Enhancing Adsorption Ice Maker Productivity using Beds of Aluminium Foam Packed with Activated Carbon

M B Elsheniti¹,²*, H Al-Ansary¹, J Orfi¹, A El-Leathy¹, M S Eissa² and O Elsamni²

¹Mechanical Engineering Department, College of Engineering, King Saud University, Riyadh 11451, Saudi Arabia
²Mechanical Engineering Department, Faculty of Engineering, Alexandria University, Alexandria 21544, Egypt.
*mbadawy.c@ksu.edu.sa

Abstract. The adsorption refrigeration system driven by solar energy is a promising sustainable solution to tackle the growing demand for cooling and meet environmental regulations as well. In addition, the increase in refrigeration needs is in phase with the increase in the abundant solar energy in the Middle East. This study aims at numerically investigating the utilization of a high adsorption performance material namely Maxsorb III, a type of activated carbon, packed in an advanced aluminium foam bed to produce ice from two-bed adsorption system. A detailed 2-D axisymmetric transient model considering mass, momentum, and energy balance equations coupled with isotherms and kinetic models in the adsorbent domain to describe the adsorption phenomena was developed and used for the simulations. Results of a typical packed bed using finned tube configuration were used as a base model to compare the performance enhancement. The aluminium foam-based system performance outperformed the base model in producing the ice by 16.8% at a high cycle time of 1200 s, regeneration temperature of 90°C, and foam thickness of 5 mm. Furthermore, the coefficient of performance and specific cooling power increased by 26.7% and 27.9%, respectively, driven by the considerable enhancement in the bed mass and heat transfer due to the use of metal foam.

1. Introduction
Solar radiation is the largest energy source and utilizing that radiation to produce cooling power will reduce CO2 emissions. Saudi Arabia's geographical location is strategic as it is sitting in the center of the so-called Sun Belt, and it has a clear sky for a long period of the year-round, which has made it a perfect possible site for employing the promising solar-cooling-technologies. In Saudi Arabia, 70% of building electricity consumption is attributed to refrigeration systems which are mainly use vapor compression systems [1]. However, these devices are highly electricity-consuming ones and use environmentally harmful refrigerants (HFCs, HCFCs). The adsorption refrigeration systems (ARSs) are considered environmentally friendly energy-saving systems where they can use low-quality thermal energy provided by solar or waste heat sources [2, 3]. Furthermore, they can utilize natural refrigerants such as water and ethanol and require low maintenance efforts due to fewer moving parts [4]. Several approaches have been studied to examine solar adsorption cooling systems. Attalla et al. [5] presented an experimental study of solar adsorption ice maker system using working pair of activated carbon and methanol. The study calculated the coefficient of performance (COP) for the tested pair to be about 0.146. Allouhi et al. [6] investigated theoretically using 1-D mathematical model for activated carbon/methanol small adsorption refrigeration unit driven by solar energy and employed for vaccine and medicine preservation. The specific cooling power and solar COP were 3.18 W/kg and 0.132, respectively. Ambarita and Kawai [7] evaluated a solar-powered adsorption refrigeration cycle with a
bed filled by different mixing ratios between activated alumina and activated carbon as adsorbents. The experimental study concluded that pure activated carbon and methanol pair achieved better COP than the mixed ratios of both adsorbents. In addition, the overall COP for all experiments was low and the main reasons was the high heat losses from the solar collector and the low heat transfer coefficient from the adsorber plate to the generator. Chekirou et al. [8] evaluated numerically the performance of activated carbon AC-35/methanol configured as nine tubular adsorbers integrated into a flat plate solar collector of 1 m² surface area. The best performance of the system attained a thermal COP of 0.424 and a solar COP of 0.143. Rupa et al. [9] studied thermodynamically the employment of activated carbon-graphene composite/ethanol pairs in adsorption systems. The adsorption characteristics were determined experimentally for different composites, samples and two kinetic model were examined. The best volumetric cooling effect reached 199.82 MJ/m³ at desorption, condensation, evaporation temperatures of 100, 30, and 10 °C, respectively. El-Sharkawy et al. [10] investigated theoretically solar silica gel/water adsorption chiller working under climate conditions of the Middle East region using actual solar data of Cairo, Aswan and Jeddah cities. The best cyclic cooling capacity reached was about 15 kW. Ghazy et al. [11] compared the use of refrigerant HFC-404a with many refrigerants that can be used with a type of activated carbon namely Maxsorb III. The maximum adsorption capacity was 2.2 kg R-404a/kg activated carbon which was the best among others. The optimum COP and SCP were 0.23 and 275 W/kg, respectively. However, HFC-134a and HFC-404a are not environmentally friendly refrigerant as they have high GWP. Regarding the adsorption bed heat and mass transfer, fins’ parameters in the packed bed adsorber have been found the most influential parameters on the adsorption system performance. Elsheniti et al. [12] developed a detailed numerical model for a two-bed adsorption chiller to examine both operational and geometrical parameters. The study highlighted the importance of adapting the physical bed parameters to match each application, particularly the fin numbers. The metal-foam-based adsorbent beds have been proposed as another way to enhance heat diffusion by conduction in the adsorbent domain via the well-structural metal connectivity. This increases the effective thermal conductivity while providing high metal-to-adsorbent contact area and vapor permeability by foam cavities, and keeping good mechanical strength at higher foam thicknesses [13, 14]. The present study introduces for the first-time comparative investigations on using an aluminum foam bed packed with Maxsorb III and the classical finned tube packed bed. A 2-D transient pressure distribution model is developed using COMSOL Multiphysics to simulate the adsorption beds, while the time variation of the condenser and evaporator pressures are considered at the valve openings. The study will help to address the effect of critical parameters such as the cycle time on the system-level performance for both approaches to reveal the enhancement conditions.

