PERFORMANCE CALCULATIONS FOR CLOSED-LOOP AIR-TO-WATER SOLAR HYBRID HEATING SYSTEMS WITH AND WITHOUT A ROCK BED IN THE SOLAR AIR HEATER

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Abstract—A transient analysis has been carried out on a hybrid solar water heater which comprises a rock bed air heater with optimum design parameters in conjunction with an air-to-water transverse fin shell-and-tube heat exchanger (mixed air and unmixed water type) in which cold water from the storage tank receives heat from the hot air coming out of the air heater which flows in the shell at right angles to the water flowing in the tubes. The system's performance has been evaluated for typical winter weather conditions in Delhi for different combinations of flow rates of air and water for different volumes of the water storage tank. No hot water is assumed to be withdrawn from the tank to serve the load. A comparative analysis of the system's performance with and without a rock bed in the air heater reveals about 11°C higher temperature of storage tank in the former at 50 kg/h m⁻² air flow rate. With both the air heater types, although the system performance was observed to increase with the rates of air and water flow, no significant improvement in system performance was achieved at \( M_w \geq 2M_a \).

1. INTRODUCTION

Water heating technology has become increasingly popular especially in the cold climatic regions of countries which are in the sun belt of the world where a substantial fraction of the energy demand can be covered by the available solar energy. However, the major disadvantage associated with the conventional solar water heaters in cold regions is the damage due to freezing of water in the collector channels. This problem can be avoided by three major ways: (1) by using a hybrid solar collector comprising an air-to-water heat exchanger [1–5]; (2) by using a solar collector with antifreeze liquid as the heat transfer fluid and liquid-to-water heat exchanger; and (3) by using "open" type water heater [6] in which the collector operates at atmospheric pressure. Water is pumped through the collector only when there is a heat gain. When the pump is stopped, water flows back to the storage due to gravitation. In this way, energy loss due to cooling of resident water is avoided and damage due to freezing is prevented.

Due to its simplicity and the absence of frosting, use of air-to-water heat exchangers was suggested by Loth [1, 2]. In a recent communication [7], we have reported a steady state analysis of a hybrid air-to-water solar hybrid heating system consisting of a conventional air heater and a longitudinal finned-tube heat exchanger. Since the pressure drop experienced by the air flowing through the conventional air heater and the heat exchanger is significantly low, no pressure drop or pumping power analysis was made in the paper.

The present paper deals with the transient analysis of a hybrid air-to-water solar collector which consists of a rock bed solar air heater and a transverse fin shell-and-tube heat exchanger through which hot air from the air heater and relatively cold water from the storage tank circulate for transfer of heat from the air to the water. Since the pressure drop experienced by the air flowing through the rock bed is relatively high, the pumping energy expended in forcing both air and water to circulate through the system has been computed. The effective energy gain of the system and the temperatures of the storage tank water have been obtained for different air and water mass flow rates and different volumes of the water storage tank. For comparison, a parallel study has also been made on a hybrid system with the rock bed air heater replaced by a conventional parallel plate empty channel air heater with cover plates above the absorber.

2. SYSTEM DESCRIPTION

2.1. System components

The different components of the hybrid air-to-water closed-loop solar heating system shown schematically
in Fig. 1(a) are (1) air heater, (2) heat exchanger and (3) water storage tank. The design features of these components are discussed below.

2.1.1. **Air heater.** The air heater in the hybrid water heating system is of the rock bed type consisting of two cover plates of glass (interspaced by 3 cm) and a wooden back plate (insulated) with the air flow passage between the inner glass cover and the back plate filled with black painted spherical rocks which absorb the solar radiation transmitted by the two cover sheets. The conventional air heater to replace the rock bed air heater for comparative analysis is assumed to be comprised of two glass covers, an absorber plate and an insulated back plate with the air flow beneath the absorber. The design parameters of the air heaters chosen for the present investigation are as per our earlier optimization studies [9] which are summarized in Table 1.

