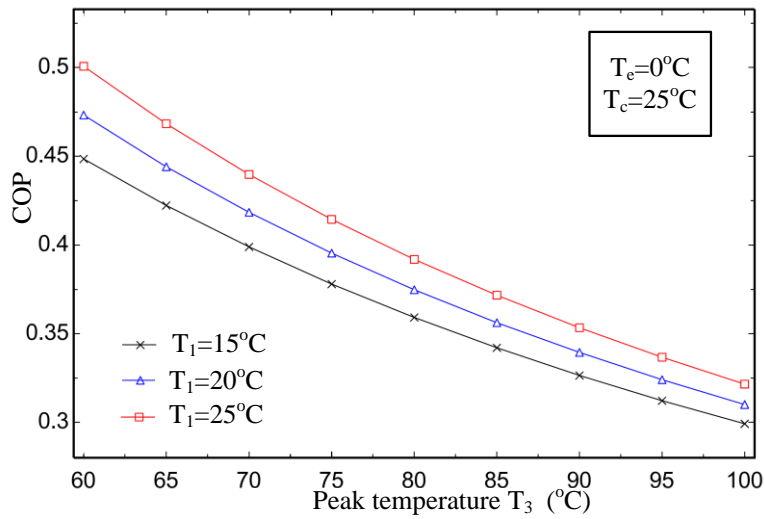
Figure (4): Effect of peak temperature on COP at different starting temperature.
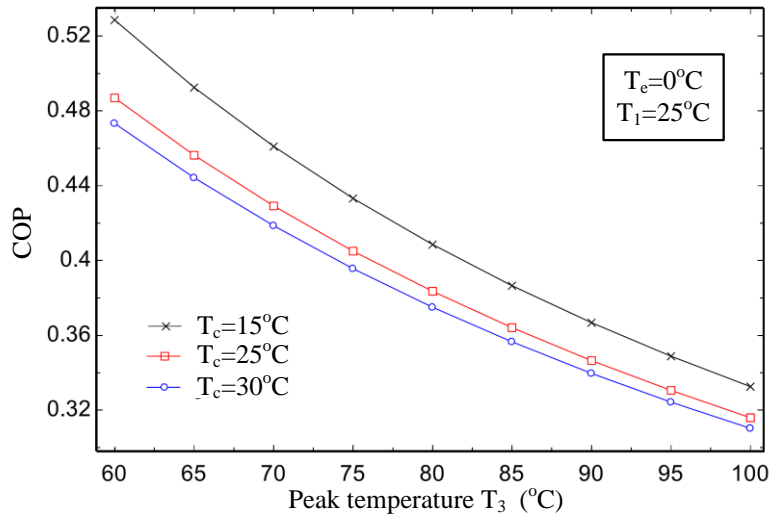
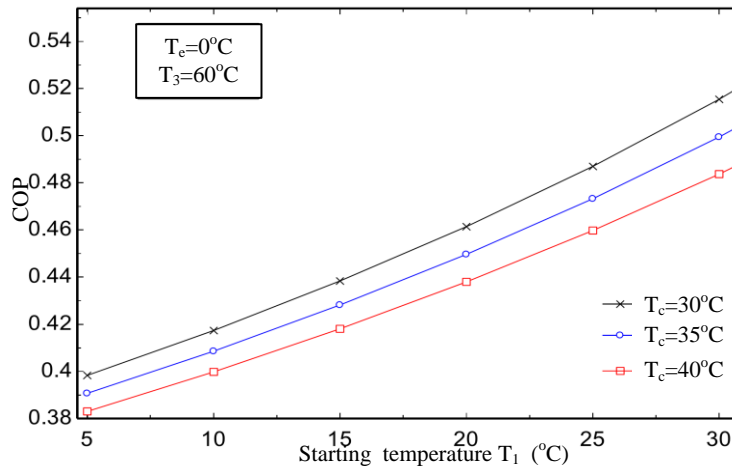Figure (5) Effect of peak temperature on COP at different condensing temp.

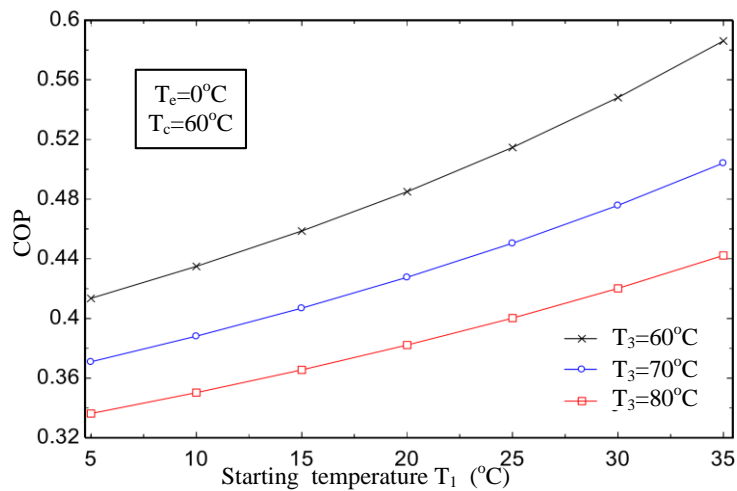Figure (6) Effect of starting temperature on COP at different peak temperature..

