ICASTOR Journal of Engineering
Vol. 3, No. 3 (2010) 269 – 281

SIMULATION AND OPTIMISATION OF A SOLAR-POWERED ADSORPTION REFRIGERATION MODULE

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ABSTRACT

Adsorption refrigeration technology has been intensively investigated in many countries of the world because of its potential for competing with conventional vapour compression refrigeration and its environmental friendliness.

A solar-powered adsorption refrigerator using activated carbon/methanol pair was designed and fabricated. A mathematical model was developed based on the thermodynamics of the adsorption process, heat and mass transfer equations of the collector/generator/absorber components and simplified idealization of the condenser and evaporator components. The partial differential equations generated from the analysis were transformed into explicit finite difference forms for numerical solution. The model was used to compute the collector plate, bond and adsorbent temperatures, and the COP. The model was validated by using data from experiments performed on a solar powered activated carbon/methanol refrigerator and from published works.

The predicted peak plate, tube and adsorbent temperatures were102, 88 and 86°C respectively which compared favourably with 109 peak plate, 95 tube, and 85°C adsorbent temperatures from published works. The COP of the modelled refrigerator using imported activated carbon ranged from 0.0340 to 0.0345 compared to 0.0300 to 0.0550 recorded in the literature while the COP achieved from the experimental rig using locally manufactured activated carbon ranged from 0.0163 to 0.0200. Reducing the tube thickness from 5 mm to 1.5 mm led to a gain of 80.0% in COP. The adsorbent parking density of 550 kg/m^3 gave an optimum COP, while a decrease of plate thickness from 1.5 mm to 1.0 mm increased the COP from 0.0338 to 0.0352.

KEYWORDS: Refrigeration, Solar, Adsorption, Activated Carbon, Methanol

INTRODUCTION

A country like Nigeria in tropical Africa has an attractive potential for solar energy applications. The demand for space cooling and refrigeration follows the pattern of availability of solar energy. The most promising applications are vaccine storage and food storage [1]. Many

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agricultural products like fruits, vegetables, meat, milk, fish, etc. can be maintained in fresh conditions for significantly longer periods of time if they are stored at low temperatures [2]. As a result of these problems, sharp differences exist in food supplies between the harvest and off harvest periods. High market value agricultural products are usually abundant and cheap during the harvest season but scarce and expensive at other times [2]. Development of an appropriate refrigeration system will help in reversing this trend.

Anyanwu *et al.*, Khattab *et al.*, Gadalla *et al.* [6−8] reported a thermodynamic design procedure for solid adsorption solar refrigerator. The mathematical model is based on the thermodynamics of the adsorption process, heat transfer in the collector plate/tube arrangement, and heat and mass transfers within the adsorbent/adsorbate layer. Temperature was also predicted to within 5°C.

Antonio Pralon Ferreira Leite *et al.* [9] presented experimental thermodynamic cycles and performance analysis of a solar-powered adsorptive icemaker that uses activated carbon-methanol pair in hot humid climate. The maximum regenerating temperatures were 100.1°C, 87.3°C and 92.7°C, with an ice production of 6.02, 2.10 and 0 kg by square meter of projected area, for cycles of clear sky, partially cloudy and overcast days.

Thus, the main focus of this study is the development and evaluation of a solar-powered adsorption refrigerator using activated carbon/methanol pair. This simulation work will help us to find out the effects of system components, adsorbent properties and local climate on the performance of the refrigerator.

1. CROSS SECTION OF SHEET AND TUBE

Figure 1 is the modelled solar adsorption refrigerator while Figure 2 shows the absorber plate of the solar collector divided into nodes.

As shown in Figure 3, the temperature distribution between tubes can be derived if we assume that the temperature gradient in the flow direction is negligible.