2.1.2. **Heat exchanger.** The heat exchanger coupled to the air heater and the water storage tank in the hybrid air-to-water solar collector system is a transverse fin shell-and-tube type as shown in Figs 1(a) and 1(b). The tubes carrying the water with transverse fins on their outer surfaces are arranged in a triangular pitch (Fig. 1(b)) inside the shell in which the air flows at right angles to the direction of water flow in the tubes (cross flow pattern). The tubes in the shell are arranged such that water flows in series from one tube bank to the next, so that there will be as many passes as banks. Since in the shell there are several banks, the air may be considered mixed and the water unmixed. The design parameters of the heat exchanger are summarized in Table 1.

2.1.3. **Water storage tank.** The tank connected to the heat exchanger for storing hot water is assumed

<table>
<thead>
<tr>
<th>Table 1. Design details of the hybrid system</th>
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| **Air heater** | \begin{tabular}{l}
Duct length = 2 m \\
Duct depth = 0.1 m \\
Duct width = 1 m \\
Rock diameter = 0.03 m \\
Packing porosity = 0.5
\end{tabular} |
| **Heat exchanger** | \begin{tabular}{l}
Duct size of shell = 1.25 x 1.25 m \\
No. of tube banks = 12 \\
No. of tubes per bank = 21 and 20 alternately (for triangular pitch) \\
Tube o.d. = 0.025 m \\
Tube i.d. = 0.02 m \\
No. of fins = 8/0.025 m \^2 \\
Fin height = 0.009 m \\
Fin thickness = 0.0009 m
\end{tabular} |
| **Water storage tank** | \begin{tabular}{l}
Volume = 0.1, 0.2 and 0.3 m \^3
\end{tabular} |
to be made up of steel, insulated with glass wool of thickness 5 cm. The water in the tank is assumed to be non-stratified, i.e. fully mixed with uniform temperature throughout for simplicity of theoretical modeling. Three water tank sizes were assumed as listed in Table 1.

2.2. Mode of operation

The closed, double-loop type hybrid air-to-water solar collector is assumed to operate during sunshine hours only when the solar flux is > 60 W/m². During this time, the hot air from the air heater and the cold water from the storage tank flow in cross flow pattern through the heat exchanger in which the hot air delivers its heat to the cold water. The relatively cool air from the heat exchanger then re-enters the air heater, receives more heat from the hot rock bed (which absorbs the solar radiation transmitted through the double glass cover) and then recirculates through the heat exchanger. Similarly, the relatively hot water from the heat exchanger enters the storage tank and after mixing with the stored water recirculates through the heat exchanger to receive more heat from the hot air. This process is repeated until the solar flux becomes very low by which time, the stored water reaches its maximum possible temperature.

3. THEORETICAL MODEL

To simplify the theoretical modelling it is assumed that the tubes coupling the three units, i.e. the rock bed/conventional air heater, the heat exchanger and the hot water storage tank are perfectly insulated and the losses through them are insignificant. In addition, the edge losses and the air leakage in the air heater are also neglected.

3.1. Energy balance equations

The time-dependent energy balance equations for the different components of the system are presented below.

3.1.1. Conventional air heater.

\[
M_1 C_1 \frac{dT_1}{dt} = x_1 I + h_{21}(T_2 - T_1) - h_{1A}(T_1 - T_A) \quad \quad (1)
\]

\[
M_2 C_2 \frac{dT_2}{dt} = \tau_1 x_2 I - h_{21}(T_2 - T_1) - h_{23}(T_2 - T_3) \quad \quad (2)
\]

\[
M_3 C_3 \frac{dT_3}{dt} = \tau_1 x_3 I - h_{23}(T_3 - T_2) - h_{34}(T_3 - T_4) - h_{3A}(T_3 - T_A) \quad \quad (3)
\]

3.1.2. Rock bed air heater.

\[
M_4 C_4 \frac{dT_4}{dt} = h_{34}(T_3 - T_4) - h_{4A}(T_4 - T_A) \quad \quad (4)
\]

3.1.3. Heat exchanger.

\[
M_k C_k \frac{dT_k}{dt} = -M_{th} C_{th}(T_{th} - T_{wa}) + A_{wa} h_{wa} \Delta T_L \quad \quad (11)
\]

