

Prototype's seizing and design of a solar refrigerator based on solid adsorption

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Abstract. Solar refrigerator machines based on solid adsorption present a highly interesting solution to the Industry of Cooling Production. In one hand, they are significantly attractive economy ways because of the abundance of the solar energy resources. In the other hand, they are environment friendly. As a result, these machines could present one of the most competitive solutions to the improvement of this very industry. The aim of this paperwork is to provide an accurate study on how to design, seize and build a prototype of an adsorption solar refrigerator using activated-carbon/ammonia pair: Firstly, we used a static model, which is based on the use of state equations (vapor/liquid) at thermodynamic equilibrium. This model computes the cycled mass and the cycle coefficient of performance (COP_c) for each four characteristic temperatures of the cycle. Secondly, we develop a dynamic simulation program based on conservation equations of energy and mass in the reactor, this program allow the calculation of the temperature, the pressure inside the reactor, the adsorbed mass and the solar coefficient of performance (COP_s). Finally, in the light of our results, we design this prototype, it would consist of the reactor: a solar panel, size 1 m² contain tubes with a diameter of 10cm, an air condenser, and a cold chamber containing an air evaporator.

Nomenclature

C	constant characteristic of the gas.
C _i	specific heat of the phase i (J kg ⁻¹ K ⁻¹)
C _{p,i}	specific thermal capacity of the refrigerant (J kg ⁻¹ K ⁻¹)
C _{psteel}	the specific thermal capacity of steel (J kg ⁻¹ K ⁻¹)
Q _s	total daily solar energy (kWh m ⁻² d ⁻¹)
h _g	specific enthalpy of ammonia at gaseous phase (J kg ⁻¹)
U ₁	overall heat loss coefficient (W m ⁻² K ⁻¹)
G	solar radiation received by the collector (W m ⁻²)
L	vaporization Latent heat of the refrigerant (J kg ⁻¹)
n	number of adsorbate molecular layers
m _a	adsorbed mass (kg (kg-ac) ⁻¹)
P	pressure (bar)
P _s	saturated vapor pressure of the adsorbate liquid (bar)
Q _F	quantity of cold produced in the evaporator (J m ⁻²)
q _m	mass flow rate of ammonia (kg s ⁻¹)
d	inner diameter of the tubes (m)
r	radial coordinate (m)
s	thickness of the tube (m)
T	temperature of adsorbent (K)
T _p	wall temperature (K)
t	time (s)
u _i	specific internal energy of the phase i (J kg ⁻¹)

Greek symbols

ΔH _{ads}	the latent heat of adsorption (J kg ⁻¹)
Δm	cycled mass of the refrigerant (kg m ⁻²)
ε	bed porosity
η ₀	optical solar efficiency of the collector
λ _c	equivalent thermal conductivity coefficient (W m ⁻¹ K ⁻¹)
ρ _i	density of the phase i (kg m ⁻³)
ρ _{steel}	the density of steel (kg m ⁻³)
α	volume fraction of the adsorbed phase

Subscripts

i	gas, solid or adsorbed gas
ads	adsorption
con	condensation, condenser
ev	evaporation, evaporator

Abbreviations

ac	activated carbon
C _{pme}	the apparent capacity of steel for a tube (J K ⁻¹)
COP	coefficient of performance
TDMA	Tri-Diagonal Matrix Algorithm

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1. Introduction

Currently, a variety of cooling technologies is available in today's market. Many of these technologies are not ecological friendly. Most of the refrigerants (CFC, HFC, HCFC...) using in these conventional technologies is not healthy for the environment and without high performance to reduce CO₂ emissions contributing to the greenhouse effect [1]. Another concern regarding this technology is the availability of electricity especially in rural areas.

Therefore, scientists are interested in alternative technologies that are dedicated to sustainability and the ecological future of our planet. Solar refrigerator systems based on solid adsorption present an interesting solution. Many theoretical and experimental studies [2] [3] have demonstrated the feasibility of such machines in the Mediterranean climate.

For this purpose, our objective is the development of a Solar refrigerator (realization in progress) based on solid adsorption principle, where the pair adsorbent/adsorbate is activated carbon/ammonia.

