Proceedings of National Conference on Multidisciplinary Engineering Sciences and Information technology (NCMESIT, 2020) in Marudhar Engineering College on the 23rd & 24th of November 2020
World Journal of Research and Review (WJRR)
ISSN: 2455-3956, Special Volume, Issue-1, November2020 Pages 15-20

Effect of Fraction of Mass Flow Rate on Collector's Efficiency on Double Flow Solar Air Heater with Longitudinal Fins

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***Abstract-* The performance of double-flow types solar air heaters with longitudinal fins attached in which air is flowing simultaneously over and under the absorbing plate, is more effective than that of the devices with only one flow channel over or under the absorbing plate because the heat-transfer area in double-flow systems is double. The effect of the fraction of mass flow rate in the upper or lower flow channel of a double-flow device on collector efficiencies, has also been investigated theoretically. Considerable improvement in collector performance is obtained by employing a double-flow type solar air heater, instead of using a single flow device, if the mass flow rates in both flow channels are kept the same.**

***Index Terms-* Solar Air Heater, Fins, Mass flow rate, solar intensity**

I. INTRODUCTION

Solar air heaters, in which energy transfer comes from a distant source of radiant energy to air, may be used for space heating, drying and paint spraying operations. Without optical concentration, the flux of incident radiation is approximately from 830 W/m2 up to 1100 W/m2 and flat-plate solar collectors are designed for applications requiring energy delivery at moderate temperatures. The solar air heater occupies an important place among solar heating systems because of minimal use of materials, and the direct use of air as the working substance reduces the number of required system components. In many industrial applications where air recirculation is not practical because of contaminants, heated outside air is used, especially for supplying fresh air to hospitals. Further, heating of ambient air is an ideal operation for a collector as it operates very close to ambient temperature.

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II. THEORY:

Model construction and working:

In present study, the solar air heater constitutes a wooden box whose bottom and sides are insulated. The absorber plate has longitudinal fins on the both sides. The cover plate comprises of two glass sheet. The air flows through ducts above and below the absorber plate.

The longitudinal fins attached above and below the absorber plate increase the surface area. This increased surface area serves to transfer more heat. When air flows through the ducts it absorbs heat from surface area and gets heated.

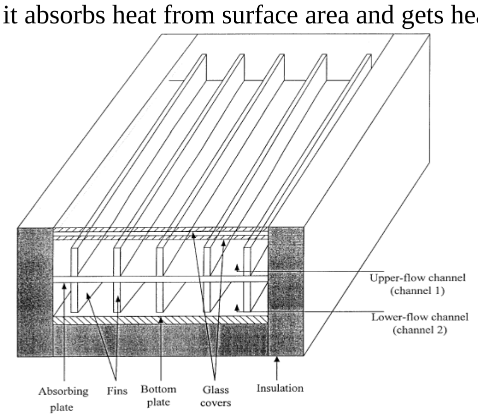

Fig 1 Double flow solar air heater with fin attached

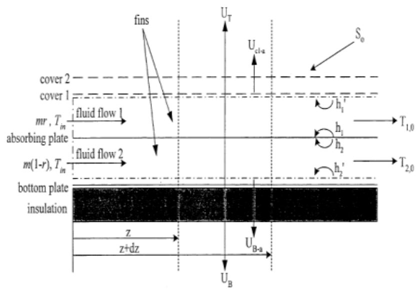

Fig 2 Energy flow diagram for double flow solar air heater with fin attached

III. ENERGY BALANCE EQUATION FOR DIFFERENTIAL LENGTH DZ:

The structure of a double-flow type solar air heater with fins attached may be illustrated by the schematic diagram of Fig. 1 while the energy-flow diagram of such a device is presented in Fig. 2 As seen in Fig.2 two air streams (fluid 1 and fluid 2) of different flow rates but with total flow rate fixed, are flowing steadily and simultaneously through two separate channels (above and below the absorbing plate) for heating. The energy balances will be taken under the following assumptions.

1. Flow is steady.

2. The temperatures of absorbing plate, bottom plate, and bulk fluids are function of flow direction only.
3. Both the glass covers and fluid do not absorb radiant energy.
4. From the fig it is clear that a strip of length dz is considered at a distance of z from the entrance of the air whose temperature is T_p. the mean temperatures of the air streams are T_{f1} and T_{f2} respectively.

