

# Optimal thermohydraulic performance of a wire mesh packed solar air heater

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## Abstract

This paper is concerned with thermohydraulic investigations on a packed bed solar air heater having its duct packed with blackened wire screen matrices of different geometrical parameters (wire diameter and pitch). The thermohydraulic performance of an air heater in terms of effective efficiency is determined on the basis of actual thermal energy gain subtracted by the primary energy required to generate power needed for pumping air through the packed bed. Based on energy transfer mechanism in the bed, a mathematical model is developed to compute effective efficiency. A design criterion is also suggested to select a matrix for packing the air flow duct of a solar air heater which results in the best thermal efficiency with minimum pumping power penalty. Resulting values of effective efficiency clearly indicate that the packed bed solar air heater investigated is thermohydraulically efficient as compared to flat plate collectors.

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*Keywords:* Solar air heater; Packed bed; Thermohydraulic performance; Matrix; Wire screen matrix

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## 1. Introduction

Flat-plate solar collectors are extensively used in low temperature energy technology and have attracted the attention of a large number of investigators. Several designs of solar air heaters have been developed over the years in order to improve their performance. Use of porous packings inside the duct, such as glass beads (Hasatani et al., 1985), iron

turnings (Cheema and Mannan, 1979), crushed glass (Collier, 1979), slit and expanded aluminium foils (Chiou et al., 1965; Shoemaker, 1961), wire mesh screens (Varshney and Saini, 1998; Ahmad et al., 1996; Prasad and Saini, 1993; Sharma et al., 1991; Beckman, 1968), hollow spheres (Swartman and Ogunlade, 1966), iron shavings and wires (Singh, 1978), iron chips, aluminium chips and pebbles (Misra and Sharma, 1981), and Raschig rings made of hard plastic (Demirel and Kunc, 1987) are some of the common reported methods to enhance the thermal performance of a solar air heaters. There has been significant interest in solar collectors with

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**Nomenclature**

|            |  |                 |   |
|------------|--|-----------------|---|
| $A_c$      | collector plate area, $m^2$  | $R_1$           | radiosity at the bottom surface of lower glass cover, $W/m^2$ |
| $A_f$      | frontal area of collector bed, $m^2$   | $R_2$           | radiosity at the bottom plate, $W/m^2$                        |
| $a_v$      | heat transfer area per unit volume of bed, $m^{-1}$                            | $R_y$           | radiosity at a distance $y$ from top surface, $W/m^2$         |
| $C$        | conversion factor  | $Re_p$          | packed bed Reynolds number ( $=2G_o D_e / 3(1 - P)\mu$ )      |
| $c_p$      | specific heat of air, $kJ/(kg\ K)$   | $r_c$           | reflectivity of glass cover                                   |
| $D$        | depth of bed, $m$  | $r_h$           | hydraulic radius ( $=Pd_w/4(1 - P)$ ), $m$                    |
| $D_e$      | equivalent diameter of particle ( $=6/a_v$ ), $m$                              | $St$            | Stanton number ( $=h_c/(c_p G_o)$ )                           |
| $d_w$      | wire diameter of screen, $m$   | $t_a$           | ambient temperature, $^{\circ}C$                              |
| $f_p$      | friction factor in packed bed  | $t_b$           | bed temperature, $^{\circ}C$                                  |
| $G$        | air mass flow rate per unit collector area, $kg/(s\ m^2)$                      | $t_g$           | air temperature, $^{\circ}C$                                  |
| $G_o$      | mass velocity of air ( $=\frac{m}{A_f P}$ ), $kg/(s\ m^2)$                     | $t_i$           | air inlet temperature, $^{\circ}C$                            |
| $h_c$      | convection heat transfer coefficient between air and matrices, $W/(m^2\ K)$    | $t_o$           | air outlet temperature, $^{\circ}C$                           |
| $h_v$      | volumetric heat transfer coefficient, $W/(m^3\ K)$                             | $t_p$           | temperature of packing material, $^{\circ}C$                  |
| $I$        | intensity of solar radiation, $W/m^2$  | $U_t$           | top loss coefficient, $W/(m^2\ K)$                            |
| $I_1$      | irradiation at the inner surface of lower glass cover, $W/m^2$                 | $V$             | velocity of air in the duct, $m/s$                            |
| $I_2$      | irradiation at the bottom plate of packed bed collector, $W/m^2$               | $x$             | distance in horizontal direction from inlet, $m$              |
| $I_y$      | intensity of solar radiation at depth $y$ from top surface of the bed, $W/m^2$ | $y$             | distance in vertical direction from top surface, $m$          |
| $J_h$      | Colburn $J$ -factor ( $=St Pr^{2/3}$ )   | $\beta$         | extinction coefficient, $m^{-1}$                              |
| $k_a$      | thermal conductivity of air, $W/(m\ K)$  | $\varepsilon_p$ | emissivity of back plate                                      |
| $k_e$      | effective thermal conductivity of packed bed, $W/(m\ K)$                       | $\mu$           | dynamic viscosity of fluid, $N\ s/m^2$                        |
| $L$        | length of collector bed, $m$   | $\rho$          | density of air, $kg/m^3$                                      |
| $m$        | mass flow rate of air, $kg/s$  | $\eta_{eff}$    | effective efficiency  |
| $n$        | number of screens in a matrix  | $\eta_f$        | efficiency of fan or blower                                   |
| $P$        | porosity   | $\eta_m$        | efficiency of the electric motor used for driving fan         |
| $P_m$      | mechanical power, $W$  | $\eta_{tr}$     | efficiency of electrical transmission from power plant        |
| $Pr$       | Prandtl number ( $=\mu c_p/k_a$ )  | $\eta_{th}$     | thermal conversion efficiency of power plant                  |
| $p_t$      | pitch of wire mesh, $m$  | $\tau$          | optical depth   |
| $\Delta p$ | pressure drop in the duct, $N/m^2$   | $\tau_o$        | optical depth at $y = D$ ( $=\beta D$ )                       |
| $Q$        | volume flow rate, $m^3/s$  | $\tau'$         | transmissivity of cover glass                                 |
| $Q_r$      | radiative heat flux at a distance $y$ , $W/m^2$                                | $(\tau)_{eff}$  | effective transmittance for double glass cover system         |
| $q_u$      | useful heat gain, $W$  |                 |   |
| $R$        | radiosity at the top surface of upper glass cover, $W/m^2$                     |                 |   |