In this paperwork, the first model identifies the influence of the four characteristic temperatures of the cycle (T_1, T_2, T_3, T_4) Figure.1 on the performance and the operating limits of the machine using the cycled mass and the COP_c. The second model find out the geometrical parameter of the reactor (adsorber diameter), which maximize the machine performance, using the COP_s as an optimization criterion. Finally, in the light of the models results' we have designed our machine. This prototype would consist of: - A solar panel size: 1m², it can reach a maximal temperature between 80° and 90° and it contain tubes with the optimum diameter of 10cm. An air condenser consists of a tube with a diameter of 2cm and contains 58 fins; the length of the tube is 30.65 cm. In addition, a cold chamber containing an air evaporator in the form of a cylindrical tube with a diameter of 1cm and a length of the tube equal to 74,72cm.

2. Basic considerations on cooling adsorption

2.1 The physical process:

Adsorption is the process through which a substance, originally present in one phase (gas or liquid), is removed from that phase by accumulation at the interface between that phase and a separate (solid) phase. We distinguish between two types of adsorption, chemical or physical depending on the type of gas molecules binding and energies put in to play. In the case of the chemical adsorption, the fixing of adsorbate molecules on the surface of the adsorbent is carried out by strong chemical bonds this bonding energy is high, in the range of 50 to 400 kJ/mol. Whereas, the physical adsorption, which is the process that results in lower physical bonds with the interaction energies of the order of 20 kJ/mol. This attachment of the molecules of the fluid on the surface of the solid is namely by Van der Waals forces. Furthermore, the physical adsorption is completely reversible, which means, that the adsorbed molecules can be easily desorbed by increasing the temperature. In this paper, physical adsorption is considered to be a transformation of State of

the fluid from a gas phase to adsorbed phase. This transformation occurs at constant temperature and pressure.

2.2 The technical approach:

In an adsorption solar cooling machine, the essential elements of the unit are the solar panel (reactor), the condenser and the evaporator.

The coupling of a mass of adsorbent with sources at different temperatures (the condenser and the evaporator) allows the realization of intermittent cycles where the adsorber acts as a chemical compressor, driving back the adsorbate to the condenser while it is heated and aspirating adsorbate from the evaporator as it is cooled. Each cycle consists of two main phases for the operation: a phase of the adsorbent/adsorbate mixture heating and a cooling of the same mixture.

2.2.1 Heating phase (day):

The adsorber is isolated along the transformation (1→2) Figure 1. Under the effect of heating the pressure and the temperature of the mixture increased, while the total mass of the refrigerant adsorbed remains constant and equal to m_{max} pressurization phase ends as soon as the pressure becomes equal to that prevailing in the condenser P_c (point2). The adsorber is connected with the condenser and the desorption of the refrigerant begins (2→3), the adsorber is then in high pressure and following the Isobar imposed by the condenser. While heating, the temperature of the mixture in the adsorber increases up to the maximum temperature T_3 . This phase is usually called generation because it is the one that makes the adsorber conducive to a new phase of cold production.

2.2.2 Cooling phase (Night):

As opposed to the first phase, the cooling of the adsorbent/adsorbate mixture begins at the point 3 (3→4), where the temperature and pressure decrease until the pressure becomes equal to that prevailing in the evaporator. The total mass of the adsorbed fluid remains constant in this phase and is equal to m_{min} .

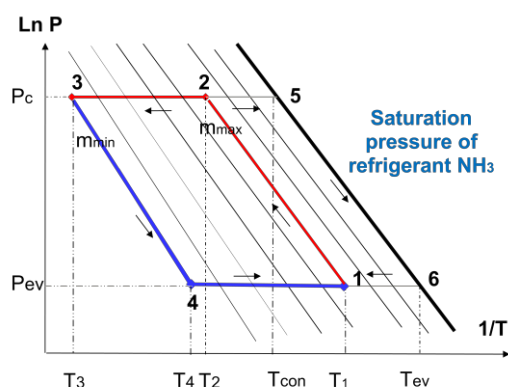


Fig. 1. Thermodynamic-path of the ideal cycle in the Clapeyron diagram.

In point 4, begins the evaporation of the refrigerant by producing the cold in the evaporator. The steam produced

adsorbed in the adsorber, until the temperature of the mixture of adsorbent/adsorbate is minimal T_1 . The system follows the Isobar imposed by the evaporator, and which matches the saturation pressure of the refrigerant at evaporation temperature. At this time, the machine is ready for a new cycle.

