COMPUTER SIMULATION OF AN AAUTOMBILE ADSORPATION AIR CONDITIONING BED [+]

برنامج محاكات لمولد البخار لمنظومة تبريد سيارة تعمل بالأمتزاز

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Abstract:

This paper is aimed to develop simulation software of automobile adsorption air conditioner bed based on the adsorption principle which utilizes only the waste heat of the vehicle. The system incorporates one adsorbent bed and utilizes an adsorbant-refrigerant working pair to provide chilled water for conditioning the air in the small space equivalent to the cab of vehicle.

The mathematical model involves a first order system for the unsteady heat transfer, thermodynamics and adsorption process. The theoretical results have an acceptable agreement with the experimental data from other researches.

Keywords: Adsorption, Air conditioner, Activated carbon, Methanol, Heat transfer.

المستخلص :

يهدف هذا البحث الى تطوير برنامج محاكاة لمولد بخار لمنظومة تبريد سيارة امتزازية التي تستفيد من الحرارة الضائعة للمحرك. يتكون النظام من مولد بخار واحد وزوج من مائع التبريد والمادة الممتزة لتجهيز ماء بارد لغرض تبريد الهواء في حيز صغير يماثل كابينة السيارة. النموذج الرياضي تضمن نظاماً من الدرجة الأولى لأنتقال الحرارة غير المستقر والثرموديناميك وعملية الأمتزاز.النتائج النظرية كانت مقاربة للنتائج العملية المستخلصة من اعمال سابقة.

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List of Symbols:

Symbol	Definition	Unit	Symbol	Definition	Unit
A	Area	m^2	Q_{in}	Heat added.	W
C_b	Specific heat capacity of bed.	$\frac{J}{kg.K}$	Q_{st}	Heat storage.	W
C_{AC}	Specific heat capacity of active carbon.	$\frac{J}{kg.K}$	T	Temperature.	K
C_M	Specific heat capacity of methanol.	$\frac{J}{kg.K}$	T_i	Initial temperature.	K
C_r	Ratio of specific heat capacity.	----	T_s	Saturation temperature.	K
h_{ad}	Enthalpy of adsorption.	$\frac{J}{kg}$	T_∞	Environment temperature.	K
m_b	Mass of bed	kg	t	Time.	s
$\dot{m}_M$	Mass low rate of methanol.	$\frac{kg}{s}$	U	Over all heat transfer.	$\frac{W}{m^2.K}$
P	Pressure.	bar	x_o	Max. mass concentration of methanol.	$\frac{kg}{kg}$
P_s	Saturation pressure.	bar	x_1	Mass concentration of methanol.	$\frac{kg}{kg}$
Q_{ad}	Heat adsorption.	W	τ_o	Characteristic system time.	s^{-1}

Introduction:

Vapor adsorption by a solid adsorbent was first discovered by Araday in 1848, and these systems were first commercialized in the 1920s and hold great promise for overcoming the limitation of the engine driven vapor compression and the liquid/vapor absorption systems. The basic solid/vapor adsorption cycle utilized in the period from 1920 to 1940 was simple and inefficient. With the advent of the mechanical compressor and the application of the CFC's which are much more efficient, further investigations of the adsorption cycles were at a standstill until the mid 1970s. After the oil crisis during the 1970s, new development of this research work were resumed with the aim of a rational use of energy, which was to use either solar energy or waste heat to drive refrigerators, air conditioners and heat pumps. The main argument in favor are that sorption systems are quiet, long lasting, easy to maintain and environmentally benign.

Natural refrigerants such as water, ammonia, methanol, etc. are the promising candidates for future refrigeration and air conditioning applications. The use of waste heat and solar energy for refrigeration and air conditioning purposes has been accepted by people; accordingly various adsorption systems have been developed for different applications. Water can be used as a refrigerant which is quite attractive for the environmental protection as it possess neither ozone depletion nor a global warming potential. Adsorption refrigeration cycle powered by solar energy or waste heat exhausted from engines has been successfully used for ice making or cold production. For examples, a carbon - methanol solar ice maker, Zeolite - water solar cold storage system, activated carbon - ammonia solar refrigerator for vaccine cooling and silica gel–water adsorption refrigeration cycle driven by exhaust heat of near ambient

temperature have been reported[1,2,3,4,5].

