Mathematical Modelling of Single Glazed and Double Glazed Solar Air Heater

Bhawna Agrawal, Pallavi Agrawal, Suman Agrawal

Abstract*: *This paper focuses on Mathematical Modelling of Single Glazed and Double Glazed Solar air heater (SAH) which is special kind of heat exchanger that transfers thermal energy from the solar radiation to the fluid flowing inside of the collector. The most potential applications of SAH is the supply of hot air for heating of buildings, to maintain a comfortable environment especially in the winter season, air preheating, desiccant refrigeration, and drying of vegetables, fruits, meat, textile and marine products. Solar radiation intensity is less in the morning that increase gradually till noon and again decrease from noon to evening. During simulations it is observed that the heat gain is directly proportional to the mass flow rate. It is maximum for the counter flow SAH and is least for transpired solar air heater. The efficiency of the SAH is directly proportional to mass flow rate. The thermal efficiency is maximum for the counter flow SAH, The useful heat gain increases is highest in the clear days of summer month particularly in the month of April-May and lowest in the cloudy days of winter month particularly in the month of December. The results are in conformation with theoretical aspects.

***Keywords*: Absorbers, Double Glazed SAH, Duct, Glazing, Single Glazed SAH, Solar Air Heater,**

I. INTRODUCTION

Solar air heater (SAH) is special kind of heat exchanger that transfers thermal energy from the solar radiation to the fluid flowing inside of the collector. The most potential applications of SAH is the supply of hot air for heating of buildings to maintain a comfortable environment especially in the winter season, air preheating, desiccant refrigeration, and drying of vegetables, fruits, meat, textile and marine products. With the development of computer, hardware and numerical methodology, advanced mathematical models are being used to carry out critical investigations on SAH. The purpose of this work is to review the present state of mathematical modelling of SAHs. The validation is an important step in mathematical modelling development, and therefore comparisons with actual experimental values or theoretical results have been included where possible. Previous works in this area goes down to the basic theory of solar heating is simple. The technique with the “hot box” and the water tanks were combined in the world’s first commercial solar water heater, which was patented in 1891 by Clarence Kemp from Baltimore, USA.

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This collector was made of black painted metal tanks that were put in boxes with glass lids, capturing the sunlight. In 1909 William J. Bailey developed a system similar to the solar systems used today, where the tank and the solar collector were separated into two units and the insulated storage tank could be placed inside the house, keeping the water hot much longer than previously used systems [1]. In 1959 Colorado Solar House achieved efficiency of 30% by using a glass and metal collector with many glazing staggered on top of each other. [2] evaluated the effect of absorber plate geometry and glazing materials on the performance of flat plate collector. They interpreted that cost of SAH can be reduced by increasing the collector efficiency. They suggested that for the restricted surface transfer area the heat collection surface area can be optimized by varying the geometry of absorber plate. In [3] it has been proposed a design of multipurpose solar heating system. They combined solar water heater and solar air heater, evaluated the performance of solar air heater of this type of multipurpose solar heating system. The air heater was placed over the riser tubes of water heating system. [4] studied the performance of single pass SAH with and without fins. It was found that efficiency depends on inlet air temperature, distance between the absorber plate and cover. They further concluded that SAH with fins have high efficiency compared to that of without fins. Modifications were made in the geometry of absorber plate and instead of a conventional flat absorber plate; absorber having dimple pockets was used to increase the heat transfer capacity of the system. Based on the comparative study [5] it was concluded that there is significant rise in average surface temperature of absorber plate because of use of dimples in the absorber plate.

II. MATHEMATICAL MODELLING OF SINGLE GLAZED SAH

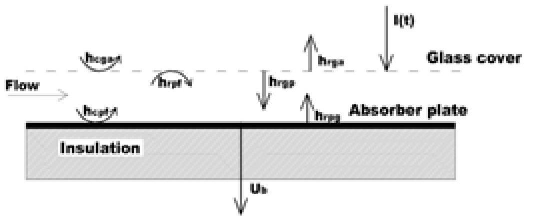

Fig. 4 Cross sectional view and nomenclature

The functioning of glazed SAH is similar to that of transpired SAH. In this system, a glazing usually a glass cover is used to increase the efficiency. Glazed solar air heater has a glass cover having an absorptivity of 0.05 and transmissivity of 0.95. The absorber plate is 2.4 m long, 0.6 m wide and has a duct of depth 7 mm. An effective absorber surface made of black painted aluminum sheet having absorptivity of 0.83.

A. Energy balance equation for glass cover

The energy balance equations for the glass cover can be written as:
Rate of heat absorbed by the Glass cover = Heat capacity of glass+ Rate of heat transfer from Glass cover to the fluid by convection + Rate of heat transfer from the glass cover to the absorber plate by radiation+ Rate of heat loss to ambient air by convection and radiation

$$\alpha_g I(t) = m_g C_{pg}\left(\frac{dT_g}{dt}\right) + h_{cgf}\left(T_g - T_f\right) + h_{rgp}\left(T_g - T_p\right) + \left(h_{cga} + h_{rga}\right)\left(T_g - T_a\right) \quad (11)$$

Neglecting the heat capacity of the glass cover and simplifying equation, the temperature of the glass cover is given by

$$T_g = \frac{\alpha_g I(t) + \left(h_{cga} + h_{rga}\right)T_a + h_{cgf}T_f + h_{rgp}T_p}{h_{cga} + h_{rga} + h_{cgf} + h_{rgp}} \quad (12)$$

B. Energy balance equation for absorber plate

The energy balance equations for the absorber plate with insulation can be written as:
Rate of heat absorbed by absorber plate + Rate of heat transfer from the glass cover to the absorber plate by radiation = Heat capacity of absorber plate + Rate of heat convected to the fluid from absorber plate + Bottom loss

$$\tau_g \alpha_p I(t) + h_{rgp}\left(T_g - T_p\right) = m_p C_p\left(\frac{dT_p}{dt}\right) + h_{cpf}(T_p - T_f) + U_b(T_p - T_a) \quad (13)$$

Neglecting the heat capacity of absorber plate and substituting the value of T_g from equations the absorber plate temperature is given by

$$T_p = \frac{\begin{array}{l}\alpha_g I(t) h_{rgp} + \alpha_p \tau_g I(t)\left(h_{cga} + h_{rga} + h_{rgp} + h_{cgf}\right) \\ + \left\{\left(h_{cga} + h_{rga}\right)h_{rgp} + \left(h_{cga} + h_{cgf} + h_{rgp} + h_{cgf}\right)U_b\right\}T_a \\ + \left\{h_{rgp}h_{cgf} + \left(h_{cga} + h_{rga} + h_{cgf} + h_{rgp}\right)h_{cpf}\right\}T_f\end{array}}{\left(h_{cga} + h_{rga} + h_{cgf} + h_{rgp}\right)\left(h_{cpf} + h_{rgp} + U_b\right) - h_{rgp}^2} \quad (14)$$

