Advance Physics Letter

A Technology to Improve Air-Heating Performance of Flat Metallic Solar Collector

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Abstract — In this paper, two kinds of solar collectors, namely the flat plate (FP) and the transpired plate (TP) as an improved version of flat plate, made-up of blackened aluminum sheet, are analyzed and compared. The transpired plate solar collector can be operated more efficiently for air heating. To evaluate the effects of two solar collectors for air heating simplified steady state mathematical models are developed to calculate useful heat delivered, air temperature rise as well as the system efficiencies. Through comparison, it is found that transpired type is a better design in terms of heat transfer. For instance, systems that operate at $500\,\mathrm{W\,m^{-2}}$ of solar radiation intensity, $300\,\mathrm{m^3 hr^{-1}}$ of airflow rate and air temperature of 293K give 3K and 8K in air temperature rise for flat and transpired plates respectively. Useful heat delivered and efficiency of transpired plate collector is comparatively high for transpired plate collector. The transpired plate collector has therefore more potential for air heating and incorporation of transpired plate system will result better quality in solar energy management for air heating applications.

Keywords: flat plate, transpired plate, heat flux, temperature.

I. INTRODUCTION

Two types of plates are selected in this study, i.e. flat and transpired plates for heating of air. Heat removal from habitation by using a metallic solar wall (MSW) [1] and mathematical modeling and thermal performance analysis of unglazed transpired solar collectors (UTC) [2] have already been studied.

Figure-1 illustrates the side views of the heating systems. The plates were made of black painted aluminum sheets. They are used as solar collectors that absorb heat from the solar radiation. Thermal performances of these systems were investigated through mathematical models. The energy balance equations were established based on steady state one-dimensional heat transfer. In addition, heat transfer mechanisms including the thermal performance analyses for both designs were discussed in further sections. Finally in the section 3, the thermal performances between these two designs are compared.

II. MATHEMATICAL MODEL

A simplified steady state mathematical model is developed to calculate temperature rise of the air, as well as the system efficiencies and useful heat delivered [3,4,5]. Therefore, the temperature equations are obtained by writing the energy balance equations in a matrix form, and solved by the matrix inversion method. The matrix algorithm is carried out using the MATLAB program.

2.1Mathematical model of flat plate collector

2.1.1 Assumptions of heating system with flat plate collector:

Some assumptions have been made while developing the mathematical model. They are as follows:

(i) The airflow rate is assumed to be constant throughout the plenum.

(ii) There is no heat loss through the side walls that are attached with aluminum plate and insulate wall into conditioned space.

(iii) All the temperatures that are used in the energy balance equations are assumed to be constant temperatures after reaching the steady stage.

2.1.2 Energy balance equations for flat plate collector

Equation (1) shows the energy balance equations for the system under a steady state and the related parameters are as shown in Figure-2:

$$\dot{m}c_p(T_{out} - T_{in}) = I\alpha_p A_p - Q_c - Q_r \qquad (1)$$

The left-hand side of this equation represents the useful energy collected. The first term on the right-hand side is the solar energy absorbed by the aluminum black plate. The second and third terms are the heat losses to the surrounding via radiation and convection respectively.

The flat plate:

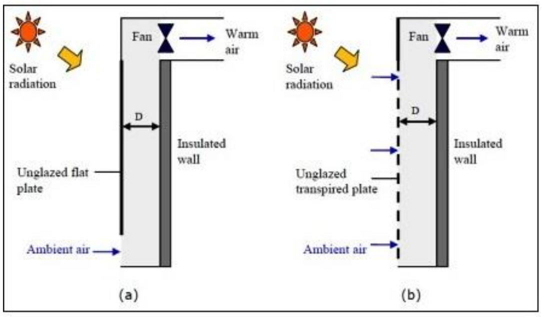

Fig. 1: Side views of schematic diagrams of heating systems with unglazed (a) flat and (b) transpired plates as solar collector.

