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PERFORMANCE AND COST ANALYSIS OF DOUBLE FLOW SOLAR AIR HEATERS

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ABSTRACT
This study involves a model to investigate the effect of mass flow, channel depth and collector length on the thermal performance and cost benefit ratio for solar collector in double flow mode with and without porous media by using a developed internet based expert system. The thermal performance was determined over a wide range of operating conditions, and the optimum operating conditions were determined. It is found that for drying purpose the designed air flow rate should be in the range of 0.03 to 0.04 kg/s, and the thermal efficiency increased by 5% in double flow after using porous media. On the other hand it is concluded that the higher in cost energy for any particular combination of flow depth, collector length and mass flow rate is at short collector length, small flow depth with high quantity of mass flow rate. Moreover, the values of duct lengths and depths for which the cost of solar energy is minimized are different for different values of mass flow rates

KEY WORDS
Solar air heaters, Thermal performance, Cost of solar energy, Expert system.

1. Introduction

Extensive investigations have been carried out on the optimum design of conventional and modified solar air heaters, in order to search for efficient and inexpensive designs suitable for mass production for different practical applications. The researchers have give their attention to the effects of design and operational parameters, type of flow passes, number of glazing and type of absorber flat, corrugated or finned on the thermal performance of solar air heaters [1-4]. Additional studies to determine the thermal performance of solar air heaters have been conducted theoretically and/or experimentally, and different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and air [5-9]. However, the availability of a tool to be used for supporting the designs of solar air heaters, application of mathematical models for the analysis of descriptive data from the domain experts would facilitate the task.

A developed internet based expert system to predict the thermal performance for different designs of solar air heaters was presented [10] and [11]. This study uses the aforementioned developed program to find the influence of different parameters, such as mass flow rate, flow channel depth and collector length on the system thermal performance and the cost benefit for flat plate collector in double passes with and without using a porous media.

2. Expert System Approach

A prototype internet based computer program is developed for selection and design of solar air heaters. Dreamweaver combined with the Active Server Pages (ASP) as an extension environment, have been chosen as development software tool. The programmatic code was written by VBScript as scripting language to execute commands on a computer. In order to represent the knowledge for design, rules have been used [11]. The knowledge base contains two different data-bases, a static and dynamic one. The required data are classified into three groups. The first group is called general input data containing the metrological condition, the second group is called collector characteristics which contains all specific manufacturing attributes related to the collector. The third group is called energy characteristics which contain measured data related to the transfer media inside the collector [10]. The aforementioned internet based computer program is in the process of continual development, but it can be access through the website http://www.eng.upm.edu.my/home/bashria/public_html/default.asp

3. Theoretical Analysis

In this section, the heat transfer and the cost factors of the double pass air heaters with and without porous media are discussed, which lead to annual thermal energy gain (ATEG) and annual cost (AC) of the systems. The cross sectional views and the thermal network of the solar air heaters are illustrated in Figure 1. In the two types, the air heaters are composed of three plates, the cover, the absorber and the rear or back plate. The air is flowing in the upper channel depth between the cover and the absorber plate and then it is turned to continue flowing in the lower duct between the absorber and the rear plate. The lower duct has been packed with a porous media of 0.80 porosity in type 2. The various heat transfer coefficients at different components of the air heaters are illustrated in the thermal network in Figure 1.

To model the solar air types and obtain there relative equations, a number of simplifying assumptions has been made to lay the foundation without obscuring the basic physical situation. These assumptions are as follows:

1- Performance is steady state
2- There is no absorption of solar energy by a cover insofar as it affects losses from the collector
3- Heat transfer fluid is considered a non-participating medium
4- The radiation coefficient between the two air duct surfaces is found by assuming a mean radiant temperature equal to the mean fluid temperature
5- Loss through front and back are to the same ambient temperature

Therefore the energy balance equations for the plates and the flowing air yields the following equations:

a) Type -1

Collector cover

$$h_1(T_{f1}-T_c)+h_{r1}(T_p-T_c)=U_t(T_c-T_a) \qquad (1)$$

Fluid 1

$$h_2(T_p-T_{f1})=(\frac{\dot{m}C_{p1}}{W})(\frac{dT_{f1}}{dx})+h_1(T_{f1}-T_c) \qquad (2)$$