C. Energy balance equation for air flowing in the duct

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat transfer from Glass cover to the fluid by convection + Rate of heat convected to the fluid from absorber plate = Heat capacity of the air + heat gain

$$h_{cgf}\left(T_g - T_f\right) + h_{cpf}\left(T_p - T_f\right) = \rho_f t_f C_{pf}\left(\frac{\partial T_f}{\partial t}\right) + \frac{m_f C_{pf}}{W}\frac{\partial T_f}{\partial x} \quad (15)$$

Where $m_f = \rho_f t_f W V_f$

Neglecting the heat capacity of the air flowing through the duct and eliminating,
T_g and T_p ,

$$m_f C_{pf}\frac{\partial T_f}{\partial x} = WF'\left[\tau_g \alpha_p I(t) - U_L\left(T_f - T_a\right)\right] \quad (16)$$

Where F' and U_L are the collector efficiency factor and Overall heat transfer coefficient.
Collector efficiency factor (F') is given by

$$F' = \frac{h_{rpg}h_{cgf} + h_{cpf}\left(h_{cga} + h_{rga}\right) + h_{cpf}h_{rpg} + h_{cgf}h_{cpf}}{\left(U_b + h_{rgp} + h_{cpf}\right)\left(h_{cga} + h_{rga} + h_{cpf} + h_{rgp}\right) - (h_{rgp})^2} \quad (17)$$

Overall heat transfer coefficient of the collector (U_L) is given by

$$U_L = \frac{\left(h_{cga} + h_{rga} + U_b\right)\left(h_{cgf}h_{cpf} + h_{cgf}h_{rgp} + h_{cpf}h_{rgp}\right) + U_b\left(h_{cga} + h_{rga}\right)\left(h_{cpf} + h_{cgf}\right)}{h_{cpf}\left(h_{cga} + h_{rgp} + h_{cgf}\right) + h_{cgf}h_{rpg} + h_{cgf}h_{rga}} \quad (18)$$

Assuming U_L and F' to be constant and applying boundary conditions $T_f = T_{fi}$ at $x = 0$, the solution of equation is given by

$$\frac{T_f - T_a - \left(\tau_g \alpha_P I(t) / U_L\right)}{T_{fi} - T_a - \left(\tau_g \alpha_P I(t) / U_L\right)} = \exp\left(\frac{-U_L F' W x}{m_f C_{pf}}\right) \quad (19)$$

The outlet fluid temperature is obtained by substituting $T_f = T_{fo}$ at $x = L$

$$T_{fo} = \frac{\left(\tau_g \alpha_p I(t) + U_L T_a\right)}{U_L}\left[1 - \exp\frac{\left(-U_L W F'\right)}{m_f c_{pf}} L\right] + T_{fi}\exp\left(\frac{-U_L W F' L}{m_f c_{pf}}\right)$$

The mean air temperature is given by

$$T_{fm} = \frac{\left(\tau_g \alpha_P I(t) + U_L T_a\right)}{U_L}\left[1 - \frac{\left(1 - \exp\frac{\left(-U_L W F'\right)}{m_f c_{pf}} L\right)}{\frac{\left(U_L W F'\right)}{m_f c_{pf}}}\right] + T_{fi}\exp\left[\frac{\left(1 - \exp\frac{\left(-U_L W F'\right)}{m_f c_{pf}} L\right)}{\frac{\left(U_L W F'\right)}{m_f c_{pf}}}\right] \quad (20)$$

III. MATHEMATICAL MODELLING OF DOUBLE GLAZED SINGLE CHANNEL SAH

A double glazed SAH is made of two glass covers and a single channel air flow between the second cover and absorber plate, and insulation. The double glazing enhances greenhouse effect and there by efficiency.

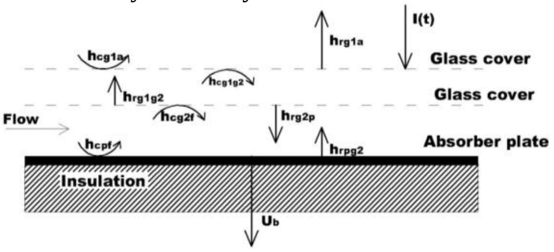

Fig.5 Cross sectional view and nomenclature

In double glazed solar air heater with single channel. Both the glass covers are made of same material and have an absorptivity of 0.05 and transmissivity of 0.95. They are placed one below the other with a gap of 5 mm.

A. Energy balance equation for top glass cover

The energy balance equations for the glass cover can be written as: Rate of heat absorbed by the Glass cover = Heat capacity of glass+ Rate of heat transfer by first glass cover to the second glass by convection + Rate of heat transfer from the first glass cover to second glass cover by radiation+ Rate of heat loss to ambient air by convection and radiation from first glass cover

$$\alpha_{g1} I(t) = m_{g1} C_{pg1} \left(\frac{dT_g}{dt} \right) + h_{cg1g2} \left(T_{g1} - T_{g2} \right) + h_{rg1g2} \left(T_{g1} - T_{g2} \right) + \left(h_{cg1a} + h_{rg1a} \right) \left(T_{g1} - T_a \right) \quad (21)$$

Neglecting the heat capacity of first glass cover and simplifying equation, the temperature of the glass cover is given by

$$T_{g1} = \frac{\alpha_{g1} I(t) + \left(h_{cg1a} + h_{rg1a} \right) T_a + h_{cg1g2} T_{g2} + h_{rg1g2} T_{g2}}{h_{cg1a} + h_{rg1a} + h_{cg1g2} + h_{rg1g2}} \quad (22)$$

B. Energy balance equation for second glass cover

The energy balance equations for the glass cover can be written as:

Rate of heat absorbed by second glass cover = Heat capacity of glass+ Rate of heat transfer from second glass cover to the fluid by convection + Rate of heat transfer from second glass cover to the absorber plate by radiation+ Rate of heat received from first glass cover by convection and radiation

$$\tau_{g1} \alpha_{g2} I(t) = m_{g2} C_{pg2} \left(\frac{dT_g}{dt} \right) + h_{cg2f} \left(T_{g2} - T_f \right) + h_{rg2p} \left(T_{g2} - T_p \right) + h_{cg1g2} \left(T_{g1} - T_{g2} \right) + h_{rg1g2} \left(T_{g1} - T_{g2} \right) \quad (23)$$

Neglecting the heat capacity of second glass cover and simplifying eqn (3.25), the temperature of the glass cover is given by

$$T_{g2} = \frac{\alpha_{g2} \tau_{g1} I(t) + h_{cg2f} T_f + \left(h_{cg1g2} + h_{rg1g2} \right) T_{g1} + h_{rg2p} T_P}{h_{cg2f} + h_{cg1g2} + h_{rg1g2} + h_{rg2p}} \quad (24)$$

C. Energy balance equation for absorber plate

The energy balance equations for the absorber plate with insulation can be written as:

Rate of heat absorbed by absorber plate=Heat capacity of absorber plate + Rate of heat convected to the fluid from absorber plate+ Rate of heat radiated to the fluid from absorber plate + Bottom loss

$$\alpha_p \tau_{g1} \tau_{g2} I(t) = m_p C_{pp} \left(\frac{dT_p}{dt} \right) + h_{cpf} \left(T_p - T_f \right) + h_{rpg2} \left(T_p - T_{g2} \right) + U_b \left(T_p - T_a \right) \quad (25)$$

Neglecting the heat capacity of absorber plate and simplifying equation, the temperature of the glass cover is given by

$$T_p = \frac{\alpha_p \tau_{g1} \tau_{g2} I(t) + h_{cpf} T_f + h_{rg2p} T_{g2} + U_b T_a}{U_b + h_{cpf} + h_{rg2p}} \quad (26)$$

D. Energy balance equation for air flowing in the duct

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat transfer from Glass cover to the fluid by convection + Rate of heat convected to the fluid from absorber plate = Heat capacity of the air + heat gain

$$h_{cg2f} \left(T_{g2} - T_f \right) + h_{cpf} \left(T_p - T_f \right) = \rho_f t_f C_{pf} \left(\frac{\partial T_f}{\partial t} \right) + \frac{m_f C_{pf}}{W} \frac{\partial T_f}{\partial x} \quad (27)$$

Where $m_f = \rho_f t_f W V_f$

Neglecting the heat capacity of the air flowing through the duct and eliminating, T_g *and* T_p,

$$m_f C_{pf} \frac{\partial T_f}{\partial x} = W F' \left[(\tau\alpha)_{eff} I(t) - U_L \left(T_f - T_a \right) \right] \quad (28)$$

$$(\tau\alpha)_{eff} = \frac{\tau_g \alpha_p}{1 - (1 - \alpha_p) \rho_g}, \quad \tau_g = \frac{\tau_{g1} \tau_{g2}}{1 - \rho_{g1} \rho_{g2}}, \quad \rho_g = \rho_{g1} + \frac{\rho_{g2} \tau_{g1}^2}{1 - \rho_{g1} \rho_{g2}},$$

$$\alpha_g = 1 - \rho_g - \tau_g$$

Where F' *and* U_L are the collector efficiency factor and *Overall heat transfer coefficient Collector efficiency factor (F') is given by*

$$F' = \frac{h_{rpg2} h_{cg2f} + h_{cpf} \left(h_{cg1a} + h_{rg1a} \right) \left(h_{cg1g2} + h_{rg1g2} \right) + h_{cpf} h_{rpg2} + h_{cg2f} h_{cpf}}{\left(U_b + h_{rpg2} + h_{cpf} \right) \left(h_{cg1a} + h_{rg1a} + h_{cg1g2} + h_{rg1g2} + h_{cpf} + h_{rpg2} \right) - (h_{rpg2})^2} \quad (29)$$

Overall heat transfer coefficient of the collector (U_L) *is given by*

$$U_L = \frac{\left(h_{cga} + h_{rga} + h_{cg1a} + h_{rg1a} + U_b \right) \left(h_{cg2f} h_{cpf} + h_{cg2f} h_{rpg2} + h_{cpf} h_{rg2p} \right) + U_b \left(h_{cg1a} + h_{rg1a} \right) \left(h_{cpf} + h_{cg2f} \right) \left(h_{rg1g2} + h_{cg1g2} \right)}{h_{cg2f} h_{rpg2} + h_{cpf} \left(h_{cg1a} + h_{rg1a} \right) \left(h_{rg1g2} + h_{cg1g2} \right) + h_{cpf} h_{rpg2} + h_{cg2f} h_{cpf} + h_{cg2f} \left(h_{cg1a} + h_{rg1g2} \right) \left(h_{cg1a} + h_{rg1a} \right) + h_{rg1a} h_{cg1g2} + h_{rg1g2} h_{cg1a}} \quad (30)$$

Assuming U_L *and* F' to be constant and applying boundary conditions $T_f = T_{fi}$ *at* $x = 0$, the solution of equation is given by

$$\frac{T_f - T_a - \left(\frac{(\tau\alpha)_{eff} I(t)}{U_L} \right)}{T_{fi} - T_a - \left(\frac{(\tau\alpha)_{eff} I(t)}{U_L} \right)} = \exp\left(\frac{-U_L F' W x}{\dot{m}_f C_{pf}} \right) \quad (31)$$

The outlet fluid temperature is obtained by substituting $T_f = T_{fo}$ *at* $x = L$

$$T_{fo} = \frac{\left(\tau_g \alpha_p I(t) + U_L T_a \right)}{U_L} \left[1 - \exp\left(\frac{(-U_L W F')}{m_f c_{pf}} L \right) \right] + T_{fi} \exp\left(\frac{-U_L W F' L}{m_f c_{pf}} \right) \quad (32)$$

IV. MATHEMATICAL MODELLING OF SINGLE GLAZED DOUBLE CHANNEL PARALLEL FLOW SAH

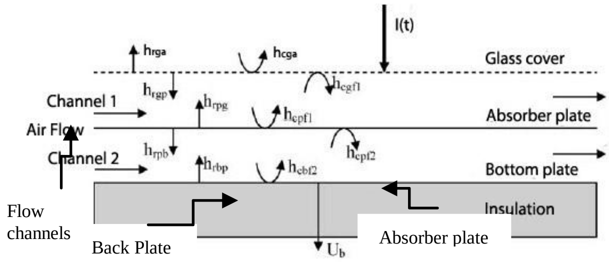

Fig 6 Cross sectional view and nomenclature

A single glazed double channel parallel flow SAH designed a glass cover, double air flows between glass cover and absorber plate and between absorber and bottom plates in parallel direction, and with insulation provided. Many papers have investigated this design [7][8]. A single glazed double channel parallel flow SAH has a glass cover having an absorptivity of 0.05 and transmissivity of 0.95. Fig. 6 shows the cross-sectional view along with all necessary nomenclatures.