2. System description
The conventional two-bed adsorption system configuration using activated carbon/ethanol pair is adopted to produce ice by cooling ethylene glycol under −2 °C as shown in Figure 1. The activated carbon is considered in granular form packed in the aluminum foam structure. The configuration of the classical finned-tube adsorption beds is also used as a base for performance comparisons. Heating/cooling water is switched between the two beds during the desorption/adsorption mode while each bed undergoes four processes (preheating, desorption, precooling and adsorption) during a complete cycle [15]. Switching valves are used to connect the evaporator and condenser with the sorption beds as shown in Figure 1. Ice is produced by feeding the ethylene glycol that continuously leaves the evaporator to an ice production tank.

3. Mathematical modelling
Unsteady fully coupled numerical model is employed for the current study to simulate the performance of the two-bed adsorption ice production system performance. Figure 2 shows 2-D axisymmetric design model containing the four domains: thermal fluid, plain copper tubes, Aluminum foam or fins packed with activated carbon, and vacuum space. The developed model embraces the volume averaged approach for solving the conservation in the adsorbent domain. The current model is taking into consideration the non-ideal condenser and evaporator pressures by applying their energy balances at the valve opening as boundary conditions. Emphasizing that the capability of the model to detect the local
changes of all variables as pressure, temperature, and amount of adsorbate within the investigated adsorbent tube by solving the governing equations simultaneously. Each adsorbent bed comprises of a total number of identical adsorbent tube used in the simulations which reflects the overall performance of the beds connected to the evaporator or condenser depending on the mode of operation.

3.1. Governing equations

(a) Thermal fluid

Using RANS equations to describe the turbulent flow scheme for incompressible thermal fluid flowing inside the copper tubes lead to mass and momentum equations as follows:

\[
\rho_f \nabla \cdot \mathbf{u}_f = 0 \tag{1}
\]

\[
\rho_f \frac{\partial \mathbf{u}_f}{\partial t} + \rho_f \mathbf{u}_f \cdot (\nabla \mathbf{u}_f) = -\nabla p + \nabla \cdot \left[ \left( \mu + \mu_t \right) (\nabla \mathbf{u}_f + (\nabla \mathbf{u}_f)^T) - \frac{2}{3} \rho_f \kappa \mathbf{I} \right] \tag{2}
\]

For the energy balance equation, the effect of turbulent flow on enhancing the thermal conductivity \( k_T \) is being considered [12] and it can be written as:

\[
\rho_f C_{pf} \frac{\partial T_f}{\partial t} + \nabla \cdot \left( \rho_f C_{pf} \mathbf{u}_f T_f \right) - \nabla \cdot (k_T \nabla T_f) = 0 \tag{3}
\]

(b) Thermal tube and fins

The energy conservation equation for the thermal tube and fins can be written as:

\[
\rho_{met} C_{p,met} \frac{\partial T_{met}}{\partial t} - \nabla \cdot (k_{met} \nabla T_{met}) = 0 \tag{4}
\]