The exhaust heat energy utilization in solid adsorption refrigeration system may be best suited for refrigeration purposes. It is a well-known fact that significant amount of energy released in the engine is wasted through exhaust gas [1]. Thus; if there is a possibility to design a system to utilize the thermal energy of an engine exhaust to run some air conditioning system innovation technologies. One of such innovations is the exhaust gas powered adsorption system.

Zhang et al. [6], established a theoretical model of the heat-transfer processes in a solid adsorbent packed bed. Based on discretized energy control equations of the fluid in the heat-transfer coil and transient heat conduction equations of the adsorbent in the bed with unsteady boundary conditions, a numerical analysis was made. Through numerical computation, a coupling temperature distribution in the adsorbent and the heat-transfer coil is obtained and solved to obtain an optimal design of the solid adsorbent packed bed.

Chekirou et al. [7], established a theoretical model for the heat and mass transfer processes in a tubular adsorber of a solid adsorption solar refrigerator. The modeling and the analysis of the adsorber was the key of such studies because of the complex coupled heat and mass transfer phenomena that occur during the working refrigeration cycle. This model consists of the energy equation in the adsorbent layers, the energy balance equation of the adsorber wall, and state equation of the bi-variant solid vapour equilibrium using the Dubinin-Astakhov model to describe the phenomena of adsorption with the activated carbon AC-35/ methanol as an adsorbent/adsorbate pair. The influence of the main parameters on the system was also discussed. Their numerical simulations have shown that the cooling power and solar performance of coefficient COPs increase with increasing the evaporation temperature and decrease with the increase of the adsorption and condensation temperatures.

Liu et al. [8], proposed a new transient two-dimensional model with non-constant condensing pressure for a zeolite/water adsorption cooling cycle. This numerical model focuses on the heat and mass transfer behaviors in the adsorber and was solved by the control volume method. Due to the heat transfer limitation in the condenser, the simulated pressure during the isobaric generation phase of the cycle is not constant and will decrease with time. Compared with the model of constant condensing pressure, the cycle duration and cycled adsorbate for the base case are increased. Furthermore, the effect of mass flow rate of condenser cooling water on system performance is also investigated. It is found that both performance of coefficient COP and specific cooling power SCP increase with an increase in the mass flow rate of cooling water in the condenser.

Recently many researches contributed to the efforts of building more efficient adsorbtion systems. Ali H. Jabbar [9] has built an experimental rig to test a possible implant of a vehicle adsorption A/C system. His experimental rig gave a maximum cycle COP of about 0.31 when the hot gases temperature is about 120 ^{o}C ,the theoretical part of his work was simple and did not took into account the mass balance principle instead he assumed that the mass of methanol is constant at full charge which is obviously an over estimated value. This pitfall was eliminated in the current model by taking the mass balance across the cycle and consequently a more suitable value for the quantity of methanol is predicted.

A second defect in the model of Jabber [9] was that he has taken only the adsorption/ desorption processes in the model isolated from the whole cycle which is also avoided in the current model by taking the whole cycle into consideration. Most of the previous researches focused on the detailed numerical analysis of the adsorption bed. Paying less attention to bed optimization in regard with timing sequences and control strategy. In this work, an attempt was made to enlighten such aspects.