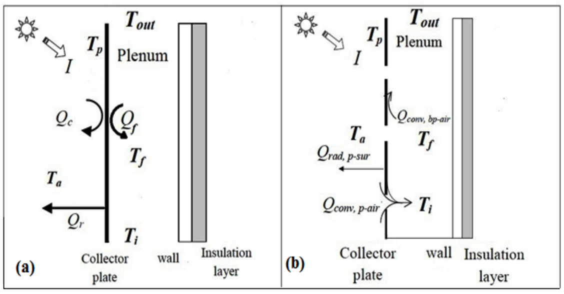

Fig. 2: Heat transfer of plate solar collector (a) flat plate (b) transpired plate

The fan-assisted system draws the ambient air through the opening at the bottom into the plenum. Energy balance equation of the plate is as shown in Equation (2).

$$I\alpha_p A_p = Q_c + Q_r + Q_f \qquad (2)$$

The plenum air:

The ambient air is heated by the back-of-plate while the air is flowing throughout the plenum. The energy balance equation of the air inside the plenum is shown in Equation (3).

$$\sum \dot{m} c_p (T_{out} - T_{in}) = Q_f \qquad (3)$$

Heat flux equations:

From the above equations, heat flux equations for the plate and plenum can be rewritten as:

The flat plate:

$$S = h_c(T_p - T_a) + h_f(T_p - T_f) + h_r(T_p - T_a) \qquad (4)$$

Where $T_a = T_{in}$ and $S = I\alpha_p$

The plenum air:

$$M\,(T_{out} - T_a) = h_f(T_p - T_f), \qquad (5)$$

Where $M = (\sum \dot{m} c_p)/A_p$; $A_p = HL$; $\dot{m} = \rho_f v_f DL$ and $v_f = V_{fan}/DL$ $T_f = (T_{out} + T_{in})/2$

Equations (4) and (5) can be simplified to Equations (6) and (7) respectively, and form a 2x2 matrix equation (8) as follows:

$$(h_c + h_f + h_r)T_p - h_f T_f = S + h_c T_a + h_r T_a \quad (6)$$

$$h_f T_p - (h_f + 2M)T_f = -2MT_a \qquad (7)$$

$$\begin{bmatrix} h_c + h_f + h_r & -h_f \\ h_f & -(h_f + 2M) \end{bmatrix} \begin{bmatrix} T_p \\ T_f \end{bmatrix} = \begin{bmatrix} S + h_c T_a + T_r T_a \\ -2MT_a \end{bmatrix} \quad (8)$$

Equation (8) is then solved by using matrix inversion method and the iteration process is continued until the convergence value is smaller than 10^{-6}

2.1.3. Heat transfer coefficients

The radiation heat transfer coefficient:

The radiation coefficient is given in Equation (10) where the emissivity (ε_p) of black paint is taken as 0.95 [6] (Incropera, 2002).

$$q_r = h_r(T_p - T_a) = \sigma\varepsilon_p(T_p^4 - T_a^4) \qquad (9)$$

$$h_r = \sigma\varepsilon_p(T_p + T_a)(T_p^2 + T_a^2) \qquad (10)$$

The convection heat transfer coefficient from the plate to the ambient:

The natural convection heat transfer coefficient from the plate to the ambient air is as shown in Equations (11) where for turbulent flow and Prvalues near to 1.0, correlation between the average $Nu_{c,fp}$ and the $Ra>10^9$ are given as Equation (12) [7](Jaluria, 2003):

$$h_c = Nu_{c,fp} K_f / H \qquad (11)$$

$$Nu_{c,fp} = \left\{0.825 + \frac{0.387\,Ra^{1/6}}{[(1+(0.492/Pr)^{9/16})^{8/27}]}\right\}^2 \qquad (12)$$

$$\text{and, } Ra = 9.807\beta H^3 (T_p - T_a)/K_f v_f \qquad (13)$$

The heat transfer coefficient from back-of-plate to the air in the plenum:

The coefficient is correlated with the $Nu_{f,fp}$ as follows:

$$h_{c,fp} = Nu_{c,fp} K_f / H \qquad (14)$$

Where $Nu_{c,fp} = 0.664 Re_{f,fp}^{1/2} Pr^{1/3}$

Plenum geometry with air gap dimension (D) over 10 cm and air velocity lower than $0.5\,m\,s^{-1}$, the free convection proportion cannot be neglected even with fan-driven forced ventilation. It shout be treated as mixed convection condition, which the $Re_{f,fp}$ should consists both from free and forced convections as in Equation (15) [8](Eicker, 2003).

$$Re_{f,fp} = \sqrt{Re_{free}^2 + Re_{forced}^2}\ , \qquad (15)$$

Where $Re_{free} = \sqrt{\frac{Gr}{2.5}}$, and $Re_{forced} = v_f \rho_f H/\mu_f$

Due to the fact that the maximum air velocity for present study is taken $0.69\,ms^{-1}$; the flows are treated as laminar modes ($Re<2x10^{-5}$) for all the operation conditions of measurements. In addition, the air velocity in the plenum is taken at D/2 and hence the measured velocity was the maximum velocity for laminar flow. The mean plenum air velocity can be calculated by using Equation (16). For flow between two parallel plates [9] (Nakayama, 2000),

$$v_f = (1/1.5)v_{max} \quad (16)$$

Solar radiation:

The solar radiation intensity that absorbed by the plate is given as below where the absorptivity (α_p) for black paint is taken as 0.95 [10] (ASHRAE, 2009):

$$S = I\alpha_p \quad (17)$$

System efficiency:

The efficiency of the heating system is the ratio of the useful energy delivered to the total solar energy input on the plate, and it is given as:

$$\eta = \sum \dot{m} c_p (T_{out} - T_a)/IA_p \quad (18)$$

2.2 Heat transfer of transpired solar collector

2.2.1. Assumptions of heating system with transpired plate collector

Due to the complication of heat transfer process of the unglazed transpired plate, some assumptions have been made as follows:

(i) The radiation heat loss over the surface of the transpired plate is everywhere constant as this has been proved that it has a modest effect on the flow distribution [11].

(ii) There is no heat loss through the insulated wall and side walls that attached with the aluminum plate.

(iii) The convection losses to the ambient are negligible and have been verified by previous study [12].

(iv) No reverse flow over the plate as the face velocities in this study are taken higher than $0.0125 ms^{-1}$ [11].

(v) The temperatures that are taken in the study which have the same height with the first row of holes from the bottom are taken as air temperatures reach the plenum after passing through the holes.

(vi)The air properties maintain the same throughout the plenum.

2.2.2 Energy balance equations for transpired plate collector

Energy balance equations are established for two components of the system, i.e. unglazed transpired plate and the air in the plenum. The heat transfers of the system are as shown in Figure 2(B).

The unglazed transpired collector:

The fans-assisted system draws the ambient air through the holes so that heat, which wouldotherwise be lost by convection, is captured by the airflow into the plenum. Thus, there will have very small amount of convection heat loss to the ambient which indeed can be neglected [2,13,14] (Augustus, 2007, Dymond, 1997, Kutscher, 1993). Energy balance equation on an unglazed transpired plate is as shown in Equation (19).

$$I\alpha_p A_p = Q_{conv,p-air} + Q_{conv,bp-air} + Q_{rad,p-sur} \quad (19)$$

The plenum air:

The ambient air which heated by the front and hole of the transpired plate is further heated by the back-of-plate when flowing throughout the plenum. Energy balance equation of the air in the plenum is shown in Equation (20)

$$\Sigma \dot{m} c_p (T_{out} - T_i) = Q_{conv,bp-air} \quad (20)$$

2.2.3 Heat flux and coefficient of heat transfer for transpired plate collector

Convection heat transfer from the front and hole of transpired plate to the ambient air:

The convection heat transfer equations can be written in term of mass flow rate and heat flux as shown in Equations (21) and (22) respectively.

$$Q_{conv,p-air} = \Sigma \dot{m} c_p (T_i - T_a)/A_p \quad (21)$$

$$Q_{conv,p-air} = h_{c,tp}(T_p - T_i) \quad (22)$$

The convection coefficient is defined as Equation (23) where the $Nu_{c,tp}$ correlation is taken from reference [15].

$$h_{c,tp} = Nu_{c,tp} \times \frac{k_a}{d} \quad (23)$$

Convection heat transfer from the back of the transpired plate to the air in the plenum:

The coefficient is shown in Equation (24) and (25) and the$Nu_{f,tp}$ correlation is taken from referees [15].

$$q_{convb,p-air} = h_{f,tp}(T_p - T_f) \quad (24)$$

Where $T_f = (T_i + T_{out})/2$

$$h_{f,tp} = Nu_{f,tp} \times \frac{k_f}{H} \quad (25)$$

Radiation heat transfer from the front of transpired plate to the surrounding:

The radiation coefficient is the same as discussed in earlier section Equation (10).

Solar radiation:

The solar radiation intensity that absorbed by the transpired plate is given as Equation (17).

System efficiency:

The efficiency of the heating system is given as in Equation (18).

Heat exchange effectiveness:

The heat exchange effectiveness of the solar collector is defined as the ratio of the actual temperature rise of air to the maximum possible temperature rise (Kutscher, 1994):

$$\varepsilon_{HX} = \frac{(T_{out} - T_a)}{(T_p - T_a)} \quad (26)$$

substituting Equations (10) (22) and (24) for equation (19), the heat balance equation for the unglazed transpired collector gives:

$$(h_{c,tp} + h_{f,tp} + h_r)T_p - \left(h_{c,tp} + \frac{h_{f,tp}}{2}\right)T_i - (h_{f,tp}/2) \quad T_{out} = I\,\alpha_p + h_r T_a \quad (27)$$

For the plenum air, substituting Equation (23) for Equation (20) gives:

$$h_{f,tp} + (G\,c_p - h_{f,tp}/2)T_i - \left(G\,c_p + \frac{h_{f,tp}}{2}\right)T_{out} = 0, \quad (28)$$

Where $G = \Sigma\dot{m}c_p/A_p$ and $T_f = (T_i + T_{out})/2$

finally, Equations (21) and (22) are combined and give:

$$h_{c,tp}T_p - (\,G\,c_p + h_c)\,T_i = -Gc_pT_a \quad (29)$$

Thus, Equations (27) to (29) can be written in a 3x3 matrix form:

$$\begin{bmatrix} (h_{c,tp} + h_{f,tp} + h_r) & -(h_{c,tp} + h_{f,tp}/2) & -(h_{f,tp}/2) \\ h_{f,tp} & (G\,c_p - h_{f,tp}/2) & -(G\,c_p + h_{f,tp}/2) \\ h_{c,tp} & -(G\,c_p + h_c) & 0 \end{bmatrix} \quad (30)$$

Physical properties of air:

The required physical properties of air are calculated by using linear interpolation for air properties between suitable temperature ranges. [6]

TABLE-1: PARAMETERS FOR THE STUDY

Parameter	Value/ range
Solar radiation intensity(Wm^{-2})	300-800
Volume air flow rate(m^3hr^{-1})	300
Plenum depth (m)	0.20-0.30
Porosity (ratio of hole area to total surface area), %	0.84
Height of the transpired plate(m)	2.0
Width of the transpired plate(m)	1.0
Area of the transpired plate (m^2)	2.0
Plate thickness(m)	0.01

III. THEORETICAL RESULTS

In this section, the thermal performances of both flat and transpired plates are evaluated by using the models that have been developed. The input parameters for both plates are the same, i.e. solar radiation intensity, ambient air temperature, airflow rate, plate area and plenum depth.

