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"ANALYSIS OF A SCREEN - TYPE SOLAR AIR COLLECTOR WITHOUT TRANSPARENT COVER"

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ABSTRACT

The performance of a screen type solar air collector of different uniform mesh sizes and without a glass cover is investigated theoretically. Formulation of simultaneous radiation and convection heat transfer for a semi-steady condition in an isotropic, absorbing set of wire screens is presented. Both colimated and diffuse radiant heat fluxes are incident on the screens, causing absorption, reflection (scattering), emission and heating, the screens, which in turn heat the circulated air flowing normal to their surfaces. In the analysis, the interspacing of screens is chosen such that natural convection currents are prohibited. A set of theoretical equations is developed and solved numerically to obtain the temperature of the air at each screen level, and that leaving the collector. The results indicate that the performance of this type of collector could be further improved if the screens are designed such that the porosity of the upper layers is higher than that of the lower ones.

1. INTRODUCTION

This investigation is a theoretical study of heat transfer in a matrix type solar air heater in which its absorbing element consists of a set of wire screens of different mesh spacing. The main objective of the study is to gain insight on the mechanics of radiative and convective transfer, thus providing a sound basis for the optimum design of a solar collector that may prove to be simple, economical and efficient.

A review of literature on the type of absorbers for solar air collectors reveal that they fall into two categories:

(a) Plate type absorber, which is a thin blackened metal plate that absorbs the incoming solar radiation at its top surface. The search of the literature indicates that there is a wealth of information on the performance of plate type absorbers in both theoretical and experimental treatments.

(b) Matrix type absorber, which can be either a thick porous darkened bed or a set of painted-black wire screens. In both types, the incident solar radiation is attenuated throughout the matrix and not just at its top surface.

Heat transfer through porous media has been the interest of numerous investigatiors due to its wide application in chemical industries, petroleum processes, transpiration cooling and solar systems. However, due to the irregular flow patterns encountered most of these studies have been experimental and correlations of specific data. There have been many theoretical investigations dealing with various aspects of the heat transfer in thick porous beds that are heated by solar radiation, but a comprehensive solution has not yet been fully assessed [1−2]. In addition, experimental studies with thick porous absorbers were investigated in connection with matrix collectors by Choi et al [3] and with honeycomb cellular structures by Lalude et al [4]. The efficiency of the honeycomb-porous bed solar air heater was found to perform as other air heaters at moderate temperatures but in a much superior manner at temperatures of the order of 90°C. A black porous matrix has provided better results in the last few years. Lansing and Clarke [5] studied the use of porous construction to achieve efficient heat extraction by obtaining a steady state temperature distribution within the porous plate, where the top surface was glazed and exposed to solar radiation. Bharadwaj et al [6] designed and fabricated a matrix type air heater and tested its performance. Singh et al [7] studied the thermal performance of a matrix air heater with cold air flowing in the opposite direction to the heat penetration. The applicability of this latter configuration has been viewed as questionable since the heated air flows upwards, contacting the transparent cover. This should cause significant upper heat losses. Recently, a simpler theoretical approach for analysing matrix solar collectors was conducted by Singh and Bansal [8] which was found to be accurate as a previous analysis conducted by Khe and Henderson [9].

The performance of air-cooled radiatively heated screen matrices were investigated both theoretically and experimentally by Khe [10], and Hamid and Beckman [11] for low temperature air heaters. In these analyses, the coverless screens were assumed to be of uniform mesh, constantly, facing perpendicularly to the sun, and exposed to steady-state solar radiation. Recently, an in-depth study of the performance of a screen-type solar collector, of uniform screen size was performed theoretically and experimentally by Soliman, Sorour and this writer [12] for moderate temperature drying applications.

The present study is an extension of the above noted works on matrix type absorbers of uniform wire-screen sizes. In this study, the effect of varying the size of the screens in the absorbing matrix on the thermal heat extract from the collector is investigated. In the model, the collector is assumed to be without cover to reduce the pressure drop which is found to be relatively high in this type of collector.

2. THE THEORETICAL MODEL

The solar collector under investigation consists of a set of wire screens, as the absorber, without transparent cover plates. The collector is insulated so that the heat losses through the side of the collector can be neglected. The air is drawn perpendicularly through the screens by means of a fan as shown in Figure (1). The screens are equally spaced and the spacing between any two adjacent screens, d1, is chosen such that the natural convection currents are prohibited in the entire flow field [13] The number of wires/cm or number of mesh/cm^2 is uniform in one screen but it may vary from one layer to another, so that the screens are of variable square mesh.

The collector is placed facing the south direction (for latitude, L = 31.2°N) at an inclined angle s, measured from the horizontal, equal to the difference between the angles of the local latitude and the declination of the sun D.

When the solar radiation (beam + diffuse) fall on the screen of the absorbing matrix, some of it is absorbed, some passes through the holes, and some is reflected and scattered. Radiation heat exchange occurs beween the absorbing matrix and the environment, between the matrix and the frontal surface of the floor, and between the parts of the screens which can see each other inside the matrix itself.

