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Mathematical Modelling of Solar Air Heater

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Abstract

Mathematical modeling has become an accepted tool for predicting the performance, and optimization of thermal processes. Solving the mathematical models representing solar air heating process and systems is one of the most tedious and repetitive problems. Mathematical modeling of conventional solar air heater with single glass cover is presented. Calculations for a collector of aperture area 2m^2 have been done for Chennai. It has been found that at solar insolation of 734 W/m^2, average temperature of outlet air is 328.352K (55.32˚C) i.e. rise in temperature of air through the collector is 8˚C for air flow rate of 440 kg/h, Instantaneous efficiency of collector is found as 51.8 % and the pressure drop is 36.982 Pa. Results are closely matching with such experimental results. Finite Difference Method can also be used to solve heat transfer equations for conventional solar air heaters. These equations have also been presented in this work.

KEYWORDS: *FLAT PLATE SOLAR AIR HEATER (FPSAH), SOLAR INSOLATION, FINITE DIFFERENCE METHOD (FDM)*

I. INTRODUCTION

In present's world the prosperity of nation is measured by the energy consumption of that nation, the GDP of country is directly linked with energy consumption. Therefore demand for energy resources is increasing day by day. There are various types of energy resources, but mainly they are divided in to two forms, these are renewable energy resources (solar, air, wind) and non-renewable energy resources (coal, petroleum). The industrial growth is accelerated by non-renewable energy resources, but there stock is limited in nature.

The necessity to move into a new energetic model of non conventional energy utilization is being progressively assumed in our societies due to climatic changes and expected increase in oil prices as we run out of fossil fuels. A new green tech revolution has been promoted during last decade and a great academic and industrial effort has contributed to turn to non-polluting renewable energy sources. Solar energy is the major source of such kind. The greatest advantage of solar energy as compared with other forms of energy is that it is clean and can be supplied without any environmental pollution. Over the past century fossil fuels have provided most of our energy because these are much cheaper and more convenient than energy from alternative energy sources, and until recently environmental pollution has been of little concern.

Most research into the use of solar energy in recent years has been on photovoltaic technology, where sunlight is converted directly into electricity. Other than this there are many applications of solar thermal energy such as heating, drying and water distillation. Solar radiation arrives on the surface of the earth with a maximum power of approximately 1 kWh/m^2. The actual usable radiation component varies depending on geographical location, cloud cover, hours of sunlight each day, etc. Solar radiation received is either as *direct* radiation or scattered or *diffuse* radiation, the ratio depends on atmospheric conditions. Both direct and diffuse radiation is useful, but diffuse radiation cannot be concentrated. The solar collector is one of the devices which can absorb and transfer energy of the sun to a usable and/or storable form in many applications such as industrial and space heating, drying agro, textile and marine products.

Several designs of the solar thermal collectors have been built and tested depending on their applications. Figure 1 and 2 shows the exploded and pictorial view of a typical Flat Plate Solar Collector which is further used for mathematical modeling and analysis.

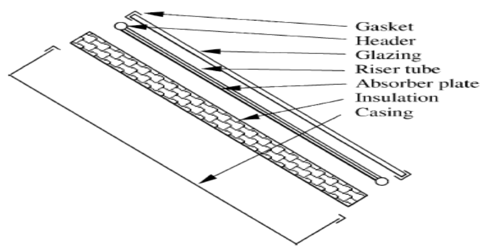

Fig.1: Exploded View of Typical Flat Plate Collector

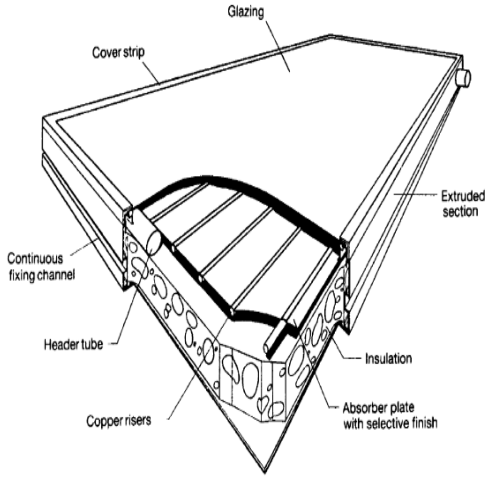

Fig.2: Pictorial View of Typical Flat Plate Collector

Numerical simulation has been revealed as an important tool in the past years in order to provide an insight into physical industrial processes, improving systems performance and process optimization.