1.1. ENERGY BALANCE EQUATION FOR COLLECTOR/GENERATOR/ ADSORBER.

The energy balance equation for this component is given by:

$$\rho C_p \frac{\partial T}{\partial t} = Q_n - U_L(T - T_{amb}) + \dot{Q}_g + K\frac{\partial^2 T}{\partial x^2} \tag{1.1}$$

$$\dot{Q}_g = h_{sg} m_{ad} \frac{dX}{dt} \tag{1.2}$$

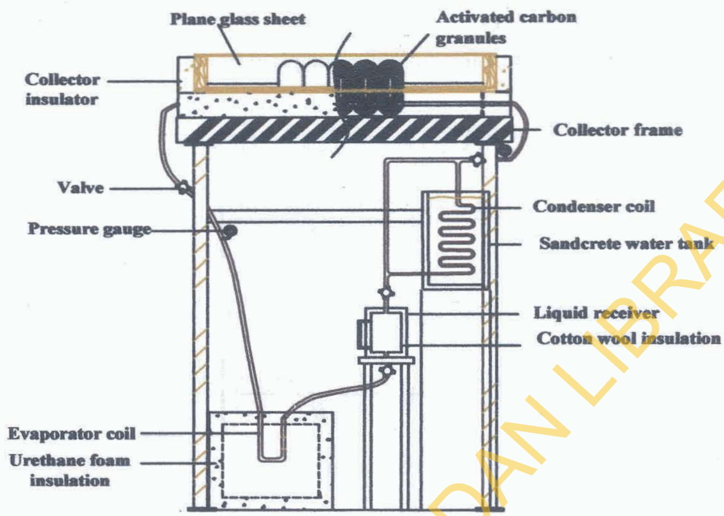

Figure 1. Sectional view of the adsorption solar refrigerator

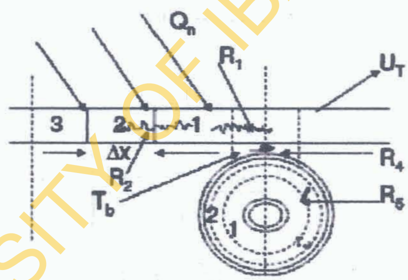

Figure 2. The Collector/Generator/Adsorber Nodes

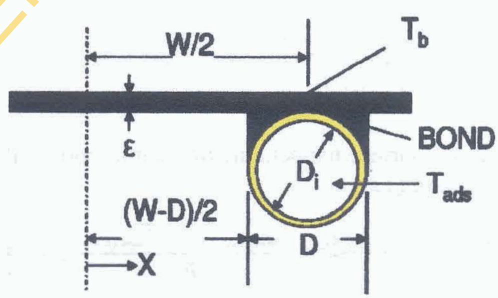

Figure 3. Sheet and Tube Dimensions

1.2. THE COLLECTOR PLATE

The finite difference equation for the collector plate is:

$$T_{p(t+1,i+1)} = T_{p(t,i+1)}\frac{\Delta t}{C_1}\left[\left[Q_n - U_L\left(T_{p(t,i+1)} - T_{amb(t)}\right)\right]A_n + \frac{T_{p(t,i+2)} - T_{p(t,i+1)}}{R_2} + \frac{T_{b(t)} - T_{p(t,i+1)}}{R_1}\right] \quad (1.3)$$

$$T_{p(t+1,i+1)} = T_{p(t,i+1)}\frac{\Delta t}{C_1}\left[\left[Q_n - U_L\left(T_{p(t,i+1)} - T_{amb(t)}\right)\right]A_n + \frac{T_{p(t,i+2)} - T_{p(t,i+1)}}{R_2} + \frac{T_{b(t)} - T_{p(t,i+1)}}{R_1}\right] \quad (1.4)$$

$$C_1 = \rho_p C_{pp} \Delta x \varepsilon_p \quad (1.5)$$

1.3. FDE FOR THE MID-PORTION (BOND)

The finite difference form of the energy equation for the mid-portion will be obtained using assumption (iv), (v) and (ix). The FDE becomes:

$$T_{b(t+1,i)} = T_{b(t,i)}\frac{\Delta t}{C_2}\left[\left[Q_n - U_L\left(T^*{}_{b(t)} - T_{amb(t)}\right)\right]A_c + \frac{2\left(T_{p(t,i+1)} - T_{b(t)}\right)}{R_1}\frac{\left(T_{ad(t,e)} - T_{b(t)}\right)}{R_4}\right] \quad (1.6)$$