(c) Adsorbent domain

Two forms of mass balance equations [16, 17]:

\[
\varepsilon_t \frac{\partial p_{\text{ads}}}{\partial t} + (1 - \varepsilon_t) \rho_s \frac{\partial X}{\partial t} + \nabla \cdot (\rho_v \mathbf{u}) = 0 \quad \text{Finned tube bed} \tag{5}
\]
\[
\varepsilon_b \frac{\partial \rho_v}{\partial t} + (1 - \varepsilon_t) \rho_s \frac{\partial X}{\partial t} + \nabla \cdot (\rho_v u) = 0
\]  
Aluminum foam bed (6)

Where \(\varepsilon_t\) and \(\varepsilon_b\) are the total and the bed porosities, \(X\) is the instantaneous adsorbate, \(\rho_v\) is the density of ethanol vapor, \(\rho_s\) is the density of the solid adsorbent and \(u\) is the averaged velocity.

Two forms of momentum equations [12]:

Finned tube bed
\[
\frac{\rho_v}{\varepsilon_t} \left[ \frac{\partial u}{\partial t} + \frac{1}{\varepsilon_t} u \cdot (\nabla u) \right] = \nabla \cdot \left[ -p I + \frac{\mu}{\varepsilon_t} (\nabla u + (\nabla u)^T) - \frac{2}{3} \frac{\mu}{\varepsilon_t} (\nabla u) I \right] - \left[ \frac{\mu}{k_p} + \frac{q_m}{\varepsilon_t} \right] u
\]  
Aluminum foam bed (7)

\[
\frac{\rho_v}{\varepsilon_b} \left[ \frac{\partial u}{\partial t} + \frac{1}{\varepsilon_b} u \cdot (\nabla u) \right] = \nabla \cdot \left[ -p I + \frac{\mu}{\varepsilon_b} (\nabla u + (\nabla u)^T) - \frac{2}{3} \frac{\mu}{\varepsilon_b} (\nabla u) I \right] - \left[ \frac{\mu}{k_p} + \frac{q_m}{\varepsilon_b} \right] u
\]  
Aluminum foam bed (8)

\(Q_m\) is the source term used in the mass balance equations:
\[
Q_m = -(1 - \varepsilon_t) \rho_s \frac{\partial X}{\partial t}
\]  
(9)

The energy equation for the adsorbent domain can be expressed as:
\[
\frac{\partial}{\partial t} (\rho C_p T) + \nabla \cdot (\rho_v u C_{p,v} T) = \nabla \cdot (\kappa_{comp} \nabla T) + (1 - \varepsilon_t) \rho_s H_{ads} \frac{\partial X}{\partial t}
\]  
While \(\rho C_p = (1 - \varepsilon_t) \rho_s (C_{p,s} + X C_{p,a}) + \varepsilon_t \rho_v C_{p,v}\) Finned tube bed (10)

\(\rho C_p = (1 - \varepsilon_t) \rho_s (C_{p,s} + X C_{p,a}) + \varepsilon_b \rho_v C_{p,v} + \rho_f_0 C_f\) Aluminum foam bed (11)

\(H_{ads}\) is the isosteric heat of adsorption and \(k_{comp}\) is the composite thermal conductivity of the porous domain.

(d) Vacuum chamber

Applying the conservation equations in the vacuum space:

The mass balance equation:
\[
\frac{\partial \rho_v}{\partial t} + \nabla \cdot (\rho_v u) = 0
\]  
(13)

The momentum balance equations:
\[
\rho_v \frac{\partial u}{\partial t} + \rho_v u \cdot \nabla (u) = -p I + \nabla \cdot \left[ \frac{\mu}{3} (\nabla u + (\nabla u)^T) - \frac{\mu}{3} (\nabla u) I \right]
\]  
(14)

The energy balance equation:
\[
\frac{\partial (\rho_v C_{p,v} T_v)}{\partial t} + \nabla \cdot \left[ \rho_v u C_{p,v} T_v \cdot \nabla (k_v \nabla T_v) \right] = 0
\]  
(15)

(e) Evaporator model

The evaporator heat balance equation is used to determine the temperature of the evaporation.
\[
\left[ M_{eva,rl} C_{p,eva,rl} + M_{eva,met} C_{p,eva,met} \right] \frac{dT_{eva}}{dt} = \dot{m}_{eva} C_{p,Eth,Gly} \dot{e}_{eva} \left( T_{Eth,Gly,i} - T_{eva} \right)
\]
\[- (1 - \alpha) \dot{m}_{eva} N_{tube,adsorber} \left[ L H_{eva} - C_{p,rl} (T_{cond} - T_{eva}) \right]
\]
\[\varepsilon_{eva} = 1 - \exp \left( \frac{-u_{eva}}{m_{eva} C_{p,Eth,Gly}} \right)\]

\(N_{tube,adsorber}\) is the total number of tubes in the adsorvent bed.