3. Quantitative approach by modelling

3.1 The static model:

The model developed simulates the operation of the machine for different temperatures of the cycle, and by calculating the COP_c evaluates the performance and the operating limits of the machine. The model used in this study has been validated by experimentation in the initial work that was done by A. Mimet [4] on a cylindrical reactor consisting of a double stainless steel envelope heated by thermal oil.

3.1.1 Model assumption:

The temperature of the adsorber is assumed uniform and the condensation and evaporation are carried out at constant temperature.

The thermodynamic equilibrium properties of the reactive mixture components (i.e. activated carbon [5], ammonia gas [6] and the adsorbed ammonia [4]) are necessary for the model resolution. As concern the adsorption isotherm, the BET model has been adopted Eq. (1).

3.1.2 The coefficient of performance of the cycle:

Before the determination of COP_c , the model calculates the adsorbed mass m_a by using the BET model:

$$\frac{m_a}{m_0} = \frac{c \cdot x}{(1-x)} \times \frac{1 - (n+1) \cdot x^n + n \cdot x^{n+1}}{1 + (c-1) \cdot x - c \cdot x^{n+1}} \quad (1)$$

where n is the number of the adsorbate molecular layers, $x = P/P_s$ is the relative pressure, P is the pressure at the equilibrium, P_s is the saturated vapor pressure of the adsorbate liquid, C is a constant characteristic of the gas, m_a is the adsorbed mass at relative pressure P/P_s and m_0 is the adsorbed mass necessary to form a monomolecular layer [4]. Therefore the cycled mass Δm is given by:

$$\Delta m = m_{\max} - m_{\min} = m_1 - m_3 \quad (2)$$

The COP_c of the machine is defined as follow:

$$COP_c = \frac{Q_F}{Q_C}$$

Where Q_F is the quantity of cold produced in the evaporator and of Q_C : the amount of heat supplied to the adsorber.

$$Q_F = \Delta m \left[L(T_{ev}) - \int_{T_{ev}}^{T_{con}} C_{P,i} dT \right] \quad (3)$$

With $L(T_{ev})$ is the vaporization latent heat of the refrigerant at temperature T_{ev} and $C_{p,i}$ is the specific thermal capacity of the refrigerant.

$$Q_C = Q_1 + Q_2 + Q_3 + Q_4 \quad (4)$$

Where Q_1 : Heating of the metal part, Q_2 : Activated-carbon heating, Q_3 : Adsorbed ammonia heating, Q_4 : the necessary heat to the desorption corresponding to the mass of desorbed ammonia.

3.1.3 Results of the first simulation:

The Figure2 presents the variation of the COP_c as a function of the maximal temperature T_3 and for two evaporation temperatures 0° and -5° , the minimum temperature T_1 and the condensation temperature T_{con} are set respectively to 20° and 30° . It is noted that the COP_c increase with the evaporation temperature, and reaches an optimum value of 0.209 when the maximum temperature T_3 is equal to 90° .

Based on these results, the optimum temperatures for our prototype are: $T_{ev}=0^\circ$, $T_{con}=30^\circ$, $T_1=20^\circ$ and $T_3=90^\circ$.

3.2 The dynamic model:

This model simulates the transitory behavior of the adsorber. It makes use of the conservation equations of energy and mass of refrigerant in the porous medium, and computes the solar COP_s to finds out the optimum geometrical parameter of the reactor (adsorber diameter).

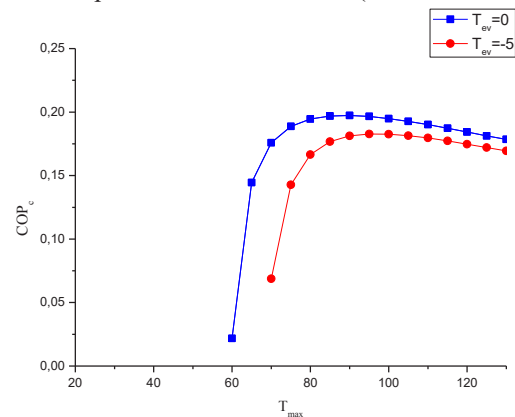


Fig. 2. Variation of the cycle COP_c

In addition, this model of heat and mass transfer used in this paper has been validated by experimentation in the work that was done by A. Mimet [4].

3.2.1 Model assumption:

- For this heat and mass transfer modeling, the following assumptions have been adopted: - The porous medium properties have a cylindrical symmetry.
- During the period time, all phases are in local thermal, mechanical and chemical equilibrium.
- The pressure is uniform.
- The heat transfer is radial and the convection heat transfer due to the radial mass transfer is neglected.