$$C_2 = \rho_p c_{pp} W_p \varepsilon_p + \rho_p C_{pb} W_b \varepsilon_b + \rho_{pe} c_{ppe} \pi \frac{(D_o^2 - D_i^2)}{4} \quad (1.7)$$

$$T_{b(t)}{}^* = T_{b(t)} + \left[T_{p(t,i+1)} - T_{b(t)}\right]\frac{\left[\frac{\varepsilon_p}{K_p} + \frac{\varepsilon_b}{K_b}\right]}{W_b R_1} \quad (1.8)$$

Equation (1.8) gives the exposed surface temperature of the mid-portion Iloeje.*et al* (1995) Boundary condition for equation (1.6) is:

$$T_{ad(t+1,i)} = T_{ad(t,i)} + \frac{\Delta t}{C_3}\left[\frac{T_{b(t)} - T_{ad(t,e)}}{R_4} + \frac{T_{ad(t,e-1)} - T_{ad(t,e)}}{R_5} + Q_g\right] \quad (1.9)$$

$$C_3 = \rho_{ad}\left[c_{p(ad)} + \left(X + c_{pr}\right)\right]V_n \quad (1.10)$$

1.4. FDE FOR THE ADSORBENT

The FDE for the adsorbent is given by:

$$T_{ad(t+1,i)} = T_{ad(t,i)} + \frac{\Delta t}{C_4}\left[\frac{K_{ad}2\pi}{\Delta r}\left(T_{ad(t,i+1)} - T_{ad(t,i)}\right)r_p + \left(\left(T_{ad(t,i-1)} - T_{ad(t,i)}\right)r_m\right) + Q_g\right] \quad (1.11)$$

$$C_4 = \rho_{ad}\left[c_{p(ad)} + \left(X + c_{pr}\right)\right]V_{ni} \quad (1.12)$$

A relationship exists between the concentration, pressure and temperature via the Duibinni-Astakhov (D-A) equation in the form stated below (Anyanwu *et al.*, 2005)

$$X = \rho_r(T)W_0 exp\left[-D\left(T\, ln\frac{P_5(T)}{P}\right)^n\right] \quad (1.13)$$

This equation (1.13) can be easily differentiated with respect to time in order to obtain the rate of change f concentration. This will give:

$$\frac{dx}{dt} = \left[nX_{e0}DT^n\left(ln\frac{P_s(T)}{P}\right)^{n-1} \quad exp\left(-D\left(Tln\frac{P_s(T)}{P}\right)^n\right)\right]\left[\frac{dlnP}{dt} - \frac{h_{sg}}{RT}\frac{\left(T_{(t+1)} - T_{(t)}\right)}{\Delta t}\right] \quad (1.14)$$

The pressure field is given by the equation (Anyanwu, 2005)

$$P = exp(7.509329 - 1004.576T_{ad}^{-1} - 246199T_{ad}^{-2} + 1.91914 \times 10^7 T_{ad}^{-3})(mbar) \quad (1.15)$$

1.5. THE CONDENSER

Water-cooled condenser is employed because of its effectiveness.

$$T_{cond} = 159.6 + 0.4575T_{amb} \quad (1.16)$$

The variation of saturation pressure with condenser temperature for methanol is given by this equation, Khattab (2006):

$$P_{cond} = exp\left(18.587 - \frac{3626.55}{T_{cond} - 34.29}\right) \quad (1.17)$$