Figure (7) Effect of starting temperature on COP at different peak temperature.
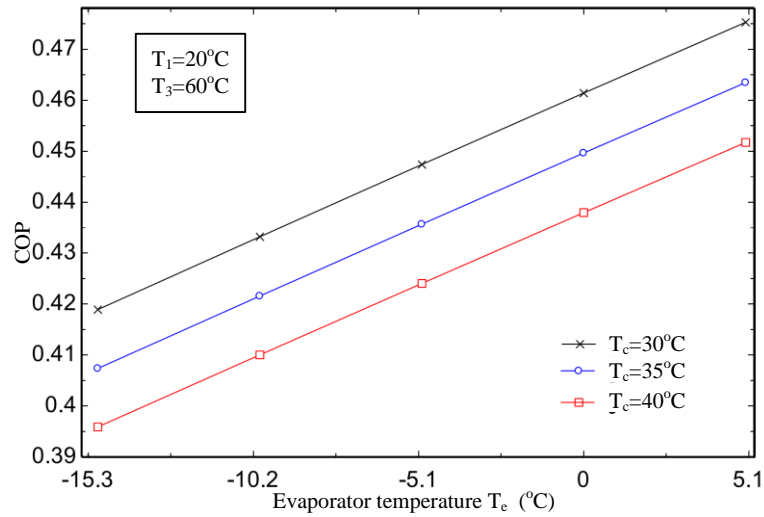
Figure (8) Effect of evaporator temperature on COP at different condensing temperature.
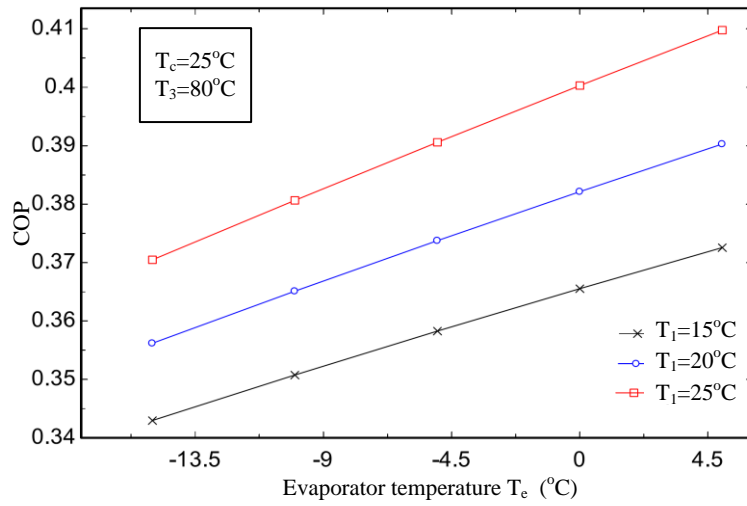

Figure (9) Effect of evaporator temperature on COP at different starting temperature.
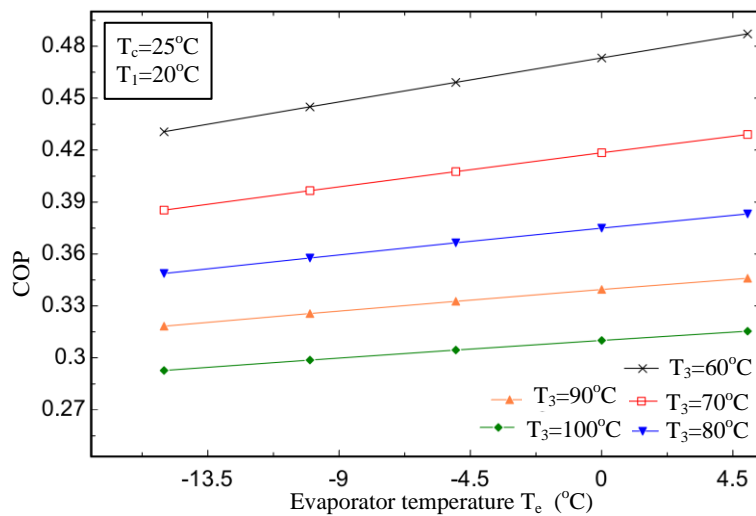

Figure (10): Effect of evaporator temperature on COP at different peak temp.

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