As shown in Figure 3, In contrast to the flat plate, transpired type gives higher rise in air temperature. This indicates that the transpired plate has better heat transfer compared to the flat plate. In order to further investigate the heat losses of the plates, Table -2 shows the fractions of heat fluxes of the plates. The useful heat flux that delivered by the system with flat plate is only about 30%of total solar heat flux that been absorbed by the plate while for the transpired plate it is about 84%. These represent the total heat losses for flat plate is about 70% while for transpired plate is only 20% of total energy absorbed. Table-2 shows the rises in heat fluxes of flat plate when the solar radiation intensity is increased from 300 to $800Wm^{-2}$ at a constant airflow rate of $300\,m^3hr^{-1}$. The losses to the ambient are increased and particularly the $q_{r,fp}$ has the greatest increase. As for the $q_{f,fp}$, though it is increased with the solar radiation intensity, the amount of rise is smaller than the losses (sum of $q_{r,fp}$ and $q_{c,fp}$). These explain why only small rise in air temperature occurs. On the other hand, for transpired plate, the rise in $q_{c,tp}$ and $q_{f,tp}$ are greater than the $q_{r,tp}$. This results in great increase of $q_{d.tp}$(sum of $q_{c,tp}$ and $q_{f,tp}$). The heat losses to ambient of flat plate are partly caused by convection heat loss while for transpired plate, the ambient convection heat loss can be ignored, whereby most of the heat which indeed would loss to the ambient has been sucked into the plenum through the small holes. This reduces the total heat losses and hence has better heat transfer with transpired design.

In terms of System efficiency as shown in figure (3); at various solar radiation intensities, the efficiencies for both plates are almost constant and high for transpired plate collector system.

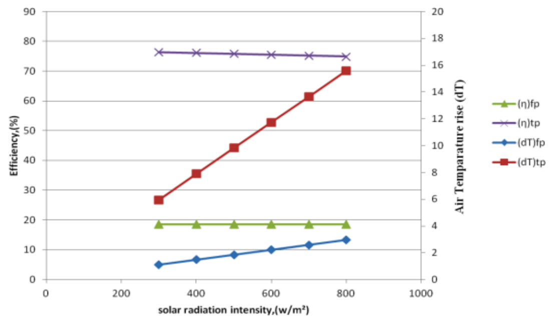

Fig. 3: Typical performance (air temperature rise and efficiencies) results at different solar radiation intensity

IV. CONCLUSION

In this study the results of heating systems with flat and transpired plate are discussed. Mathematical model for the thermal performance has been developed. The main findings can be summarized as follows: Transpired type is a better design in terms of heat transfer. For instance, systems that operate at $500W/m^2$ of solar radiation intensity, $300m^3hr^{-1}$ of airflow rate and air temperature of 293K give 3K and 8K in air temperature rise for flat and transpired plates respectively. At various solar radiation intensities, the efficiencies for both plates are almost constant and high for transpired plate as shown in graph. Useful heat delivered to the air is high for transpired plate heating system for same operating conditions. Thus it can be concluded that the transpired plate collector has more potential for air heating and incorporation of transpired plate system will result better quality in solar energy management for air heating applications.

Here in the table-2, S is the total solar heat flux absorb by the plate and the heat fluxes that based on previously used equations in unit of $W\ m^{-2}$ (their values in percentage with respect to S is shown in brackets) where, $q_{r,fp}$ and $q_{c,fp}$ are the losses to ambient through radiation and convection by flat plate respectively. $q_{f,fp}$ is the convection of back-of-plate which is also the useful heat flux that delivered by flat plate. Similarly $q_{r,tp}$ is radiation heat loss; $q_{d,tp}$ is the sum of $q_{c,tp}$ and $q_{f,tp}$, also the useful heat flux that delivered by the transpired plate.