The total mass of methanol generated is given by the equation:

$$m_r = \left[X_{initial} - X_{final}\right]APDV_{pi}n_{pi} \quad (1.18)$$

1.6. OVERALL HEAT TRANSFER COEFFICIENT

This is given as defined by Khattab (2006):

$$U_T = \left\{ \frac{n_g}{{}^{C}/_{T_p} \left[\frac{T_p - T_{amb}}{n_g + f} \right]^e} + \frac{1}{h_w} \right\}^{-1} + \frac{\sigma(T_p + T_{amb})(T_p{}^2 + T_{amb}{}^2)}{(E_p + 0.00591 n_g h_w) + \frac{2n_g + f - 1 + 0.133E_p}{E_g} - n_g} \quad (1.19)$$

$$f = (1 + 0.089h_w - 0.1166h_w E_p) \quad (1.20)$$

$$e = 0.43\left(1 - \frac{100}{T_p}\right) \quad (1.21)$$

$$C = 520(1 - 0.000051\beta^2), \text{ for } 0 < \beta < 70°, \text{ for } 70° < \beta < 90°, \text{ use } \beta = 70° \quad (1.22)$$

$$h_w = 2.8 + 3.0xWS \quad (1.23)$$

2. RESULTS AND DISCUSSION

The mathematical model developed was employed in a computer algorithm using FORTRAN programming coding language. This was then run on a personal compute with sufficient memory facilities to carry out the simulation exercise. The hourly solar radiation data obtained at the test location in Ibadan was used for the program run. The average condensing temperature of the evaporative condenser used was 24°C while the evaporating temperature of the evaporator was -10°C. The simulator was used to predict the refrigerator performance using the insolation data obtained for ten days in Ibadan. The peak plate (102° C), tube surface (85° C) temperatures and commencement of methanol generation were accurately predicted. The results of the sensitivity tests are presented in Figures 4-12.The predicted plate, tube and adsorbent temperatures are shown in Figure 4. For the same number of tubes of constant thickness, Figure 5 is pointing to the fact that there is significant improvement in refrigerator performance with decrease in plate thickness. Figure 6 demonstrates the effect of tube thickness on the performance of the refrigerator.

An increase in tube thickness with all other parameters remaining constant leads to a decrease in the efficiency of the refrigerator. Figure 7 shows a direct relationship between the refrigerator coefficient of performance and adsorbent packing density, APD. Figure 8 shows the dependence of the refrigerator performance on the overall heat transfer coefficient (U_L). It indicates that for the system to perform optimally the collector must be designed such that leakages (heat losses) are minimised during the heating operation of the activated carbon and methanol. The information from Figure 9 is that the performance of the refrigerator is not a strong function of adsorbent thermal conductivity when other parameters are held constant. Figure 10 shows the dependence of the system performance on plate thermal conductivity up to a particular point while other parameters are held constant. It means that a material with thermal conductivity in the range of 41-50 W/m.K will be appropriate for the machine. Figure 11 reveals a direct relationship between plate absorptivity and transmittance of the glass cover. The predicted plate temperature and measured temperature are shown in Figure 12. They follow the same pattern.