packed bed because they absorb solar radiation in depth, which reduces the top losses. Such collectors have a high ratio of heat transfer area to volume resulting in high heat transfer capability. Moreover, the heat transfer coefficient between a packing ele-

ment and flowing air is also increased due to increased turbulence of flowing air. An attempt to enhance the heat transfer is always accompanied with an increase in pressure drop, and thus the pumping power requirement is increased.

It is, therefore, desirable to optimize the system to maximize the heat transfer while keeping frictional losses at a minimum possible level.

In the present work, a methodology has been presented for thermohydraulic optimization of a packed bed solar air heater having its air flow passage packed with wire screen matrices. For thermohydraulic optimization of the solar air heater, a system with a set of six different matrices, as reported by Varshney and Saini (1998), has been chosen. The schematic diagram of the packed bed solar collector and the geometry of the wire screens packed inside the flow duct are shown in Figs. 1 and 2, respectively. To form a matrix, the desired number of layers of wire mesh screens are stacked on top of each other, parallel to the back plate of the collector, at equal distances along the depth. The results of thermohydraulic optimization obtained from packed bed collector have been compared with that of a conventional flat plate collector, as shown in Fig. 3.

**2. Mathematical model for thermal performance**

Heat balance equations, as those suggested by Hasatani et al. (1985) for transient conditions, have been modified for quasi steady state conditions for a packed bed collector. The following assumptions have been made in the formulation:

- (a) edge and back losses have been neglected,
- (b) environmental temperature and wind velocity have been assumed to be constant.