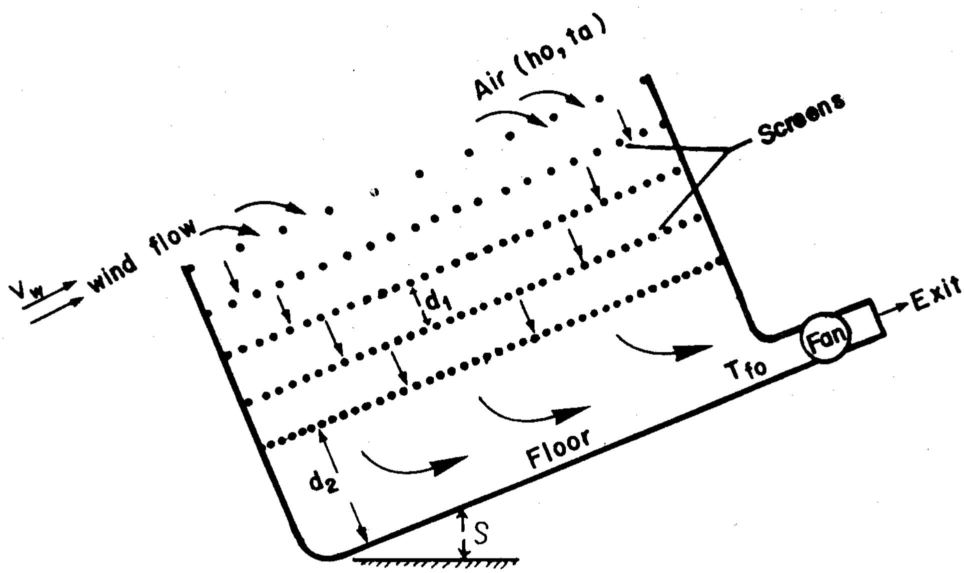

Fig. (1) : Sketch of the heater.

If the screen is viewed from one angle, it appears to have less openings than when it faces the observer directly. For diffuse radiation, either solar diffuse or longwave radiation, the percentage opaque area relative to the total area of the screen is a function of the ray angles θ and Φ and the physical dimensions of the wire screen. This percentage, defined as the spectral extinction factor r_i, can be obtained by following the procedures outlined in Reference [14].

$$r_i = 1 - \left(\frac{A_i}{1+A_i}\right)^2 \left[1 - \frac{\tan\theta}{A_i}(\cos\theta + \sin\phi) + \frac{\tan^2\theta}{2A_i^2}\sin 2\phi\right] \quad (1)$$

In equation (1), A_i is the average spacing between two adjacent wires divided by the thickness of wire. This ratio is proportional to the number of wires/cm in each screen, i. The angles of the ray, θ and ϕ are measured from the normal to the collector and the south directions respectively. The polar angle ϕ has a maximum limit at which $r_i=1$, i.e. the screen appears totally apaque to incident radiation. For isotropic diffuse radiation, the parameter r_i is constant with time.

For beam solar radiation, the optical opening area relative to the total area of the screen i, is given according to reference [14] as:

$$S_i = \left(\frac{A_i}{1+A_i}\right)\left[1 - \frac{\tan\theta_b}{A_i}(\cos\phi_b + \sin\phi_b) + \frac{\tan^2\theta_b}{2A_i^2}\sin 2\phi_b\right] \quad (2)$$

where, θ_b and ϕ_b are the angles of the beam radiation given by:

$$\cos\theta_b = \cos D \cos(L-s)\cos H + \sin D \sin(L-s) \quad (3)$$

$$\sin\phi_b = \cos D \sin H/\sin\phi H \quad (4)$$

Here, H = 15x(No. of hours measured from Noon), $\theta H = \theta_b$ (at s=0).
As shown in the above equations, S_i is a time-dependent parameter, increasing as θ_b decreases and reaches a maximum value at $\theta_b = 0$ (at noon time).

For statistcal purposes a screen i (i=,2, ..., n) can be viewed as a flat surface having transmissivity $(1-r_i)$ and absorptivity αr_i, as well as emitting radiation at an intensity of ϵI_i, where α and ϵ are the absorptivity and the emissivity of the paint on the screen respectively. I_i is the radiative intensity of a black surface having the same temperature T_i as that of the screen i, i.e. $I_i = \sigma T_i^4/\pi$.

In the analysis, the temperature of each screen at time t, is assumed uniform throughout its area and equal to that of the flowing air as long as the thickness of the wires is much less than the spacing d_l. Under these considerations, the whole system is treated as a series of partially transparent parallel flat plates, each having its own transmissivity, absorptivity and reflectivity that vary with the angle of incidence of the radiation.

The heat transfer interactions at a given screen i, other than the first screen, is subject to:

1. Reradiation, as a result of its temperature at an energy rate of:

$$q_i^{(1)} = 2\varepsilon \int_0^{2\pi} \int_0^{\pi/2} r_i \cos\theta \sin\theta \, d\theta \, d\phi \tag{5}$$

2. Radiation coming from above (forward), due to longwave emissions of the environment (ϵI_s), the (i−1) screens above, and the diffuse solar radiation at an energy rate of:

$$q_i^{(2)} = \alpha \int_0^{2\pi} \int_0^{\pi/2} r_i I_i \cos\theta \sin\theta \, d\theta \, d\phi \tag{6}$$

The forward radiation intensity I^+_i is given by

$$I_i^+ \doteq \left(I_s + \frac{Q_D}{\pi}\right) \prod_{k=1}^{i-1} (1-r_k) + \varepsilon \sum_{k=1}^{i-1} I_k r_k \prod_{j=1}^{i-1} (1-r_j) \tag{7}$$

where, $I_s = \frac{\sigma}{\pi} (T^4_s)$, T_s = sky temperature which is a function of the ambient condition as reported in reference [12].

3. Beam solar radiation that passes through the top (i−1) screens and is absorbed by the screen i at an energy rate of:

$$q_i^{(3)} = \tau_i \, (1-S_i) \prod_{k=1}^{i-1} S_k \cdot Q_B \tag{8}$$

Here, the transmissivity τ_i stands for the shading effect of the collector side walls on the screens. This parameter can be calculated using Equation (2) after replacing the design parameter of the screens A_i by an equivalent one for the collector.