- The conduction heat transfer in the medium can be characterized by an equivalent thermal conductivity coefficient λ_e . [4].

3.2.2 General heat and mass transfer equations in the adsorbent bed:

This equation is obtained by the combination of the energy balance equation (5) and the mass balance equation (6) for a layer of porous medium of width dr and a length of one meter:

$$\begin{aligned} \frac{\partial}{\partial t} [2\pi r dr [(1 - \epsilon)\rho_s u_s + (\epsilon - \alpha)\rho_g u_g + \alpha\rho_a u_a]] \\ + q_m(r, t) \cdot h_g(T(r), P) \\ - q_m(r + dr, t) \cdot h_g(T(r + dr), P) \\ = 2\pi r dr \lambda_e \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right] \end{aligned} \quad (5)$$

Where r is the radial coordinate, ϵ is the bed porosity, ρ_i is the density of the phase i , u_i is the specific internal energy of the phase i , gas, solid or adsorbed, α is the volume fraction of the adsorbed phase, q_m is the mass flow rate of ammonia, h_g is the specific enthalpy of ammonia at gaseous phase, P is the pressure, t is the time, T is the temperature and λ_e is the equivalent thermal conductivity of the porous medium.

The mass conservation of ammonia in the control volume of the bed (a layer with radial coordinate r and thickness dr) is given by:

$$\begin{aligned} \frac{\partial}{\partial t} [2\pi r dr [(\epsilon - \alpha)\rho_g + \alpha\rho_a]] \\ = -q_m(r, t) + q_m(r + dr, t) \\ = \left(\frac{\partial q_m}{\partial r} \right) dr \end{aligned} \quad (6)$$

Combining Eq. (5) and (6) allow the determination of the general heat and mass transfer equation as:

$$\begin{aligned} [(1 - \epsilon)\rho_s C_s + (\epsilon - \alpha)\rho_g C_g + \alpha\rho_a C_a] \frac{\partial T}{\partial t} \\ = \lambda_e \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right] \\ + \frac{\partial}{\partial t} [(\epsilon - \alpha)\rho_g] \frac{P}{\rho_g} + \frac{1}{2\pi r dr} \left(\frac{P}{\rho_a} + \Delta H_{ads} \right) \left(\frac{\partial m_a}{\partial t} \right) \end{aligned} \quad (7)$$

Where C_i is the specific heat of the phase i , (i = solid, gas or adsorbed) and ΔH_{ads} is the latent heat of adsorption. ΔH_{ads} is calculated by applying the Clausius–Clapeyron equation [7]:

$$\Delta H_{ads} = RT^2 \left(\frac{\partial \ln P}{\partial T} \right)_{m_a} \quad (8)$$

With R is the gas constant of the adsorbate, m_a is the adsorbed mass of the refrigerant, it is estimated by using the BET model Eq. (1).

Eq. (7) has been written for all the layers contained in the cylinder giving a nonlinear system of partial derivative equations, which is completed by initial and boundary conditions. This system is solved numerically using the implicit finite difference method. The discretized equations are solved using the Tri-Diagonal Matrix Algorithm

(TDMA) and the nonlinearity of the equations is solved by iterative techniques. A computer program written in FORTRAN and based on this numerical scheme has been developed in order to simulate the behavior of the machine.

3.2.3 The initial and boundary conditions:

To complete the mathematical formulation of the model, the initial and boundary conditions are reported as below:

-Initial condition :

$$T(r, 0) = T_1 = 20^\circ \quad (r=0, \dots, R_{tube})$$

Where $T(r, 0)$ is the temperature of the layer of the porous medium of radius r at the time $t = 0$, T_1 is the minimal temperature and R_{tube} is the radius of the cylinder.