Aim and objective of this work is to review and understand the design procedure of flat plate solar air heaters, FPSAH and develop a mathematical model for thermal analysis of FPSAH using CAD software to predict the Thermal efficiency, Useful heat gain, Collector Efficiency, Collector heat removal factor and Pressure drop parameter for Flat Plate solar air heater. And also to investigate the effect of temperature rise parameter on effective efficiency other important parameters.

II. Literature Review

In any solar-thermal collection device, the principle usually followed is to expose a dark surface to solar to solar radiation so that most of the solar radiation is absorbed and converted heat energy. A part of this heat energy is then transferred to a fluid like water or air. When no optical concentration of radiation is done, the device in which the collection is achieved is called the flat plate collector. In other words, when the area of interception of solar radiation is same as that of absorption then the collection device is called flat-plate collector else it is concentrating collector. The flat plate collector is the most important type of solar collector because it is simple in design, has no moving parts and requires little maintenance. It can be used for variety of applications in which temperature of required heat energy ranges from 40-100^{o}C.

When the temperature above100^{o}C are required, it usually becomes necessary to concentrate the radiation. This is achieved by concentrating collector. The devices which are used for this purpose are known as solar collectors. Fig 3 shows classification of solar collectors.

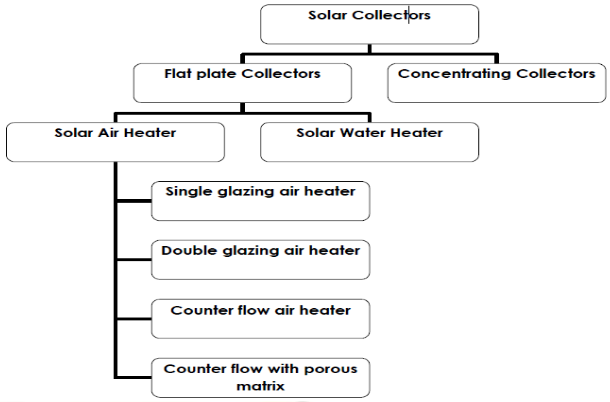

Fig. 3: Classification of Solar Collectors.

Flat-plate collectors are very common and are available as liquid based and air-based collectors. These collectors are better suited for moderate temperature applications where the demand temperature is 30- 70^{o}C and for applications that require heat during the winter months. The air-based collectors are used for the heating of buildings, ventilation air and crop-drying. In this type of collector a flat absorber plate efficiently transforms sunlight into heat. To minimize heat escaping, the plate is located between a glazing (glass pane or transparent material) and an insulating panel. The glazing is chosen so that a maximum amount of sunlight will pass though it and reach absorber.

A flat plate collector basically consists of a flat surface with high absorptivity for solar radiations, called the absorbing surface; typically a metal plate, painted black. The energy is transferred from the absorber plate to a carrier fluid circulating across the collector. Thermal insulation is usually placed on the rear side to prevent heat losses. The front side has transparent covers, generally glass that allows transmission of incoming solar radiations but is opaque to the infrared radiations form the absorber plate. Flat plate collectors are usually permanently fixed in position and require no tracking of the sun. The collector should be oriented directly towards the equator, facing south in the northern hemisphere and facing north in the southern hemisphere. Fig 4 Schematic of smooth flat plate collector.

