THERMAL PERFORMANCE ANALYSIS OF SOLAR AIR HEATERS

Mohamad Jamil
Fakulti Kejuruteraan Jentera,
Universiti Teknologi Malaysia.

Prof. Madya Amer Nordin Darus
Fakulti Kejuruteraan Jentera,
Universiti Teknologi Malaysia.

Synopsis:

A computer routine to calculate the thermal performance of several different low temperature types of flat-plate air heaters is to be discussed. Analysis of each type is also described. The programme accepts as input real or simulated flux, collector geometry, air flow rate and enviromental data. It computes temperatures and extracts energy as a function of time of the day. The programme evaluates radiation, convection, conduction and wind losses, and the radiation exchange with the enviromental conditions. The procedure used in the derivation of the governing equations is also described. The prediction of performance provided by this programme is particularly useful in comparing performances of different collectors and for studying a specific collector's performance with changes in enviroment and design parameters which can be controlled to some extent by the designer.

Introduction

In the study of the thermal performance of a solar collector its steady state of behaviour has to be taken into account. As criteria, the efficiency and effectivity of collectors are calculated with respect to their great number of parameters. These are grouped into "Collector parameters", "Operation parameters" and "Meteorological parameters". By their individual variation in the computer programme, their particular effects are obtained as well as their relation to other paramaters.

Due to the periodic and intermittent supply of sunshine the collection of solar energy consequently is periodic and intermittent with physical processes which are basically transient and only in special cases follow steady state condition. In any case these processes depend upon a multitude of parameters and conditions. Those which are not interrelated with each other are, the geometric configuration of the absorber plate and the transparent cover as well as their thermal and radiative properties. These are pertaining to the collector directly, are named as collector parameters. Other parameters such as mass flowrate of the heat carrier and its temperature are considered as operation parameters. Finally, solar irradiation, ambient temperature, wind speed and sky temperature are grouped together as meteorological paramaters.

For the evaluation and rating of the combined action of all these parameters, the so-called thermal performance data of each collector are considered. The thermal performance data are in the form of efficiency, heat loss term, outlet temperature, etc. The flexibility and usefulness of the computational procedure may be judged by the simplicity of adjustment of the various collector parameters. Any heater under consideration may be tested for a wide range of consrruction, materials processes and enviromental conditions by a simple adjustment of the input data. The simple parameter adjustment whithin the programme would allow one to make a systematic decision as to the best of all the design components under a wide range of operating conditions.

Description of the Air Heaters

Many different types of solar air heaters have been designed before. Fig. (1) shows ten such types of solar air heaters which have been selected for this analysis. Each of them is namely identified as type 1 to type 10. It can be seen from Fig. (1a) that the type 1 heater represents the simplest design as it is basically a flow box formed from the two essential plates, that is, the transparent cover and the combined collector/base plate. In types 2 and 4, an attempt is made to lower the overall heat loss by raising the heater plate from the floor of the duct thus reducing the downwards conduction loss from the unit. The crented gap between the heater plate and the cover plate may now be either used to form a stagnant air gap with the aim of reducing the upward convection loss to the cover or to form a second flow duct with the advantage of increasing the heat transfer surface to the flowing air stream.

In types 3, 5 and 8, their absorber plate were modified in an attempt to increase the collector performance. It can be seen clearly from Fig. (1) that those two basic collectors (types 2 and 4) were modified into the form of either vee-corrugated plate or finned plate. Thus this increases their flow surfaces as well as the value of the heat transfer coefficient, h. Type 6 is actually heater type 4 but without transparent cover. Type 7 is a triangular duct collector, which was designed mainly for residential heating, restrict transport fluid flow to triangular duct as shown in Fig. (1g). Type 9 is a matrix absorber air heater and was studied thoroughly by Hamid and Beckman (6). The absorber of this type is then called porous absorber since the air can penetrate through it. Hence, it increases the performance of the heater. Type 10 is a clipped V through heater and was developed and studied by Bordoloi and Bynum (1). This type of air heater is shown in Fig. (1j).