A. Energy balance equation for Glass cover

The energy balance equations for the glass cover can be written as:

Rate of heat absorbed by the Glass cover = Heat capacity of glass+ Rate of heat transfer from glass cover to the fluid by convection + Rate of heat transfer from the glass cover to the absorber plate by radiation+ Rate of heat loss to ambient air by convection and radiation

$$\alpha_g I(t) = m_g C_{pg}\left(\frac{dT_g}{dt}\right) + h_{cgf1}\left(T_g - T_{f1}\right) + h_{rgp}\left(T_g - T_p\right) + \left(h_{cga} + h_{rga}\right)\left(T_g - T_a\right) \qquad (33)$$

Neglecting the heat capacity of glass cover and simplifying equation, the temperature of the glass cover is given by

$$T_g = \frac{\left(h_{cga} + h_{rga}\right)T_a + h_{cgf1}T_{f1} + h_{rgp}T_p}{h_{cga} + h_{rga} + h_{cgf1} + h_{rgp}} \qquad (34)$$

B. Energy balance equation for back plate

The energy balance equations for the back plate with insulation can be written as:

Rate of heat radiated from absorber plate = Bottom loss + Rate of heat convected by the plate to fluid in second channel

$$h_{rpb}\left(T_p - T_b\right) = U_b\left(T_b - T_a\right) + h_{cbf2}\left(T_b - T_{f2}\right)$$

The temperature of the back plate is given by

$$T_b = \frac{U_b T_a + h_{cbf2}T_{f2} + h_{rpb}T_p}{U_b + h_{cbf2} + h_{rbp}} \qquad (35)$$

C. Energy balance equation for absorber plate

The energy balance equations for the absorber plate can be written as:

Rate of heat absorbed by absorber plate + Rate of heat transfer from the glass cover to the absorber plate by radiation =Heat capacity of absorber plate +Rate of heat convected to the fluid in second channel by absorber plate+ Rate of heat radiated by absorber plate to the bottom plate + Rate of heat convected to the fluid in first channel by absorber plate

$$\tau_g \alpha_p I(t) + h_{rgp}\left(T_g - T_p\right) = m_p C_{pp}\left(\frac{dT_p}{dt}\right) + h_{cpf2}(T_p - T_{f2}) + h_{rpb}(T_p - T_b) + h_{cpf1}\left(T_p - T_{f1}\right) \qquad (36)$$

Neglecting the heat capacity of absorber plate and substituting the values of T_g *and* T_b in equation above, . The absorber plate temperature is given by

$$T_p = \frac{I(t)\sigma_1\sigma_2 + \left(h_{cpf1}\sigma_1 + h_{cgf1}h_{rgp}\right)\sigma_2 T_{f1} + \left(h_{cpf2}\sigma_2 + h_{cbf2}h_{rpb}\right)\sigma_1 T_{f2} + \left(h_{rgp}\left(h_{cga} + h_{rga}\right)\sigma_2 + h_{rpb}U_b\sigma_1\right)T_a}{\left(h_{cpf1} + h_{cpf2}\right)\sigma_1\sigma_2 + h_{rpg}\left(h_{cga} + h_{rga} + h_{cgf1}\right)\sigma_2 + h_{rpb}(U_b + h_{cbf2})\sigma_1} \qquad (37)$$

where $\sigma_1 = h_{cga} + h_{rga} + h_{cgf1} + h_{rgp}$, $\sigma_2 = U_b + h_{cbf2} + h_{rpb}$

D. Energy balance equation for air flowing in first channel

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat convected by glass cover + Rate of heat convected by the absorber plate = Heat capacity of the air +Heat gain

$$h_{cgf1}\left(T_g - T_{f1}\right) + h_{cpf1}\left(T_p - T_{f1}\right) = \rho_f t_f C_{pf}\left(\frac{\partial T_f}{\partial t}\right) + \frac{m_{f1}C_{pf}}{W}\frac{\partial T_{f1}}{\partial x} \qquad (38)$$

E. Energy balance equation for air flowing in second channel

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat convected by absorber plate + Rate of heat convected by the bottom plate = Heat capacity of the air + Heat gain

$$h_{cpf2}\left(T_p - T_{f2}\right) + h_{cbf2}\left(T_b - T_{f2}\right) = \rho_f t_f C_{pf}\left(\frac{\partial T_f}{\partial t}\right) + \frac{m_{f2}C_{pf}}{W}\frac{\partial T_{f2}}{\partial x} \qquad (39)$$

Neglecting the heat capacity of the air and substituting the values of T_b, T_p, T_g in equations and adding them the useful heat gain is given by

$$m_{f1}C_{pf}\frac{\partial T_{f1}}{\partial x} + m_{f2}C_{pf}\frac{\partial T_{f2}}{\partial x} = WF'\left[(\tau\alpha)_{eff} I(t) - U_{01}\left(T_{f1} - T_a\right) - U_{02}\left(T_{f2} - T_a\right)\right]$$

Where F' *and* U_L are the collector efficiency factor and overall loss coefficient.

Collector efficiency factor (F') is given by

$$F' = \frac{(h_{cpf1} + h_{cpf2})\sigma_1\sigma_2 + h_{cgf1}h_{rgp}\sigma_2 + h_{cbf2}h_{rpb}\sigma_1}{\sigma_3} \qquad (40)$$

$$U_{01} = \frac{h_{cgf1}U_t\sigma_3 + \left(h_{cpf1}\sigma_1 + h_{cgf1}h_{rgp}\right)\left(h_{rgp}U_t\sigma_2 + h_{rpb}U_b\sigma_1\right)}{\left[\left(h_{cpf1} + h_{cpf2}\right)\sigma_1\sigma_2 + h_{cgf1}h_{rgp}\sigma_2 + h_{cbf2}h_{rpb}\sigma_1\right]\sigma_1} \qquad (41)$$

(3.37)

$$U_{02} = \frac{h_{cbf2}U_b\sigma_3 + \left(h_{cpf2}\sigma_2 + h_{cbf2}h_{rpb}\right)\left(h_{rgp}U_t\sigma_2 + h_{rpb}U_b\sigma_1\right)}{\left[\left(h_{cpf1} + h_{cpf2}\right)\sigma_1\sigma_2 + h_{cgf1}h_{rgp}\sigma_2 + h_{cbf2}h_{rpb}\sigma_1\right]\sigma_1} \qquad (42)$$

$$\sigma_3 = \left(h_{cpf1} + h_{cpf2}\right)\sigma_1\sigma_2 + h_{rgp}\left(U_t + h_{cgf1}\right)\sigma_2 + h_{rpb}\left(U_b + h_{cbf2}\right)\sigma_1 \qquad (43)$$

Overall heat transfer coefficient U_L is given by

$$U_L = U_{01} + U_{02}$$

Equation (3.43) can be written as

$$m_f C_{pf}\frac{\partial T_f}{\partial x} = WF'\left[(\tau\alpha)_{eff} I(t) - U_L\left(T_f - T_a\right)\right] \qquad (44)$$

Where $T_f = \frac{U_{01}T_{f1} + U_{02}T_{f2}}{U_L}$

To estimate each convection heat transfer coefficients inside both channels of the collector, it is necessary to calculate the mean values of T_{f1} and T_{f2} between the positions of the air inlet and outlet according to these authors recommendation. The values of T_{f1} and T_{f2} is obtained by differentiating the equations below