If the absorbing surfaces are attached by fins, then heat transfer by fins to fluid

$q_1 = h_1 \phi_1 A_C (T_p - T_{f1})$

and

$q_2 = h_2 \phi_2 A_c (T_p - T_{f2})$,

Energy in = Energy out

Accordingly, for glass cover 1 (inner cover),

$$h_{r.p-c_1}(T_p - T_{c_1}) + h_1'(T_{f_1} - T_{c_1}) = U_{c_1-a}(T_{c_1} - T_a) \quad (1)$$

for the absorbing plate

$$S_0 \alpha_p \tau_g^2 U_T = (T_p - T_a) + U_B(T_p - T_a) + \phi_1 h_1 (T_p - T_{f_1}) + h_2 \phi_2 (T_p - T_{f_2}) \quad (2)$$

Where fin effectiveness and fin efficiency

$$fin\,effectiveness\,\emptyset_i = 1 + \left(\frac{A_{f,i}}{A_c}\right) \cdot \eta_{f,i}$$

$$fin\,efficiency\,\eta_{f,i} = \frac{\tanh\sqrt{2h_i/ktw_2}}{\sqrt{2h_i/ktw_2}}$$

for the bottom plate,

$$h_2'(T_{f_2} - T_R) + h_{r.p-R}(T_p - T_R) = U_{B-a}(T_R - T_a) \quad (3)$$

for fluid 1 (fluid in upper channel)

$$\phi_1 h_1 (T_p - T_{f1}) = \frac{m.r.C_p}{W}\left(\frac{dT_{f1}}{dz}\right) + h_1'(T_{f1} - T_{cl}) \quad (4)$$

for fluid 2 (fluid in lower channel)

$$\phi_2 h_2 (T_p - T_{f_2}) = \frac{m(1-r)C_p}{W}\left(\frac{dT_{f_2}}{dz}\right) + h_2'(T_{f_2} - T_R) \quad (5)$$

A. Collector efficiency:

The useful gains of energy brought out by fluid 1 and fluid 2 in the upper and lower flow channels, respectively are,

$$Q_{u_1} = mrC_p(T_{1,0} - T_{in}) = MrA_C(T_{1,0} - T_{in}) \quad (6)$$

$$Q_{u_2} = m(1-r)C_p(T_{2,0} - T_{in}) = M(1-r)A_C(T_{2,0} - T_{in}) \quad (7)$$

Where $T_{1,0}$ and $T_{2,0}$ are outlet temperature of fluid 1 and fluid 2 respectively

$T_{1,0}$ and $T_{2,0}$ can be obtained by putting $\xi = 1$ and $Z = L$

$$T_{1,0} = \frac{Y_1 - B_5/(1-r)}{B_4/(1-r)} C_1 e^{(y_1/M)} + \frac{Y_2 - B_5/(1-r)}{B_4/(1-r)} C_2 e^{(y_2/M)} - \frac{B_5}{B_4}\frac{(B_3B_4 - B_1B_6)}{(B_1B_5 - B_2B_4)} - \frac{B_6}{B_4} + T_a \quad (8)$$

$$T_{2,0} = C_1 e^{(y_1/M)} + C_2 e^{(y_2/M)} + \frac{(B_3B_4 - B_1B_6)}{(B_1B_5 - B_2B_4)} + T_a \quad (9)$$

The total energy gain is

$$Q_u = Q_{u_1} + Q_{u_2} \quad (10)$$

The collector efficiency is obtained

$$\eta = \frac{Q_u / A_c}{S_0}$$

Q_u/A_c = Total solar energy gain per unit collector area

$$Q_u = Q_{u_1} + Q_{u_2}$$
$$= MrA_C(T_{1,0} - T_{in}) + M(1-r)A_C(T_{2,0} - T_{in})$$
$$= MA_C\{rT_{1,0} + (1-r)T_{2,0} - T_{in}\} \quad (11)$$

$$\eta = \frac{MA_C(rT_{1,0} + (1-r)T_{2,0} - T_{in})/A_C}{S_0}$$

$$\therefore \eta = (M/S_0)[(rT_{1,0} + (1-r)T_{2,0} - T_{in})] \quad (12)$$

B. Heat transfer coefficient

The resistances to energy loss through the bottom and edges of the collector and form the surfaces of the bottom and edges of the collector to the ambient are mainly the resistance to heat flow through the insulation by conduction, i.e.