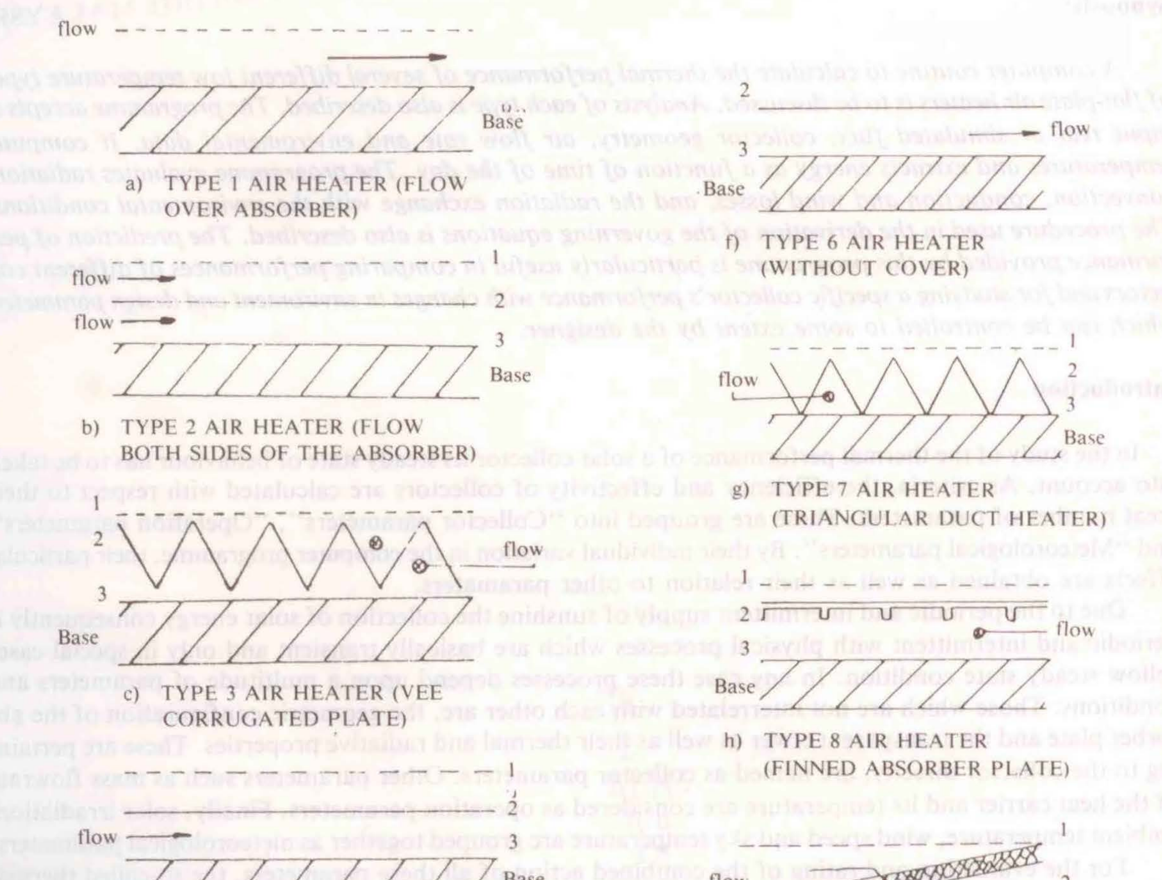


FIG. 1.0 TYPE OF SOLAR AIR HEATERS

Mathematical Formulation

The steady state governing equations of the solar air heaters are very important in an attempt to study and analyse their thermal performance by means of computer simulation. Different types of air heaters can be analysed by varying the pertinent variables of the air heaters within the computer programme. Thus, it enables one to compare the various types of air heaters. In this respect, the following assumptions have been made for simplicity. Although the analysis of the actual heat transfer behaviour is complex, with this rather simple analysis, sufficiently accurate result could still be obtained.

The pertinent assumptions are as follows;

i. Uniform temperature across any section of the flow duct, that is, a temperature gradient only exists in the flow direction along the duct.
ii. The incident insolation is taken to be normal to the cover glass.
iii. All edge effects are negligible.
iv. The collector system is in equalibrium and performance is in a steady state.
v. Surface and transmittent properties are constant over the range to which they are applied.
vi. Duct and dirt on collector are negligible.

The Hottel-Bliss generalised equation is used for calculating the thermal performance for flat-plate solar air heaters. For the fluid circulating through the collector the useful energy gained or thermal power collected, Q_U, is expressed as:

$$Q_U = F_R A_c \left((\tau\alpha)_e I - U_L(T_{p,m} - T_a)\right) \qquad \text{.......(1)}$$

where

$$F_R = \frac{GC_p}{A_c U_L}\left(1 - e^{-(A_c U_L F'/GC_p)}\right) \qquad \text{.......(2)}$$

The collector efficiency and loss factors, F' and U_L are different from one collector to another. For all of the solar collectors, both factors are derived in a steady state condition (5, 9) and are set out in Appendix (i). The heat removal factor of the collector can be conveniently expressed in term of flow factor, F'', this

$$F'' = F_R/F' = \frac{GC_p}{A_c U_L F'}\left(1 - e^{-(A_c U_L F'/GC_p)}\right) \qquad \text{.......(3)}$$

The effective transmittent-absorptance product, $(\tau\alpha)_e$, was approximated by Duffie and Beckman (5) as

$$(\tau\alpha)_e = 1.02\ \tau\alpha \qquad \text{.......(4)}$$

For vee-corrugated plates the effective absorptance is given by Kreider (10) as

$$\alpha_e = 1 - r^n = 1 - (1 - \alpha)^n \qquad \text{.......(5)}$$

For vee angle of 60°, n = 3, therefore

$$\alpha_e = 1 - r^3 = 1 - (1 - \alpha)^3 \qquad \text{.......(6)}$$

Heat input Q_{IN}, and heat loss Q_L, of the solar collector are respectively expressed as

$$Q_{IN} = A_c I \qquad \text{.......(7)}$$

and

$$Q_L = A_c U_L (T_{p,m} - T_a) \qquad \text{.......(8)}$$

The efficiency η, of the collector is given by the ratio of the heat gain, Q_U, to the heat input Q_{IN}, therefore

$$\eta = Q_U / Q_{IN} = F_R((\tau\alpha)_e - \frac{U_L}{I}(T_{p,m} - T_a)) \qquad \text{.......(9)}$$

The heat loss coefficient by conductivity through the insulated collector base U_b, is given, for an insulation thermal conductivity, K, and insulation thickness D_b, by

$$U_b = K/D_b \qquad \text{.......(10)}$$

The radiation heat transfer coefficient between heater plate and the cover, h_r, and the combined radiative and convective coefficient for heat loss through the top cover, U_t, are dependent upon plate temperature (the mean value, $T_{p,m}$). The $T_{p,m}$ in turn must be iteratively solved by applying later equations relating that temperature to useful energy gain, Q_U, equation (1).

The convective heat transfer coefficient between heated plate and flowing air, h, is calculated on the basis of air flowing between a heated plate and unheated plate which are parallel with the same Reynold number for the flow along the channel of the collector. The convective heat transfer coefficient, h, is obtained by applying the usual defination of the Nusselt number, Nu, and then using an emphirical correlation which expressed Nu, as a function of one or more other standard nondimensional groups which include the Prandtl number, Pr, Grashof number, Gr, and the Reynold number, Re. The radiation heat transfer coefficient is then given by

$$h_r = \frac{4\sigma\bar{T}^3}{(1/\varepsilon_p + 1/\varepsilon_c - 1)} \qquad \text{.......(11)}$$

where T, is the mean temperature for radiation between plate and cover. This in turn is assumed equal to the mean fluid temperature, $T_{f,m}$, with T_a and $T_{p,m}$, being ambient and plate temperature respectively expressed in Kelvin, to be compatible with Klein's emphirical expression for U_t. Thus U_t is expressed as

$$U_t = \left| \frac{N}{\frac{C}{T_{p,m}} \left(\frac{(T_{p,m} - T_a)}{(N+f)}\right)^e} + \frac{1}{h_w} \right|^{-1}$$

$$+ \frac{\sigma (T_{p,m} + T_a)(T^2_{p,m} + T^2_a)}{(E_p + 0.00591Nh_w) + (2N + f - 1 + 0.133E_p) / (E_c - N)} \qquad \text{.......(12)}$$