4. **Radiation coming from below (backward), due to long-wave emissions of the (n-i) screens below and the floor plate, at an energy rate of:**

$$q_i^{(4)} = \alpha \int_0^{2\pi} \int_0^{\pi/2} r_i \, I_i^- \cos\theta \, \sin\theta \, d\theta \, d\phi \qquad (9)$$

where the backward radiation intensity is given by.

$$I_i^- = \varepsilon \sum_{k=i+1}^{n} r_k I_k \prod_{j=i+1}^{k-1} (1-r_j) + \varepsilon_f I_f \prod_{k=i+1}^{n} (i-r_k) \qquad (10)$$

5. **Reflected energy: the rate of energy absorbed by the i^{th} screen due to reflected radiation just above and below the screen is:**

$$q_i^{(5)} = \int_0^{2\pi} \int_0^{\pi/2} I_r \cos\theta \, \sin\theta \, d\theta \, d\phi \qquad (11)$$

$$I_r = \frac{\rho}{\pi} [g(q_{i-1}^{(2)} + q_{i+1}^{(4)}) + (1-g)(q_{i+1}^{(2)} + q_{i-1}^{(4)}) + g_b \, q_{i-1}^{(3)} + (1-g_b) q_{i+1}^{(3)}] \qquad (12)$$

In the above equation, g is the percentage of the reflected energy that is scattered in the direction of the rays. Since g is found to vary to a very narrow range with the angles θ & ϕ, in the analysis, it will be assumed to be constant. g_b is the value of g when ($\theta = \theta_b$, $\Phi = \Phi_b$).

6. Heat convection to the flowing air:
   The heat transferred by forced convection from the screen i to the flowing air is given by the relation:

$$q_i^{(6)} = h_l(T_{i+1} - T_i) - h_u(T_i - T_{i-1}) \tag{13}$$

The coefficients, h_l and h_u at the lower and upper surfaces of screen i are given in Reference [12]. For the first screen h_u is replaced by the outside heat coefficient, h_o which is a function of the wind speed and the mass flow rate of the air inside the collector [15].

In summary, the net heat balance equation for screen i of negligible heat capacity is:

$$q_i^{(1)} - q_i^{(2)} - q_i^{(3)} - q_i^{(4)} - q_i^{(5)} - q_i^{(6)} = 0 \tag{14}$$

Using the following notations:

$$F_{1,k} = 4 \int_0^{\pi/4} \int_0^{\pi/2} r_k \quad \sin 2\theta \quad d\theta \quad d\phi \tag{15}$$

$$F_{2,k} = 4 \int_0^{\pi/4} \int_0^{\pi/2} r_k \prod_{j=1}^{k-1} (1-r_j) \sin 2\theta \quad d\theta \quad d\phi \tag{16}$$

$$F_{3,k} = 4 \int_0^{\pi/4} \int_0^{\pi/2} r_k \prod_{j=k+1}^{n} (1-r_j) \sin 2\theta \quad d\theta \quad d\phi$$

$$G_{i,k} = 4 \int_0^{\pi/4} \int_0^{\pi/2} r_i \prod_{j=i+1}^{k-1} (1-r_j) \quad r_k \sin 2\theta \quad d\theta \quad d\phi \tag{18}$$

$$G_{k,i} = G_{i,k} \quad \text{and} \quad G_{i,i} = 0 \tag{19}$$

$$H_k = \tau_k (1-S_k) \prod_{j=1}^{k-1} S_j \tag{20}$$

$$H_f = \tau_f \prod_{j=1}^{n} S_j \tag{21}$$

$$M_n = 4 \int_0^{\pi/4} \int_0^{\pi/2} [\prod_{k=1}^{n} (1-r_k)] \sin 2\theta \quad d\theta \quad d\phi \tag{22}$$

$$R_i = \frac{\rho g \; F_{1,i}}{\pi} , \quad R_i^* = \frac{\rho(1-g)}{\pi} F_{1,i} \tag{23}$$

and introducing the expressions (15-23) into Equation (14) and re-arranging the terms, the end result leads to:

$$\alpha\varepsilon \sum_{k=1}^{i-2} (G_{i,k} + R_i G_{i-1,k} + R_i^* G_{i+1,k}) I_k$$

$$+ \alpha\varepsilon (G_{i,i-1} + R^*_i G_{i+1,i-1}) I_{i-1}$$

$$+ \alpha\varepsilon (R_i^* G_{i+1,i} + R_i^* G_{i+1,i-1} - 2 F_{1,i}/\alpha) I_i$$

$$+ \alpha\varepsilon (G_{i,i+1} + R_i^* G_{i-1,i+1}) I_{i+1}$$

$$+ \alpha\varepsilon \sum_{k=i+2}^{n} (G_{i,k} + R_i^* G_{i-1,k} + R_i G_{i+1,k}) I_k$$

$$+ \alpha\varepsilon_f (F_{3,i} + R_i^* F_{3,i-1} + R_i F_{3,i+1}) I_f$$

$$+ \alpha \quad (H_i + Rb_i H_{i-1} + Rb_i^* \; H_{i+1}) \; QB$$

$$+ \alpha\tau_{di} \quad (F_{2,i} + R_i \; F_{2,i-1} + R_i^* \; F_{2,i+1}) \; (\frac{QD}{\pi} + \varepsilon \, I_s)$$

$$+ \quad h_1(T_{i+1} - T_i) - h_u(T_i - T_{i-1}) = 0 \qquad (24)$$

Equation (24) is applicable for the screens ($1< i <n$). For the first screen, the last screen and the floor equation (24) needs a few modifications because of their special location and also the floor plate is completely opaque (i.e. $r_f = 1$).