(f) Condenser model

The heat balance equation for the condenser is used to determine the ethanol condensing temperature.
\[
\left[ M_{cond,rl} C_{p,cond,rl} + M_{cond,met} C_{p,cond,met} \right] \frac{dT_{cond}}{dt} = -\dot{m}_{cw} C_{p,cw} \varepsilon_{cond} \left( T_{cond} - T_{cw,1} \right)
\]
\[- (1 - \beta) \dot{m}_{e,cond} N_{tube,adsorber} \left[ L H_{cond} + C_{p,cv} (T_{V,out} - T_{cond}) \right]
\]
\[\varepsilon_{cond} = 1 - \exp \left( \frac{-u_{e,cond}}{m_{cw} C_{p,cw}} \right)\]

Where \(\alpha\) and \(\beta\) are flags used to simulate the current states (on/off) of connecting valves that connect the sorption beds with the evaporator and condenser.

(g) Adsorption isotherms and kinetics

The linear-driving-force model is suitable to represent the resistance to the intraparticle mass transfer [9, 18]:
\[ \frac{dx}{dt} = K_{LDF} (X_{eq} - X) \]  

(20)

where \( K_{LDF} \) is the mass transfer coefficient and can be expressed as: In case of activated carbon in granular

\[ K_{LDF} = A \exp \left( \frac{-E_a}{R_u T} \right) \]  

(21)

\( A \) is pre-exponential factor, \( R_u \) is the universal gas constant and \( E_a \) is the activation energy.

For the equilibrium adsorption uptake of AC/ethanol (\( X_{eq} \)) [9, 18].

\[ X_{eq} = X_{max} \exp \left[ -\left( \frac{RT}{E} \ln \left( \frac{P_s}{p} \right) \right)^{n} \right] \]  

(22)

Where, \( X_{max} \) is the maximum adsorption uptake, \( R \) is the gas constant, \( T \) is the adsorbent temperature, \( E \) is the adsorption characteristics parameter, \( n \) is the heterogeneity parameter, \( P \) is local pressure and \( P_s \) is the saturation pressure that can be calculated from Antoine’s equation for ethanol:

\[ P_s = 0.1333 \times 10^{A - \frac{B}{T + C}} \]  

(23)

Where, \( A, B \) and \( C \) are equation constants, \( T \) is the temperature in °C and \( P_s \) is the pressure in kPa.

### 3.2. Performance indicators

The system performance is investigated by evaluating the coefficient of performance (COP), specific cooling power (SCP) and daily ice production (DIP) as follows:

\[ Q_{eva} = \frac{1}{t_{cycle}} \int_{0}^{t_{cycle}} \dot{m}_{Eth,Gly} C_{Eth,Gly} (T_{Eth,Gly,i} - T_{Eth,Gly,out}) dt \]  

(24)

\[ Q_{heat} = \frac{1}{t_{cycle}} \int_{0}^{t_{cycle}} \dot{m}_{hw} C_{hw} (T_{hw,i} - T_{hw,out}) dt \]  

(25)

\[ COP = \frac{Q_{eva}}{Q_{heat}} \]  

(26)

\[ SCP = \frac{Q_{eva}}{M_s} \]  

(27)

\[ DIP = \sum_{i=1}^{n} \int_{0}^{t_{cycle}} \frac{\dot{m}_{Eth,Gly} C_{Eth,Gly} (T_{Eth,Gly,i} - T_{Eth,Gly,out})}{c_p(T_{w,in} - T_{freezing}) + \dot{h}_{fg} + \dot{c} p_{ice} (T_{freezing} - T_{ice,out})} dt \]  

(28)

Where \( M_s \) is adsorbent mass in the two sorption beds and \( n \) is the total cycles in 8 hours per day as the system proposed to be driven by solar energy.

All thermophysical properties, operational parameters and constant parameters for isotherms used for simulations are furnished in Table 1, while the parameters for the evaporator and condenser simulations are similar to that have been used in reference [12]. Figure 2 illustrates the dimensions of the simulated bed tube.