$$U_{B-a} \approx K_s / l_s \quad (13)$$

$$\frac{1}{U_B} = \frac{1}{U_{B-a}} + \frac{1}{h_2'}\frac{l_B}{k_B} \quad (14)$$

An empirical equation for U_T was developed by Klein, following the basic procedure of Hottel and Woertz **.** For the horizontal collector with two glasses covers shows in Figs. 3 and 4

$$U_T = \left[\frac{2(T_{pm}/520)}{\left[\dfrac{(T_{pm} - T_a)}{2 + (1 + 0.089 h_w - 0.1166 h_w \epsilon_p)(1 + 0.07866 x 2)}\right]} + \frac{1}{h_w}\right]^{-1} +$$
$$\sigma(T_{pm} + T_a)(T_{pm}^2 + T_a^2)/\{(\epsilon_p + 2 x 0.00591 h_w)^{-1} + [2 x 2 + (0.089 h_w$$
$$- 0.1166\ h_w \epsilon_p)(1 + 0.07866\ x\ 2) - 1 + 0.133 \epsilon_p]/\epsilon_g - 2\} \quad (15)$$

The heat transfer coefficient between air and two and the duct walls may be assumed to be equal i.e.

$$h_1 = h_1' \text{ and } h_2 = h_2' \quad (16)$$

In the study of the solar air heater and collector-storage walls, it is necessary to know the forced convection heat transfer coefficient between flat plates. For air the of lowing correlation may be derived from Kay's [**13**] data with the modification of Mc Adams [**16**] for turbulent flow in a short conduit:

$$Nu_i = h_i De/k = 0.0158\ R_{e.i}^{0.8}\left[1+(De/L)^{0.7}\right] \qquad i=1,2 \tag{17}$$

While for laminar flow, the equation presented by Heaton et al.[**15**] may be used

$$Nu_i = 4.4 + \frac{0.00398(0.7R_{e.i}\ De/L)^{1.66}}{1+0.0114(0.7R_{e.i}\ De/L)^{1.12}} \tag{18}$$

The characteristic length is the equivalent diameter of the duct:

$$De = 4\,(HW - t\,H)/2\,(W+H) \tag{19}$$

The Reynolds number for the rectangular ducts are then defined by

$$Re_1 = \frac{De\ v_1\rho_1}{\mu_1} = \frac{[4(HW-tH)/2(W+H)][mr/(\rho_1 WH)\rho_1]}{\mu_1} = \frac{2mr}{\mu_1(W+H)} \tag{20}$$

and

$$Re_2 = \frac{De\ v_2\rho_2}{\mu_2} = \frac{2m(1-r)}{\mu_2} \tag{21}$$

The thermal resistance from glass cover 1 through glass cover 2 to the ambient, may be expressed as

$$\frac{1}{U_{c_1-a}} = \frac{1}{U_T} + \frac{1}{h_{r.p-c_1}+h_1} \tag{22}$$

or

$$\frac{1}{U_{C_1-a}} = \frac{1}{h_w + h_{r.c_2-a}} + \frac{1}{h_{c_1-c_2}+h_{rc_1-c_2}} \tag{23}$$

Where the heat transfer coefficient for free convection of air between two glass covers may be estimated by Hottel's empirical equation as

$$h_{c_1-c_2} = 1.25\left(T_{c_1m} - T_{c_2m}\right)^{0.25} \tag{24}$$

and the convective heat transfer coefficient for flowing over the outside surface of glass cover 2, will be calculated by the following empirical equation given by Mc Adams as

$$h_w = 5.7 + 3.8\,V \tag{25}$$

Finally, the radiation coefficients between the two air duct surfaces may be estimated by assuming a mean radiant temperature equal to the mean fluid temperature, viz.

$$h_{r.p-C_1} \approx \frac{4\sigma T_{1.m}3}{\left[(1/\in_p)+(1/\in_{C_1})-1\right]} \tag{26}$$

$$h_{r.p-R} \approx \frac{4\sigma T_{2.m}^2}{\left[(1/\in_p)+(1/\in_R)-1\right]} \tag{27}$$

While those between two glass covers and from cover 2 to the ambient are, respectively,

$$h_{r.C_1-C_2} \approx \frac{\sigma\left(T_{C_1.m}^2 + T_{C_2.m}^2\right)\left(T_{C_1.m}+T_{C_2.m}\right)}{(1/\in_g)+(1/\in_g)-1} \tag{28}$$

$$h_{r.C_2-a} \approx \in_g \sigma\left(T_{C_2.m}^2 + T_a^2\right)\left(T_{C_2.m}+T_a\right) \tag{29}$$