For the First Screen:

$$\alpha\varepsilon(R_i^* \; G_{2,1} - 2 \; F_{1,1}/\alpha) \; I_1 + \alpha\varepsilon \, G_{1,2} I_2 + \alpha\varepsilon \sum_{k=3}^{n} (G_{1,k} + R_1 G_{2,k}) \; I_k$$

$$+ \quad \alpha\varepsilon_f(F_{3,1} + R_1 F_{3,2}) \; I_f + \quad (H_1 + Rb_1^* \; H_2) \; QB + \alpha \; \tau_1(F_{1,1} + Rb^* \; F_{2,2})$$

$$(\frac{QD}{\pi} + \varepsilon \, I_s) + h_1(T_2 - T_1) - h_o(T_1 - T_a) = 0 \qquad (25)$$

For the last Screen:

$$\alpha\varepsilon \quad \sum_{k=1}^{n-2} \quad (G_{n,k} + R_n G_{n-1,k} + \frac{\rho_f \; F_{1,n}}{\pi} \quad F_{3,k}) \; I_k$$

$$+ \, \alpha\varepsilon \quad (G_{n-1,n} + \frac{\rho_f \; F_{1,n}}{\pi} \, F_{3,n-1}) \; I_{n-1}'$$

$$+ \alpha\varepsilon \; (R^*_n \, G_{n-1,n} + \frac{\rho_f \, F_{1,n}}{\pi} \, F_{3,n-1} - 2 \, F_{1,n}/\alpha \,) \, I_n$$

$$+ \alpha\varepsilon_f \; (F_{3,n} + R^*_n \, F_{3,n-1}) \, I_f + \alpha (H_n + Rb_n \, H_{n-1} + \frac{\rho_f \, F_{1,n}}{\pi} \, H_f) \; QB$$

$$+ \alpha\tau_{dn} \; (F_{2,n} + R_n F_{2,n-1} + \frac{\rho_f \, F_{1,n}}{\pi} \, M_n)(\frac{QD}{\pi} + \varepsilon I_s)$$

$$+ \quad hl_f (T_f - T_n) - h_u (T_n - T_{n-1}) = 0 \qquad (26)$$

For the Floor:

$$\alpha_f \varepsilon \sum_{k=1}^{n-1} (F_{3,k} + g \, G_{n,k}) \, I_k + \alpha_f \varepsilon \, F_{3,n} \, I_n$$

$$+ \quad \alpha_f \, \varepsilon_f \; (R^*_n - 1/\,\alpha_f) \; I_f + \alpha_f (H_f + g_b H_n) \; Q_B \qquad (27)$$

$$+ \quad \alpha_f \tau_{df} (M_n + gF_{2,n}) \; (\frac{Q_D}{\pi} + \varepsilon I_s) - h_f (T_f - T_n) - U_b (T_f - T_a) = 0$$

Here, h_f and U_b are the convection coefficient and the conductance over and below the surface of the floor. Equations (24–27) for the screens (i=l to n) and the floor are the set of equations that describe the heat transfer in the matrix collector. They constitute a system of n+1 equations with n+1 unknowns, the unknowns being the temperatures of the screens and the floor plate temperature. This system of equations can be written in matrix form as:

$$[A] \quad \langle T \rangle \; + \; [B] \quad \langle T^4 \rangle \; + \; \langle C \rangle \; = 0 \qquad (28)$$

where, [A] and [B] are (n+l) x (n+l) square matrices and $< C >$ and $< T >$ are (n+l) dimensional constant and vector respectively. The above set of the non-linear equations (28) is solved by an iterative scheme, after linearizing the non-linear term $< T^4 >$ by using the Taylor series expansion. The numerical results are found to converge very rapidly, at most, after 3 iterations. A sample of the comutational results are presented in a graphical form in the next section.

3. RESULTS AND DISCUSSIONS

To analyse the effects of environmental and operating conditions and variations in the system design parameters on the thermal performance of the collector, the daily efficiency, η_d is used, η_d is defined by:

$$\eta_d = \int_{t=t1}^{t2} \dot{m}\, C_p (T_{fo} - T_{fi})\, dt \Big/ \int_{t1}^{t2} [Q_B(t) + Q_D(t)]\, dt \qquad (29)$$

where:

$T_{fi} = T_a = 28^oC$, $D = 23.1^o$, $L = 31.2^oN$, $d_1 = 1$ cm, $d_2 = 4$ cm, $d_{ins} =$ 10 cm, $d_w = 1$ mm, $\alpha = \varepsilon = 0.9$, $V_w = 4$ m/s, $k_{ins} = 0.05$ W/m.k,

$g = 0.3$ & $g_b = 0.2$

Figure (2) shows the average measured hourly rate of beam and diffuse solar radiation during the month of June in Alexandria, Egypt for the time from 9 am to 3 pm. The amount of diffuse radiation relative to the total incident radiation is about 20%.

Figure (3) illustrates the influence of the wind speed V_w on the film coefficient h_o at the upper surface of the first screen for various values of the withdrawal air flow rate, m. Although, the heat coefficient h_o increases as V_w or m, the thermal efficiency n_d is found to decrease with an increase in V_w while increasing with the increase of the flow rate, as shown in Figure (4). In this figure, two collector models are presented. The first has one screen of (1 wire/cm) while the other has five screens of different uniform mesh sizes.