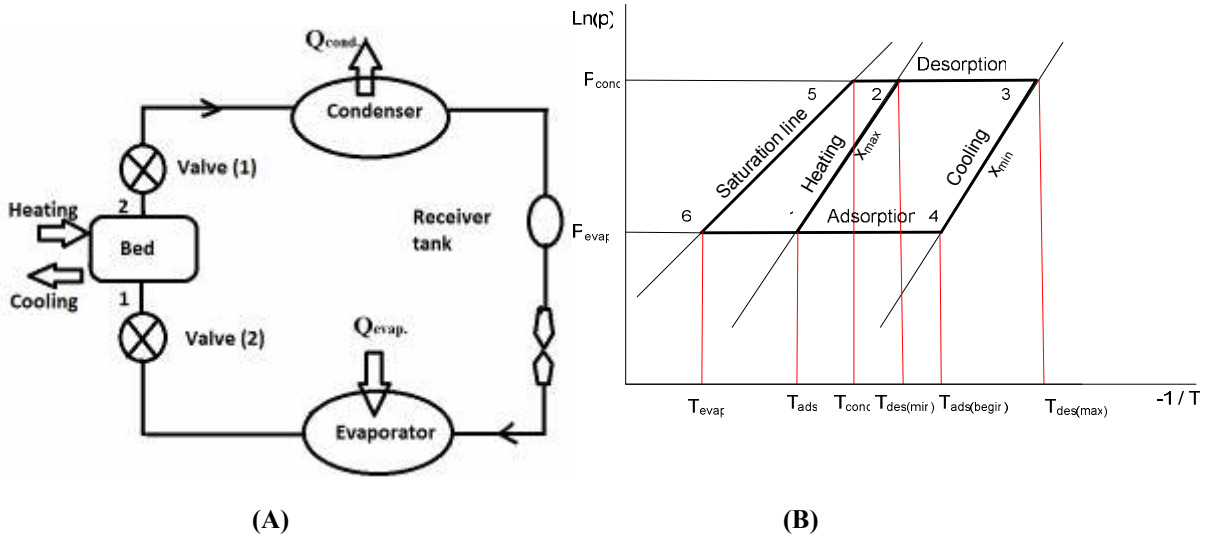
Thus the aim of the current work can be summarized as:

1- Build a more sophisticated mathematical model to simulate the operation of an adsorption system capable of prediction and analysis the design parameters.

2- Validate the mathematical model with the available experimental data from other researches.

Physical Model:

Figure (1-A) represent a schematic of an adsorption / desorption mechanism. It consists of a bed isolated and connected to the condenser and evaporator by valves. When heat is added to the generator (bed) the temperature and pressure increase until they reach the desorption threshold pressure and the generator (bed) and condenser are then connected by opening valve(1), while keeping valve (2) closed. At this moment the (heating/desorption) process starts. During this process, the temperature of the generator (bed) rises, keeping the pressure constant at condenser pressure.



(A) **(B)**

Figure (1) (A) Adsorption and desorption mechanism and (B) the P-T-X diagram of the ideal basic adsorption cycle

During this constant pressure process, the adsorbate (refrigerant) is driven off until the minimum concentration of the refrigerant is reached in the generator/bed (corresponding to the maximum temperature).

The adsorption / desorption cycle consists of two isosters and two isobars, as illustrated in the diagram Figure (1-B). The process starts at point 1, where the adsorbent is at a low temperature T_{ads} (adsorption temperature) and at low pressure P_{evap} (evaporation pressure). While the adsorbent is heated by a hot gases, the temperature and the pressure increase along the isoster during which the mass of the adsorbate in the adsorbent remains constant at m_{max} . The adsorber still isolated until the pressure reaches the condenser pressure, point 2 (the limit point of desorption $T_{des(min)}$). At this time, the adsorber is connected with the condenser and the progressive heating of the adsorbent from point 2 to 3 causes a desorption of methanol and its vapour is condensed in the condenser and collected in a receiver. When the adsorbent reached its maximum temperature value $T_{des(max)}$ (regenerating temperature) and the adsorbed mass decreases to its minimum value m_{min} (point 3), the adsorbent starts cooling along the isoster at a constant mass m_{min} to point 4 (the limit point of adsorption $T_{ads(begin)}$). During this isosteric cooling phase, the adsorbent pressure decrease until it reaches the evaporator pressure P_{evap} . After that, the adsorber is connected to the evaporator, and both adsorption and evaporation occur while the adsorbent is cooled from point 4 to 1. In this phase, the adsorbed mass increases up to its maximum m_{max} at point 1. The adsorbent is cooled until the adsorption temperature T_{ads} by rejecting the sensible heat and the heat of adsorption. During this phase the cold is produced.

Mathematical Model :

Seeking for an optimal timing design requires better understanding for the interrelationship between (x, P, T, $\dot{m}_M$, t). This requirement could be sufficiently fulfilled by a direct algebraic relations between the relevant parameter in addition algebraic relations can be easily obtained.