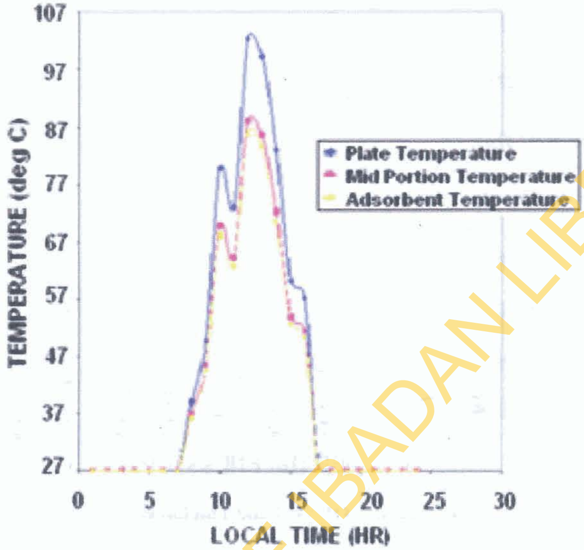

Figure 4. Predicted Plate, Tube (Mid-Portion) and Adsorbent Temperatures

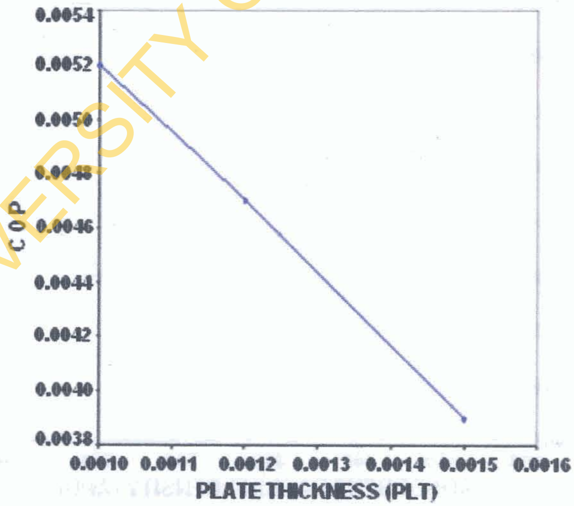

Figure 5. COP vs. Plate Thickness

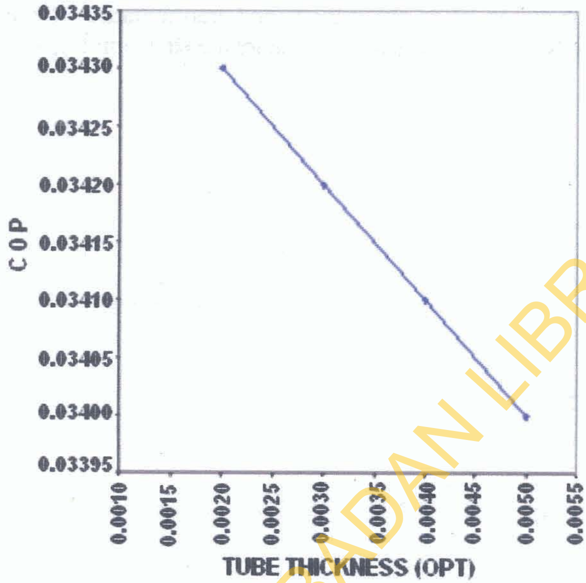

Figure 6. COP vs. Tube Thickness

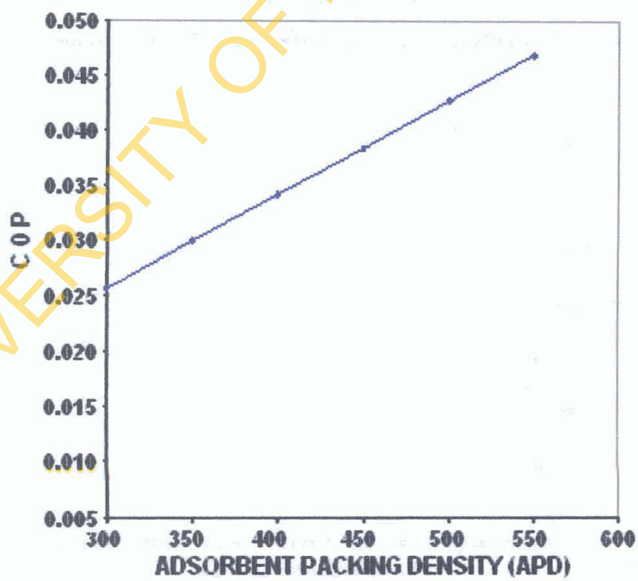

Figure 7. COP vs. Adsorbent Packing Density, APD, kg/m^3

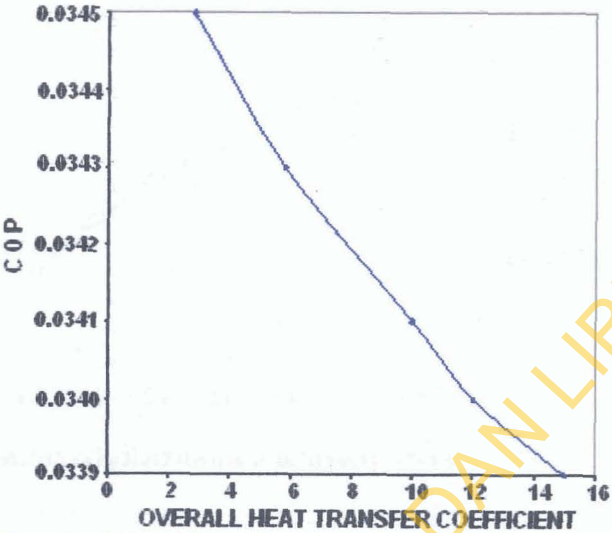

Figure 8. COP vs. Overall Heat Transfer Coefficient (U_L) (W/m^2.K

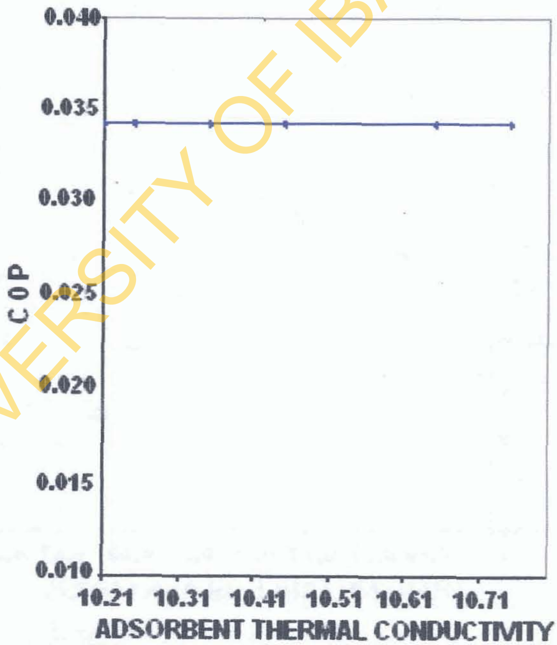

Figure 9. COP vs. Adsorbent Thermal Conductivity

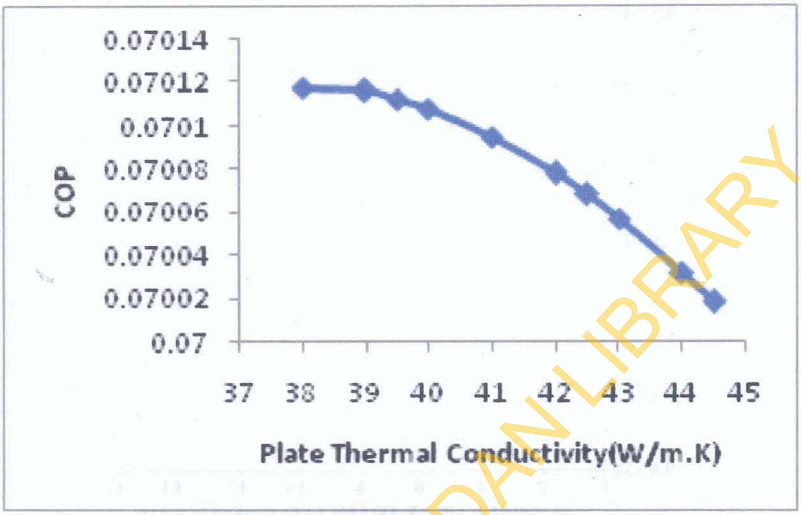

Figure 10. COP vs. Plate Thermal Conductivity

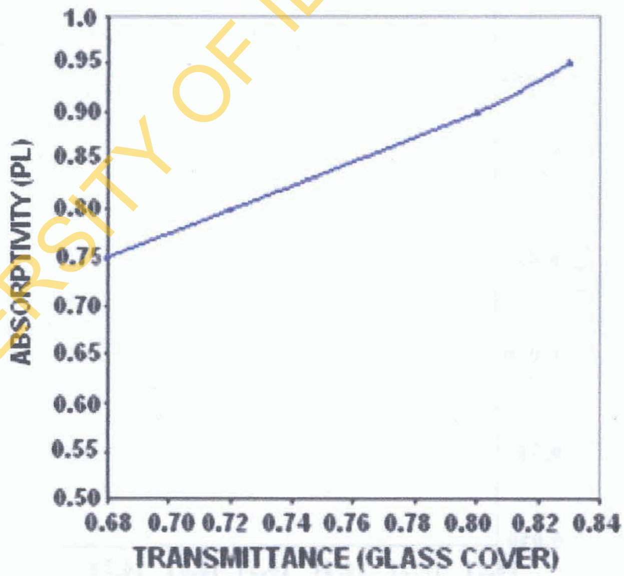

Figure 11. Plate Absorptivity vs. Transmittance of Glass Cover

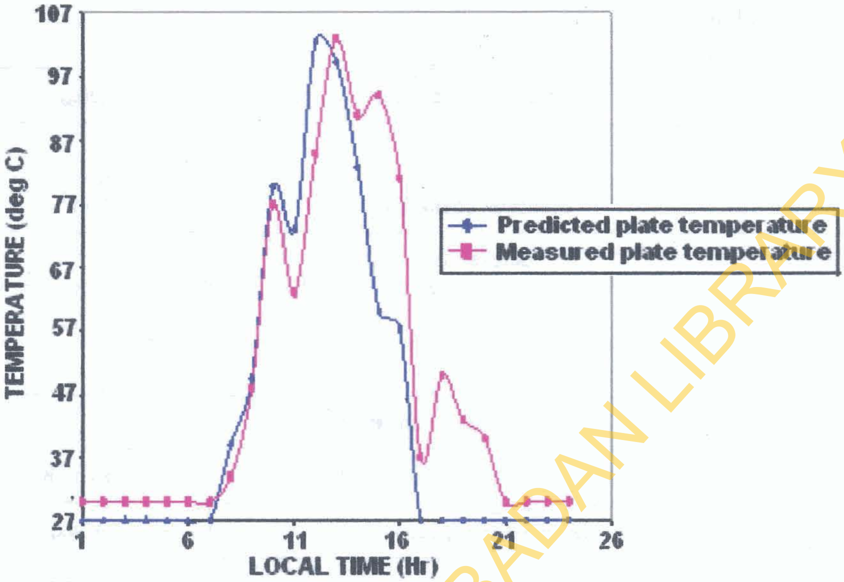

Figure 12. Predicted Plate Temperature vs. Measured Plate Temperature

CONCLUSIONS