By applying boundary conditions; $T_{f1}(0)=T_{f2}(0)=T_i$ the solution of these equations are given below

$$T_{f1}(x) = K_1\exp\left(\alpha_1 x\right) + K_2\exp\left(\alpha_2 x\right) + C \qquad (45$$

$$T_{f2}(x) = K_1\exp\left(\alpha_1 x\right) - \frac{U_{01}}{U_{02}}K_2\exp\left(\alpha_2 x\right) + D \qquad (46)$$

where $\alpha_1 = -\frac{A_c F' U_L}{m_f C_{pf} L}$, $\alpha_2 = -\frac{A_c F' U_L}{m_f C_{pf} L}\left(1 + \frac{U_L U_{12}^1}{U_{02}U_{01}'}\right)$,

$$K_1 = T_i - T_a - \frac{I(t)}{U_L}\left[\frac{U_{01}'U_{02}' + U_L(U_{12}^1 + U_{12}^2)}{U_{01}'U_{02}' + U_{12}^1 U_{02}' + U_{12}^2 U_{01}'}\right],$$

$$K_2 = \frac{I(t)}{U_L}\left[\frac{U_{02}\left(U_{01}' - U_{02}'\right)}{U_{01}'U_{02}' + U_{12}^1 U_{02}' + U_{12}^2 U_{01}'}\right]$$

$$C = T_a + \left(\frac{U_{02}' + U_{12}^1 + U_{12}^2}{U_{01}'U_{02}' + U_{12}^1 U_{02}' + U_{12}^2 U_{01}'}\right)I(t),$$

$$D = T_a + \left(\frac{U_{01}^{'} + U_{12}^{1} + U_{12}^{2}}{U_{01}^{'}U_{02}^{'} + U_{12}^{1}U_{02}^{'} + U_{12}^{2}U_{01}^{'}} \right) I(t) \quad (47)$$

$$F_1^{'} = \frac{h_{cpf1}\sigma_1\sigma_2 + h_{cgf1}h_{rgp}\sigma_2}{\sigma_3}, \quad F_2^{'} = \frac{h_{cpf2}\sigma_1\sigma_2 + h_{cbf2}h_{rpb}\sigma_1}{\sigma_3}$$

$$U_{01}^{'} = \frac{F^{'}U_{01}}{F_1^{'}}, \quad U_{02}^{'} = \frac{F^{'}U_{02}}{F_2^{'}}, \quad U_{12}^{1} = \frac{h_{cpf2}\sigma_2 + h_{cbf2}h_{rpb}}{\sigma_2},$$

$$U_{12}^{2} = \frac{h_{cpf1}\sigma_1 + h_{cgf1}h_{rgp}}{\sigma_1}$$

Outlet air temperature is obtained by differentiating eqn (4.49) and the solution of equation is given by

$$\frac{T_f - T_a - \left((\tau\alpha)_{eff}\, I(t) \big/ U_L \right)}{T_{fi} - T_a - \left((\tau\alpha)_{eff}\, I(t) \big/ U_L \right)} = \exp\left(\frac{-U_L F' W x}{\dot{m}_f C_{pf}} \right) \quad (48)$$

The outlet fluid temperature is obtained by substituting $T_f = T_{fo}\ at\ x = L$

$$T_{fo} = \frac{\left(\tau_g \alpha_p I(t) + U_L T_a\right)}{U_L} \left[1 - \exp\frac{\left(-U_L W F^{'}\right)}{m_f c_{pf}} L \right] + T_{fi} \exp\left(\frac{-U_L W F^{'} L}{m_f c_{pf}} \right) \quad (49)$$

V. MATHEMATICAL MODELLING OF SINGLE GLAZED DOUBLE CHANNEL COUNTER FLOW SAH

A single glazed double channel counter flow SAH [9] [10] designed a glass cover, double air flows between glass cover and absorber plate and between absorber and bottom plates in opposite direction, and with insulation provided.

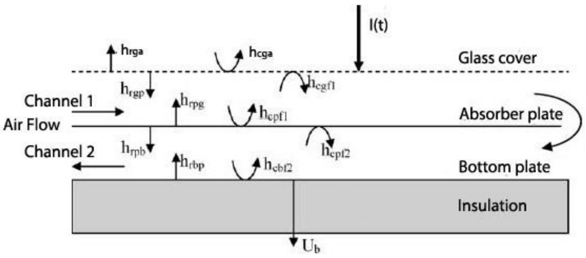

Fig. 7 Cross sectional view and nomenclature

In a single glazed double channel counter flow SAH, the absorber plate is 2.4 m long, 0.6 m wide and is placed 5 mm below the glass cover and 7 mm above back plate. Fig. 7 shows the cross-sectional view along with all necessary nomenclatures.

A. Energy balance equation for Glass cover

The energy balance equations for the glass cover can be written as:

Rate of heat absorbed by the Glass cover = Heat capacity of glass+ Rate of heat transfer from glass cover to the fluid by convection + Rate of heat transfer from the glass cover to the absorber plate by radiation+ Rate of heat loss to ambient air by convection and radiation

$$\alpha_g I(t) = m_g C_{pg}\left(\frac{dT_g}{dt}\right) + h_{cgf1}\left(T_g - T_{f1}\right) + h_{rgp}\left(T_g - T_p\right) + \left(h_{cga} + h_{rga}\right)\left(T_g - T_a\right) \quad (50)$$

Neglecting the heat capacity of glass cover and simplifying eqn (3.54), the temperature of the glass cover is given by

$$T_g = \frac{\left(h_{cga} + h_{rga}\right)T_a + h_{cgf1}T_{f1} + h_{rgp}T_p}{h_{cga} + h_{rga} + h_{cgf1} + h_{rgp}} \quad (51)$$

B. Energy balance equation for back plate

The energy balance equations for the back plate with insulation can be written as:

Rate of heat radiated from absorber plate = Bottom loss + Rate of heat convected by the plate to fluid in second channel

$$h_{rpb}\left(T_p - T_b\right) = U_b\left(T_b - T_a\right) + h_{cbf2}\left(T_b - T_{f2}\right)$$

The temperature of the back plate is given by

$$T_b = \frac{U_b T_a + h_{cbf2}T_{f2} + h_{rpb}T_p}{U_b + h_{cbf2} + h_{rbp}} \quad (52)$$

C. Energy balance equation for absorber plate

The energy balance equations for the absorber plate can be written as:

Rate of heat absorbed by absorber plate + Rate of heat transfer from the glass cover to the absorber plate by radiation =Heat capacity of absorber plate +Rate of heat convected to the fluid in second channel by absorber plate+ Rate of heat radiated by absorber plate to the bottom plate + Rate of heat convected to the fluid in first channel by absorber plate

$$\tau_g \alpha_p I(t) + h_{rgp}\left(T_g - T_p\right) = m_p C_{pp}\left(\frac{dT_p}{dt}\right) + h_{cpf2}(T_p - T_{f2}) + h_{rpb}(T_p - T_b) + h_{cpf1}\left(T_p - T_{f1}\right) \quad (53)$$

Neglecting the heat capacity of absorber plate and substituting the values of Tg and T_b in equation above, the absorber plate temperature is given by

$$T_p = \frac{I(t)\sigma_1\sigma_2 + \left(h_{cpf1}\sigma_1 + h_{cgf1}h_{rgp}\right)\sigma_2 T_{f1} + \left(h_{cpf2}\sigma_2 + h_{cbf2}h_{rpb}\right)\sigma_1 T_{f2} + \left(h_{rgp}\left(h_{cga} + h_{rga}\right)\sigma_2 + h_{rpb}U_b\sigma_1\right)T_a}{\left(h_{cpf1} + h_{cpf2}\right)\sigma_1\sigma_2 + h_{rpg}\left(h_{cga} + h_{rga} + h_{cgf1}\right)\sigma_2 + h_{rpb}(U_b + h_{cbf2})\sigma_1} \quad (54)$$