### 3.3. Initial and boundary conditions

Before any analysis is being investigated, reaching the condition of steady state is being verified after six complete cycles to vanish the effect of the initial conditions on the performance indicators. The boundary conditions are set as described by Elsheniti et al [12].

**Table 1.** The main parameters used in the simulations.

| Parameter                  | Value | SI Unit       |
|----------------------------|-------|---------------|
| Activated carbon density (\( \rho_s \)) | 2200  | (kg.m\(^{-3}\)) |
| Activated carbon specific heat (\( C_{p,s} \)) | 1375  | (J.kg\(^{-1}\).K\(^{-1}\)) |
| Aluminium foam density (\( \rho_{fo} \)) | 270   | (kg.m\(^{-3}\)) |
Table 2. Parameters of Alumimum foam and Activated carbon associated with the adsorption process.

| Parameter                                      | Value       |
|------------------------------------------------|-------------|
| Aluminium foam specific heat ($C_{p,fo}$)     | 895 [J kg$^{-1}$ K$^{-1}$] |
| Bulk density ($\rho_{bulk}$)                   | 275 [kg m$^{-3}$]   |
| Bed permeability ($K_p$) for the foamed structure | $3 \times 10^{-12}$ [m$^2$] |
| Bed permeability ($K_p$) for the packed form   | $6.296 \times 10^{-12}$ [m$^2$] |
| Particle porosity ($\varepsilon_p$)            | 0.789       |
| Bed porosity ($\varepsilon_b$) for the foamed structure | 0.3416 |
| Bed porosity ($\varepsilon_b$) for the packed form | 0.4075 |
| Total porosity ($\varepsilon_t$) for the foamed structure | 0.861 |
| Total porosity ($\varepsilon_t$) for the packed form | 0.875 |
| Foam porosity ($\varepsilon_fo$)              | 0.9         |
| Thermal conductivity of aluminium foam ($K_{fo}$) | 6 [W m$^{-1}$ K$^{-1}$] |
| Thermal conductivity of Activated carbon ($K_{ac}$) | 0.2 [W m$^{-1}$ K$^{-1}$] |
| Pre-exponential factor ($A$)                   | 132.89 [s$^{-1}$] |
| Activation energy ($E_a$)                     | 22.97 [kJ mol$^{-1}$] |
| Isosteric heat of adsorption at ($H_{ads}$)    | 1002 [kJ kg$^{-1}$] |
| Maximum uptake ($X_{max}$)                    | 1.2 [kg kg$^{-1}$] |
| Adsorption characteristics parameter ($\varepsilon$) | 139.5 [kJ kg$^{-1}$] |
| Heterogeneity parameter ($n$)                  | 1.8         |

3.4. Validation

The numerical finned tube based model using COMSOL Multiphysics was validated with the experimental work in references [12, 19]. The numerical results showed good agreement with the experimental results. Additionally, the numerical model system of equations was validated in reference [20] with copper foamed beds coated with advanced adsorbents.

4. Results and discussion

The base model is a classical finned two-bed adsorption system packed with the Maxsorb grains. The main idea is to set the basic configurations of the adsorber tubes and change the way that the adsorbent material is packed in the sorption reactor to study the net effect of using a bed constructed from aluminium foam applied on the outer surface of the heat transfer tube. The fin height is set to 5 mm which is also considered the foam thickness. The cycle time is ranged from 400 s to 1200 s with equal desorption and adsorption times in this investigation. For each form, the mass of the adsorbent is different as the foam structure occupies more space than the straight fins. The total amount of adsorbent mass used in the two beds is calculated to be 4 kg for the finned tube beds and 3.8 kg for the aluminium foam beds while considering the same dimensions for both configurations.

The absolute values of the performance indicators of the finned tube base model are illustrated in Table 2. At a smaller cycle time of 400s, the best SCP and DIP are attained with 435 W·kg$\text{ads}^{-1}$ and 118 kgIce·day$^{-1}$, respectively, with 1 mm fin spacing. Similar results were achieved at 2 mm fin spacing for the SCP and DIP. However, the COP enhanced considerably at a higher cycle time of 1200 s to about 19 % of that at 400 s for fin spacing of 2 mm, due to the reduction in the thermal mass of the bed at lower fin numbers and longer switching time.

