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Thermal Performance of an Air Heating Storing System

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***Abstract*—**Owing to the lack of synchronization between the solar energy availability and the heat demands in a specific application, the energy storing sub-system is necessary to maintain the continuity of thermal process. The present work is dealing with an active solar heating storing system in which an air solar collector is connected to storing unit where this energy is distributed and provided to the heated space in a controlled manner. The solar collector is a box type absorber where the air flows between a number of vanes attached between the collector absorber and the bottom plate. This design can improve the efficiency due to increasing the heat transfer area exposed to the flowing air, as well as the heat conduction through the metal vanes from the top absorbing surface. The storing unit is a packed bed type where the air is coming from the air collector and circulated through the bed in order to add/remove the energy through the charging / discharging processes, respectively. The major advantage of the packed bed storage is its high degree of thermal stratification. Numerical solution of the packed bed energy storage is considered through dividing the bed into a number of equal segments for the bed particles and solved the energy equation for each segment depending on the neighbor ones. The studied design and performance parameters in the developed simulation model including, particle size, void fraction, etc. The final results showed that the collector efficiency was fluctuated between 55%-61% in winter season (January) under the climatic conditions of Misurata in Libya. Maximum temperature of 52ºC is attained at the top of the bed while the lower one is 25ºC at the end of the charging process of hot air into the bed. This distribution can satisfy the required load for the most house heating in Libya.

***Keywords*—**Solar energy, thermal process, performance, collector, packed bed, numerical analysis, simulation.

I. Introduction

In recent years, interest in active and passive solar design of buildings has been increased. In some designs, it is possible to use a part of the building structure as a solar radiation absorber to heat the inlet air to the building i.e. Trombe walls [1], [2]. In other instances, solar-assisted heating has been provided by using part of the building structure as a large area solar collector to pre-heat air prior to its passage over a heat pump evaporator [3]. The developed of using an air solar collector coupled by a packed bed storage has also received attention.

Thus, among different designs, the present study is developed and is dealing with an air solar collector coupled by packed bed storage. The selection of the air solar collector used in the study is a box type design [4]. This selection is based on the previous and published advantages of different configurations as: matrix type design [5]-[7]; single pass corrugated bare plate solar air collector [8]; and an artificial roughness surface in ducts flow of air [9], [10]. Accordingly, the box type solar collector is considered in the study due to the features of air flow channels enclosed by an upper absorbing surface, a back surface, and metal vanes between the two surfaces.

On the other hand, the selection of storage type is based on the available published works on the heating and cooling of packed beds where it is more suitable under Libyan climatic conditions. Also, it is cheaper and available with various particle sizes. Moreover, the packed bed storage can satisfy an acceptable degree of thermal stratification compared to other types i.e. phase change materials.

The first modeling of the packed bed storage was developed by Schumann [11], where the governing equations describing the packed bed are often referred to as the Schumann model. The model is based on the assumptions of: one dimensional plug flow; no axial conduction or dispersion; constant properties; no heat loss to the environment; and finally no temperature gradients within the solid particles. However, for long term study of solar energy systems, the analytical solutions are not useful and in such cases, numerical techniques must be employed. For this purpose, Kuhn et al [12] investigated a large number of finite difference scheme to numerically solve the equations developed by Schumann and concluded that the "complicated effectiveness NTU" method of Hughes [13] was the best suited for solar system simulation.

II. Mathematical Modeling

The developed system is mainly composed of a box type solar collector and packed bed storage as shown in Fig. 1.

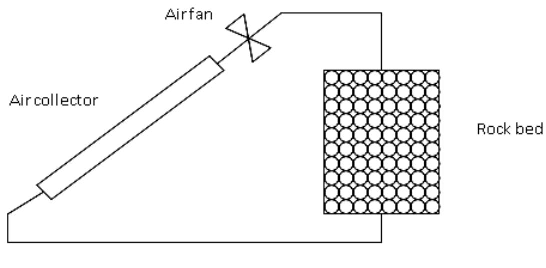

Fig. 1 Schematic of a solar air collector coupled by a packed bed storage

Modeling of the air box collector is based on the model developed in [4] including the detailed derivation and the output useful heat of the collector as function of a number of performance factors i.e. F1-F6. These factors are given as:

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$$F_1 = \frac{N\sqrt{2h_c\, l_c\, K\, A}}{A_c \sinh mD} \tag{1}$$

$$F_2 = U_t + h_1 + h_r + F_1 (\cosh mD) \tag{2}$$

$$F_3 = h_r + F_1 \tag{3}$$

$$F_4 = h_r + F_1 (\cosh mD) + h_3 + U_b \tag{4}$$

$$F_5 = h_1 + F_1 [(\cosh mD) - 1] \tag{5}$$

$$F_6 = h_3 + F_1 [(\cosh mD) - 1] \tag{6}$$

Thus, as a function of the given factors, the useful heat from the collector can be calculated by [4], [14]:

$$Q_u = A_c\, F_R [S - U_L (T_{fi} - T_a)] \tag{7}$$

$$F_R = \frac{m C_p}{A_c U_L}\left[1 - \exp\left(-\frac{A_c U_L F^{\backslash}}{m C_p}\right)\right] \tag{8}$$

$$F^{\backslash} = \left(\frac{F_4 F_5 + F_3 F_6}{F_2 F_4 - F_3^2}\right) \tag{9}$$

$$U_L = \frac{U_t (F_4 F_5 + F_3 F_6) + U_b (F_3 F_5 + F_2 F_6)}{F_4 F_5 + F_3 F_6} \tag{10}$$

The symbols in the previous equations are given in the Nomenclature list. Solution of the air collector model is based on assuming an initial value of the collector absorber. Heat transfer coefficients and collector parameters are calculated on the basis of the guessed absorber temperature and hence the outlet temperature is calculated and considered as the inlet one to the packed bed storage. The assumed absorber temperature is checked by [14]:

$$T_{pm} = T_{fi} + \frac{Q_u / A_c}{F_R U_L}(1 - F_R) \tag{11}$$

The iterative procedure is repeated until all consecutive mean temperatures and parameters are differed by less than 0.1-0.2 %. The modeling of the packed bed storage is based on the approach used by Schumann [11] who analyzed and made an energy balance for the fluid and bed in a differential form as:

$$\left(\rho C_p\right)_f \varepsilon \frac{dT_f}{dt} = -\frac{(m C_p)_f}{A_b}\frac{dT_f}{dx} + h_v \left(T_b - T_f\right) \tag{12}$$

$$\left(\rho C_p\right)_b (1-\varepsilon)\frac{dT_b}{dt} = h_v (T_f - T_b) \tag{13}$$

The volumetric heat transfer coefficient h_v in W/m^3C in (13) is given as:

$$h_v = 650\left(\frac{G_o}{D_p}\right)^{0.7} \tag{14}$$

where G_o is the mass velocity in $kg/m^2.s$ while D_p is the particle diameter in meters.

III. Results and Discussion

The box type air collector was simulated with the incident solar radiation and ambient air temperature as input data. The solar radiation is calculated using clear sky model (ASHRAE model) in [15]. The ambient air temperature is obtained from the available meteorological data for Misurata city (latitude angle 32.16 degree).

The results included the most performance parameters for the collector and storage unit under the given operating conditions. The simulation is carried out with the following dimensions and specifications: collector area =2m^2; air mass flow rate =0.03kg/s; specific heat of air =1005 J/kg.K; Number of collector metal vanes =10; air duct depth =0.1m; length in flow direction of the bed =1m; cross section of the bed =2m^2; equivalent diameter of pebbles =0.02m; density of the pebble material (rock) =1350 kg/m^3; specific heat of pebbles =900 J/kg C; thermal conductivity of pebble material =0.85 W/m.C; void fraction =30%; Number of segments of the bed =10.

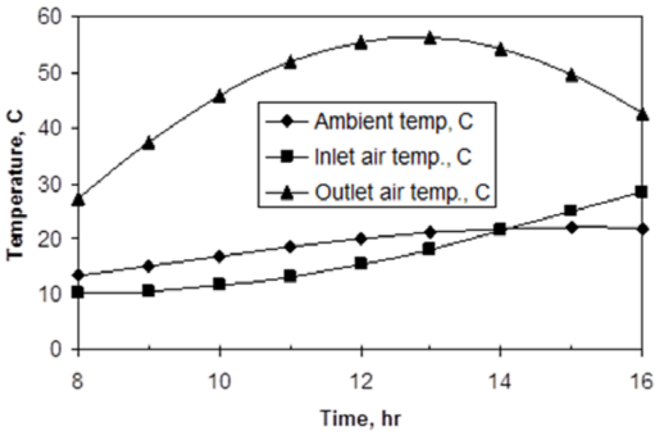

Fig. 2 Variation of the temperatures of ambient air, inlet air and outlet air of collector along the day through the charging process

Fig. 2 shows the inlet and outlet temperatures of the box type solar collector against the time of day based on the data given previously. The ambient air temperature is given also in the figure. It is important to note that the inlet temperature to the collector is the same temperature of the lower side of the packed bed. From the figure, it is clear that the trend of the inlet temperature is lower than the ambient one particularly at morning times. At 2 p.m., the inlet temperature increased than the ambient due to the heating of the lower segment of the packed bed while the ambient temperature begins to decrease. The outlet temperature is the highest for the whole hours of the day. Maximum temperature difference between the inlet and ambient air of about 5ºC is observed while the maximum temperature of the air exit from the collector is 55ºC.