where,

$f = (1 + 0.089h_w - 0.1166h_w E_p)(1 + 07866N)$

$C = 520(1 - 0.000051N\beta^2)$

$e = 0.43(1 - 100/T_{p,m})$

$\sigma = 5.67 * 10^{-8}$

$h_w = 5.7 + 3.8\bar{V}_w$

The convective heat transfer coefficient, h, is determined from the correlation between Nu and Re for turbulent fully developed flow of air between parallel plates where one of them is heated. This correlation is acquired from Kay's (8) data

$$Nu = 0.0158Re^{0.8} \qquad ....(13)$$

Also, since Re is given by

$$Re = V_f D_H/\mu = GD_H/A_f/\mu \qquad ....(14)$$

therefore, the convective heat transfer coefficient is given as,

$$h = K_a Nu/D_H \qquad ....(15)$$

in which,

$$D_H = 4A_f/Prm \qquad ....(16)$$

The equations for mean plate and mean fluid temperatures, $T_{p,m}$ and $T_{f,m}$, are used in the iteration together with this equation for Q_U, in which

$$T_{p,m} = T_{f,i} + (Q_U/A_c)(1 - F_R/F')/GC_p \qquad ....(17)$$

and

$$T_{f,m} = T_{f,i} + ((Q_U/A_c)/(U_L F_R))(1 - F'') \qquad ....(18)$$

Also, the outlet temperature is expressed as

$$T_{f,o} = T_{f,i} + (Q_U/A_c)/GC_p \qquad ....(19)$$

Variables in Air Heater Performance

The main factors that determine the efficiency of heat collection of a solar air heater operating at a given air inlet temperature are,

i) air heater configuration, that is, it includes the length, depth and breadth of the air heater
ii) air mass flow through the heater
iii) spectral reflectance properties of the absorber plate
iv) spectral reflectance-transmittance properties of the cover plate
v) insulation at the absorber base
vi) solar insolation rate

Only the first four factors will be considered in this analysis. The variation of these factors in a computer routine enable one to predict the thermal performance of each type of solar air heaters described before. The insulation material used is saw dust with its thickness arbitrarily fixed. The solar insolation rate global value was obtained from weather bureau data measured and published work (2) on solar radiation in Kuala Lumpur.

Basis for Heater Comparison

For comparative purpose of the solar air heaters the configuration for each type of the solar air heaters should be the same. For air heaters with stagnant air gap, represented by air heaters types 4, 5 and 8, the depth of the stagnant air gap is taken as a ratio to the overall depth. Similarly, for the air heaters with two flow channels, it has been assumed that the heater plate is centred in the flow duct to allow equal mass flows through the top and bottom flow ducts on either sides of the absorber plate. The value of the convective heat transfer coefficients whether between the air and the cover plate or between the air and the absorber plate, for each individual type of the plane surface absorber air heaters are assumed equal to one another. In the case of the vee-corrugated plate air heaters, for vee angle of 60°, the convective heat transfer coefficient is found to be twice of that plane surface absorber air heater (3). Similarly, for absorber plate with fins, the convective heat transfer coefficient is also twice of that plane surface absorber plate, since adding fins increased both heat transfer surface as well as the turbulent rate of the airflow through the heater.

Most of the solar air heaters described before have absorber plates with different forms of geometries. In order to evaluate the hydraulic diameter for the calculation of the convective heat transfer coefficient for each type of the solar air heaters, the flow depth, flow area and the perimeter of those air heaters have to be obtained first. The geometrical relationships of the flow depth, flow are and the perimeter can be found in Appendix (ii).

Method of Numerical Solution

The Q_U, equation (1), is first evaluated with an assumed values for mean plate and mean fluid temperatures, as is h_r, U_L, F', F'', F_R and U_t, the mean fluid and plate temperatures are then evaluated. An absolute error comparison is made between the i^{th} and $(i+1)^{th}$ value of $T_{p,m}$ at a particular time step, with the iteration continued by plugging the new $T_{p,m}$ value back into Q_U and repeating cycle, until the tolerance is met. The value of important changing variables are then given and the next time step is taken. Fig. (2) shows the flowchart of the Fortran programme which is used in this analysis.