The effect of cycle time on the effective uptake for both finned tube-based bed and aluminium foam-based bed is shown in Figure 3. At a smaller cycle time, the difference between the minimum and maximum uptakes is higher for finned tube-based bed compared to the aluminium foam-based bed. This difference is eliminated at higher cycle time which can reflect the enhancement in the latter system performance at higher cycle time.
Table 2 - The performance indicators for the base model at 1 mm fin spacing.

| Cycle time (s) | COP   | SCP (W. kg⁻¹) | DIP (kg. day⁻¹) | COP   | SCP (W. kg⁻¹) | DIP (kg. day⁻¹) |
|----------------|-------|---------------|-----------------|-------|---------------|-----------------|
| 400            | 0.213 | 435           | 118             | 0.237 | 422           | 114             |
| 600            | 0.277 | 380           | 103             | 0.293 | 385           | 104             |
| 800            | 0.278 | 326           | 88.3            | 0.318 | 340           | 92.3            |
| 1200           | 0.289 | 250           | 67.7            | 0.344 | 269           | 72.4            |

a) Finned tube bed (1mm fin spacing)  
b) Aluminium foam bed

![Graph1](image1)
![Graph2](image2)

**Figure 3.** The P-T-X relation at different cycle times for the finned tube and aluminium foam beds.

(a) 1 mm fin spacing  
b) 2 mm fin spacing  

![Graph3](image3)
![Graph4](image4)
Figure 4. Effect of the cycle time on the percentage change of the COP, SCP, and DIP due to use aluminium foam bed.

As shown in Figure 4-a.1 the COP of the ice maker adsorption system of foamed form at 5 mm thickness outperformed the finned tube model with 1 mm fin spacing at all investigated cycle times and reached the maximum increase by 26.7% at 1200 s cycle time. However, the COP of the system that uses 2 mm fin spacing is showing better performance compared to the aluminium foam-based system as revealed by Figure 4-b.1 at cycle time of 400s. The specific cooling power of the aluminium foam-based system is higher than that of the finned tube-based system at cycle times of 600, 800, and 1200 s at both fin spacings of 1mm and 2 mm. The best SCP attained 27.9% and 19.7% higher than that produced by finned tube-based system at 1200 s for 1mm and 2mm fin spacing, respectively, as indicated in Figure 4-a.2 and b.2. This is attributed to the enhancement in the effective uptake while using a less amount of adsorbent material in the foamed beds. The daily ice production from the aluminium foam-based adsorption system increases with increasing the cycle time compared with the finned tube-based adsorption system. At a higher cycle time of 1200 s, the DIP of the foam-based system increased by 16.8% and 9.3% compared to finned tube-based systems with 1 mm and 2 mm fin spacing respectively.

5. Conclusions
The performance of adsorption ice maker using the emerging aluminium foam-based bed packed with activated carbon type Maxsorb III is numerically investigated in this study on a system level. The
obtained results have been compared to those of the finned tube-based adsorption system. It can be concluded that using aluminium foam structure design in the adsorbent bed for an adsorption ice maker associated with a higher cycle time of 1200 s can enhance all the performance indicators and increase the potential use of solar energy to drive the system.

Acknowledgments
This Project was funded by the National Plan for Science, Technology and Innovation (MAARIFAH), King Abdulaziz City for Science and Technology, Kingdom of Saudi Arabia, Award Number (II-ENE1845-02)