Absorber

$$I\tau\alpha=h_{r1}(T_p-T_c)+h_2(T_p-T_{f1})+h_3(T_p-T_{f2})+h_{r2}(T_p-T_r) \qquad (3)$$

Fluid 2

$$h_3(T_p-T_{f2})=(\frac{\dot{m}C_{p2}}{W})(\frac{dT_{f2}}{dx})+h_4(T_{f2}-T_r) \qquad (4)$$

Bottom Plate

$$h_4(T_{f2}-T_r)+h_{r2}(T_p-T_r)=U_b(T_r-T_a) \qquad (5)$$

b) Type -2

Collector cover

$$h_1(T_{f1}-T_c)+h_{r1}(T_p-T_c)=U_t(T_c-T_a) \qquad (6)$$

Fluid 1

$$h_2(T_p-T_{f1})=(\frac{\dot{m}C_{p1}}{W})(\frac{dT_{f1}}{dx})+h_1(T_{f1}-T_c) \qquad (7)$$

Absorber

$$I\tau\alpha=h_{r1}(T_p-T_c)+h_2(T_p-T_{f1})+h_3(T_p-T_{f2})+h_{r2}(T_p-T_{pr}) \qquad (8)$$

Fluid 2

$$h_3(T_p-T_{f2})=(\frac{mC_{p2}}{W})(\frac{dT_{f2}}{dx})+h_4(T_{pr}-T_{f2})+h_5(T_{f2}-T_r) \qquad (9)$$

Porous media

$$h_{r2}(T_p-T_{pr})+h_6(T_{pr}-T_r)=h_4(T_{pr}-T_{f2}) \qquad (10)$$

Bottom plate

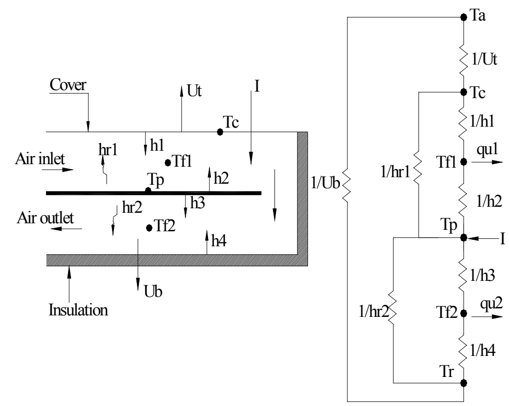
$$h_5(T_{f2}-T_r)+h_6(T_{pr}-T_r)=U_b(T_r-T_a) \qquad (11)$$

The useful heat obtained from the collector can be determined using the following equation:

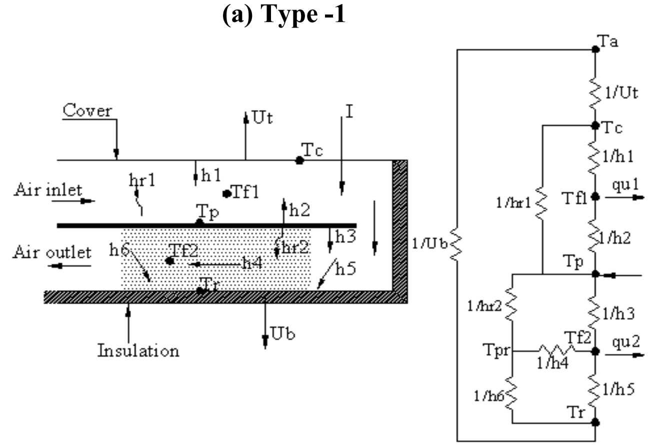
$$Q=\dot{m}C_p(T_o-T_i) \qquad (12)$$

The efficiency of the collector which defined as the ratio of the useful energy to the total incident solar radiation is expressed by

$$\eta=\dot{m}C_p(T_o-T)\big/I \qquad (13)$$



(a) Type -1



(b) Type -2

Figure 1: Schematic diagram of the solar air heaters with thermal network

3.1 Annual Energy Gain

The annual thermal energy gain (ATEG) available from the collector can be obtained by multiplying the useful heat by the number of operating days in a year and the number of hours per day during which useful sunshine is available (Choudhury *et al.,* 1995).