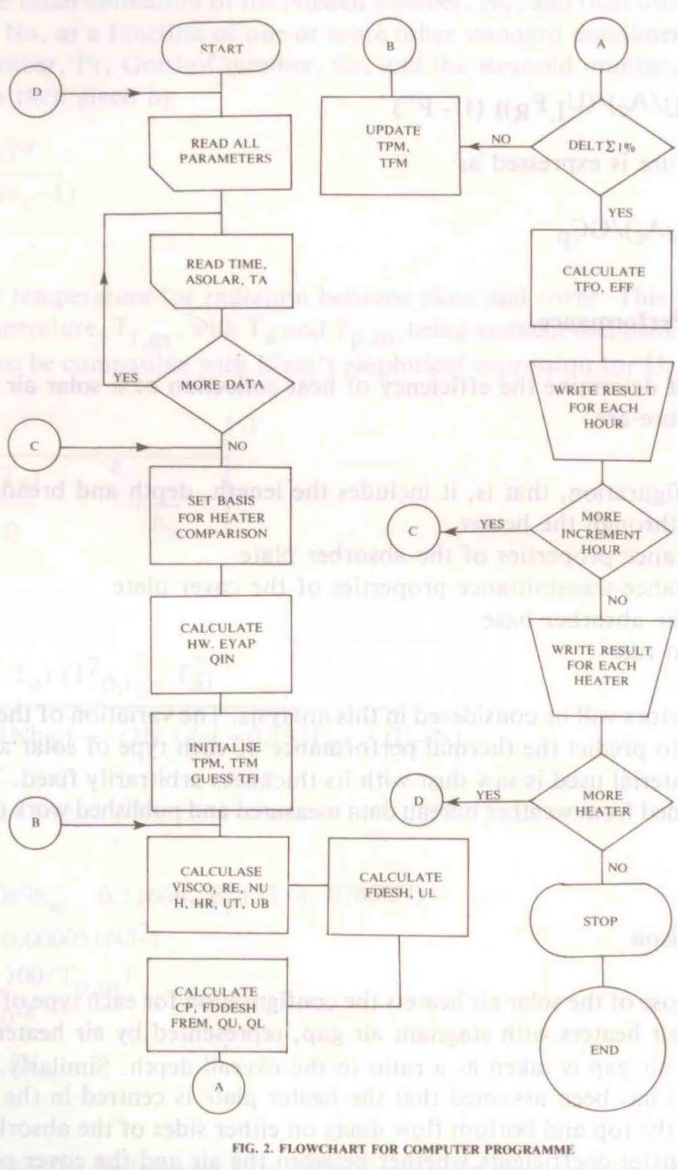


FIG. 2. FLOWCHART FOR COMPUTER PROGRAMME

Result and Discussion

The effects of various parameters are investigated individually by varying one parameter and leaving the others constant. The effects of collectors physical parameters, in the construction of solar collectors, are of great interest. These are greatly influenced by the choice of configuration, material or geometry. The emissivity of the glass cover, for example, effects the overall heat loss of the collector. Their influence upon collectors efficiency is presented in Fig. (3). As a parameter, the infrared emissivity is varied between $\varepsilon_c = 0.1$ and $\varepsilon_c = ,0.9$. The infrared transmission through the glass cover was assumed constant. As can be expected, the heat losses from glass cover decrease and thus the efficiency increases when the emissivity assumes small values.

The radiation heat losses from the absorber plate can be reduced when the emissivity of the absorber in the infrared is reduced. Its absorbitivity, on the other hand, should be as high as possible. The effects of these parameters on collectors efficiency are shown in Fig. (4) and Fig. (6). The effect of different mass flow rates upon collectors efficiency is presented in Fig. (9). Consequently the efficiency is increased with the increased of mass flow rate.

The effects of collector depth upon the efficiency and the outlet temperature are shown in Fig. (7) and Fig. (8). The efficiency is also dependent upon the length of the collector. This is shown in Fig. (10). As the length increases, the efficiency of the collectors is decreased. This is due to the increases of collector surface area, thus it also increases the overall heat loss from the collector.

The variation of the collectors efficiency with the time of the day is clearly shown in Fig. (11). The optimum efficiency of all the collectors occurs between 12 noon and 1 p.m. The best type air heater is type 8 and the worst is type 6. It was also found that the vee corrugated air heaters performs better than the plane surface type air heaters. The highest outlet temperatures were produced by heaters with two flow channels, that is, heaters types 2 and 3.

NOMENCLATURE

A_c	collector surface area
A_f	cross-sectional area of flow duct
B	breadth of the collector
C_p	specific heat to air
D	overall collector depth
D_b	depth of insulation material
D_f	depth of flow duct
D_H	hydraulic diameter
ε_c	emissivity of cover plate
ε_p,	emissivity of absorber plate
F'	collector efficiency factor
F''	collector flow factor
F_R	collector heat removal factor
G	mass flow rate
h	convective heat transfer coefficient
h_r	radiative heat transfer coefficient
I	incident solar insolation on collector cover
K	thermal conductivity of the insulation material
K_a	thermal conductivity of air
L	length of the collector
N	number of transparent cover
n	number of reflection by the absorber plate
Q_{IN}	heat input to the collector
Q_L	heat loss from the collector
Q_U	useful heat gain by the collector
r	reflection factor of the absorber plate
T_a	ambient air temperature
$T_{f,i}$	inlet fluid temperature
$T_{f,m}$	mean fluid temperature
$T_{f,o}$	outlet fluid temperature

$T_{p,m}$ mean plate temperature
U_b heat transmission coefficient from the absorber plate to the ambient air under collector
U_L collector overall loss coefficient
U_t top loss coefficient
β collector tilt (degree)
σ Stefan-boltsman constant
τ transmittance property of the cover plate
$\propto$ absorptance property of the absorber plate
θ vee angle of corrugated plate
Φ channel angle of clipped v heater

Appendix (i)

Efficiency and Loss Factors of the Solar Air Heaters

i) Flow Both Sides of the Absorber

$$F' = D/(2h_1h_2P + 2h_2U_bU_t + h_{r21}((h_1 + h_{r23})(P + 2h_2) + U_b(2h_2 + U_t) + h_1h_{r23}(h_1(P + 2h_2) + 2h_2U_t + U_bU_t))$$

$$\begin{aligned}U_L = {} & (4h_1h_2U_bU_t + h_{r21}((h_1 + h_2)(h_{r23}U_t + U_bU_t \\ & + h_{r23}U_b) + h_1h_2U_t) + 2h_{r23}U_b(h_1h_2 + h_1U_t + h_2U_t) \\ & + (1\text{-}n)h_1U_t(h_1(2h_2 + h_{r21}) + h_{r21}Q) \\ & + (1 + n)h_1U_b(h_1(h_{r23} + 2h_2) + h_{r21}Q)\end{aligned}$$

where

$$D = (2h_1h_2P + 2h_2U_bU_t + h_{r21}(Q(h_1 + h_{r23} + U_b) + h_1h_{r23}) + h_{r23}Q(h_1 + U_t))$$

$$P = h_1 + U_b + U_t$$

$$Q = h_1 + 2h_2$$

$$n = 1\text{-}(T_{ft}\text{-}T_a)/(T_{fb}\text{-}T_a)$$

For type 3, h_2 is replaced by $h_2/\mathrm{Sin}(\theta/2)$

ii) Flow Over the Absorber

$$F' = C/(h_1(h_2 + U_b) + h_2U_t + h_{r21}(h_1 + h_2 + U_b + U_t) + U_bU_t)$$

$$U_L = (h_1(h_2U_b + h_2U_t + U_bU_t) + h_2U_bU_t + h_{r21}(h_1U_b + h_1U_t + h_2U_b + h_2U_b))/C$$

where

$$C = h_1h_2 + h_2U_t + h_{r21}(h_1 + h_2)$$

iii) Flow Under the Absorber

$$F' = E/(H + U_b(h_2 + h_{r23} + U_t) + U_t(h_{r23} + h_3))$$

$$U_L = (H(U_b + U_t) + (h_2 + h_3)U_bU_t)/E$$

where

$$E = H + h_2U_b$$

$$H = (h_2h_3 + h_2h_{r23} + h_3h_{r23})$$

For type 5, h_2 is replaced by $h_2/\text{Sin}\ (\theta/2)$

iv) Triangular Duct Air Heater

$$F' = h_2/(h_2 + U_L)$$

$$U_L = U_b + U_t$$

v) Finned Absorber Plate Air Heater

$$F' = F'_o\ 1 + \frac{(1/F'_o - 1)}{(1/F_d + (1/F_bF'_o)\ (LK_1M_1/WK_1M_1)\ (c/b)^2}$$