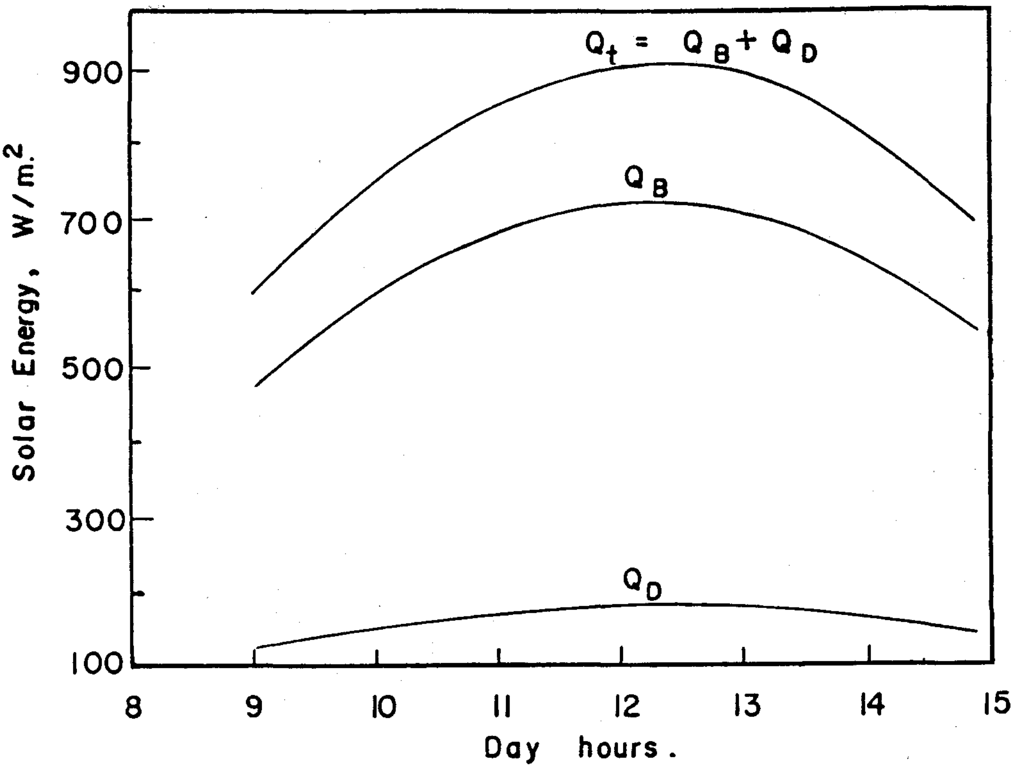

Fig. (2) : Measured hourly beam & diffuse solar radiation for (L = 31.2°, D = 23.2° & S = 8.2°)

The collector efficiences are plotted against the number of screens in the absorber for number of wires/cm = 1,2,3 and 5. For screens having 1 wire/cm, the absorber is very efficient, as far as heat collection is concerned, if it has 4 screens. However, as the number of wires/cm increases say from 2 to 5, the optimum number of screens of uniform mesh is decreased to 2. A collector of 2 screens, each having 3x3 mesh/cm^2, will be the most efficient one among all the other selections presented in the figure. From Figure (5), we conclude that if the number of large mesh screens is too small, the absorber is not a good heat trap because radiation can pass through the absorber readily, inward and outward. On the other hand, if the number of small mesh screens is too large, the lower layers receive little or do not receive any incoming radiation.

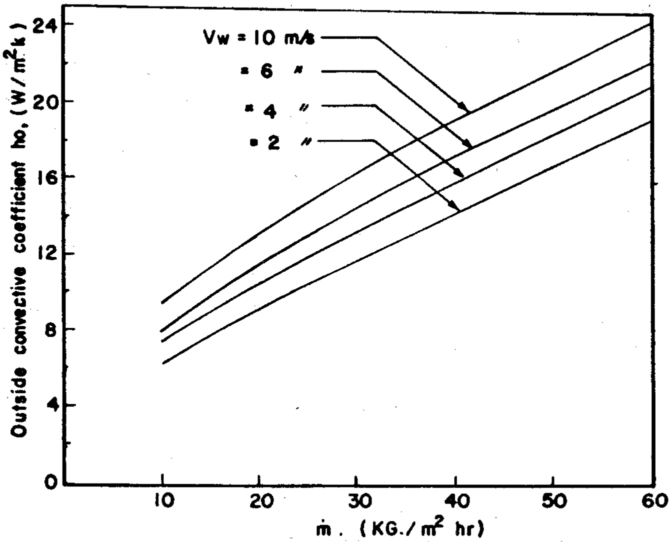

Fig. (3) : Outside convective coefficient ho as a function of the wind speed Vw and the air mass flow rate m.

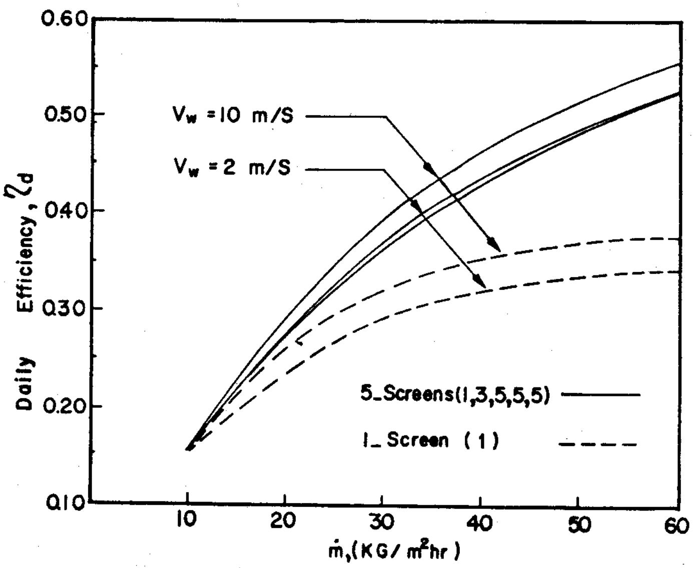

Fig. (4) : Effect of wind speed on at the daily efficiency for various flow rates.