The following relations can be written for radiosity,  $R$ , and irradiation,  $I$ , based on the transmitted

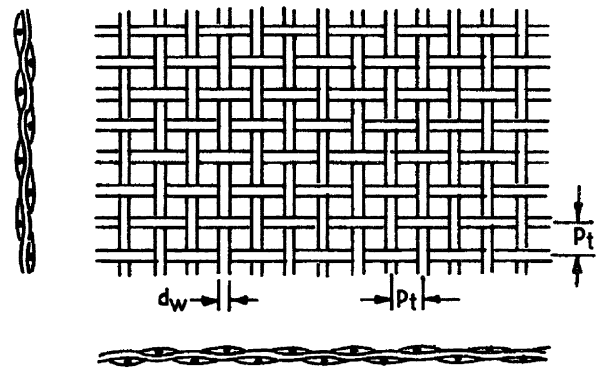


Fig. 2. Schematic of wire screen used as packing element.

and reflected radiation inside the bed as has been shown in Fig. 1.

$$R = r_c I + I_1(\tau)_{\text{eff}} \tag{1}$$

$$R_1 = I(\tau)_{\text{eff}} + r_c I_1 \tag{2}$$

$$I_1 = R_2 e^{-\tau_0} \tag{3}$$

$$I_2 = R_1 e^{-\tau_0} \tag{4}$$

$$R_2 = (1 - \epsilon_p) I_2 \tag{5}$$

$$I_y = R_1 e^{-\tau} \tag{6}$$

$$R_y = R_2 e^{-(\tau_0 - \tau)} \tag{7}$$

$$Q_r = I_y - R_y = R_1 e^{-\tau} - R_2 e^{-(\tau_0 - \tau)} \tag{8}$$

For the packed bed the following steady state heat balance equations can be written:

$$\begin{aligned} & \text{(Net heat flux entering the element due to conduction)} \\ & + \text{(Net heat flux due to radiation)} \\ & = \text{(Rate of heat gained by the air by means} \\ & \quad \text{of convection from matrix to air)} \end{aligned} \tag{9}$$

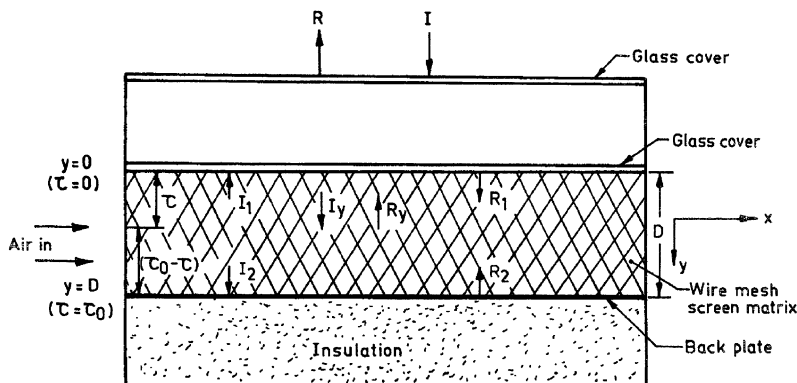


Fig. 1. Schematic of packed bed solar air heater.

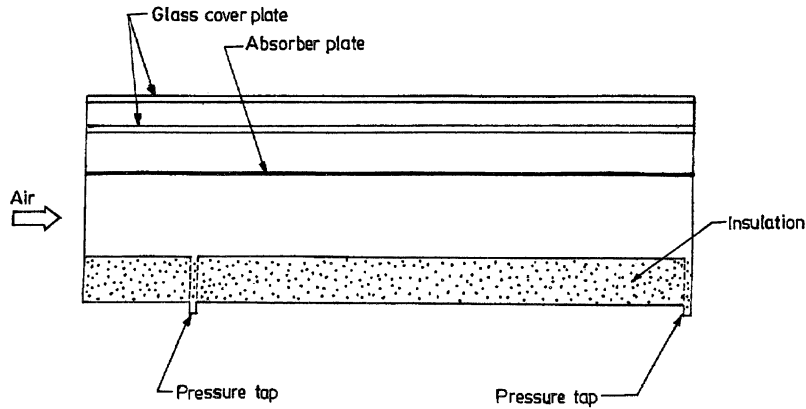


Fig. 3. Schematic of smooth flat plate collector.