$$U_L = U_t + U_b$$

where

$$F'_o = 1/(1 + U_L/h_1)$$

$$F_b = (1/b)\ \tanh\ (b)$$

$$F_d = (2/d)\ \tanh\ (d/2)$$

and

$$a^2 = W^2U_L/K_1M_1$$

$$b^2 = 2L^2h_2/K_2M_2$$

$$c^2 = W^2h_1/K_1M_1$$

$$d^2 = a^2 + c^2$$

vi) Wire Mesh Air Heater

$$F' = 1/(1 + (U_ch_2)$$

$$U_L = U_a + U_b + U_c$$

$$U_a = 1/(1/h_1 + 1/U_t)$$

$$U_b = 1/(1/h_{r21} + 1/U_t)$$

vii) Clipped-v Air Heater

$$F' = 1/(1 + \frac{h_{r21}U_t}{Ah_2^2\ th_2(h_{r21} + A(U_t + h_{r21}))})$$

$$U_L = \frac{U_t + U_b}{1 + \frac{U_t + U_b}{h_2 + h_{r21}}(1 + 1/A)}$$

where

$$A = l_2/l_1 + (1 - l_2/l_1)/\text{Sin}\ (\Phi/2)$$

Appendix (ii)

For a given overall depth, D, ratio of the stagnant air gap, a, and the base thickness, D_b, the geometrical relationship for each types of the solar air heaters are obtained as follows:

1. Types 1, 6 and 9 (Rectangular ducts air heaters)

   $D_f = D - D_b$

   $A_f = B * D_f$

   $Prm = 2(B + D_f)$

2. Type 2 (Rectangular duct air heater with two flow channels)

   $D_f = 0.5(D - D_b)$

   $A_f = B * D_f$

   $Prm = 2(B + D_f)$

3. Types 3 and 5 (vee corrugated air heaters)

   $D_f = (1-a)D - D_b$

   $A_f = 0.5B(D_f + aD)$

   $Prm = B + 2aD + B(\text{Cos}(\theta/2)\text{Tan}(\theta/2))$

4. Type 4 (Rectangular duct air heater with stagnant air gap)

   $D_f = (1-a)D - D_b$

   $A_f = B * D_f$

   $Prm = 2(B + D_f)$

5. Type 7 (Triangular duct air heater)

   $D_f = (1-a)D - D_b$

   $A_f = 0.5BD_f$

   $Prm = B(1 + \text{Cos}(\theta/2)/\text{Tan}(\theta/2))$

6. Type 8 (Finned absorber plate)

   $D_f = (1-a)D - D_b$

   $A_f = BD_f - (B/W)(MIAL)$

   $Prm = 2((BAL/W) + B + D_f)$

7. Type 10 (Clipped v through air heater)

   $D_f = D - D_b$

   $A_f = (BD_f/L1)(D_f\text{Tan}\,\Phi + L2)$

   $Prm = B((2/AL1)(2D\,\text{Cos}(\Phi) + AL2) + 1)$

Appendix (iii)

The values of the parameters used in this analysis were:

$L = 2.5$ m, $B = 0.5$ m, $D = 0.08$m, $V_w = 5$ m/s

$G = 200$ Kg/hr, $\alpha = 0.9$, $\tau = 0.9$, $E_c = 0.9$, $E_p = 0.3$,

$\beta = 0^o$, $\theta = 60^o$, $\Phi = 45^o$

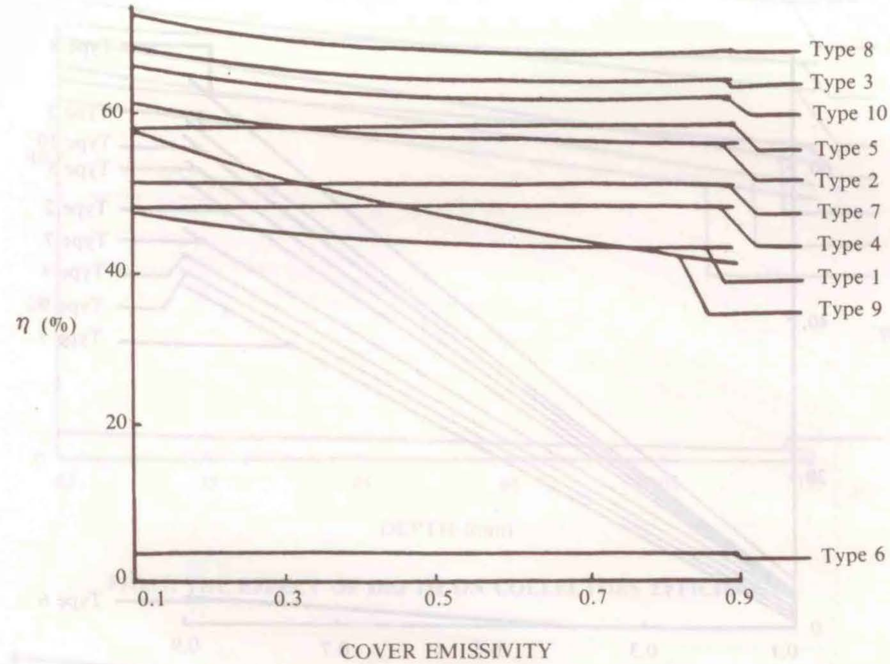


FIG. 3. THE EFFECT OF COVER EMISSIVITY ON COLLECTORS EFFICIENCY

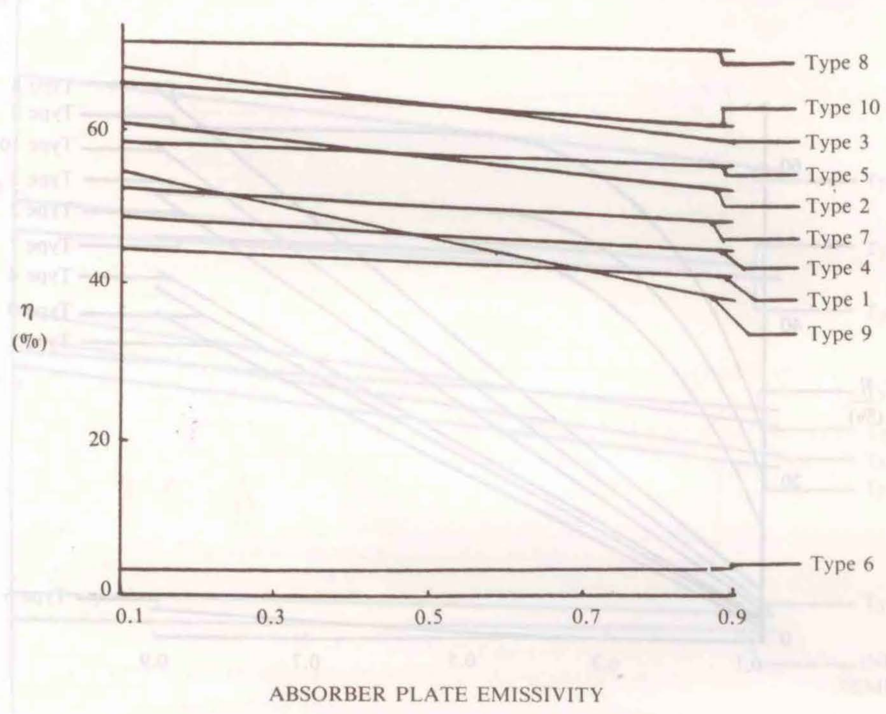
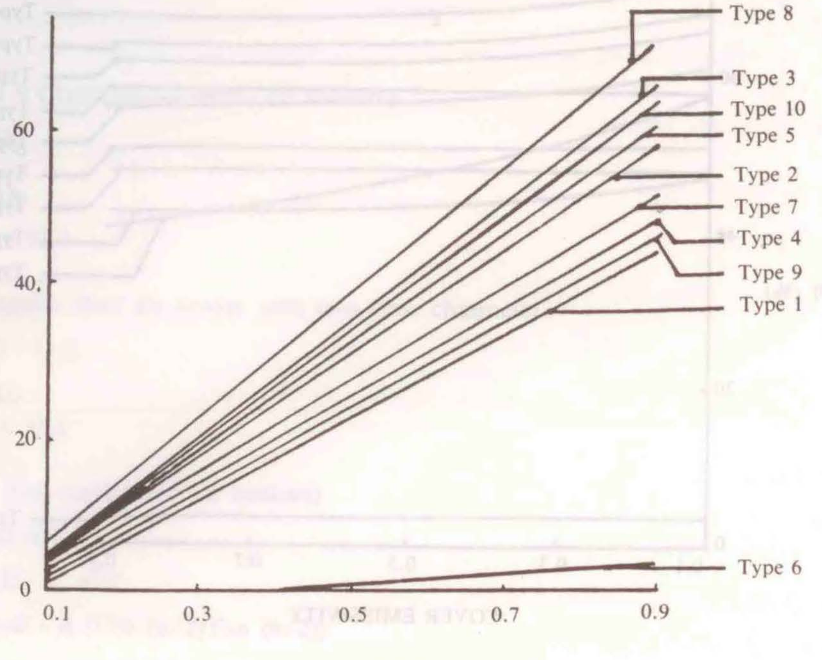