Here,

\[
\Delta T_L = K(\Delta T_1 + \Delta T_2)/2,
\]

where

\[
\Delta T_1 = T_n - T_{wa},
\]

and

\[
\Delta T_2 = T_m - T_{wa}.
\]

3.1.4. Water storage tank.

\[
M_w C_w \frac{dT_w}{dt} = M_w C_w(T_{wa} - T_w) - U_{wa} A_k(T_s - T_A). \quad \quad (13)
\]

Here,

\[
T_s = T_{wa}.
\]
To obtain the temperatures of the different components of the hybrid water heating system and the hot water storage tank, eqs (1)–(13) were solved by using a finite difference technique.

3.2. Effective energy gain

By obtaining the temperature of the water in the storage tank, rock bed, etc., the effective energy gained by the hybrid system is obtained as

\[ E.E.G.(W) = M_c C_w (T_w - T_{w1}) - \int_0^T \dot{M}_w \Delta \rho_w \, dt / \rho_w \]

\[ - \int_0^T \dot{M}_w \Delta \rho_w \, dt / \rho_w \]  

(14)

\[ E.E.G.(R) = M_p C_R (T_R - T(1))L \]

\[ - \int_0^T \dot{M}_R \Delta \rho_R \, dt / \rho_R \]  

(15)

\[ E.E.G.(A) = M_A C_A (T_A - T(1))L \]

\[ - \int_0^T \dot{M}_A \Delta \rho_A \, dt / \rho_A \]  

(16)

3.3. Heat transfer coefficients

3.3.1. Air heater. For the above computations, the heat transfer coefficients \( h_{1,1}, h_{2,1}, h_{1,2}, h_{n,1}, h_{n,2}, h_{ox}, h_{2,3}, \) and \( h_{3,3} \) were obtained by using the empirical relations from refs [4], [8] and [9].

3.3.2. Heat exchanger. The air to water heat transfer coefficients \( h_{nw} \) in the heat exchanger for different air and water mass flow rates were obtained by using the following relations [10]:

\[ h_{nw} = \frac{h^*_w h^*_a}{h^*_w + h^*_a} \]  

(17)

where \( \Omega \) is obtained from Fig. A1 (Appendix), and

\[ h^*_w = \frac{h_w h_{nw}}{h_w + h_{nw}} \]  

(19)

\[ h^*_a = \frac{h_a h_{nw}}{h_a + h_{nw}} \]  

(20)

\[ h_0 = \frac{1}{R} \]  

(21)

where \( R \) is the fouling factor and is assumed as 0.003 (km²/W):

\[ h_{nw} = j_w (k_a D_0) \frac{(C_s \mu_s)^{1/3}}{k_a} \]  

(22)

where \( j_w = f(Re_{nw}) \) is obtained from Fig. A2 (Appendix) where

\[ Re_{nw} = D_0 G_{nw} / \mu_s \]  

(23) and \( h_w = f(V_w) \) is obtained from Fig. A3 (Appendix).

3.4. Pressure drop

The pressure drop experienced by the air flowing through the rock bed solar air heater is

\[ \Delta P_{aw} = 150 \left[ \frac{1 - \varepsilon}{Re_{aw}} + 1.75 \right] G^2 / \rho_a D_R \left[ 1 - \varepsilon \right] \]  

(24)

where \( Re_{aw} = GD_{aw} / \mu_a \)  

(25) and that for the air flowing through the conventional air heater is

\[ \Delta P_{ac} = 0.059 \frac{Re_{aw}^{1.2}}{2} \frac{G^2}{\rho_a} L \]  

(26)

where \( Re_{ac} = GD_{ac} / \mu_a \)  

(27)

The pressure drop experienced by the air flowing through the shell in the heat exchanger is

\[ P_{nh} = \frac{f_{sh} G_s^2 L_{sh}}{5.22 \times 10^{10} \times D_{sh} \left( \frac{D_s}{d} \right) \left( \frac{P_{w}^{0.6}}{d} \right)} \]  

(28)

where \( f_{sh} \), the friction factor for air, is obtained from Fig. A2 of the Appendix where

\[ f_{sh} = f(Re_{sh}) \]  