The mathematical model was developed under the following assumptions:

1. The adsorbent / adsorbate system is in thermodynamic equilibrium at all point of the adsorber in any given moment;
2. The pressure is assumed to be uniform in the bed (grad P = 0);
3. The system is considered to be lumped system;
4. Uniform the thermo physical properties;
5. During the heating/cooling processes, the mass of the bed is held constant since the adsorbate is teemed in the bed;
6. All other properties are assumed to be constant (U,A,C_b) ,since they are almost unchanged over the small working temperature range (about 50^{o}C).

The governing equations on which the model is based are the unsteady heat equation and the first law of thermodynamics.

A- Heating and cooling process:

The unsteady heat transfer equation was used to model the bed undergoing these processes.

$$\frac{d}{dt}.m_b\,.\,C_b.T = U\,.A.\left(T_o - T_i\right) \qquad ....(1)$$

Finally equation (1)

$$T = T_\infty - \left(T_\infty - T_i\right).e^{-\beta_1 .t} \qquad ....(2)$$

Where (β_1) is a new parameter defined as:

$$\beta_1 = \frac{U.A}{m_b.C_b} = \frac{1}{\tau_o} \qquad ....(3)$$

$$\frac{T_\infty - T}{T_\infty - T_i} = e^{-\beta_1 .t} \qquad ....(4)$$

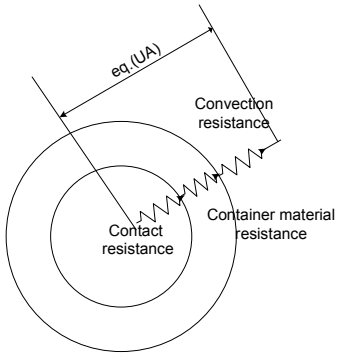


Figure (2) the diagram of bed thermal resistances

One of the main objectives of the current work is to identify and simulate an existing working automobile adsorption system of Jabber [9], the parameter β_1 should be evaluated from the working conditions of the above mentioned system.

Also during this phase the pressure can be calculated from Doubnin-Astakhov (DA) equation [3].

$$\left[\frac{1}{D}\ln\frac{x_o}{x_1}\right]^{\frac{1}{n}} = T\ln\left(\frac{P_s}{P}\right) \qquad ....(5)$$

Where (D) and (n) are constant adsorption parameters for each adsorption pair.

The LHS of equation (5) is constant during heating/cooling process, since there is no adsorption/desorption of methanol during this phase. For P_s, an equation devised by Wang [3] is used:

$$\ln\left(P_s\right) = A - \frac{B}{T} \qquad ....(6)$$

Where

$$A = C_1 + \frac{C_2}{T^2} C_1 \qquad \ldots(7)$$

$$B = C_3 + 2\frac{C_2}{T} C_3 \qquad \ldots(8)$$

Then the RHS of equation (5) could be simplified to be

$$\therefore T.\ln\left(\frac{P_s}{P}\right) = T.\ln(P_s) - T.\ln(P) \qquad \ldots(9)$$

$$T.\ln\left(\frac{P_s}{P}\right) = A.T - B - T.\ln(P) \qquad \ldots(10)$$

Using equation (6) or by using equations (7) and (8).

$$T.\ln\left(\frac{P_s}{P}\right) = C_1.T + \frac{C_2}{T} - C_3 - 2\frac{C_2}{T} - T.\ln(P) \qquad \ldots(11)$$

$$T.\ln\left(\frac{P_s}{P}\right) = T[C_1 - \ln(p)] - C_3 - \frac{C_2}{T} \qquad \ldots(12)$$

To apply the model on the existing system in Technical College Baghdad Laboratories, the following operational specifications of the system are used
m=3kg, D= 0.00009394 , n= 1.38.
Also, since the bed is not fully changed (saturated), then it will not follow line (6-5) on Figure (1-B). But other will follow 1-2 line and thus it is expected that:

$x_{max} < x_{sat.}$

$P < P_{sat.}$

In which for the active carbon-methanol pair: $x_{sat.}=0.29$
To check out the system, two arbitrary points are selected, for model evaluation:
At t=0, T= 25^{o}C=298.15 K , P=0.05 bar(measured)
From equation (6)
P_s=16.77 kPa which is higher than(P) as expected, meanwhile (x) could be calculated using equation (5)
At t=20 min=1200s ,T=52 ^{o}C=325.15 K, P=0.34 bar (measured) P_s=60.2 bar
P_s again is higher than P.
For the cooling period, β_1 should be corrected for the different values of (x) during the heating and cooling processes as follows since Wang [3]:

$m_b C_b = m_{AC}\,(C_{AC} + xC_M)$ ….(13)

$m_b C_{bed} = m_b\,(1000 + 2536.56x)$ for current system. Hence it is suggested in the current work that β_1 should be corrected as follows:

$$\beta_{1_{cooling}} = \beta_{1_{heating}} \frac{(m_b C_b)_{heating}}{(m_b C_b)_{cooling}} \qquad \ldots(14)$$

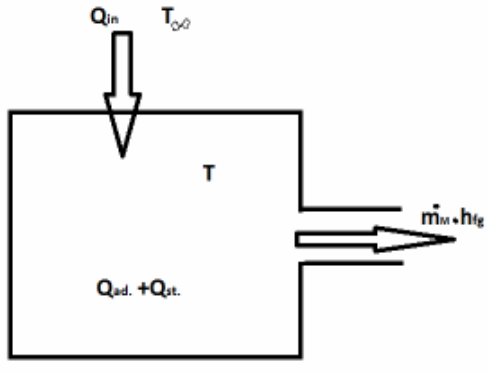


Figure (3) the diagram of heat transfer for the bed system

B- **Desorption and adsorption process**:
Applying the conservation of energy to the bed system gives:

$$Q_{in} - m_M^{\bullet}.h_g = Q_{ad} + Q_{st} \qquad \ldots(15)$$

Where

$$Q_{in} = U.A(T_\infty - T) \quad \ldots(16)$$

$$m_M^{\bullet}.h_g = -m_{AC}\frac{dx}{dt}h_g \quad \ldots(17)$$

$$Q_{ad} = -m_{AC}\frac{dx}{dt}h_g \quad \ldots(18)$$

$$Q_{st} = -m_b.C_b\frac{dT}{dt} \quad \ldots(19)$$

Substitution equations (16, 17, 18 and19) into equation (15) gives:

$$U.A(T_\infty - T) + m_{AC}\frac{dx}{dt}h_g = -m_{AC}\frac{dx}{dt}h_g + m_bC_b\frac{dT}{dt} \quad \ldots(20)$$

$$m_{bed}.C_{bed} = m_{AC}(C_{AC} + xC_M) \quad \ldots(21)$$

Since (x) is a function of both T and P then:

$$\frac{dx}{dt} = \left(\frac{\partial x}{\partial T}\right)_P.\frac{\partial T}{\partial t} + \left(\frac{\partial x}{\partial P}\right)_T.\frac{\partial P}{\partial t} \quad \ldots(22)$$

Since the desorption process can be considered as an isobaric process, for an isolated bed then, $\frac{\partial P}{\partial t} \approx 0$ and

$$\frac{dx}{dt} = \left(\frac{\partial x}{\partial T}\right)_P.\frac{\partial T}{\partial t} \quad \ldots(23)$$

Defining

$$\beta_2 = \frac{U.A}{m_{AC}.C_{refrence}} \quad \ldots(24)$$

And if $C_{refrence}$ is taken as C_M average value, then;

$$\beta_2 = \frac{U.A}{m_{AC}.C_M} \quad \ldots(25)$$

From known thermodynamics relations for h_{fg} and h_{ad}

$$h_{fg} = R.\frac{d[\ln(P_s)]}{d\left[-\frac{1}{T_s}\right]} = R\left(C_3 + \frac{2C_2}{T_s}\right) \quad \ldots(26)$$

$$h_{ad} = R.\left(\frac{\partial[\ln(P)]}{\partial\left(-\frac{1}{Y}\right)}\right)_x \quad \ldots(27)$$

$$h_{ad} = [T.\ln(P_s) - T.\ln(P) + B].R \quad \ldots(28)$$

Applying equations (6) to (12) into the above equation (29) yields
or

$$h_{ad} = T.[A - \ln(P)].R \quad \ldots(29)$$

$$h_{ad} = R.\left[T.(C_1 - \ln(P)) + \frac{C_2}{T} \right] \quad \ldots.(30)$$

Since

$$h_g = h_f + h_{fg} \quad \ldots.(31)$$

$$\left(\frac{\partial x}{\partial T} \right)_P = -n.D\left[T.\ln\left(\frac{P_s}{P} \right) \right]^n . \frac{x}{T} \quad \ldots.(32)$$

$$\left(\frac{\partial x}{\partial T} \right)_P = \frac{n.x}{T} \ln\left(\frac{x_o}{x} \right) \quad \ldots.(33)$$