$$ATEG = \dot{m}C_p(T_o - T_i)t_{op} \quad (14)$$

3.2 Annual Cost

In order to estimate the annual cost of the collector (AC) of the solar air different cost factors have to be calculated. This includes the annual pump cost or running cost (ARC), annual capital cost (ACC), annual maintenance cost (MC) and annual salvage value (ASV). The annual running cost is calculated as follows [1] and [2]:

$$ARC = (m_v \Delta p)t_{op}CE \quad (15)$$

Where ΔP is the pressure drop (Pa), t_{op} is the time of operation and CE is the cost of electricity (RM/KWh)

$$\Delta p = f\left(\frac{\dot{m}^2}{\rho}\right)\left(\frac{L}{D}\right)^3 \quad (16)$$

$$f = f_o + y\left(\frac{D}{L}\right) \quad (17)$$

The values of f_o and y are

$f_o = 24/\mathrm{Re}$, $y = 0.9$ for laminar flow (Re<2550)

$f_o = 0.0094$, $y = 2.92\,\mathrm{Re}^{-0.15}$ for transitional flow ($2550<\mathrm{Re}<10^4$)

$f_o = 0.059\,\mathrm{Re}^{-0.2}$, $y = 0.73$ for turbulent flow ($10^4< \mathrm{Re}<10^5$)

The annual capital cost (ACC) is given by the following relations:

$$ACC = CRFxCI \quad (18)$$

$$CI = CC + CSSC + FC \quad (19)$$

$$CRF = i(i+1)^n/[(i+1)^n - 1] \quad (20)$$

$$CC = WL*(X1+Y1) + (2D+W)*LZ1 \quad (21)$$

The annual salvage value (ASV) is given as:

$$ASV = SFFxSV \quad (22)$$

$$SFF = i/[(i+1)^n - 1] \quad (23)$$

$$SV = 0.1CI \quad (24)$$

Where CI is the Capital investment, SFF is the salvage fund factor and SV is the salvage value

Therefore, the annual cost of the collector (AC) is calculated as

$$AC = ACC + MC + ARC - ASV \quad (25)$$

4. Results and Discussion

To compare the performance of the double flow flat plate collector with and without porous media the input data have been entered to give the same configuration. The detailed specification of the entered data for solar air heaters are given in Table 1. The variation of efficiency and outlet temperature with different mass flow double pass with and without porous media are shown in Figures 2 and 3. With the increase in mass flow rate the thermal efficiency of the system increases, in contrast the outlet temperature is decreased with the increase of mass flow rate. By comparing the values of the efficiency and outlet temperature for double pass with and without porous media, it is found that the use of porous media in lower duct increases the system efficiency by 2-6 % and the outlet temperature from 2-5°C.

Table 1: Specification for Solar air Heaters

Collector tilt angle (degree)	10
Collector length (m)	2.5 & 1.5
Collector width (m)	1
Upper depth (m)	0.03
Lower depth (m)	0.03, 0.04, 0,05, 0.06 and 0.07
Plate type	Flat plate
Absorber material	Black steel, α = 0.9 and ε = 0.85
Cover material	Ordinary clear glass, τ = 0.85
Number of cover	1
Insulation material	Fiber glass , k = 0.045 W/m.k
Back insulation thickness (m)	0.05
Edge insulation thickness (m)	0.05
Porous media	Glass-wool of 0.8 porosity for double pass with porous media in lower duct