-Boundary conditions:

For ($r=0$):

$$\left(\frac{\partial T}{\partial r} \right)_{r=0} = 0$$

For ($r=R_{tube}$):

The boundary conditions at enclose of the tube represent an energy balance in the corral of the steel tube. The energy received by the outer wall of the tube coming from the sun is equal to the sum of the energy losses, heat capacity of steel and thermal conduction in porous medium. It is reported as follows:

$$\eta_0(d + 2s)G = U_l \frac{\pi(d+2s)}{2} (T_p - T_1) + C_{pme} \frac{\partial T_p}{\partial t} + 2\pi r \lambda_e \left(\frac{\partial T}{\partial r} \right) \quad (9)$$

With $C_{pme} = C_{p_{steel}} \cdot \rho_{steel} \cdot V_{steel}$

where $G(t)$ is the global solar irradiance, U_l is the loss coefficient, C_{pme} is the apparent capacity of steel for a tube, T_p is the wall temperature, d is the diameter of the tubes and s is the thickness of the tube, η_0 is the optical solar efficiency of the collector, $C_{p_{steel}}$ is the specific thermal capacity of steel, ρ_{steel} is the density of steel, V_{steel} is the volume of steel contained between two cylinders of diameters respectively d and $d + 2s$.

3.2.4 The solar coefficient of performance:

This parameter is defined by the following formula:

$$COP_s = \frac{Q_F}{Q_s}$$

Where Q_F is the cooling production at the evaporator during adsorption cycle Eq. (3) and Q_s is the total daily solar energy.

$$Q_s = \int_{sunrise}^{sunset} G(t) dt \quad (10)$$

3.2.5 Results of the second simulation:

The figure3 shows the variation of the COP_s as a function of the diameter of the tube, it can be seen that the COP_s increase to an maximum value of 0,109, it correspond to an optimum diameter of the cylinder equal to 10cm.

On the light of this result, it has been decided to choose this value of 10cm to build our prototype.

The next section will focus on the design of the reactor and the sizing of the others important parts of the machine: the condenser and the evaporator.

4. Application to a prototype design

In the reactor, if the objective is to reach the temperature about 80 to 90 °C, a normal solar collector with simple glazing is sufficient.

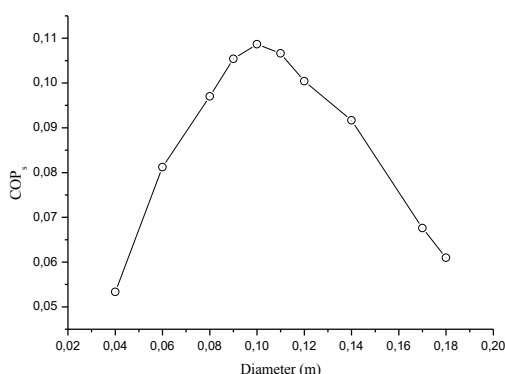


Fig. 3. The variation of the solar COP_s

The prototype is composed with the following:

- 1: A solar panel normal size 1mx1m, containing a set of cylindrical steel tubes of diameter d = 10cm.
- 2: An air condenser.
- 3: A chamber containing a cold air evaporator.
- 4: Pipe connection

4.1 Solar panel (reactor-adsorber):

The solar panel figure4 sizes 1mx1m is composed of:

The box:

A rectangular (aluminum, wood ...) for storing heat with in the sensor. The insulation material used is glass wool or Polyurethane.

Glass:

To create greenhouses affect inside the box.
 Required physical properties: excellent transparency, reflection coefficient closes to zero, low absorption coefficient and high thermal insulation material.

The heat adsorber:

Is made of stainless steel cylindrical tubes of length 1m and 10cm in diameter, filled with activated carbon, closed at the ends by a fine mesh, to retain the activated carbon within the cylinders. The outer surface of the cylinders is tinted in paint selectively (color Black); the latter is

connected to other parts of the machine by a welded connection in the tube end.

Our solar panel uses 9 tubes; the total mass of activated carbon contained in the tubes is 35.3 kg.

The mass of ammonia cycled for 1kg of activated carbon in the system is given experimentally by Δm=0,078 Kg/Kg-ac [4].And the calculated cycled mass of ammonia in the machine for 35,3 kg of activated carbon is Δm=2,75 Kg.

4.2 Condenser:

Based on the first simulation results for the condensation, the temperature and the saturated vapor pressure of the condenser chosen are T_{con}=30° and P_c=11.66 bar.