(Rate of rise in the sensible heat of air)  
 = (Rate of energy carried away by the air) (10)

The mathematical expressions of Eqs. (9) and (10) can be written as

$$\frac{\partial}{\partial y} \left[ k_c \frac{\partial t_b}{\partial y} - Q_r \right] = h_c a_v (t_p - t_g) \quad (11)$$

$$G_o c_p \frac{\partial t_g}{\partial x} = h_c a_v (t_p - t_g) \quad (12)$$

The bed temperature,  $t_b$  is assumed to be average of matrix temperature,  $t_p$  and air temperature,  $t_g$ . Therefore, Eqs. (11) and (12) can be rewritten as

$$\frac{\partial}{\partial y} \left[ k_c \frac{\partial t_b}{\partial y} - Q_r \right] = 2h_c a_v (t_b - t_g) \quad (13)$$

$$G_o c_p \frac{\partial t_g}{\partial x} = 2h_c a_v (t_b - t_g) \quad (14)$$

The effective thermal conductivity,  $k_c$ , which is a function of thermo-physical properties of flowing fluid and matrix material as well as that of the porosity of the bed, has been calculated from the relation proposed by Yagi and Kunii (1957). The convective heat transfer coefficient between bed element and air,  $h_c = (J_h c_p G_o) / Pr^{2/3}$ , has been calculated using the following correlation reported by Varshney and Saini (1998) based on their experimental investigations.

$$J_h = 0.647 \left[ \frac{1}{nP} \left( \frac{P_t}{d_w} \right) \right]^{2.104} Re_p^{-0.55} \quad (15)$$

Applying boundary conditions for Eq. (13), the following additional equations are obtained:

At the top of the boundary i.e., at  $y = 0$

$$k_c \frac{\partial t_b}{\partial y} - U_t (t_b - t_a) = 0 \quad (16)$$

At the bottom of bed i.e., at  $y = D$

$$k_c \frac{\partial t_b}{\partial y} - Q_r = 0 \quad (17)$$

Boundary condition for Eq. (14), at the entry i.e., at  $x = 0$  yields:

$$t_g = t_i \quad (18)$$

Eqs. (13)–(18) have been normalized introducing the following non-dimensional parameters.

$$\tau = \beta y \quad (19)$$

$$t_b^* = t_b / t_i \quad (20)$$

$$t_g^* = t_g / t_i \quad (21)$$

$$t_a^* = t_a / t_i \quad (22)$$

$$\bar{x} = x / L \quad (23)$$

$$h_v^* = 2h_v L / (G_o c_p) \quad (24)$$

$$h_v^{**} = 2h_v / (k_c \beta^2) \quad (25)$$

$$Q_r^* = Q_r / (k_c \beta t_i) \quad (26)$$

$$U_t^* = U_t / (k_c \beta) \quad (27)$$

$$R_1^* = R_1 / (k_c \beta t_i) \quad (28)$$

$$R_2^* = R_2 / (k_c \beta t_i) \quad (29)$$

The resulting normalized equations have been converted to finite difference form and solved using the Gauss elimination method to estimate bed and air temperatures. The bed and air temperatures for given operating conditions are used to calculate the thermal efficiency using the following relation:

$$\eta = \frac{\dot{m} c_p (t_o - t_i)}{I A_c} \quad (30)$$

**3. Performance optimization**

Cortes and Piacentini (1990) have suggested that the performance of a solar collector cannot be optimized by simply subtracting the fan power from the thermal output of the collectors, because the former is produced mainly from thermal power plant and then transmitted, losing a considerable part of the energy in conversion and transmission. In order to evaluate the real economic performance of the collector, the following expression for effective efficiency has been used in the present analysis.

$$\eta_{\text{eff}} = \frac{q_u - \frac{P_m}{C}}{IA_C} \tag{31}$$

where  $C = \eta_f \eta_m \eta_{tr} \eta_{th}$ , is the conversion factor accounting for net conversion efficiency from primary energy to mechanical energy. The value of  $C$  as recommended by Cortes and Piacentini (1990) is 0.18 (typical values of efficiency factors being:  $\eta_f = 0.65$ ,  $\eta_m = 0.88$ ,  $\eta_{tr} = 0.92$ ,  $\eta_{th} = 0.35$ ).

The rate of the useful thermal energy gain,  $q_u$ , of the collector can be expressed as follows:

$$q_u = \dot{m}c_p(t_o - t_i) \tag{32}$$

Here, the outlet temperature,  $t_o$ , is determined by solving the heat balance equations formulated for the matrix element.