Thus, the heated air as it passes through the absorber will lose some of its heat energy to the relatively cooler lower layers. This tends to reduce the efficiency of the collector.

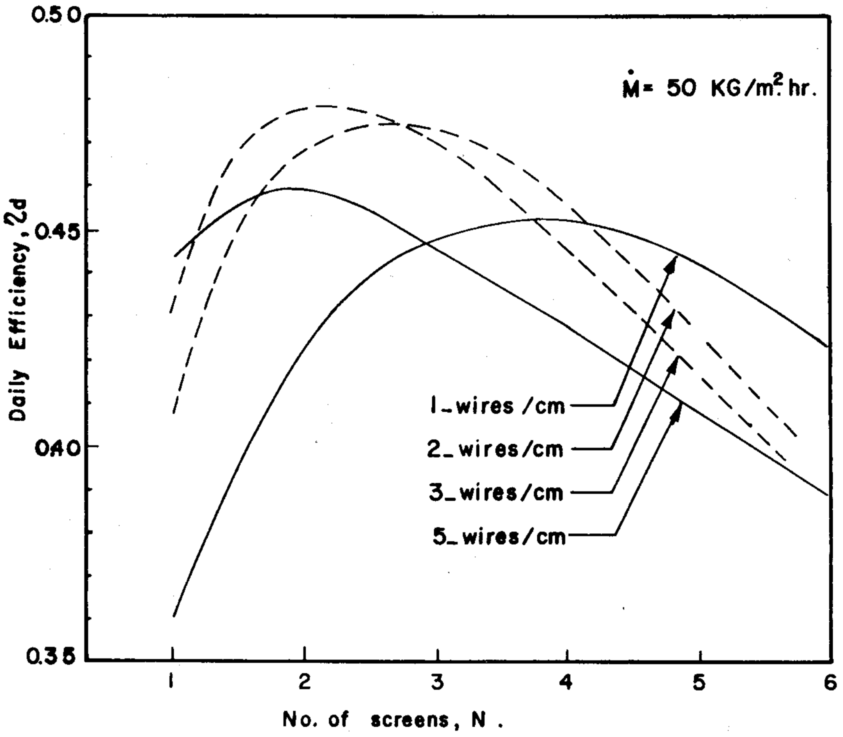

Fig. (5) : Effect of the number of screens and the number of wires / cm on the collector thermal performance.

To further improve the thermal performance of the absorber, it is suggested to use screens of variable size (spaces) as shown in Figure (6–9). Figure (6), shows a comparison between the efficiency of uniform-mesh screen collectors and that for non-uniform-mesh screen collectors, where each absorber has 4 screens. Clearly, the later ones have always higher efficiencies than the former ones in all of the air flow rates of concern. The difference is found to increase with the increase of the flow rate.

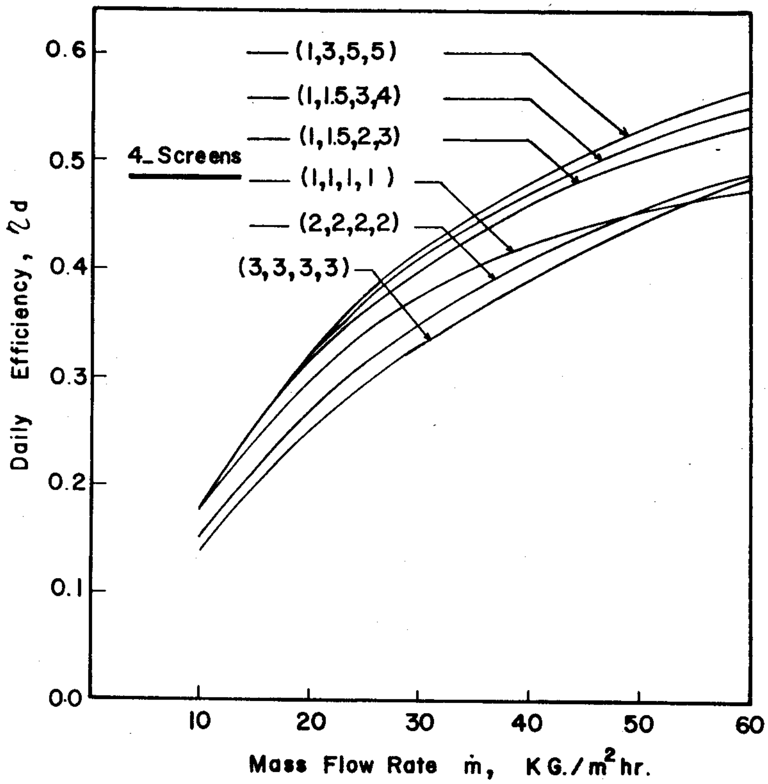

Fig. (6) : Effect of the variation of the number of wires/cm from screen to another on the collector thermal performance.

The effect of using variable screen sizes on the collector performance can be explained by the use of Figure (7). As is shown, the absorbers having larger number of wires/cm (smaller porosity) in their lower layers, absorb more energy than those of smaller number of wires/cm (larger porosity) in these layers. In addition, the intermediate layers are found to absorb more energy than the top and bottom screens, as the porosity of the lower layers decreases. This, will tend to reduce the heat losses via reradiation and convection to the surrounding atmosphere, and thus resulting in higher efficiency collectors. If the number of screens is increased over 5 or 6, the reverse will come true, since the lower layers will not receive any effctive radiation.