The transient analysis of a solar powered solid adsorption refrigerator using activated carbon/methanol pair has been undertaken in a simulation exercise. The predicted peak plate temperature is 103°C while the measured peak temperature was 102°C. This is in good agreement with the works of Anyanwu *et al.* (2001) and Iloeje *et al.* (1995) who obtained peak plate temperatures of 109°C and 104°C respectively. Khattab *et al.* (2004) using reflector arrangement to enhance heat transfer in the collector obtained peak temperature of 110°C. Khattab *et al.* (2006) with the aid of glass shells within the adsorptive bed obtained a peak temperature of 130°C. Antonio Pralon *et al.* (2007) reported a peak temperature of 100°C for a solar-powered adsorptive icemaker in hot humid climate. This program can therefore be used for parametric and design optimization studies of a solar-powered adsorption refrigerator.

ACKNOWLEDGEMENT

This work was carried out at School of Engineering and Design, Brunel University, Middlesex, United Kingdom. The author is grateful to the School for the facilities made available and Professor T. G. Karayiannis for his helpful comments.

NOMENCLATURE

C_p	specific heat capacity	kJ/kg/K
E	emmitance	
E_k	extinction coefficient	m^{-1}
F_d	diffuse ratio	
F_D	dust factor	
f_s	shading factor	
D	constant in Dubinin equation	
h_w	wind heat transfer coefficient	$W/m^2/K$
h_{fg}	latent heat of vapourisation	kJ/kg
k	thermal conductivity	W/m/K
h_{sg}	isosteric heat of sorption	kJ/kg
L	length	m
m	mass	kg
n_g	number of glass cover	
n_{pi}	number of pipes	
P	pressure	bar
$\dot{Q}$	heat flow rate	kJ/s
Q	heat energy	kJ
R	specific gas constant	kJ/kg/K
S	tilt correction factor	
T	temperature	K
U_L	overall heat transfer coefficient	$W/m^2/K$

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