References
[1] K. A. P. S. a. R. C. (KAPSARC), "The Future of Cooling in Saudi Arabia: Technology, Market and Policy Options," 2020.
[2] M. B. Elsheniti, A. Rezk, M. Shaaban, M. Roshdy, Y. M. Nagib, O. A. Elsamni, et al., "Performance of a solar adsorption cooling and desalination system using aluminum fumarate and silica gel," Applied Thermal Engineering, vol. 194, p. 117116, 2021.
[3] E. S. Rashed, O. Ahmed Elsamni, H. A. Elkaranshawy, and M. B. Elsheniti, "Design and Performance Evaluation of Large Field of Flat Plate Solar Collectors Network for Industrial Application," pp. 1-7, 2021.
[4] M. B. Elsheniti, A. T. Abd El-Hamid, O. A. El-Samni, S. M. Elsherbiny, and E. Elsayed, "Experimental evaluation of a solar two-bed lab-scale adsorption cooling system," Alexandria Engineering Journal, vol. 60, pp. 2747-2757, 2021.
[5] M. Attalla, S. Sadek, M. Salem Ahmed, I. M. Shafie, and M. Hassan, "Experimental study of solar powered ice maker using adsorption pair of activated carbon and methanol," Applied Thermal Engineering, vol. 141, pp. 877-886, 2018.
[6] A. Allouhi, T. Kousksou, A. Jamil, Y. Agrouaz, T. Bouhal, R. Saidur, et al., "Performance evaluation of solar adsorption cooling systems for vaccine preservation in Sub-Saharan Africa," Applied Energy, vol. 170, pp. 232-241, 2016.
[7] H. Ambarita and H. Kawai, "Experimental study on solar-powered adsorption refrigeration cycle with activated alumina and activated carbon as adsorbent," Case Studies in Thermal Engineering, vol. 7, pp. 36-46, 2016.
[8] W. Chekiriou, A. Chikouche, N. Boukheit, A. Karaali, and S. Phalippou, "Dynamic modelling and simulation of the adsorber of a solid adsorption pair of activated carbon and methanol," International Journal of Refrigeration, vol. 39, pp. 137-151, 2014.
[9] M. J. Rupa, A. Pal, and B. B. Saha, "Activated carbon-graphene nanoplatelets based green cooling system: Adsorption kinetics, heat of adsorption, and thermodynamic performance," Energy, vol. 193, p. 116774, 2020.
[10] I. I. El-Sharkawy, H. AbdelMeguid, and B. B. Saha, "Potential application of solar powered adsorption cooling systems in the Middle East," Applied Energy, vol. 126, pp. 235-245, 2014.
[11] M. Ghazy, A. A. Askalany, and B. B. Saha, "Maxsorb III/HFC404a as an adsorption pair for renewable energy driven systems," International Journal of Refrigeration, vol. 120, pp. 12-21, 2020.
[12] M. B. Elsheniti, M. A. Hassab, and A.-E. Attia, "Examination of effects of operating and geometric parameters on the performance of a two-bed adsorption chiller," Applied Thermal Engineering, vol. 146, pp. 674-687, 2019.
[13] L. Schnabel, M. Tattlier, F. Schmidt, and A. Erdem-Şenatalar, "Adsorption kinetics of zeolite coatings directly crystallized on metal supports for heat pump applications (adsorption kinetics of zeolite coatings)," Applied Thermal Engineering, vol. 30, pp. 1409-1416, 2010.
[14] L. Calabrese, L. Bonaccorsi, P. Bruzzaniti, A. Frazzica, A. Freni, and E. Proverbio, "Adsorption performance and thermodynamic analysis of SAPO-34 silicone composite foams for adsorption heat pump applications," Materials for Renewable and Sustainable Energy, vol. 7, 2018.
[15] M. B. Elsheniti, O. A. Elsamni, R. K. Al-dadah, S. Mahmoud, E. Elsayed, and K. Saleh, "Adsorption Refrigeration Technologies," 2018.

[16] Y. Zhao, L. Wang, R. Wang, Z. Q. Jin, L. Jiang, and M. Fleurance, "Simulation of Heat and Mass Transfer Performance with Consolidated Composite Activated Carbon," Heat Transfer Research, vol. 46, pp. 109-122, 2015.

[17] R. H. Mohammed, O. Mesalhy, M. L. Elsayed, and L. C. Chow, "Performance enhancement of adsorption beds with silica-gel particles packed in aluminum foams," International Journal of Refrigeration, vol. 104, pp. 201-212, 2019.

[18] S. Jribi, T. Miyazaki, B. B. Saha, S. Koyama, S. Maeda, and T. Maruyama, "Corrected adsorption rate model of activated carbon–ethanol pair by means of CFD simulation," International Journal of Refrigeration, vol. 71, pp. 60-68, 2016.

[19] I. Albaik, M. Badawy Elsheniti, R. Al-Dadah, S. Mahmoud, and İ. Solmaz, "Numerical and experimental investigation of multiple heat exchanger modules in cooling and desalination adsorption system using metal organic framework," Energy Conversion and Management, vol. 251, p. 114934, 2022.

[20] M. Shaaban, M. B. Elsheniti, A. Rezk, M. Elhelw, and O. A. Elsamni, "Performance investigation of adsorption cooling and desalination systems employing thermally enhanced copper foamed bed coated with SAPO-34 and CPO-27(Ni)," Applied Thermal Engineering, vol. 205, p. 118056, 2022/03/25/ 2022.