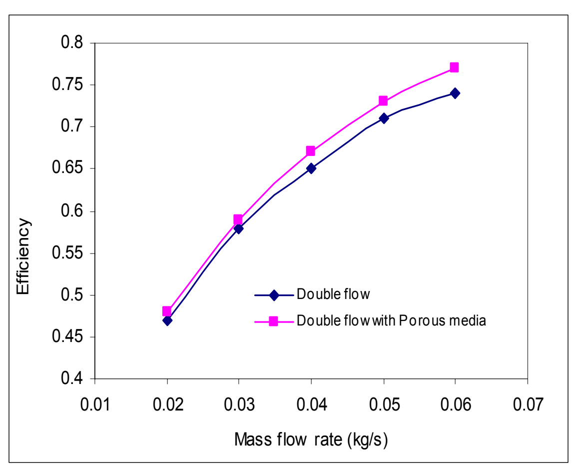


Figure 2: Efficiency Variation with Mass Flow Rate

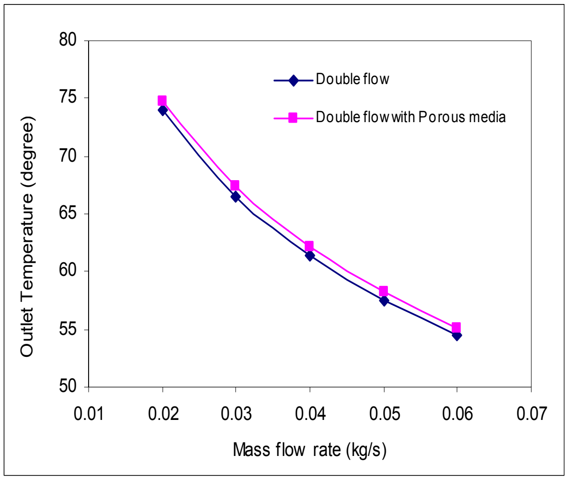


Figure 3: Outlet Temperature Variation with Mass Flow Rate

Figure 4 shows the combined variation of efficiency and outlet temperature with mass flow rate for the two types of solar heater. From these curves, the optimum operating conditions with respect to efficiency and outlet temperature can be determined. The values of these conditions are shown in Table 2. The effect of lower channel depth on the pressure drop is conducted at fixed upper flow depth of 0.03m, length of 2.5m and the optimum mass flow rates at 0.035kg/s for double flow and 0.033kg/s for double flow with porous media. It is found that with the increase of lower channel depth the pressure drop decreased as shown in Figure 5.

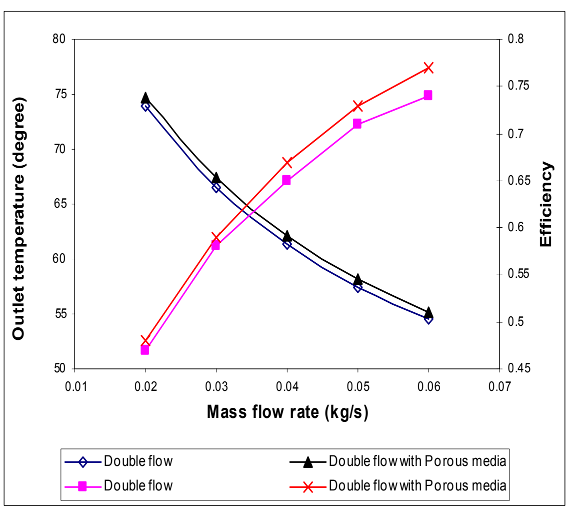


Figure 4: Combined Efficiency and Outlet Temperature Double flow with and without porous media

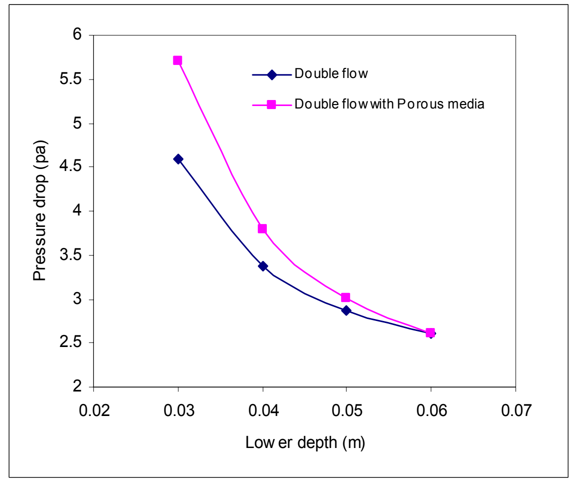


Figure 5: The Variation of Pressure Drop with Lower Channel Depth

Table 2: Designed Conditions

Collector type	Flow rate (kg/s)	Outlet temperature (^oC)	Efficiency
Double flow	0.035	63	0.61
Double flow with porous media	0.033	65	0.64