IV. CALCULATION METHOD FOR COLLECTOR EFFICIENCY:

The procedure for calculating of theoretical values of ŋ will now be described. With known collector geometries (L,W,H) and system properties (tg =0.875; ap =0.96; eg = 0.94; ep = 0.8; er = 0.94; Ub = hs / ls=0) as well as the given operating conditions (S0,Ta,V,m,r,Tin) the value of ŋ is calculating from equation (40) coupled with all appropriate equations. In the present work, the mathematical model has been adopted from the work of Ho-Ming Yeh *, etl, in "The improvement of collector efficiency in solar air heaters by simultaneously air flow over and under the absorbing plate" the values of operating parameters have been adopted from the same paper. Programming has been developed for longitudinal fins attached solar air heater and without fin solar air heater. The results obtained for both the cases are compared.

V. RESULTS AND DISCUSSION

Table 1 List of input data

S.NO.	INPUT DATA	NUMERICAL VALUE
1	Length of absorber plate (L)	30 c.m.
2	Width of absorber plate (B)	30 c.m.
3	Height of the air tunnels in a double-flow solar collector (H)	2.75 c.m.
4	Distance between fins (w_1)	6 c.m.
5	Height of the fins (w_2)	5.5 c.m.
6	Wind velocity (V)	1 m/s
7	Mass flow rate (m)	0.010-0.024 m/s
8	Solar incident (S_o)	830-1100 W/m^2
9	Thickness of fin (t)	0.1 c.m.
10	Mass fraction (r)	0-1
11	Absorptivity (a_p)	0.96
12	Emissivity of glass, absorber, bottom plate (e_g, e_p, e_r)	0.94, 0.8, 0.94
13	Transmittance of the glass cover (τ_g)	0.875
14	Thermal conductivity of fin (k)	50 (kJ h^{-1}m^{-1} K^{-1})
15	Stefan–Boltzmann constant (σ)	5.67x10^{-8}KW m^{-2} k^4
16	No of fins is used	5

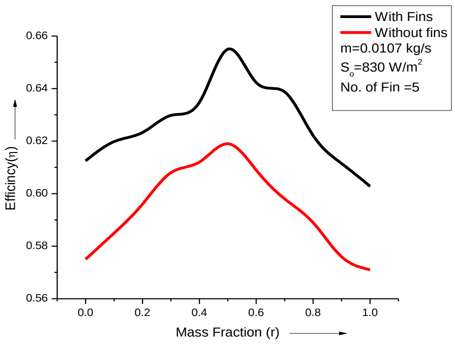

Fig 3 Effect of the fraction of mass-flow rate on collector efficiency in a double-flow type solar air heater.

Table 2 Efficiency with and without fins at different mass fraction

Mass Fraction(r)	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7
Efficiency with fins (Ƞ)	0.612	0.619	0.623	0.629	0.634	0.655	0.642	0.638
Efficiency without fins (Ƞ)	0.575	0.585	0.596	0.608	0.612	0.619	0.609	0.598

Fig 3 shows the collector efficiency Vs fraction of mass flow rate of air for double flow with fin attached type solar air heater for S_0=830 W/m^2, mass flow rate m=0.0107 kg/s. Fig represents a gradual increase in efficiency with increased mass fraction (r) up to the point r =0.5 where it acquires the maximum value and decreases thereafter.
Efficiency of solar air heater is increased for solar incident S_0=1100 W/m^2 as compared to fig 1, where solar incident S_0=830 W/m^2
The reason behind the increase in efficiency up to point r =0.5 is that as the amount of air in upper duct increases, it absorbs heat from the absorber plate as well as longitudinal fins and gets heated up well. The efficiency reaches the maximum value when the amount of air is equally divided in upper and lower duct. After the point r=0.5, the amount of air keeps increasing in upper duct resulting in poor heat absorption. This less heating of air in upper duct reduces efficiency afterwards. The red curve in the fig shows variation in efficiency with respect to fraction mass for solar air heater without fins. It is clear that the efficiency in case of without fins is much less than in case of with fins. The fig represents same variation in efficiency in case of without fins.