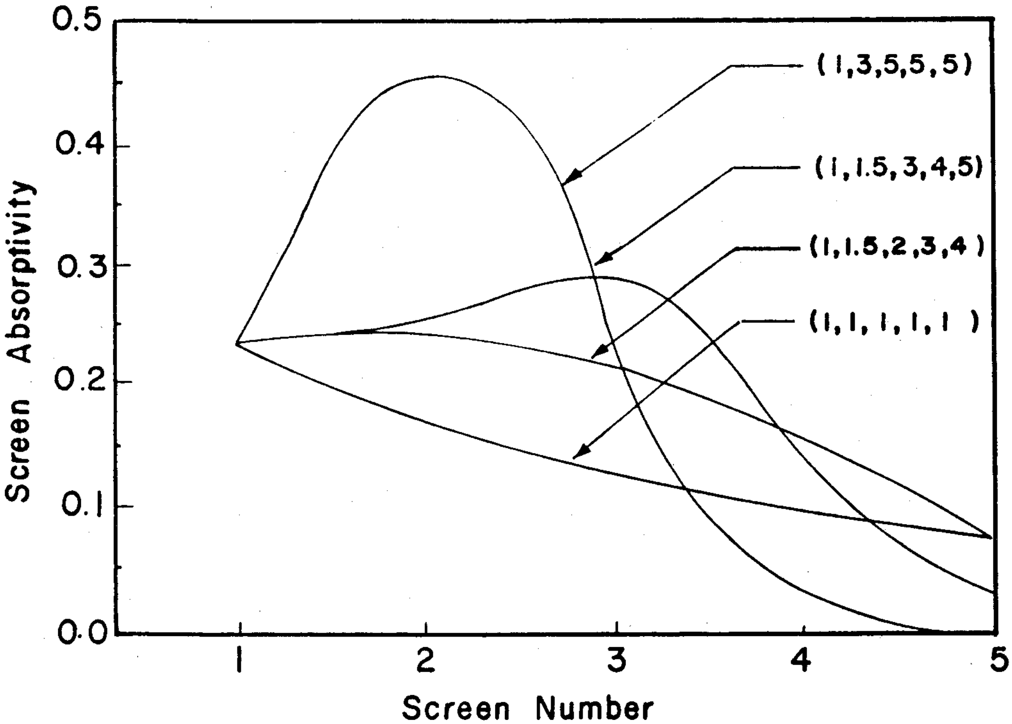

Fig. (7) : Solar absorptivity of each screen as a function of the number of wires/ cm in each screen for collectors having up to 5-screens.

Figure (8), illustrates the influence of decreasing the absorptivity of the paint on the screens on the performance of the collector for absorbers having 2 and 4 screens respectively. As is seen, as the absorptivity decreases, the efficiency is rapidly decreased for both collectors. The rate of decrease in η_d is much higher for 4−screen collector especially at higher values of the air flow rate. This is attributed to the increasing effect of the radiation scattered between the screens as the number of screens increases. The scattering effect tends to reduce the heat gain by the screens and thus reduces the efficiency of the system.

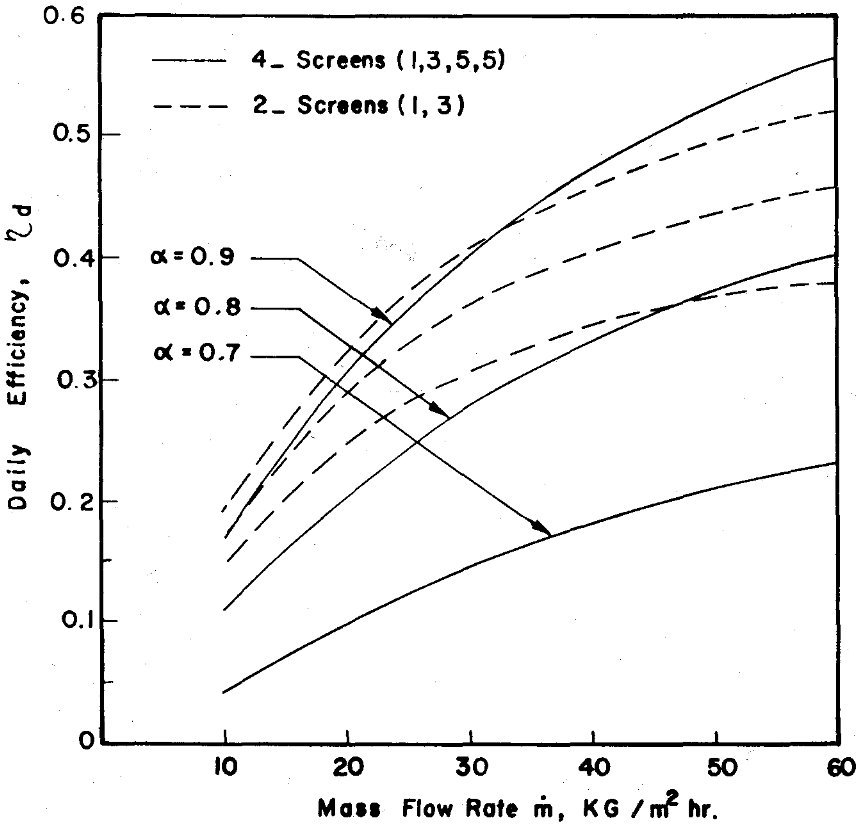

Fig. (8) : Effect of the absorption coefficient and emissivity of the screens on the collector performance.