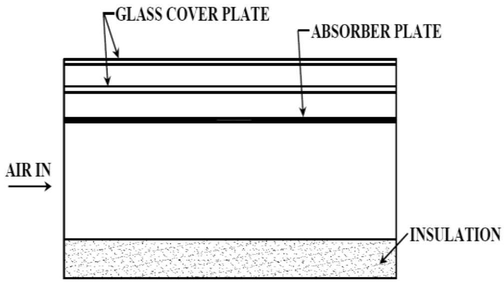

Fig.4 :Schematic of Smooth Flat Plate Collector (Choudhury et al, 1993)

For year round applications, the optimum tilt angle of collector is equal to the latitude, whereas for winter, tilt angle should be approximately 10° to 15° more than the latitude and for summer, tilt angle should be approximately 10° to 15° less than latitude. Temperature up to a maximum of about 100 °C above ambient can be achieved through flat plate collectors. Flat plate solar collectors may be divided into two main classifications based on the type of heat transfer fluid used i.e. liquid heating and air heating collectors.

There is an increasing demand for the solar collectors, especially the flat-plate solar collector. Owing to the many parameters affecting the solar collector performance, attempting to make a detailed analysis of a solar collector is a very complicated problem. Fortunately, a relatively simple analysis will yield very useful results, *Duffie Beckmann, 1991.*

One of the most potential applications of solar energy is the supply of hot air for the drying of agricultural, textile and marine products, and heating of buildings to maintain a comfortable environment especially in the winter season. Designer and potential user of these systems must consider a number of factors when comparing their merits. These can mainly be categorized as: (i) thermal performance, (ii) cost (iii) lifetime/durability, (iv) maintenance and (v) ease of installation. Thermal performance of collectors is compared by using the concept of thermal efficiency. It is generally believed that the thermal efficiency of a solar collector is the major requirement for the prediction of thermal performance of the complete solar system of which the solar air collector is a part. *Chandra and Sodha, 1991* have provided a fundamental understanding of testing procedures for solar air heaters.

Solar air heaters are broadly classified into two types: bare plate and cover plate solar energy collectors, *Ekechukwu and Norton, 1999*. Based on this classification, authors have summarized various designs of solar air heaters. Solar air heaters are simple devices to heat air by utilizing solar energy. Such heaters are implemented in many applications which require low to moderate temperature below 60 °C. The disadvantages of solar air heaters are the need for handling larger volumes of air than liquids due to the low density of air as a working substance and low thermal capacity of air. However, the efficiency of solar air heaters is low because of the low Prandtl number of air *Ucar and Inallı, 2006* and low absorber to air heat transfer coefficient *Ozturk and Demirel, 2008*. Also, in cases where thermal storage is needed, water is superior to air. Some of these disadvantages are being addressed using theoretical models. On the other hand, the most important advantages for solar air heaters include: no freezing, boiling or pressure problems; generally lower weight and low construction cost *Selcuk, 1977, Qenawy and Mohamad, 2007*.

Important design parameters are identified and their appropriateness validated through experimental results, *Karslı, 2007* and *Chow, 2010*. Different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and air. These modifications include use of absorber with fins *Garg et al., 1991*, a corrugated absorber *Choudhury et al., 1988*, a solid matrix *Sharma et al., 1991* and hollow spheres *Swartman and Ogunade, 1966*. These modifications enhance the thermal efficiency of solar collector but also increase the pressure drop, which becomes important at high volume flow rates of the air. *Mohamad, 1997* has analyzed the performance of a counter-flow solar air heater with a porous matrix and the results were compared with conventional solar air heaters, thermal efficiency of this type of collector was found to be exceeding 75% which is much higher than the efficiency of the conventional air heaters.

Several investigators have attempted to design more effective solar air heaters by changing the design characteristics, the applications of solar air heaters or using various collector types. Experimental investigation of three different solar flat plate air heaters was carried out by *Deniz et al., 2010*, two of which having fins, Type II and Type III, and the other without fins, Type I, one of the heater with a fin had single glass cover, Type III and the others had double glass covers, Type I and Type II. The energy and exergy output rates of the solar air heaters were evaluated for various air flow rates 25, 50 and 100 m^3/m^2 h, tilt angle 0, 15 and 30° and temperature conditions versus time. Based on the energy and exergy output rates they found heater with double glass covers and fins, Type

II, to be more effective and the difference between the input and output air temperature to be higher than of the others. Besides, they also found that the circulation time of air inside the heater played more important role than of the number of transparent sheet. Lower air flow rates should be preferred in the applications of which temperature differences is more important.