TABLE 2:HEAT FLUXES OF FLAT AND TRANSPIRED PLATES

I (Wm^{-2})	V_{fan} (m^3hr^{-1})	T_a (K)	D (m)	S(Wm^{-2})	$q_{r,fp}$ (%)	$q_{c,fp}$ (%)	$q_{f,fp}=q_{d,fp}$ (%)	$q_{r,tp}$ (%)	$q_{c,tp}$ (%)	$q_{f,tp}$ (%)	$q_{d,tp}$ (%)
300	300	293	0.20	291.7 (100%)	123.3 (42.3)	79.6 (27.3)	88.7 (30.4)	46.6 (16.0)	136.8 (46.9)	108.3 (37.1)	245.1 (84.0)
800	300	293	0.20	779.0 (100%)	312.1 (40.1)	222.8 (28.6)	244.1 (31.3)	122.6 (15.7)	337.5 (43.3)	318.9 (40.9)	656.4 (84.3)
300	300	293	0.25	291.7 (100%)	125.0 (42.9)	80.9 (27.7)	85.7 (29.4)	43.7 (15.7)	128.8 (46.0)	119.1 (38.3)	247.9 (84.3)
800	300	293	0.25	779.0 (100%)	314.7 (40.4)	224.7 (28.8)	239.6 (30.8)	120.0 (16.0)	330.4 (44.0)	328.8 (39.9)	659.2 (84.0)
300	300	293	0.30	291.7 (100%)	126.0 (43.2)	81.7 (28.0)	83.9 (28.8)	43.7 (15.0)	128.8 (44.2)	119.1 (40.8)	247.9 (85.0)
800	300	293	0.30	779.0 (100%)	316.2 (40.6)	225.8 (29.0)	237.0 (30.4)	119.7 (15.4)	330.4 (42.4)	328.8 (42.2)	659.2 (84.6)

No.	Nomenclature / Greek Symbol / Suffix	Description
Nomenclature		
1	A_P	Surface area of plate (m^2)
2	c_p	Heat capacity ($Jkg^{-1}K^{-1}$)
3	D	Plenum depth (m)
4	g	Acceleration due to gravity (m s–2)
5	G	suction mass flow rate per unit area ($kg\ s^{-1}m^{-2}$)
6	Gr	Grasoff number
7	h	Heat transfer coefficient ($W\ m^{-2}K^{-1}$)
8	H	Plenum height (m)
9	I	Solar irradiance ($W\ m^{-2}$)
10	K	Thermal conductivity ($W\ m^{-1}K^{-1}$)
11	L	Width of aluminum plate (m)
12	m	Mass flow rate (kg s^(-1))
13	Nu	Nusselt number
14	P	Plate porosity
15	Pr	Prandtl number
16	q	Heat flux (Wm^{-2})
17	q_d	Useful energy heat flux(Wm^{-2})
18	Q	Heat (W)
19	Re	Reynold’s number
20	Ra	Rayleigh number
21	S	Solar radiation heat flux absorbed by the plate(Wm^{-2})
22	T	Temperature (K)
23	v_f	Plenum air velocity
24	V_{fan}	Volume airflow rate (m^3hr^{-1})

Greek Symbol		
1	α_p	Absorptivity
2	Β	Coefficient of volumetric expansion (m^{-1})
3	ρ	Air density (kgm^{-3})
4	σ	Stefan–Boltzmann constant
5	ε	Emissivity
6	ε_{HX}	Heat exchange effectiveness (%)
7	Η	Efficiency (%)
8	μ	Dynamic viscosity (Nsm^{-1})
Suffix		
1	a	Ambient air.
2	bp-air	Between back-of-plate of transpired plate and plenum air
3	c; conv	Convection
4	c,fp	Convection between flat plate and ambient air
5	c,tp	Convection between transpired plate and ambient air
6	EX	Heat exchange
7	f	Convection between back-of-plate an plenum air; fluid(air)
8	fp	Flat plate
9	f,fp	Convection between back of flat plate and plenum air
10	free	Free convection
11	forced	Forced convection
12	f,tp	Convection between back of transpired plate and plenum air.
13	i	Air entering into plenum; air temperature at bottom of plenum ,K (transpired plate)
14	in	Inlet
15	out	Outlet
16	p	Plate
17	p-air	Between transpired plate and ambient air
18	p-sur	Between plate and surrounding
19	r	Radiation heat transfer from plate to surrounding
20	rad	Radiation heat transfer from plate to surrounding
21	t_p	Transpired plate