The mechanical power,  $P_m$ , required to force air through the duct is given by

$$P_m = Q\Delta p \tag{33}$$

Here, the pressure drop,  $\Delta p$ , across the duct of length,  $L$ , can be determined from the following expression:

$$\Delta p = \frac{f_p L \rho V^2}{2r_h} \tag{34}$$

where  $f_p$  has been calculated using the correlation reported by Varshney and Saini (1998):

Table 2  
Base values used in analytical calculation

| S. No. | Input data   | Numerical value           |
|--------|--|---------------------------|
| 1      | Length of collector, $L$                             | 2.39 m                    |
| 2      | Width of collector duct, $W$                         | 0.41 m                    |
| 3      | Depth of collector, $D$                              | 0.025 m                   |
| 4      | Mass flow rate of air                                | 0.005–0.05 kg/s           |
| 5      | Insolation, $I$                                      | 600 W/m <sup>2</sup>      |
| 6      | Inlet air temperature, $t_i$                         | 30 °C                     |
| 7      | Ambient temperature, $t_a$                           | 30 °C                     |
| 8      | Absorptivity of cover glass, $\alpha_c$              | 0.13                      |
| 9      | Reflectivity of cover glass, $r_c$                   | 0.0434                    |
| 10     | Emissivity of wire mesh screen, $\epsilon$           | 0.8                       |
| 11     | Emissivity of bottom plate, $\epsilon_p$             | 0.8                       |
| 12     | Thermal conductivity of matrix, $k_s$                | 62.764 W/(m K)            |
| 13     | Thermal conductivity of air, $k_a$                   | 0.0267 W/(m K)            |
| 14     | Density of air, $\rho$                               | 1.2 kg/m <sup>3</sup>     |
| 15     | Refractive index of mesh material, $n'$              | 1.0                       |
| 16     | Viscosity of air, $\mu$                              | 0.00001865 kg/(m s)       |
| 17     | Wind heat transfer coefficient, $h_w$                | 10.0 W/(m <sup>2</sup> K) |
| 18     | Eff. transmissivity of covers, $(\tau)_{\text{eff}}$ | 0.78                      |

$$f_p = 2.484 \left[ \frac{1}{nP} \left( \frac{p_t}{d_w} \right) \right]^{0.699} Re_p^{-0.44} \tag{35}$$

The effective efficiency has been evaluated using Eq. (31) for a set of matrices with the geometrical parameters given in Table 1. The values of effective efficiency have been obtained for varying air mass flow rates in the range of 0.005–0.05 kg/s along with base values given in Table 2.

**4. Results and discussion**

Values of the effective efficiency for different matrices have been obtained numerically using the above approach for smooth as well as packed bed collectors. The values of the effective efficiency have been obtained for varying air mass flow rates in the range of 0.005–0.05 kg/s with a constant value of

Table 1  
Geometrical parameters of wire screen matrices

| Matrix type | Wire diameter of matrix, $d_w$ (mm) | Pitch, $p_t$ (mm) | No. of layers, $n$ | Porosity, $P$ | Hydraulic radius, $r_h \times 10^3$ (m) | Extinction coefficient (m <sup>-1</sup> ) |
|-------------|-------------------------------------|-------------------|--------------------|---------------|---|---|
| M1          | 0.360                               | 2.72              | 14                 | 0.958         | 2.057                                   | 170.4                                     |
| M2          | 0.450                               | 2.08              | 10                 | 0.939         | 1.725                                   | 210.8                                     |
| M3          | 0.590                               | 2.23              | 10                 | 0.902         | 1.403                                   | 224.3                                     |
| M4          | 0.795                               | 3.19              | 9                  | 0.887         | 1.575                                   | 181.2                                     |
| M4(a)       | 0.795                               | 3.19              | 7                  | 0.905         | 2.023                                   | 142.6                                     |
| M4(b)       | 0.795                               | 3.19              | 5                  | 0.937         | 2.994                                   | 102.4                                     |