Rearranging

$$\beta_2 (T_\infty - T) = \left[(C_r + x) - \frac{n.x}{T} \ln\left(\frac{x_o}{x} \right) . (h_g + h_{ad}) \right] \frac{dT}{dt} \quad \ldots.(34)$$

Where $C_r = \dfrac{C_{AC}}{C_{refrence}}$

This equation can now be integrated to find variation with time.

The same equation can be used to model the adsorption and desorption process by changing the working pressure, concentration and other properties to accommodate the relevant process.

Results:

Figure (4) shows bath the experimental results [9] and the numerical simulation results for the transient temperature distribution. The matlab program was built utilizing the mass balance principle to get the mass of refrigerant cycled through the device. This figure depicts an acceptable agreement especially during the adsorbtion period while the desorbtion period shows the worst case. In general the temperature variation through the cycle is qualitative follows the experimental results at least in the trend. The slope of the rising edge of the pressurization period is almost identical but there is a discrepancy in the starting point and the period duration. The desorption period slope compared well to the experimental results as well as the duration but the accumulated error from the previous period shift s the numerical curve up. The same can be said for the cooling and adsorption periods. Figure (5) shows the pressure diagram throughout one bed cycle. Again it can be seen the first period is the source of accumulated error that affects the whole cycle. The starting overshot is caused by the valve sudden opening. Another overshot is also show to be caused by valve closing.

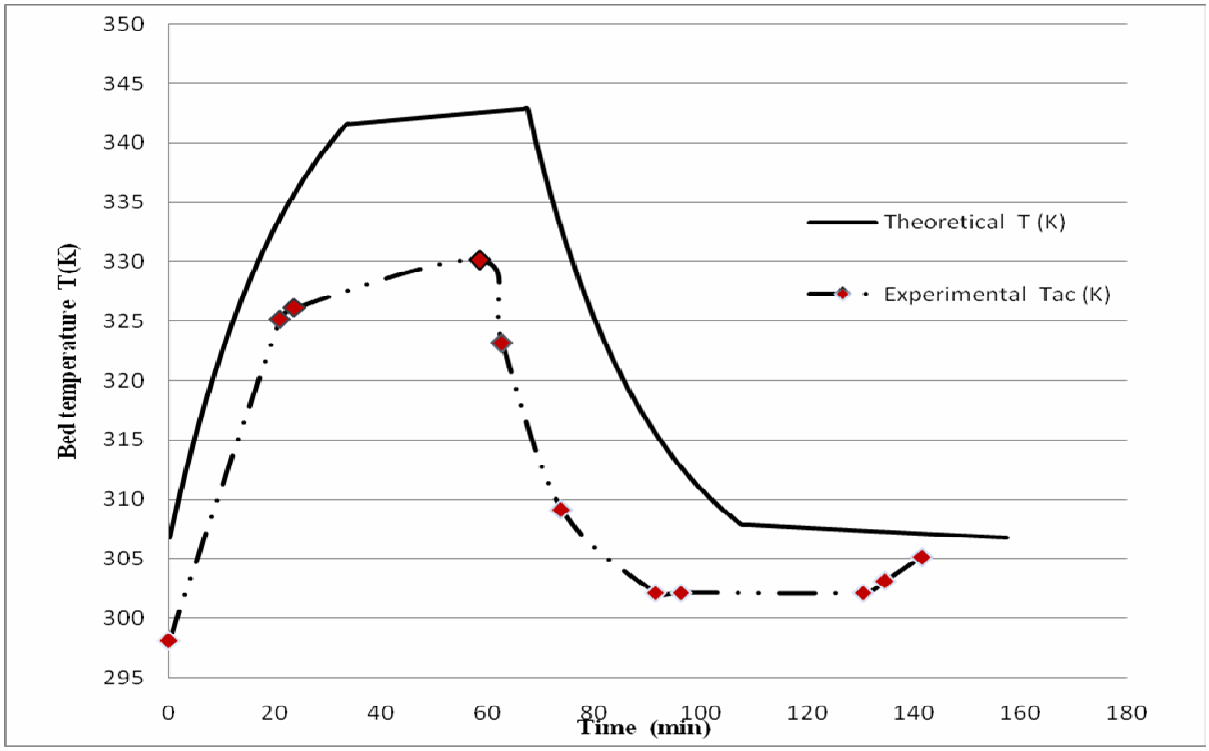


Figure (4) the adsorption bed temperature transient during one complete cycle.