By using the developed program, the cost of solar energy (i.e. ratio of the annual cost of the collector (AC) / annual thermal energy gain (ATEG) for the two solar air heaters types were computed at different flow depths and mass flow rate. Figure 6 and 7 illustrates the cost of solar energy values as a function of flow depth for double flow mode in flat plate collector with and without porous media at the optimum values of flow rate (0.035kg/s and 0.033kg/s), constant upper channel depth of 0.03m and two different channels length 1.5m and 2.5m. It is shown that the cost of solar energy at first is decreased with the increase of flow depth, and then it is increased steadily with the increase of flow depth. Both Figures 6 and 7 show that the cost of solar energy is a function in collector length; hence the cost is reduced by increasing the collector length. Figure 8 and 9 illustrate the cost of solar energy as a function of flow depth at fixed length 2.5m and different values of mass flow rate for double flow mode with and without porous media respectively. The graphs show that, as the air mass flow rate increases, the cost of solar energy increases, which is becomes the highest at small flow depths at mass flow rate 0.05 kg/s. This is a consequence of the relatively larger rate of pumping cost than the rise of energy with a decrease in duct depth and an increase in air mass flow in the system.

The graphs in Figures 6-9 reveal that the higher in cost energy for any particular combination of flow depth, collector length and mass flow rate is at short collector length and small flow depth with high quantity of mass flow rate. Moreover, the values of duct lengths and depths for which the cost of solar energy is minimized are different for different values of mass flow rates

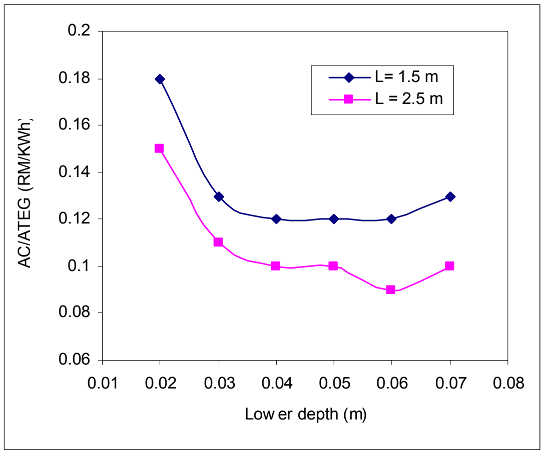


Figure 6: The Variation of AC/ATEG with Respect to Lower Flow Depth in Double Flow Mode.

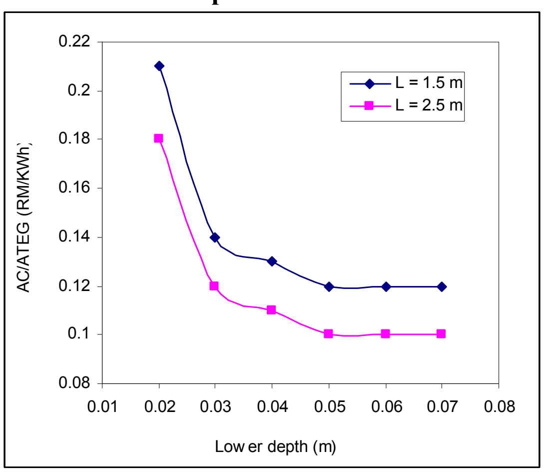


Figure 7: The Variation of AC/ATEG with Respect to Lower Flow Depth in Double Flow Flat Plate Collector with Porous Media.