FIG. 4. THE EFFECT OF ABSORBER PLATE EMISSIVITY ON COLLECTORS EFFICIENCY



FIFIG. 5. THE EFFECT OF COVER TRANSMITTANCE ON COLLECTORS EFFICIENCY

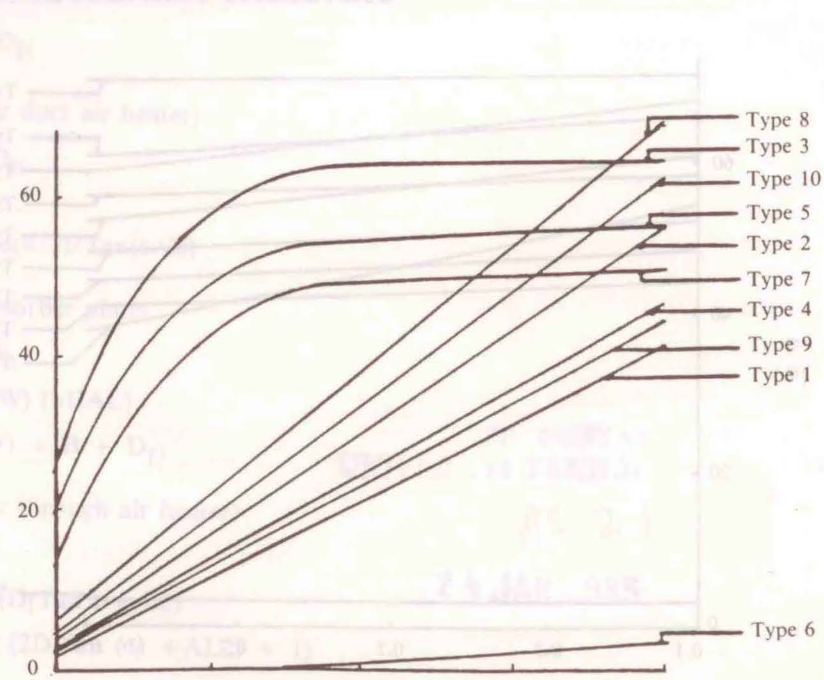


FIG. 6. THE EFFECT OF ABSORBER PLATE ABSORPTANCE ON COLLECTORS EFFICIENCY

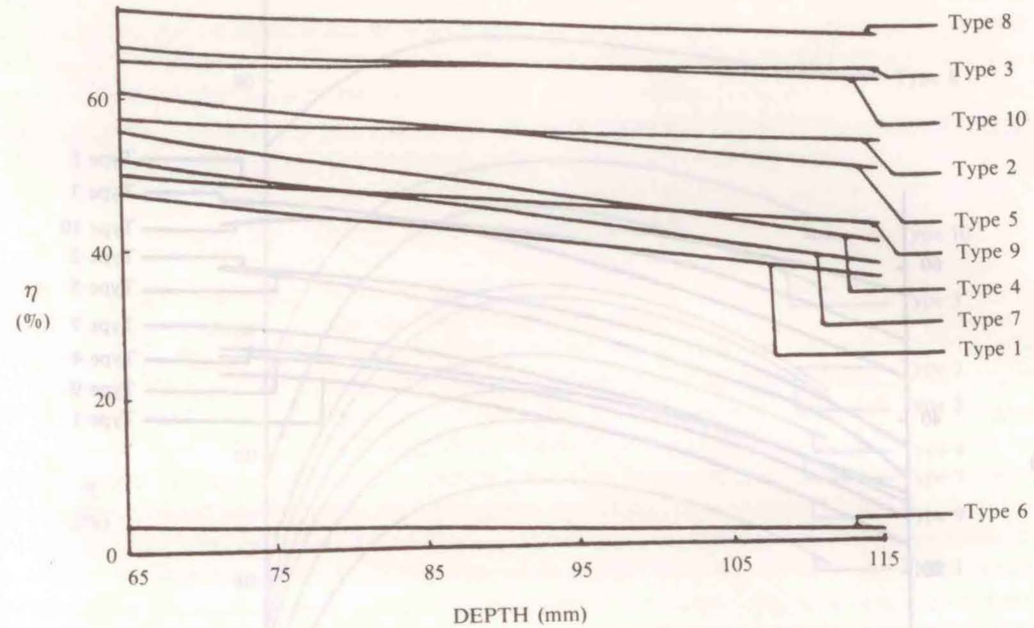


FIG. 7. THE EFFECT OF DEPTH ON COLLECTORS EFFICIENCY

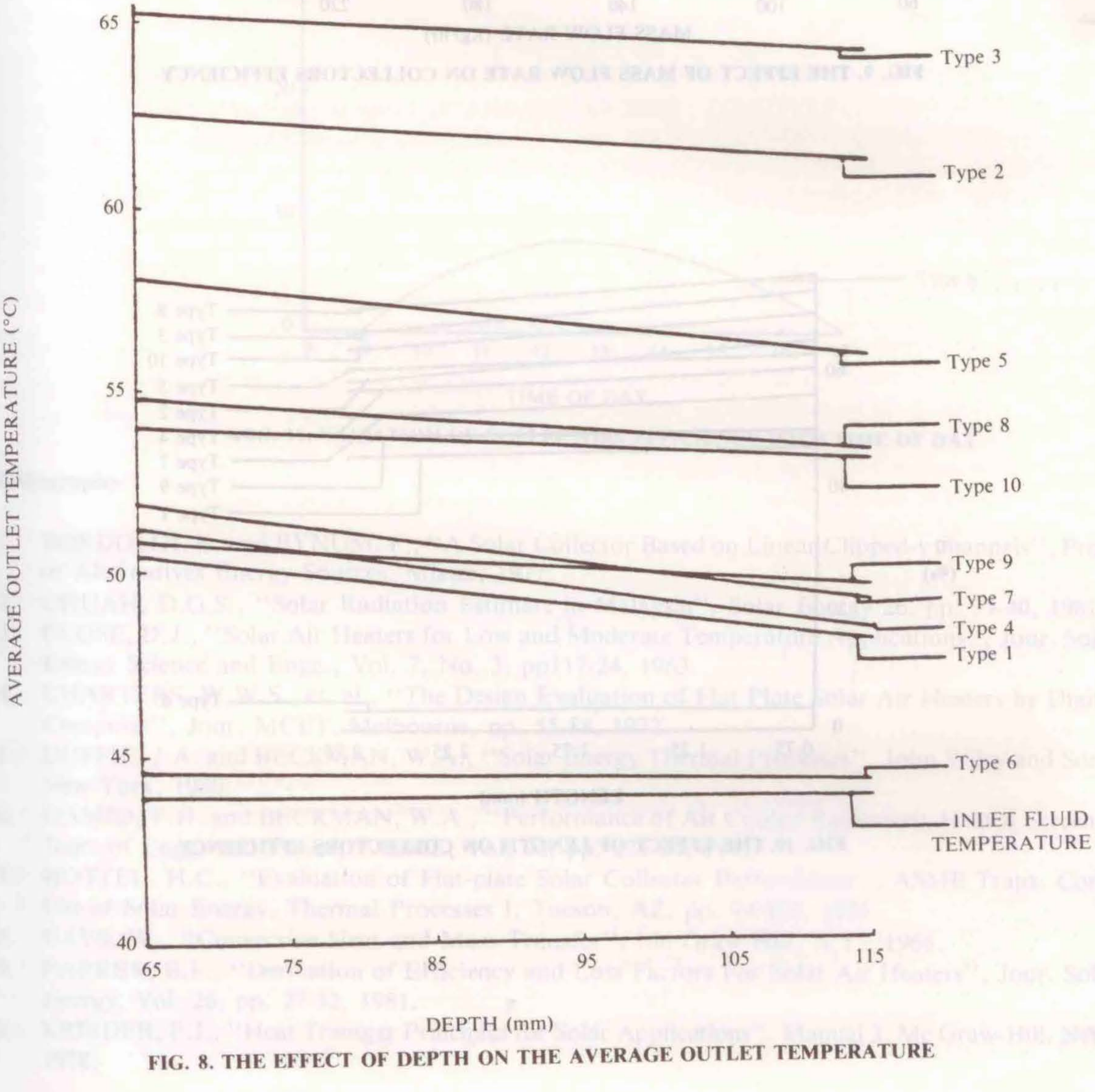