(29) and

\[ Re_{sh} = \frac{D_s G_s}{\mu_s} \]  

(30)

The pressure drop experienced by the water flowing through the tube banks in the shell is

\[ \Delta P_{aw} = \frac{f_w G_w^2 L_{aw}}{5.22 \times 10^{10} \times D_{aw}} \]  

(31)

Here \( f_w = f(Re_{aw}) \) is obtained from Fig. A4 (Appendix 1) where

\[ Re_{aw} = \frac{D_w G_w}{\mu_w} \]  

(32)

4. RESULTS AND DISCUSSION

The analysis has been made for a typical winter day in Delhi the solar flux and the ambient temperature for which are displayed in Fig. 2.
In Fig. 3 are shown the hourly values of the storage tank water temperature for different water flow rates for air flow rate 50 kg/h and storage tank volume 100 l. Similar curves for air flow rates 100 and 150 kg/h are shown in Figs 4 and 5. The curves clearly show that for fixed values of air flow rate, the storage temperature increases with an increase in the flow rate of water. However, at very large rates of water flow, i.e. $M_w > 2M_a$, the increase in water temperature is not significant. The curves in different figures also illustrate that for fixed values of water flow rate, the increase in the air flow rate results in an increase in the storage tank temperature. The maximum storage tank temperature for all air and water flow rates is observed to be attained at 5:00 pm.

The effective energy gained by the water in the storage tank and the rocks in the air heating collector, which were computed by subtracting the energy expended in pumping the fluids from the total energy gained by them for different flow rates, are presented in Figs 6-8. For a particular air flow rate, the effective energy gained by the water increases with an increase in the water flow rate, whereas the reverse is the case for the effective energy gained by the rocks. This can
be attributed to the fact that the water at a higher flow rate carries a larger amount of heat for storage in the water tank. Since at higher water flow rates more heat is carried away from the air, the outlet air from the heat exchanger, which recirculates through the air heater, is at a relatively lower temperature as compared with that when the water flow rate is low. The inlet air at a lower temperature in the rock bed collector carries more heat from the rocks which results in lower effective energy retained in the rocks. At all the air flow rates, the energy gain by water is observed not to increase significantly if the water flow rate is greater than twice the flow rate of air (i.e. $M_w > 2M_a$). The maximum energy gain by rocks is observed to be attained at 3:00 p.m., whereas that by water is observed to be attained at 6:00 p.m. With an increase in the air flow rate, for the same water flow rate, the energy gain by the water increases, whereas that by the rocks decreases. This is a consequence of the larger heat transfer from the rocks to the air at higher flow rates of air which, in turn, transfers more heat to the water in the heat exchanger.

The curves in Figs 9–12 compare the performance parameters of the hybrid system with the rock bed air heater and with the conventional air heater for different flow rates of air but for a fixed water flow rate ($M_w = 2M_a$) and fixed storage tank capacity ($M_{sw} = 100$ kg). For an air flow rate of $50$ kg/h, the curves in Fig. 9 show that the maximum storage tank temperature in the case of the rock bed air heater is $11$°C higher than that in the case of the conventional double cover air heater. This is clearly due to the larger heat gain by air flowing through the rocks due to the turbulence producing air flow path and larger heat transfer area in the rock bed collector which results in a larger heat transfer to the flowing air, which in turn gives more heat to the water in the heat exchanger as compared with that in the system with the conventional air heating collector.

The effective energy gain by the water in the storage tank and the absorber plate in the hybrid system with the conventional air heating collector and the energy stored in the water and the rock bed absorber in the case of the rock bed collector in the system, for different hours are plotted in Figs 10 and 11, respectively. For the reason discussed earlier, the energy stored in water in both cases is observed to increase with an increase in the air flow rate. However, at $M_w = 150$ kg/h, the energy gain by the water is not significantly larger than that when $M_w = 100$ kg/h. Moreover, for all air flow rates, the energy stored in the water tank in case of the rock bed collector is much higher than that in case of the conventional collector in the system. This higher energy gained by
water in the former case is due to the energy storage capacity of rocks which is much higher than that in the case of the absorber plate which is to be utilized during the next hour of operation of the system.