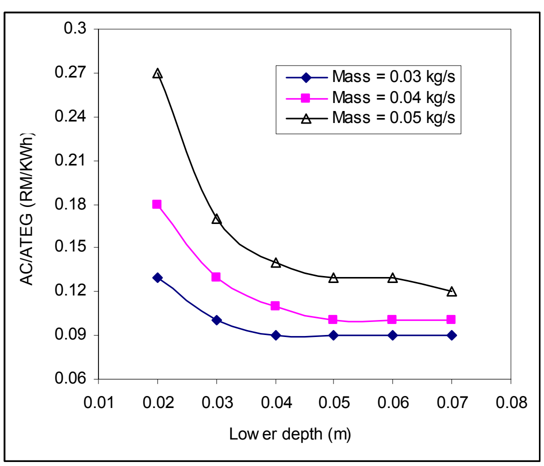


Figure 8: The variation of AC/ATEG with respect to lower flow depth in double flow flat plate collector.

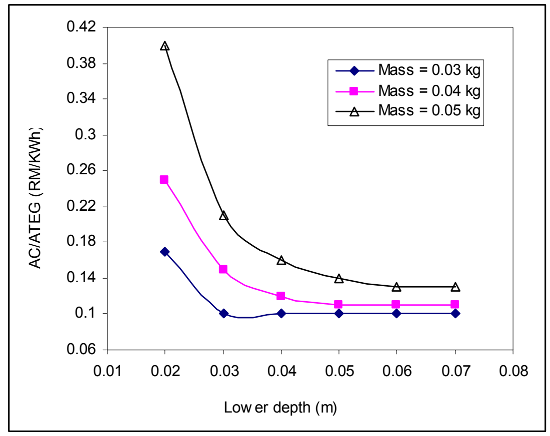


Figure 9: The variation of AC/ATEG with respect to lower flow depth in double flow flat plate collector with porous media.

5. Conclusion

Development of an internet based expert system technique to help in predicting the thermal performance according to suitable cost for of solar air heaters is presented. From the results obtained it can be concluded that:

- Increasing the mass flow rate through the solar air heaters results in increasing the efficiency as well as decreasing the outlet temperature, the estimation of optimum operating conditions with respect to required efficiency and outlet temperature can be predicted by using the developed program. Which is for drying purpose the designed air flow rate would be in the range of about 0.03- 0.04 kg/s, this range of air flow gives an outlet temperature suitable for most agricultural drying applications and the corresponding efficiency is considered reasonable.
- The system thermal efficiency increases by 5% after using porous media in the lower channel than the double flow without porous media. At the same time the pressure drop will be increased, thus increasing the pumping power expanding by about 3-4 times
- The cost of solar energy; the annual cost of the collector/the annual thermal energy gain (AC/ATEG) in double flow duct double duct flat plate collector with porous media is less than the (AC/ATEG) in double flow duct double duct flat plate collector without porous media due to the higher useful energy gained from the using of porous media which increase the heat transfer area.

Nomenclatures

ACC	The annual capital cost, RM
CC	The cost of the collector array, RM
CI	The capital investment, RM
C_p	Specific heat of working fluid, J/kg K
CRF	The capital recovery factor
CSSC	The collector support structure cost, RM
D	Flow channel depth, m
FC	The fabrication cost, RM
h	Fluid heat transfer coefficient, W/m^2 K
h_r	Radiation heat transfer coefficient, W/m^2 K
I	Solar radiation, W/m^2
i	The interest rate, assumed as 8%
L	Collector length, m
$\dot{m}$	Collector flow rate, kg/s
m_v	The volumetric rate, m^3/sec
MC	The maintenance cost, considered to be 10% of the ACC.
n	The collector life, assumed as 10 years
Q	Rate of useful energy gain, W
T_a	Ambient air temperature, K
T_c	Cover temperature, K
T_f	Fluid temperature, K
T_i	Fluid inlet temperature, K
T_o	Outlet temperature, K
T_p	Absorber plate surface temperature, K
T_{pr}	Porous media temperature, K
T_r	Bottom plate temperature, K
U_b	Back loss coefficient, W/m^2 K
U_t	Top loss coefficient, W/m^2 K
W	Collector width, m
X1	Cost of the absorber plate, RM/m^2
Y1	Cost of the cover, RM/m^2
Z1	Cost of side wall + bottom insulation, RM/m^2

Greek symbols

τ	Solar transmittance of glazing
α	Solar absorptance of collector plate
ρ	Density, kg/m^3

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