Performance studies on four different type of solar air heater, two corrugated and two mesh, was performed by *Gupta and Garg 1966* .They concluded that solar air heaters of the corrugated type of construction can be fabricated to supply hot air up to 30 ^{o}C above ambient with an overall efficiency of 60 % and mesh type of air heaters to supply air up to 20 ^{o}C above ambient with an efficiency of 50 % or more. *Ong and Than, 2002* have studied performance of flat plate solar air heater operating under natural convection. They plotted mean glass, wall and air temperatures, outlet air temperature, outlet air flow velocity and instantaneous efficiency against solar radiation. They observed that all the variables plotted increased with incident solar radiation. Wall temperature was higher than glass or air temperatures. Glass temperature was observed to be lower than air temperature initially and at low radiation levels. At higher radiation glass temperature was higher than air temperature. At 90^{o} tilt, glass temperature was always higher than air. It indicated that the air flow in the duct was non-symmetrical and that the air was initially heated up by the heat absorbing wall and then transferred some heat to the glass as it flows upwards.

Satcunanathan and Deonarine, 1973 have suggested use of two pass solar air heater to reduce the heat loss. *Whillier,1964* carried out experiments and analyze the conventional air heater consists of an absorbing plate, a rear plate, insulation below the rear plate, transparent cover on exposed side, and the air flows between the absorbing plate and rear plate. *Ranjan et al., 1983* refers to an air heater with flow above the absorber which consists of an absorber plate with a transparent cover at the top and insulation at the bottom. The cover and the plate provide the passage for the air. Solar radiation, after transmission through the cover, is absorbed by the absorber plate. *Sodha and Bansalt, 1982* have studied an air heater with flow on both sides of the absorber assuming that equal flow occurs both above and below the absorber plate and the heat transfer coefficient between the absorber plate and the air stream on either side is the same.

There are numerous works on both experimental and theoretical performance of solar air heaters. Steady state heat balance equations, involved linking plate efficiency and heat removal factors with air flow rates, surface wind heat transfer coefficients, and heat transfer coefficients between the moving air streams and the surfaces forming the flow channels also reported *Parker 1981, Vijeysundera et al., 1982, Than and Ong 1984, Biondi et al. 1988, Duffie and Beckman 1991, Verma et al. 1992, Parker et al. 1993.* Different mathematical model and solution procedure were studied by *Ong, 1995*. Finite Element Modeling of serpentine Solar Collector was presented by *Álvarez et al., 2010.* It was found that because of new topology it was possible for the fluid to have greater contact surface with the absorber plate and leads to an increase in collector efficiency. They obtained an efficiency of 84.5% with maximum plate temperature of 31.7^{o}C for stationary analysis.

For low temperature solar drying applications (temperature rise less than 40^{o}C above ambient), single covered solar air heaters are adequate. Where higher temperature rises are desired, double or triple covered solar air heaters can be used to reduce drastically the upward convective and re-radiative heat losses. However, the resulting higher temperature rise would imply more insulation than in bare-plate or single covered solar air heaters. The additional cost of constructing double or triple covered solar air heaters (with improved insulation) may not be justified economically for numerous low cost passive solar air heater designs. Because of the considerable heat losses in bare-plate solar air heaters (with temperature rise less than 10^{o}C above ambient), they would require high air velocities 15 m/s. The limitation imposed by this is the use of fans which, thus, makes bare-plate solar-energy air heaters inappropriate for natural-circulation solar dryers. For higher temperature rises, the reduction of heat losses from the absorber plate becomes necessary by the use of glazing. Thus Single cover Flat Plate Solar air Heater is selected for Mathematical Modeling.

III Problem Formulation

A literature review of solar air heaters shows that that the different type of solar air heaters have been investigated experimentally under different systems and operating conditions by several researchers. A comparison of performance of these heaters become difficult without identical values of parameters such a ambient conditions, inlet air temperature, air mass flow rate per unit collector width, gap between the absorber and cover plates, intensity of solar radiation etc. Thus to compare the performance of these heaters under identical values of parameters, their simulation becomes necessary.