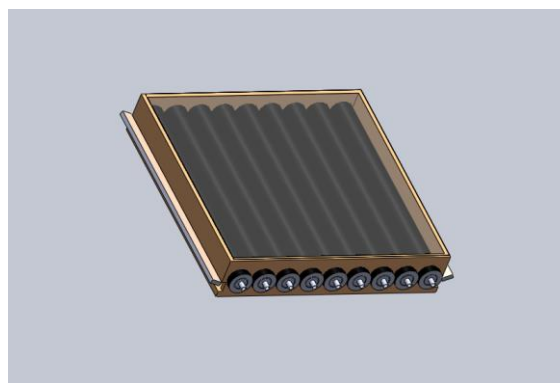


Fig. 4. Perspective view of the reactor

The proposed air condenser Figure5 consists of a tube of inner diameter d_i = 2cm and outer diameter d_e = 2.6cm and contains 58 fins spaced by 0.5cm. The fins are square shapes of 6.5cm² and 0.2cm thickness. The calculated length of the tube is 30.65cm [8].

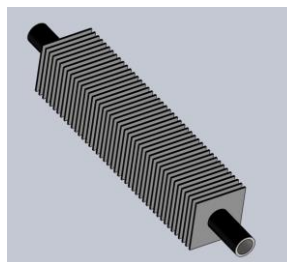


Fig. 5. Condenser

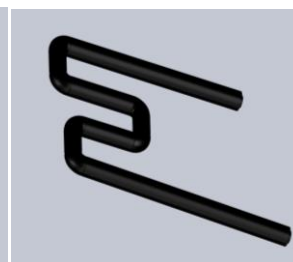


Fig. 6. Evaporator

4.3 Evaporator:

The temperature and the saturated vapor pressure of the evaporator selected are: T_{ev}=0° and P_e= 4.29 bar.

The evaporator Figure6 is in the form of a cylindrical tube of inner diameter d_i = 1cm and outer diameter d_e = 1.6cm. Moreover, the Calculated length of the tube is 74,72cm [9].

The evaporator is placed inside a cold room type of a parallelepiped size 40x40x30cm³.

4.4 The Prototype:

The assembly of all parts consist the prototype; it is shown in figure7:

1-Glass; 2-Adsorber; 3-Box; 4-Collector; 5-Support; 6-Cold room; 7-Evaporator; 8-Condenser; 9-Cooling components.

5. Conclusion

This paperwork presented the essential steps for design and dimensioning a prototype of solar adsorption refrigerator. Two models simulating the real behavior of our adsorption machine has been developed. The first model established has been used to identify the four characteristics temperatures of the cycle machine corresponding to an optimal cycle coefficient of performance. The second numerical simulation program computes the optimum diameter of the cylindrical tube of the reactor matching with the solar coefficient of performance. The results show that the cycle COP_c reaches 0.209 with operates temperatures: $T_{ev}=0^\circ$, $T_{con}=30^\circ$, $T_1=20^\circ$ and $T_3=90^\circ$. Then, the solar COP_s increase to a maximum value of 0,109, which correspond to the diameter of the cylindrical reactor equal to 10cm.

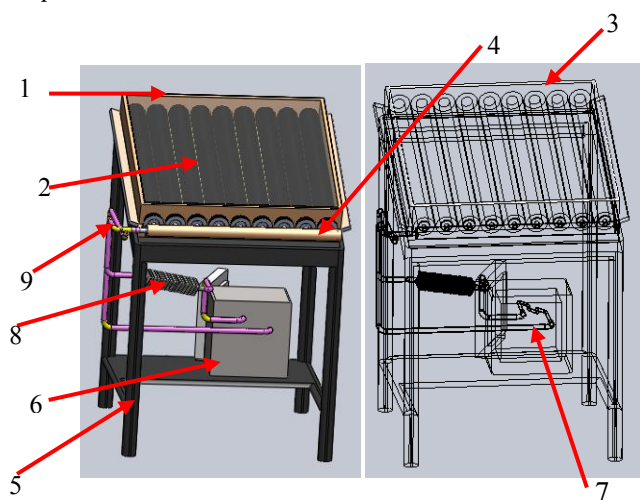


Fig. 7. Prototype of the solar adsorption Refrigeration machine

Finally, these results helped to determine the characteristics of the others elements of the machine namely the condenser which is a tube with a diameter of 2cm contains 58fins; the length of the tube is 30.65cm, and the evaporator consists of a tube with a diameter of 1cm and a length equal to 74,72cm. The prototype presented in this paperwork is simple and can demonstrate the usefulness of this technology. In fact, cost-benefit analyses of the prototype have to be conduct to prove the commercial feasibility.

Key words

Solar Refrigerator, Adsorption, Solar Energy, Heat transfer, dimensioning, Simulation.

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