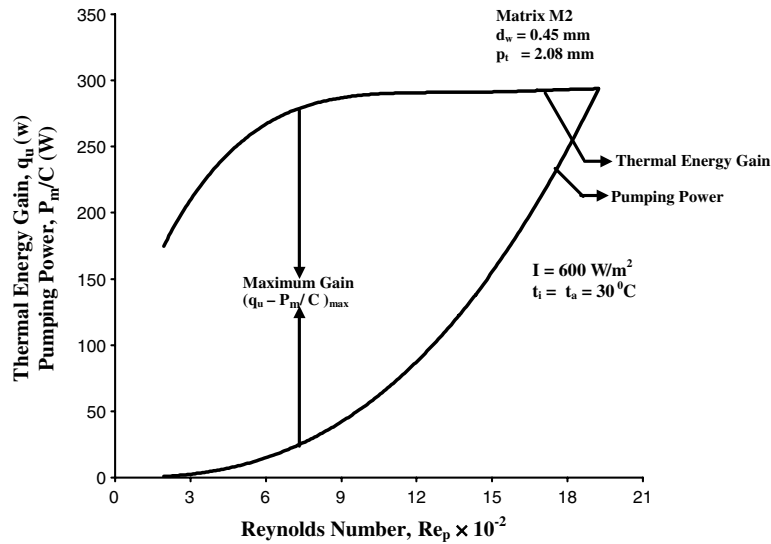


Fig. 4. Effect of Reynolds number on thermal gain and pumping power.

the insolation of  $600 \text{ W/m}^2$  and ambient air temperature of  $30 \text{ }^\circ\text{C}$ .

Fig. 4 has been drawn to show the effect of the Reynolds number on the thermal energy gain and the pumping power for a typical matrix M2. It may be observed that, at higher Reynolds numbers, the rate of increase in pumping power is very high, while the rate of the useful energy gain becomes nearly constant. It is also observed that at lower Reynolds numbers, the rate of increase in power required to force the air through the duct is low, while the rate of useful energy gain is substantial. Therefore, with the increase in Reynolds number, a stage is reached where we get the optimum value of energy gain. It is also evident from the figure that at higher value of Reynolds number, the benefit of net energy gain might eventually vanish.

Fig. 5 shows the effect of the Reynolds number on the effective efficiency. It is seen that as the air flow rate increases, the effective efficiency increases up to a threshold value of the Reynolds number, attains a maximum, and then reduces sharply. There exists an optimum value of effective efficiency for a given matrix. This effect can be attributed to the fact that the Reynolds number is a strong parameter that affects the pumping power and thermal energy gain, thereby affecting the effective efficiency.

In order to investigate the behavior of number of screens on the performance of the collector, the geometry of screen used for constructing matrix M4 has been chosen. Different number of layers of this screen have been stacked in a fixed depth of the

bed forming matrix M4, M4(a) and M4(b). It is observed that matrix M4(b) results in the best thermal as well as thermohydraulic performance as indicated in Fig. 5 itself. This behavior is probably due to the fact that with a higher number of screens forming the matrix in a fixed depth, the screens will be tightly pressed against each other and the fraction of effective heat transfer area available for contact with the flowing air will be less. Moreover, a higher number of screens offer more resistance to the flow of air, resulting in increased frictional losses. Consequently pumping power increases with an increase in the number of screens for the same operating conditions.

The effect of the porosity of a matrix on the thermohydraulic performance of the packed bed air heater is understandable from the effective efficiency plot for matrix M2 and M4 against Reynolds number, as shown in Fig. 5. The two matrices M2 and M4(b) have almost the same porosity but different geometrical parameters (wire diameter and pitch). It is seen that the effective efficiency with matrix M4(b) is higher in the entire range of Reynolds numbers investigated, in spite of the fact that the two matrices have almost the same porosity. This shows that geometry of the matrix strongly affects the performance of an air heater.

The effective efficiency of a packed bed solar air heater has been compared with a smooth solar air heater in Fig. 6. This figure reveals that matrix M4(b) is thermohydraulically most efficient in almost the entire range of flow rates investigated, i.e., from  $0.005$  to  $0.05 \text{ kg/s}$ . However, in case of