Numerical simulation has been revealed as an important tool in the past years in order to improve system performance and process optimization. Aim and objective of this work is to review and understand the design procedure of flat plate solar air heaters, FPSAH, and develop a mathematical model for thermal analysis of FPSAH. Based on the literature review simple geometry of a conventional solar air heater is selected.Mathematical simulation of Conventional flat plate solar air heater, FPSAH has been carried out.

Mathematical model for designing a FPSAH is obtained by the application of the governing conservation laws. The heat balance is accomplished across each component of FPSAH i.e., glass covers, air stream and the absorber plate. The heat balance for the air stream yields the governing differential equations and the associated boundary conditions. The finite difference method, FDM, is used to solve the differential equations and hence to simulate a given solar air heater. Study has been extended by changing the various governing parameters like the air mass flow rate, the inlet air temperature, the depth of the collector duct and the intensity of solar radiation.

Simulation in a crude sense is the method of predicting the output of a proposed system or an existing system without conducting experiment on it. System simulation involves the development of the mathematical model for a given problem which should be dynamic in nature. The mathematical model is then solved numerically. Energy balance analysis for FPSAH has been presented in this work. The analysis is based on the assumption that heat and the fluid flow are one-dimensional and in steady state. A numerical approach is applied to obtain the solution of the given problem.

In FDM method, the physical domain is first discritized into computational domain by the method of grid generation. The grid consists of the linear elements with nodes at the point of intersection. The following steps describe the whole procedure:

(i) Differential Equations can be obtained by the application of governing conservation laws over a system.
(ii) Applying finite difference schemes to transform the given differential equations into the difference equations.
(iii) Algebraic equations in nodal unknowns are thus obtained.
(iv) These simultaneous algebraic equations can be solved by any numerical method.
(v) The solution of these algebraic equations is the solution at the nodes.
(vi) Finally, quantities of interest can be calculated.

Following are the assumptions made in the analysis of the conventional FPSAH:

1. The air flow is steady and one dimensional.
2. Thermo physical properties of air are independent of temperature.
3. Air velocity in the channel at any section is constant.
4. The temperature drop across the thickness of the covers is negligible.
5. Heat loss from the sides of the collector is very small and hence neglected.
6. The temperature distribution within each element and glass cover is uniform.

IV.Mathematical Modeling and Analysis

Design procedure for solar air heater is presented below. Solar data required for simulation program has been taken from *Mani, 1981*. The equations used for calculating the collector efficiency are based on the analysis listed in *Sukhatme S.P.and Nayak J K, 2011*.

4.1 Design Procedure of Solar Collector
The simulation procedure is given below.
Determining number of the day for 21st April 2010

$$n = 31 + 28 + 31 + 21$$

$$n = 111$$

Hour angle at 11.00 am

$$\omega = ((12 - time)\cdot deg)\cdot 15$$

$$\omega = 15^{o}$$

Declination angle for the day of the year, n = 111, for 21st April

$$\delta = (23.45.deg).sin\left[\frac{360}{365}(284+n).deg\right]$$

$$\delta = 11.58^{o}$$

Cos of incident angle for β = 30° and ϕ_l = 19.12°

$$cos\,\theta = sin(\delta)\,sin(\Phi_l - \beta) + cos(\delta).cos(\Phi_l - \beta)$$

$$cos\,\theta = 0.924$$

Angle of reflection

$$\varphi_r = \sin^{-1}\left(\frac{\sin\theta}{1.50}\right)$$

$$\varphi_r = 14.77^{\circ}$$

Transmissivity of beam radiation for the values of $\tau_r = 0.922$, $\tau_{ar} = 0.969$

$$\tau = \tau_r . \tau_{ar}$$

$$\tau = 0.894$$

Transmissivity absorptivity product for $\alpha_p = 0.9$

$$\tau\alpha_b = \tau . \frac{\alpha_p}{[1-(1-\alpha_p).0.15]}$$

$$\tau\alpha_b = 0.82$$

Transmissivity of diffused radiation for the values of $\tau_d = 0.738$, $\tau_{ad} = 0.964$