Assuming the load on the system to be zero for the particular day, the water storage tank temperature and the effective-energy-gain by the water and the rock bed were computed for an air flow rate $M_a = 100$ kg/h, water flow rate $M_w = 2M_a$, and storage tank capacity $(M_s) = 100$, 200 and 300 kg which are presented in Figs 12 and 13. The water temperature and the effective energy gain attain a maximum at 6 p.m., but the temperature decreases slowly with time due to loss through the storage tank walls during the night when the air and water flow rates are zero. For obvious reasons, although the water temperature is lower for larger storage tank volumes, the effective energy gain by the storage tank water increases with an increase in $M_s$.

5. CONCLUSIONS

The transient analysis carried out in the paper shows that the hybrid air-to-water heating system performs much better when coupled with a rock bed air heater than when coupled to a conventional empty channel air heating collector. The storage tank temperature in the hybrid system is observed to be higher at higher rates of water and air flow. Since the pressure drop experienced by the water and the air recirculating through the system is not very large, the effective energy gain of the system also increases with an increase in the rates of flow of the fluid. However, for a fixed air flow rate, no significant improvement in the system’s performance can be achieved by increasing the water flow rate to a value more than twice that of the flow rate of air, i.e., for $M_w > 2M_a$. 

Fig. 10. Variation of effective energy gain with time by storage tank water (W) and absorber plate (A) in the conventional air heater for different air flow rates.

Fig. 11. Variation of effective energy gain with time by storage tank water (W) and rock bed (R) in the rock bed air heater for different air flow rates.

Fig. 12. Variation of storage tank water temperature with time for different water mass in the storage tank.

Fig. 13. Variation of effective energy gain by water (W) and rock bed (R) with time for different water mass in the storage tank.
NOMENCLATURE

\( A \) = surface area (m\(^2\))
\( C \) = specific heat capacity (J/kg °C)
\( d \) = center to center distance of tubes in shell (m)
\( D \) = equivalent diameter (m)
\( f \) = friction factor
\( G \) = mass flow velocity (kg/h m\(^2\))
\( h \) = heat transfer coefficient (W/m\(^2\) °C)
\( j \) = Colburn’s heat transfer factor
\( k \) = conductivity of air (W/m °C)
\( K \) = correction factor
\( L \) = length (m)
\( m \) = specific mass flow rate (kg/h m\(^2\))
\( M \) = mass flow rate (kg/h)
\( M \) = mass (kg/m\(^3\)) (kg for water in the storage tank)
\( n \) = number of flow passes
\( p \) = pitch (m)
\( \Delta p \) = pressure drop (Pa)
\( R \) = dirt or fouling factor (°C m\(^2\)/W)
\( Re \) = Reynolds number
\( s \) = specific gravity
\( t \) = time (s)
\( T \) = temperature (°C)
\( T(1) \) = temperature at the starting of system operation (°C)
\( V \) = velocity (m/s)
\( \alpha \) = solar absorbance
\( \tau \) = solar transmittance
\( \rho \) = density (kg/m\(^3\))
\( \mu \) = viscosity (kg/ms)
\( \varepsilon \) = rock bed porosity
\( \Omega \) = fin efficiency.

Subscripts
1 = outer cover
2 = inner cover
3 = absorber plate in conventional air heater; base (wooden) plate in rock bed air heater
4 = base (wooden) plate in conventional air heater
a = air
A = ambient
C = conventional
d = dirt (fouling)
f = fin
h = heat exchanger
I = inlet to heat exchanger
o = outer wall/outlet
IN = inlet to air heater
K = rock bed
r = radiative
s = shell/storage tank
t = tubes in shell
v = volumetric
W = water.

REFERENCES


APPENDIX

Fig. A1. \( \Omega \) as function of the finned-tube parameter.

Fig. A2. \( J_a \) and \( f_a \) as functions of \( Re_a \) and \( Re_r \), respectively.
Fig. A3. $h_w$ as a function of $V_w$.

Fig. A4. $f_w$ as a function of $Re_w$. 