Where $\sigma_1 = h_{cga} + h_{rga} + h_{cgf1} + h_{rgp}$, $\sigma_2 = U_b + h_{cbf2} + h_{rpb}$

D. Energy balance equation for air flowing in first channel

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat convected by glass cover + Rate of heat convected by the absorber plate = Heat capacity of the air +Heat gain

$$h_{cgf1}\left(T_g - T_{f1}\right) + h_{cpf1}\left(T_p - T_{f1}\right) = \rho_f t_f C_{pf}\left(\frac{\partial T_f}{\partial t}\right) + \frac{m_{f1}C_{pf}}{W}\frac{\partial T_{f1}}{\partial x} \quad (55)$$

E. Energy balance equation for air flowing in second channel

The energy balance equation for the air flowing through the duct can be written as:

Rate of heat convected by absorber plate + Rate of heat convected by the bottom plate = Heat capacity of the air +Heat gain

$$h_{cpf2}\left(T_p - T_{f2}\right) + h_{cbf2}\left(T_b - T_{f2}\right) = \rho_f t_f C_{pf}\left(\frac{\partial T_f}{\partial t}\right) + \frac{m_{f2}C_{pf}}{W}\frac{\partial T_{f2}}{\partial x} \quad (56)$$

Neglecting the heat capacity of the air and substituting the values of T_b, T_p, T_g in equations and adding them the useful heat gain is given by

$$m_{f1}C_{pf}\frac{\partial T_{f1}}{\partial x}+m_{f2}C_{pf}\frac{\partial T_{f2}}{\partial x}=WF'\left[(\tau\alpha)_{eff}I(t)-U_{01}\left(T_{f1}-T_a\right)-U_{02}\left(T_{f2}-T_a\right)\right] \quad (57)$$

where F' and U_L are the collector efficiency factor and overall loss coefficient.

Collector efficiency factor (F') is given by

$$F'=\frac{(h_{cpf1}+h_{cpf2})\sigma_1\sigma_2+h_{cgf1}h_{rgp}\sigma_2+h_{cbf2}h_{rpb}\sigma_1}{\sigma_3} \quad (58)$$

$$U_{01}=\frac{h_{cgf1}U_t\sigma_3+\left(h_{cpf1}\sigma_1+h_{cgf1}h_{rgp}\right)\left(h_{rgp}U_t\sigma_2+h_{rpb}U_b\sigma_1\right)}{\left[\left(h_{cpf1}+h_{cpf2}\right)\sigma_1\sigma_2+h_{cgf1}h_{rgp}\sigma_2+h_{cbf2}h_{rpb}\sigma_1\right]\sigma_1} \quad (59)$$

$$U_{02}=\frac{h_{cbf2}U_b\sigma_3+\left(h_{cpf2}\sigma_2+h_{cbf2}h_{rpb}\right)\left(h_{rgp}U_t\sigma_2+h_{rpb}U_b\sigma_1\right)}{\left[\left(h_{cpf1}+h_{cpf2}\right)\sigma_1\sigma_2+h_{cgf1}h_{rgp}\sigma_2+h_{cbf2}h_{rpb}\sigma_1\right]\sigma_1} \quad (60)$$

Overall heat transfer coefficient U_L is given by

$$U_L=U_{01}+U_{02}$$

Eq (4.62) can be written as

$$m_fC_{pf}\frac{\partial T_f}{\partial x}=WF'\left[(\tau\alpha)_{eff}I(t)-U_L\left(T_f-T_a\right)\right] \quad (61)$$

$$T_f=\frac{U_{01}T_{f1}+U_{02}T_{f2}}{U_L} \quad (62)$$

To estimate each convection heat transfer coefficients [11] [12] inside both channels of the collector, it is necessary to calculate the mean values of T_{f1} and T_{f2} between the positions of the air inlet and outlet according to these authors recommendation. The values of T_{f1} and T_{f2} is obtained by differentiating the equationsFig

By applying the boundary conditions

$T_{f1}(0)=T_{fi}$ and $T_{f1}(L)=T_{f2}(L)$

The solution of these equations are given below

$$T_{f1}(x)=K_1K_3\exp(\alpha_1x)+K_2K_4\exp(\alpha_2x)+C \quad (63)$$

$$T_{f2}(x)=K_1\exp(\alpha_1x)+K_2\exp(\alpha_2x)+D \quad (64)$$

where

$$\alpha_1=\frac{1A_cF'U_L}{2m_fC_{pf}L}\left[\frac{(U_{02}-U_{01})}{U_L}+\sqrt{1+\frac{4U_{01}U_{12}^1}{U_LU_{01}'}}\right],$$

$$\alpha_2=\frac{1A_cF'U_L}{2m_fC_{pf}L}\left[\frac{(U_{02}-U_{01})}{U_L}-\sqrt{1+\frac{4U_{01}U_{12}^1}{U_LU_{01}'}}\right] \quad (65)$$

$$C=T_a+\left(\frac{U_{02}'+U_{12}^1+U_{12}^2}{U_{01}'U_{02}'+U_{12}^1U_{02}'+U_{12}^2U_{01}'}\right)I(t),$$

$$D=T_a+\left(\frac{U_{01}'+U_{12}^1+U_{12}^2}{U_{01}'U_{02}'+U_{12}^1U_{02}'+U_{12}^2U_{01}'}\right)I(t) \quad (66)$$

$$Y=\left(U_{02}'-\frac{m_fC_{pf}\alpha_2}{WF_2'}\right)\left(U_{02}'+U_{12}^2-\frac{m_fC_{pf}\alpha_1}{WF_2'}\right)\exp(\alpha_2L)-\left(U_{02}'-\frac{m_fC_{pf}\alpha_1}{WF_2'}\right)\left(U_{02}'+U_{12}^2-\frac{m_fC_{pf}\alpha_2}{WF_2'}\right)\exp(\alpha_1L) \quad (67)$$

$$F_1'=\frac{h_{cpf1}\sigma_1\sigma_2+h_{cgf1}h_{rgp}\sigma_2}{\sigma_3}, \quad F_2'=\frac{h_{cpf2}\sigma_1\sigma_2+h_{cbf2}h_{rpb}\sigma_1}{\sigma_3} \quad (68)$$

$$U_{01}'=\frac{F'U_{01}}{F_1'}, \quad U_{02}'=\frac{F'U_{02}}{F_2'}, \quad U_{12}^1=\frac{h_{cpf2}\sigma_2+h_{cbf2}h_{rpb}}{\sigma_2},$$