Finally, Figure (9) shows the effect of changing size of the mesh from one screen to another on the temperature of these layers. Each absorber has 4 screens and m = 10 & 60 Kg/m^2. hr. For screens of different porosity, the rate of change of temperature is very rapid at the upper layers and very small at the lower ones. Furthermore, the temperature is found to be decreased at the lower layers of smaller mesh sizes, as described previously. In spite of that, the temperatures of different porous screens are still higher than those of uniform porous screens especially at the higher values of the air flow rate.

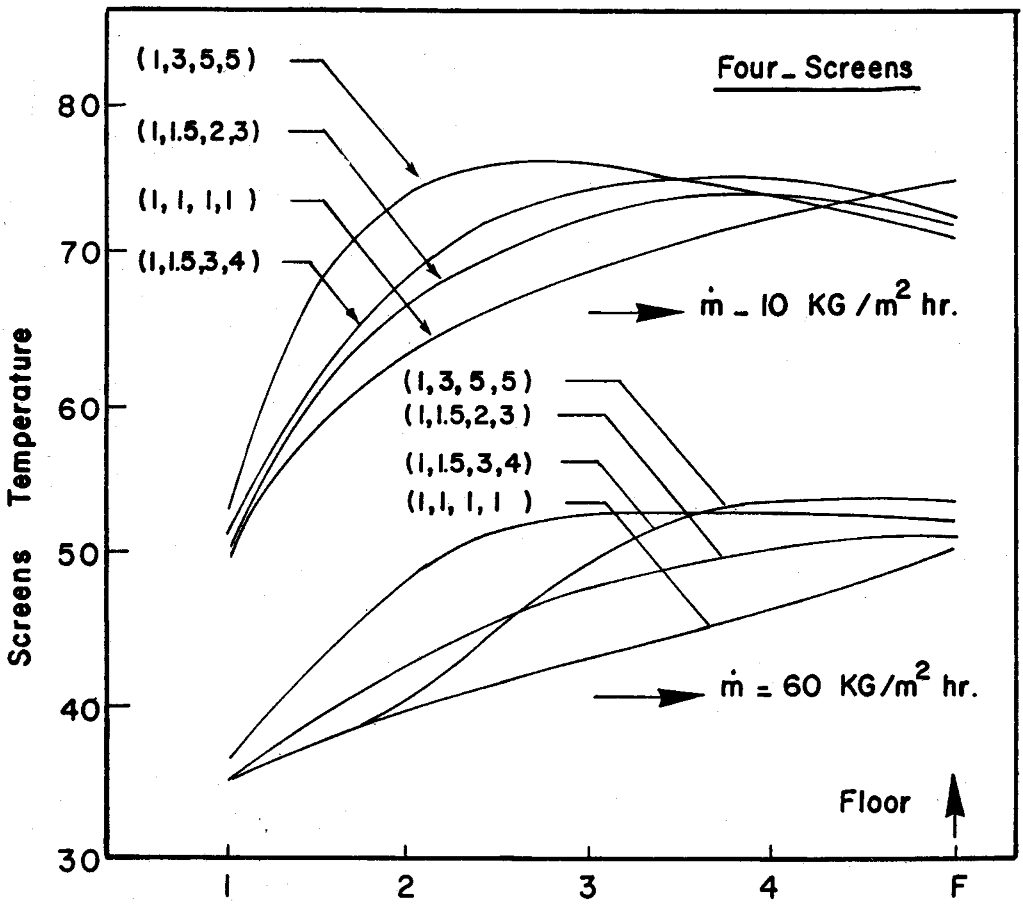

Fig. 9 - Predicted screen temperatures at (11 A.M.) for collector having four screens of different mesh sizes.

From this study, it can be concluded that the design of a matrix absorber of uniform screens each having a different mesh will lead to improved collector performance compared to that for absorbers of uniform screens all having the same mesh. A further study involving an optimization scheme is recommended to determine the number of screens and their sizes that would make the collector optimally efficient and economical over the range of the system parameters of concern.

NOMENCLATURE

C_p	=	specific heat of air, J/Kg. K
d	=	spacing between two adjacent screens, m
D	=	Declination of sun, degrees
$F_{i,j}$	=	paramenters defined by equations (19)
g	=	percentage of reflected energy that is directed downwards w.r.t. the screen, decimal
H_i	=	parameter defined by equations (20 & 21)
I_i	=	radiation intensity from black surface, W/m_2
L	=	local latitude, degrees
m	=	air flow rate per unit area of screen, $Kg/m^2.h$
M_n	=	parameter defined by equation (22)
n	=	number of screen in the matrix
q	=	heat flux, W/m^2
Q_B	=	diffuse solar radiation intensity, W/m^2
Q_D	=	diffuse solar radiation intensity, W/m^2
r_i	=	extinction factor defined by equation (1), decimal
R, R^*	=	parameters defined by equation (23)
s	=	collector tilt angle from horizontal, degrees
S_i	=	percentage of screen opening, equation (2), decimal
t	=	time, hr
T	=	temperature, °C
U_b	=	conductance of bottome insulation, $W/m^2.k$
V_w	=	wind speed, m/s

Greek Letters:

α	=	absorptivity of the screens
ϵ	=	emissivity of the screens
η_d	=	daily thermal efficiency of the collector
θ	=	polar angle, degrees
ρ	=	reflectivity of the screens
τ	=	transmission of the screens
σ	=	stefan-Boltzman constant = $5.67 \times 10^{-8}\ W/m^2.K^4$
ϕ	=	azimuth angle, degrees
Σ	=	sum of series
Π	=	product of series

Subscripts:

a	= ambient
B,b	= direct
D	= floor
i,j,k	= screen number
n	= last screen
ins	= insulation
w	= wire

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