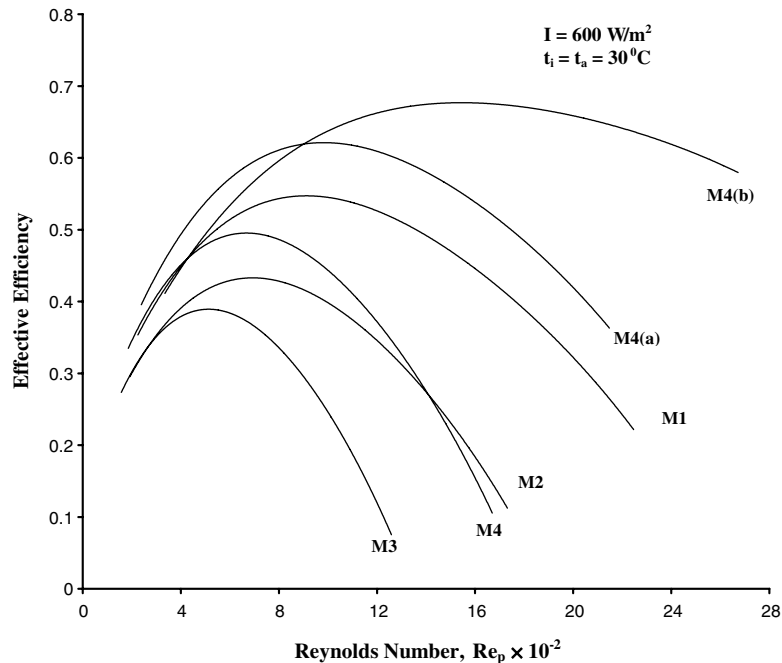


Fig. 5. Effect of Reynolds number on effective efficiency.

other matrices, it may be seen that the smooth collector shows better thermohydraulic results as compared to other matrix arrangements at higher flow rates. The thermohydraulic efficiency of a packed bed arrangement with M4(a) is better up to mass flow rates of about 0.04 kg/s. Similarly, matrices M1, M2, M4 and M3 show a better thermohydraulic performance compared to the smooth collector up to a range of about 0.035, 0.03, 0.025 and 0.02 kg/s, respectively.

The effective efficiency against temperature rise parameter,  $\frac{t_o - t_i}{I}$ , for different matrices has been plotted in Fig. 7. It is obvious that, as the mass flow rate increases, the temperature rise parameter decreases because temperature rise is inversely proportional to the mass flow rate. From the figure it is seen that as the temperature rise parameter decreases, the effective efficiency of all the matrices first increases, attains a maximum and then decreases. It is also evident that the effective efficiency of a smooth collector attains higher values as compared to packed beds corresponding to low temperature rise parameter conditions. Therefore, in the low temperature rise parameter range, packed bed air heaters are not beneficial in terms of effective efficiency, although they are beneficial from thermal performance point of view. The figure also indicates that the effective efficiency increases moderately with

decrease in the temperature rise parameter in the higher range, whereas for lower values of the temperature rise parameter the effective efficiency decreases sharply. These trends can be explained based on the fact that for lower flow rates, the rise in temperature shall be higher while the frictional losses would be lower, resulting in higher effective efficiency. For the cases, where the rise in temperature is lower, the mass flow rate would be higher; resulting in significant frictional losses and hence more pumping power would be needed. This would result in lowering the effective efficiency sharply for large values of flow rates.

Fig. 7 can also be used by a designer for selecting a matrix for packing the air flow duct of an air heater so as to give best performance in terms of effective efficiency. Out of the matrices used in this investigation, matrix M4(b) gives the best thermohydraulic performance. The corresponding mass flow rate for a desired temperature rise needed for a particular application at any location, where the intensity of solar radiation is known, can also be determined using the plot. For example, let the solar air heater is operating with inlet air at ambient temperature of 30 °C and the required outlet temperature is 50 °C, when solar radiation intensity is 600 W/m<sup>2</sup>. The temperature rise parameter  $(t_o - t_i)/I$  works out to be approximately 0.034. For this value of