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REFERENCES

[1] J.Hirunlabh, W.Kongduang, P.Namprakai, J.Khedari. Study of natural ventilation of houses by ametallic solar wall under tropical climate. Renewable Energy 1999; 18:109-119.

[2] L.M. Augustus, S. Kumar. Mathematical modeling and thermal performance analysis of unglazedtranspired solar collectors. Solar Energy 2007; 81:62–75.

[3] M. Maerefat, A.P. Haghighi. Natural cooling of stand-alone houses using solar chimney andevaporative cooling cavity. Renewable Energy 2010; 35:2040–2052.

[4] K.S. Ong. Thermal performance of solar air heaters: mathematical model and solution procedure. SolarEnergy,1995; 55:93–109.

[5] X.Q. Zhai, Y.J. Dai, R.Z. Wang. Experimental investigation on air heating and natural ventilation of asolar air collector. Energy and Buildings 2005; 37:373–381.

[6] F.P. Incropera, D.P. Dewitt. Fundamentals of Heat and Mass Transfer, 5th ed., John Wiley & Sons Inc,2002.

[7] Jaluria, Yogesh. Natural Convection: In Heat Transfer Handbook, ed. Adrian Bejan, Allan D.Kraus. New Jersey: John Wiley & Sons, Inc,2003

[8] U. Eicker. Solar Technologies for Buildings. West Sussex: John Wiley & Sons Ltd, 2003.

[9] Y.Nakayama. Flow of viscous fluid: In Introduction to Fluid Mechanics, ed. R. F. Boucher.Oxford: Butterworth-Heinemann, 2000.

[10] ASHRAE. ASHRAE Handbook: Fundamentals, SI ed., American Society of Heating, Refrigeratingand Air-Conditioning Engineers Inc., Atlanta, 2009.

[11] L.H. Gunnewiek, E. Brundrett, K.G.T. Hollands. Flow distribution in unglazed transpired plate solarair heaters of large area. Solar Energy1996; 58: 227–237.

[12] C.F. Kutscher. Heat exchange effectiveness and pressure drop for air flow through perforated

plateswith and without crosswinds. Journal of Heat Transfer 1994; 116:391–399.
[13] C. Dymond, C. Kutscher. Development of a flow distribution and design model for transpired solarCollectors. Solar Energy 1997; 60:291–300.
[14] C.F. Kutscher, C.B. Christensen, G.M. Barker. Unglazed transpired solar collectors: heat loss theory.ASME Journal of Solar Engineering 1993; 115:182–188.
[15] H.Y.Chan, S. Riffat, J. Zhu.Solar facades for heating and cooling in buildings. In SET –8th International Conference on Sustainable Energy Technologies. Aachen, Germany,2009
[16] D. Mishra, N.K. Saikhedkar. Evacuated U-Tube Solar Water Heating System - A Descriptive Study, International Journal of Innovative Research in Science, Engineering and Technology", 3 (5): 12627-12635, May 2014.
[17] D. Mishra, N.K. Saikhedkar.A Study and Theoretical Analysis of Evacuated Tube Collectors as Solar Energy Conversion Device for Water Heating, Advance Physics Letter, 1 (3): 30-39, 2014.
[18] D. Mishra. Experimental Analysis of Thermal Performance of Evacuated U-Tube Solar Collector, Advance Physics Letters, 2 (3), 2015

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