$$U_{12}^2=\frac{h_{cpf1}\sigma_1+h_{cgf1}h_{rgp}}{\sigma_1}$$

$$K_1=\frac{-U_{12}^2\left[1-\frac{U_{02}'\left(U_{02}'+U_{12}^1+U_{12}^2\right)+\left(U_{01}'-U_{02}'\right)\frac{m_fC_{pf}\alpha_2}{WF_2'}}{U_{01}'U_{02}'+U_{12}^1U_{02}'+U_{12}^2U_{01}'}\right]I(t)+U_{12}^2(T_i-C)\left(U_{02}'-\frac{m_fC_{pf}\alpha_2}{WF_2'}\right)\exp(\alpha_2L)}{Y} \quad (69)$$

$$K_2=\frac{U_{12}^2\left[1-\frac{U_{02}'\left(U_{02}'+U_{12}^1+U_{12}^2\right)+\left(U_{01}'-U_{02}'\right)\frac{m_fC_{pf}\alpha_1}{WF_2'}}{U_{01}'U_{02}'+U_{12}^1U_{02}'+U_{12}^2U_{01}'}\right]I(t)-U_{12}^2(T_i-C)\left(U_{02}'-\frac{m_fC_{pf}\alpha_1}{WF_2'}\right)\exp(\alpha_1L)}{Y} \quad (70)$$

$$K_3=1+\frac{1F'U_L}{2F_2'U_{12}^2}\left[1-\sqrt{1+\frac{4U_{01}U_{12}^1}{U_LU_{01}'}}\right] \quad (71)$$

$$K_4=1+\frac{1F'U_L}{2F_2'U_{12}^2}\left[1+\sqrt{1+\frac{4U_{01}U_{12}^1}{U_LU_{01}'}}\right] \quad (72)$$

The outlet air temperature is obtained by evaluating eqn (4.68) at $x = 0$ (3.67)

$$T_{fo}=T_{f2}(0)=K_1+K_2+D \quad (73)$$

VI. SIMULATION RESULTS

Solar radiation data and variation in ambient air temperature for the Srinagar, India for eleven years were obtained from Indian Metrological Department (IMD) Pune. These data are classified into four climatic conditions depending upon the ratio of daily diffuse to daily global radiations and number of sunshine hour, namely: Type A; Type B; Type C and Type D,

Type a: The clear days (blue sky), the ratio of daily diffuse to daily global irradiation is less than or equal to 0.25 and number of sunshine hour is greater than or equal to 9 hours.

Type b: The Hazy days (fully), the ratio of daily diffuse to daily global irradiation between 0.25-0.50 and number of sunshine hour is between 7 to 9 hours.

Type c: The Hazy and cloudy (partially) days, the ratio of daily diffuse to daily global irradiation between 0.50-0.75 and number of sunshine hour is between 5 to 7 hours.

Type d: The cloudy days (fully, the ratio of daily diffuse to daily global irradiation is more than or equal to 0.75 and number of sunshine hour is less than or equal to 5 hours**.**

The classified solar radiation and climatic condition data, were used to compare the five different configurations of SAH.

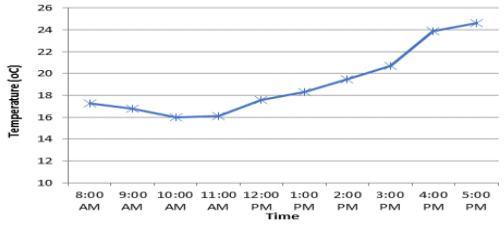

Fig.8 Hourly variations in temp of air for weather condition of type A

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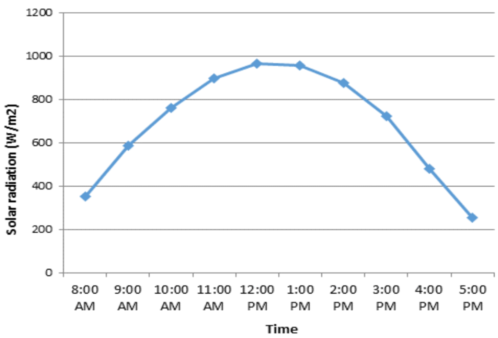

Fig. 9 Hourly variation of solar radiation for weather condition of type A

Fig 8 shows the hourly variation of ambient air temperature of the Srinagar for a typical day of weather condition of type A for the month of May. There is gradual increase in the temperature of ambient air from morning till evening.

Fig 9 shows the hourly variation of solar radiation for a typical day of type A for the month of May. Solar radiation intensity is less in the morning that increase gradually till noon and again decrease from noon to evening.

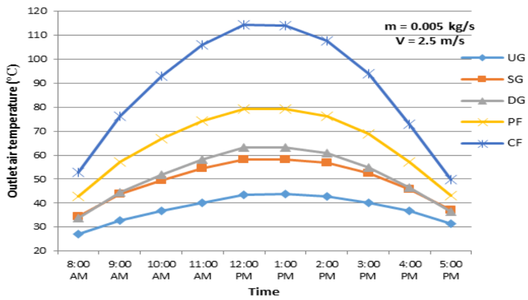

Fig. 10 Hourly variation of outlet temperature

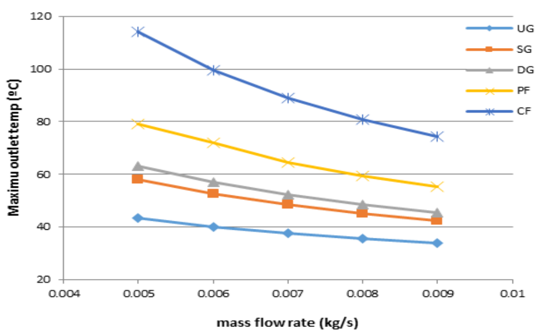

Fig. 11 Effect of mass flow rate on outlet air temperature

Fig 10 shows the hourly variation of air temperature at the outlet for different geometries of solar air heater, namely UG = transpired SAH; SG = single glazed SAH; DG = double glazed single channel SAH; PF = single glazed double channel parallel flow SAH; CF = single glazed double channel counter flow SAH, in weather conditions of Type A and month May. Outlet temperature increases gradually from morning till noon and the maximum value of outlet temperature is obtained during the time period 12:00 PM – 1:00 PM. This is due to the maximum solar radiations are absorbed during this time period increasing the temperature of absorber plate and thus the temperature of air flowing inside the duct. It is also observed that average flow velocity through the duct as 2.5 m/s, the outlet temperature for counter flow SAH is maximum at the noon and attains a value of 115°C, parallel flow SAH attains a maximum temperature of 79°C followed by double glazed SAH, single glazed SAH and transpired SAH.

Fig 11 shows the effect of air mass flow rate through duct on the outlet air temperature. It is found that for the counter flow SAH the outlet air temperature is nearly 114°C at the mass flow rate of 0.005 kg/s and falls to 74°C at the mass flow rate of 0.009 kg/s. Similarly, for the other geometric of the SAH, the outlet air temperature is highest in case of low mass flow rate of air and drops gradually with increase in the mass flow rate of air through the duct. This is owing to air entering the duct has relatively longer time to remain in contact with the surface wall.