The incident solar radiation and the energy extracted from the box type collector versus the day time in January are presented in Fig. 3. It is seen from the figure that the useful heat from collector is moving paralleling to the incident solar radiation. Maximum useful heat of 604 W/m^2 corresponding to maximum radiation of 1027 W/m^2 is observed in the figure.

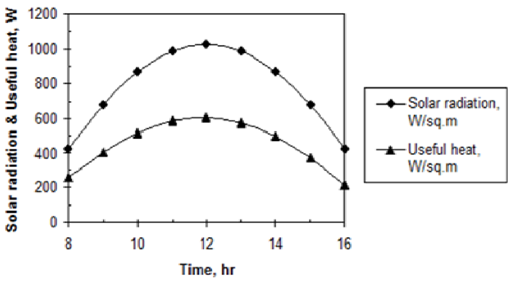

Fig. 3 Solar radiation and useful heat from the collector versus day time

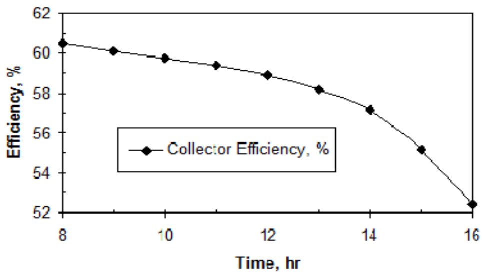

Fig. 4 Variation of the air collector efficiency along the day hours

The collector efficiency which is defined by the useful energy divided by the incident solar radiation is given in Fig. 4. In the morning, the collector efficiency having a higher value while it decreased with the day time. This behavior is referred to the lower temperatures of both, the absorbing surface and inlet air at morning times leading to decrease the collector losses. Maximum efficiency of about 61% for the collector is attained in the figure.

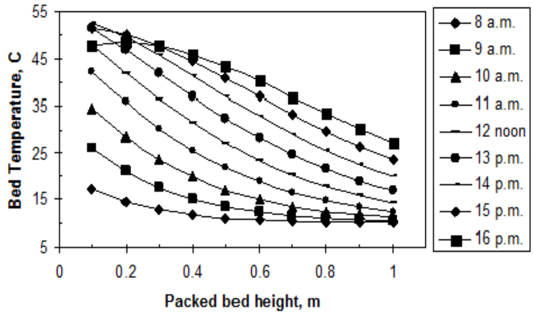

Fig. 5 Temperature distribution of the packed bed along the bed height at different hours through the charging process

Fig. 5 shows the variation of the temperatures of packed bed material (rock) versus the height of the bed at different hours of the day. This distribution shows a high degree of thermal stratification inside the bed for the most hours of the day satisfying a higher efficiency for the air collector and also higher temperature of exit air at discharging process for space heating. At the end of the day (i.e. 16 p.m.), the thermal stratification begins to destroy due to decrease of the collector outlet temperature as seen in the figure. Maximum temperature of 52ºC at the upper section of the bed is attained as seen.

At the same time, the distribution of the hot air temperature passing through the bed is shown in Fig. 6.

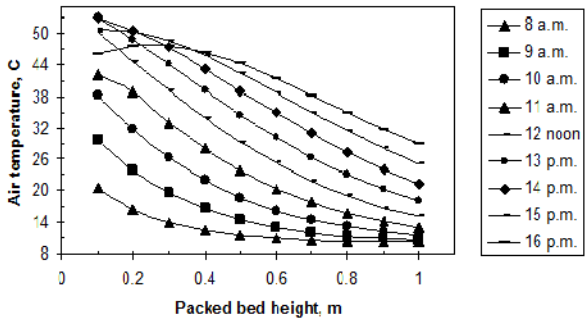

Fig. 6 Temperature distribution of hot air along the bed height at different hours through the charging process

This figure shows a decrease of the air temperature as it flows from the top section to the bottom one of the bed. This is referred to the decrease of heat transferred from the flowing air to the bed particle through the air flow inside the bed. At late time, the inlet temperature to the bed which is concerning to the collector outlet temperature is decreased as seen in the figure. Thus, cold air is exit from the lower section of the bed and going to the air collector increasing its efficiency.

IV. Conclusions

From the previous study, it can be concluded the following items:

- The box type solar collector coupling by packed bed energy storage can be used successfully in space heating applications under Libyan climatic conditions.

- High degree of thermal stratification is attained in the packed bed storage. Maximum temperature of 52ºC at the top of the bed is attained in the developed simulation study which is enough for space heating through the discharging process in the night.
- Solar air collector could be operated with a higher efficiency (55%-61%) during the charging process for the day time in January month due to the lower inlet air temperature coming from the packed bed.

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