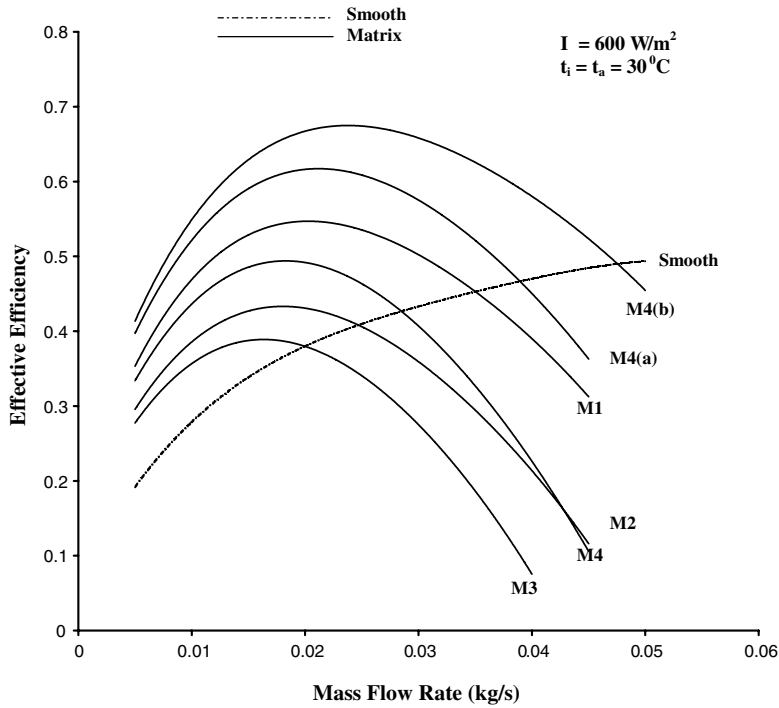


Fig. 6. Effect of mass flow rate on effective efficiency.

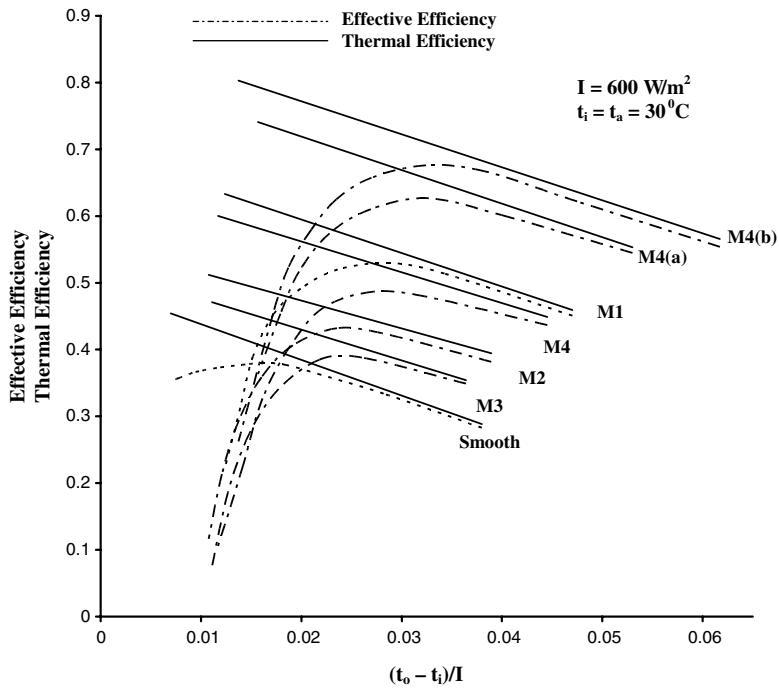


Fig. 7. Efficiency vs temperature rise parameter for different matrices.

the temperature rise parameter, Fig. 7 gives an effective efficiency of about 68% using matrix M4(b) and

a mass flow rate per unit collector area  $G = \eta I / (c_p \Delta t)$  comes out to be approximately  $0.02 \text{ kg}/(\text{m}^2 \text{ s})$ .



## 5. Conclusions

1. Thermohydraulic investigations of a packed bed solar air heater packed with wire screen matrix indicate that the thermal gain of such collectors is relatively higher as compared to smooth collectors, although the pressure drop across the duct also increases significantly. Wire mesh packed solar air heaters have been found to have a better effective efficiency as compared to conventional smooth air heaters.
2. The Reynolds number has been found to be a strong parameter affecting the effective efficiency. It has been found that there exists an optimum value of effective efficiency for a given matrix.
3. A solar air heater with matrix M4(b) is found to yield the best thermohydraulic efficiency in almost the entire range of mass flow rate investigated (0.005–0.05 kg/s).
4. It is found that for higher values of the temperature rise parameter, the effective efficiency values closely follow the thermal efficiency values, whereas there is an appreciable difference in the lower range of temperature rise parameter values.
5. It has been observed that merely the porosity of the bed does not govern the performance. The performance of a packed bed air heater is also a strong function of the geometrical parameters of the matrix.

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