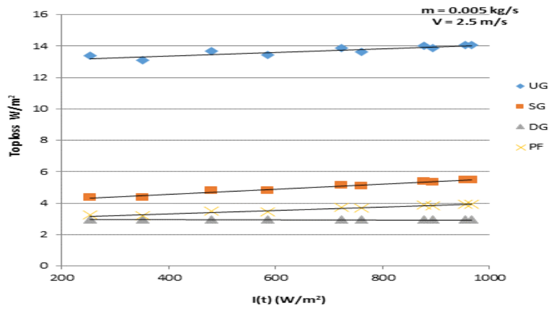

Figure 12 Top loss for different type of SAH

Fig 12 shows the top loss for different geometries of SAH. Top loss is highest for transpired SAH compared to other geometries owing to absence of greenhouse effect and therefore the performance is very poor. It increases linearly with the increase in the intensity of the solar radiation. Glazing reduces the top loss (single flow and double flow) significantly. For the double glazed SAH the top loss is minimum and do not have significant change with the intensity of the solar radiation.

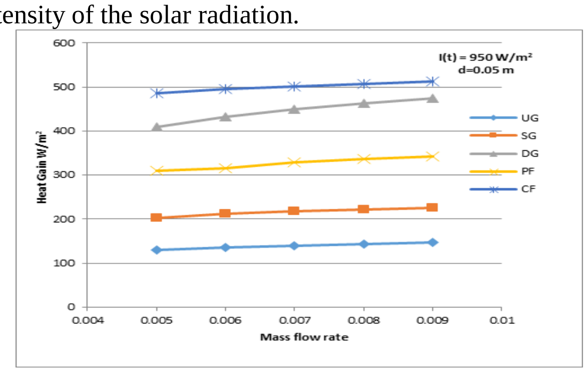

Figure 13 Effect of mass flow rate on useful heat gain

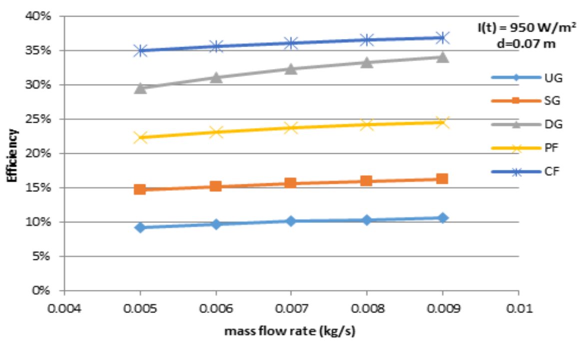

Fig 14 Efficiency variation with mass flow rate

Fig 13 shows the effect of air mass flow rate through duct on the useful heat gain. It is being observed that with an increase in mass flow rate from 0.005 kg/s – 0.009 kg/s the useful heat gain for all geometries of SAH also increases. The heat gain is directly proportional to the mass flow rate. It is maximum for the counter flow SAH and is least for transpired solar air heater.

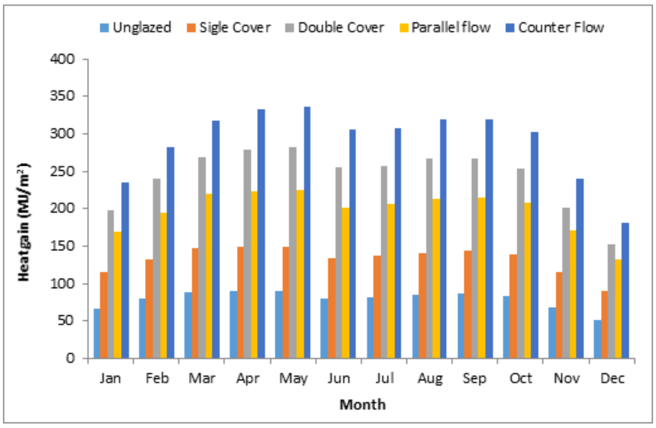

Fig 15 Annual Heat gain

Fig 14 shows the effect of air mass flow rate through duct over the efficiency of SAH. The efficiency of the SAH is directly proportional to mass flow rate. The thermal efficiency is maximum for the counter flow SAH, which ranges from 35 to 37%. It is minimum around 10% for the transpired SAH. For the other geometries it varies between the counter flow SAH and transpired SAH, 14-18% for single glazed SAH, 29-32% for double glazed and 22-24% for paraller flow SAH.

Fig 15 shows the effect of air mass flow rate through duct over the heat gain under different climatic conditions of the year. The useful heat gain increases is highest in the clear days of summer month particularly in the month of April-May and lowest in the cloudy days of winter month particularly in the month of December. It is further observed that the highest heat gain is for the counter flow arrangement, followed by double glazed arrangement.

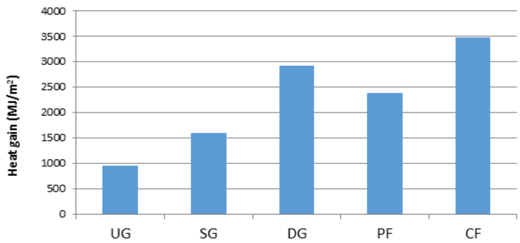

Figure 16 Useful Heat gain for different geometry of solar air heater

Fig 16 shows the annual useful heat gain for different geometries of solar air heaters. The annual heat gain for transpired SAH, single glazed SAH, double glazed SAH, parallel flow SAH and counter flow SAH are 950 MJ/m^2, 1590 MJ/m^2, 2920 MJ/m^2, 2380 MJ/m^2 and 3480 MJ/m^2 respectively. Thus the best configuration is the single glazed solar heater with counter flow arrangement.

VII. CONCLUSIONS

The efficiency and net useful heat gain for the flat plate SAH were investigated using empirical correlations. Annual heat gain for transpired SAH, single glazed SAH, double glazed SAH, parallel flow SAH and counter flow SAH are 950 MJ/m^2, 1590 MJ/m^2, 2920 MJ/m^2, 2380 MJ/m^2 and 3480 MJ/m^2 respectively. Thermal efficiency for transpired SAH, single glazed SAH, double glazed SAH, parallel flow SAH and counter flow SAH are 8-10%, 14-18%, 29-32%, 22-24% and 34-38% respectively. Methods adopted to reduce the top loss coefficient will result in higher useful heat gain and efficiency of the system. Therefore transpired SAH has lowest heat gain and efficiency whereas double glazed has relatively higher heat gain. Mass flow rate also plays significant role in performance of solar air heater. Increasing the mass flow rate increases the useful heat gain and efficiency of the solar air heater. However, it reduces the air outlet temperature. Counter flow SAH has relatively better performance as compared to other arrangement of SAH. Solar air heaters are very useful for rural areas by improving living standard of farmers by earning through crop drying and medicinal points.

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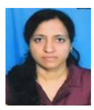

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