Table 3 Efficiency with and without fins at different mass fraction

Mass Fraction(r)	0	0.1	0.2	0.3	0.4	0.5	0.6
Efficiency With Fins(Ƞ)	0.653	0.659	0.663	0.669	0.675	0.685	0.672
Efficiency Without Fins(Ƞ)	0.625	0.638	0.648	0.657	0.663	0.668	0.660

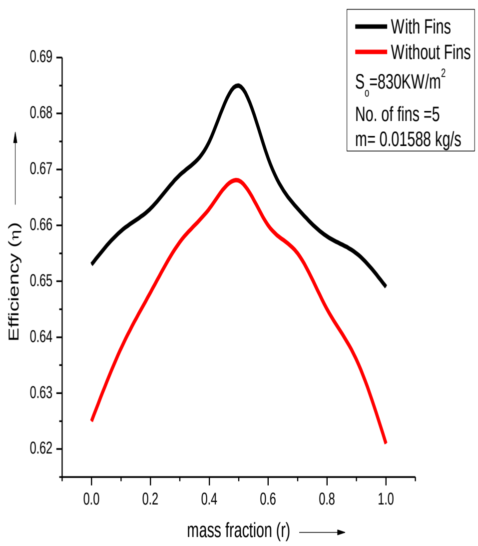

Fig 4Effect of the fraction of mass-flow rate on collector efficiency in a double-flow type solar air heater.

Fig 4shows the collector efficiency Vs fraction of mass flow rate of air for double flow with fin attached type solar air heater for S_0=830 W/m^2, mass flow rate m=0.0158 kg/s. Fig represents a gradual increase in efficiency with increased mass fraction (r) up to the point r =0.5 where it acquires the maximum value and decreases thereafter.
The reason behind the increase in efficiency up to point r =0.5 is that as the amount of air in upper duct increases, it absorbs heat from the absorber plate as well as longitudinal fins and gets heated up well. The efficiency reaches the maximum value when the amount of air is equally divided in upper and lower duct. After the point r=0.5, the amount of air keeps increasing in upper duct resulting in poor heat absorption. This less heating of air in upper duct reduces efficiency afterwards.
The red curve in the fig shows variation in efficiency with respect to fraction mass for solar air heater without fins. It is

clear that the efficiency in case of without fins is much less than in case of with fins. The fig represents same variation in efficiency in case of without fins.

Table 4 Efficiency with and without fins at different mass flow rate

Mass Flow Rate(m) kg/s	0.010	0.012	0.014	0.016	0.018	0.020	0.022	0.024
Efficiency without fins (Ƞ)	0.619	0.636	0.653	0.678	0.689	0.703	0.718	0.732
Efficiency with fins (Ƞ)	0.665	0.678	0.689	0.695	0.711	0.724	0.733	0.742

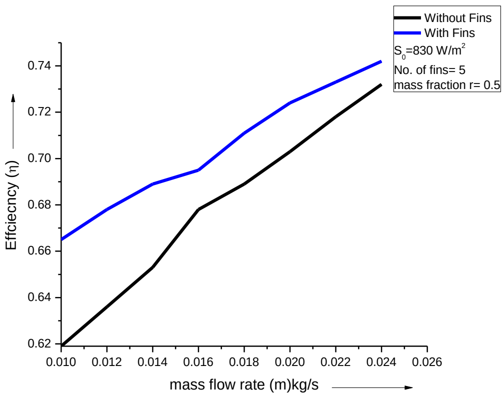

Fig 5 Effect of mass flow rate on efficiency

Fig 5 represents variation of efficiency with respect to mass flow rate, it is clear from the fig that efficiency is higher in case of attached fins than in case of without fins. The efficiency increases with increase in mass flow rate of air. The reason behind the increase in efficiency is that when mass flow rate increases Reynolds number (Re) also increases. With increase in Reynolds number the convection coefficient increases resulting in increased heat transfer. Hence along with this increased heat transfer efficiency of solar air heater increases.

VI. CONCLUSION

On the basis of analytical investigation carried out in this work in connection with the heat and fluid flow in solar air heater with double flow longitudinal fins attached absorber plate. The following conclusions are drawn.

1. An analytical model of temperature distribution of air and efficiency of double flow longitudinal fins attached solar air heater has been developed.
2. The effect of fins attached on absorber plate on efficiency of solar air heater has been investigated. Results show that the efficiency of solar air heater increases in case of fins as compared to without fins.
3. The efficiency of solar air heater found to be maximum for mass fraction r=0.5, when So=1100 W/m^2, So=830 W/m^2 with mass flow rate m=0.0107 kg/s
4. The efficiency of solar air heater found to be maximum for mass fraction r=0.5, when So=1100 W/m^2, So=830 W/m^2 with mass flow rate m=0.0158 kg/s.
5. The efficiency of solar air heater increases with increase in mass flow rate.

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