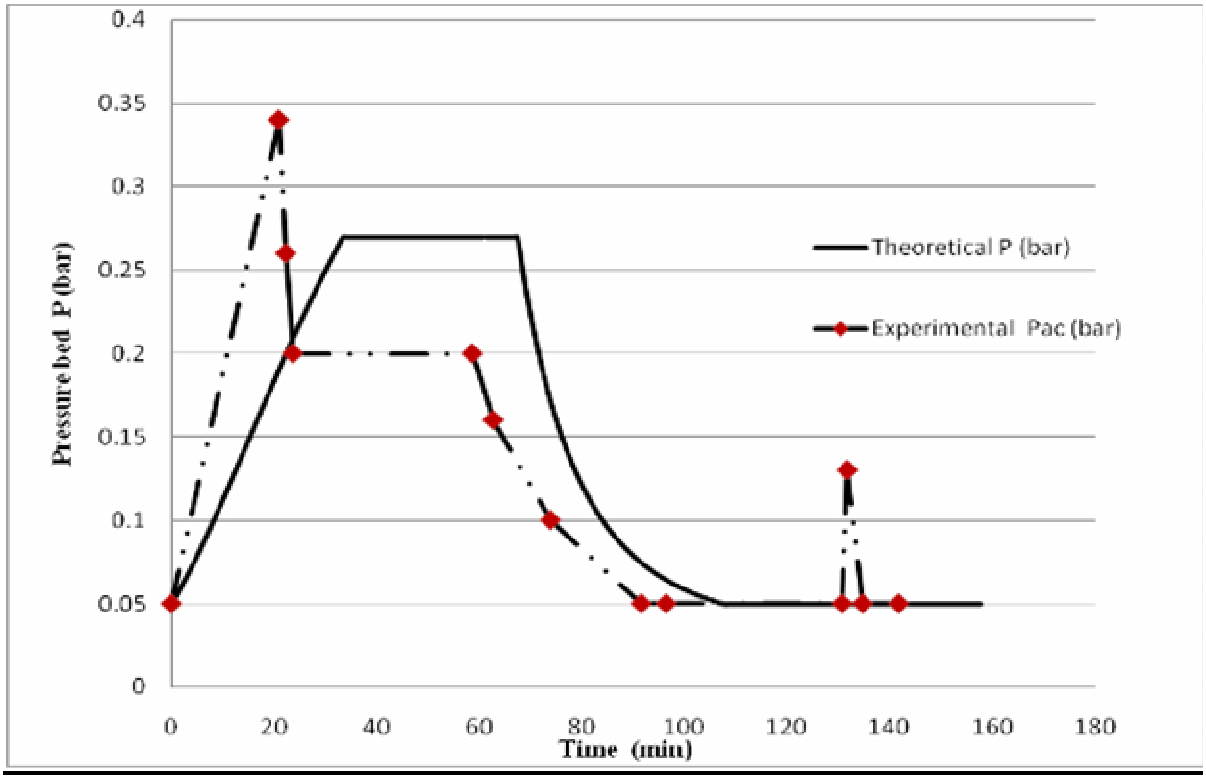


Figure (5) the adsorption bed pressure transient during one complete cycle.

Conclusions:

1- The simulation model results compare satisfactory to the experimental results at least in trend case in general.
2- The source of the error was found to be in first period (pressurization), hence it needs more sophisticated model for this period.
3- The numerical model fail to predict the two overshoots appeared in the pressure diagram. Generating a local error of about 20% at these points. This is due to that the current model does not account for the phenomenon accompanied the valve opening or closing processes.
4- The theoretical model predicts the slop and duration for the periods often than the first period satisfactory.

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