FIG. 8. THE EFFECT OF DEPTH ON THE AVERAGE OUTLET TEMPERATURE

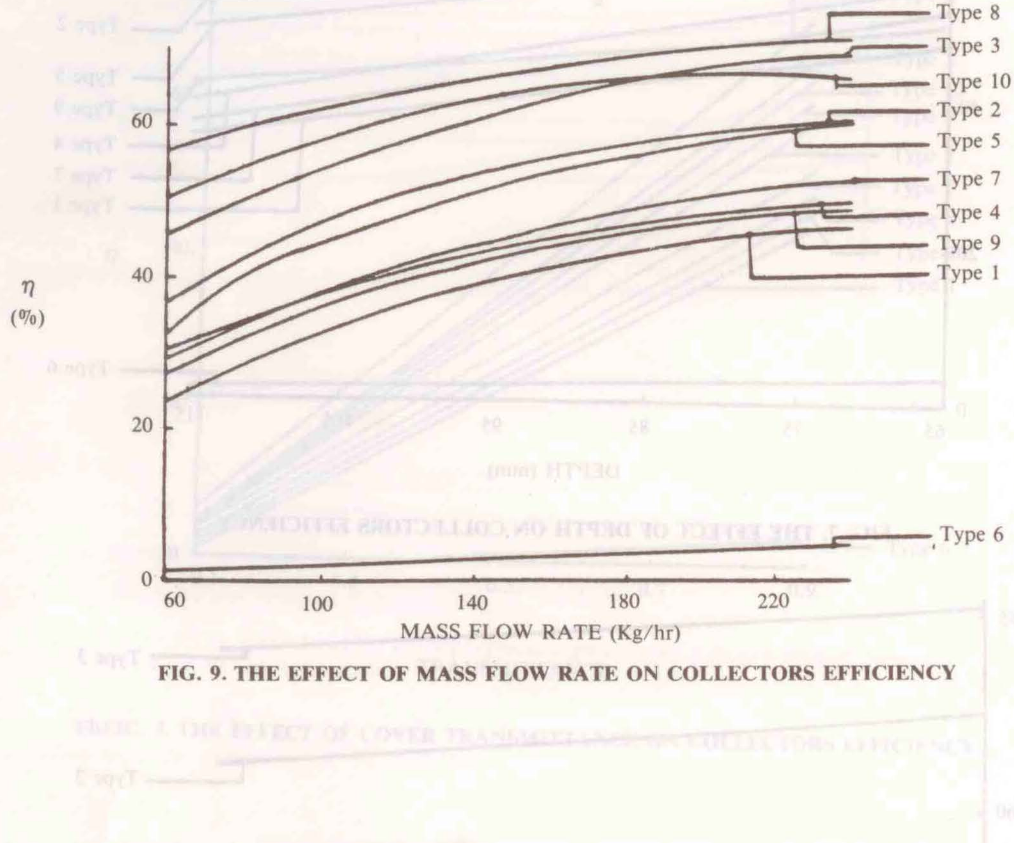


FIG. 9. THE EFFECT OF MASS FLOW RATE ON COLLECTORS EFFICIENCY

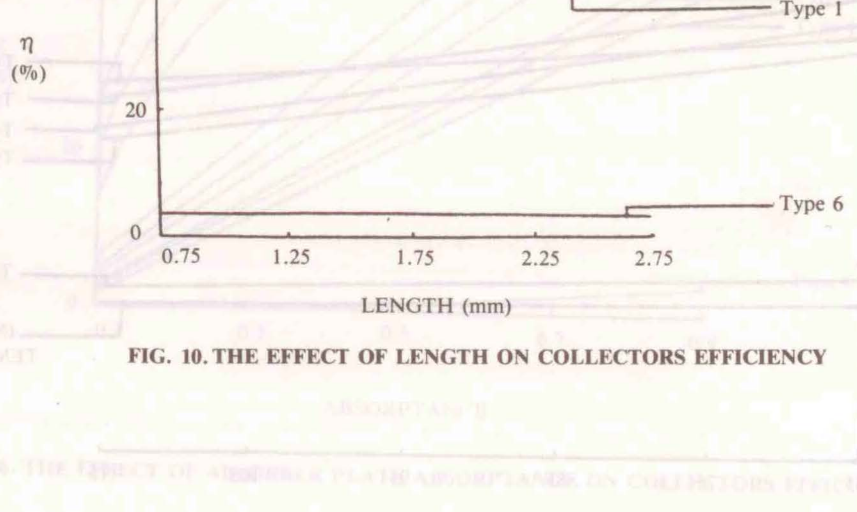


FIG. 10. THE EFFECT OF LENGTH ON COLLECTORS EFFICIENCY

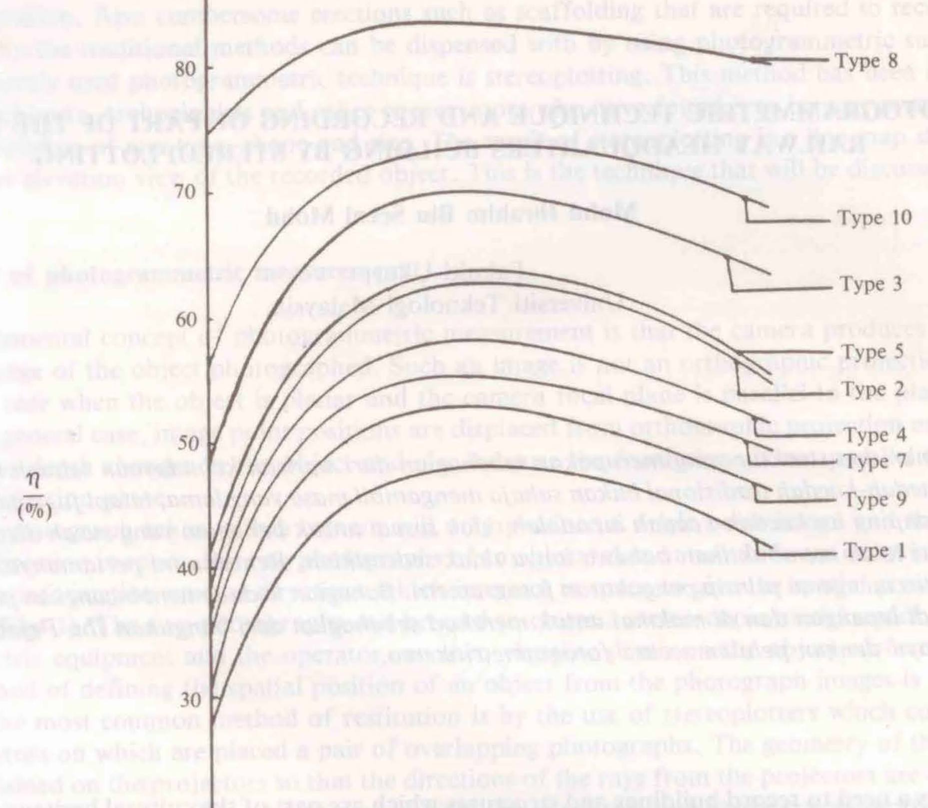


FIG. 11. VARIATION OF COLLECTORS EFFICIENCY WITH TIME OF DAY

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