$$\tau_{dif} = \tau_d . \tau_{ad}$$

$$\tau_{dif} = 0.71$$

Transmissivity and absorptivity product of diffused radiation

$$\tau\alpha_d = \tau_{diff} . \frac{\alpha_p}{[1-(1-\alpha_p).0.15]}$$

$$\tau\alpha_d = 0.65$$

Cos of zenith angle

$$\cos\Theta = \sin\Phi_l . \sin\delta + \cos\Phi_l . ccs\delta . \cos\omega$$

$$\cos\Theta = 0.96$$

Tilt factor for beam radiation

$$r_1 = \frac{\cos\theta}{\cos\Theta}$$

$$r_1 = 0.963$$

Tilt factor for diffuse radiation

$$r_2 = \frac{1+\cos\beta}{2}$$

$$r_2 = 0.933$$

Tilt factor for reflected radiation

$$r_3 = \rho_r . \frac{1-\cos\beta}{2}$$

$$r_3 = 0.013$$

Flux falling on tilted surface for the values of $I_b = 591$ W/m^2, $I_d = 254$ W/m^2

$$I_t = [I_b . r_1 + I_d . r_2 + (I_b + I_d) . r_3]$$

$$I_t = 817.34 \text{ W/m}^2$$

Assuming efficiency of solar collector as 66%

$$\eta_c = \frac{q_u}{I_t}$$

$$q_u = 539.44\, W/m^2$$

Mass flow rate of the air is calculated by assuming a difference of 30 °C in inlet and outlet air temperature of the collector.

$$q_u = m\, Cp\, dt$$

Mass flow rate of the air
m = 17.89 g/s or 64.4 kg/h.

Mass flow rate of the air can be controlled by adjusting fan speed or using dampers or flow controllers.

Energy Balance analysis have been done for FPSAH. The analysis is based on the assumption that heat and the fluid flow are one-dimensional and in steady state. A numerical approach is applied to obtain the solution of the given problem. In the present study first a mathematical model is obtained by the application of the governing conservation laws. The heat balance is accomplished across each component of a given air heater, i.e., the glass covers, the air streams and the absorber plate. The heat balance for the air stream yields the governing differential equations and the associated boundary conditions. It is because of the radiation heat exchange terms that render the problem non-linear hence making the exact solution cumbersome. So a numerical approach which would give a solution with a fairly good accuracy is needed.
The finite difference (FDM) has been employed to solve the differential equations. In FDM technique the first step involves the transformation of the actual physical domain

into the computational grid. Second step is to transform the differential equations into difference equations. Which along with the equations obtained by heat balance across the covers and the absorber are the simultaneous non linear algebraic equations. The next step is to solve those numerically using gauss elimination method. The solution is obtained in the form of nodal temperatures for the covers, the air streams and the absorber. Study has been extended by changing the various governing parameters like the air mass flow rate, the inlet air temperature, the depth of the collector duct and the intensity of solar radiation and finally the performance characteristics have been obtained.

Under steady state operating conditions, the energy balance for the Conventional FPSAH and applying the Finite Difference Method on proposed model are as below:

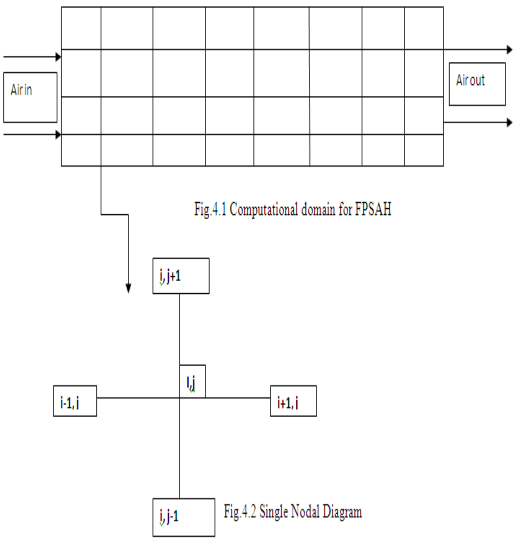

Fig 4.1 and 4.2 Computational Domain for FPSAH

For i^{th} node

1. For Absorber plate:

Governing Differential Equation is

$$I\ \tau_b\alpha_b = U_l\ (T_{pm}-T_a) + h_{fp}\ (T_{pm}-T_f) + h_r\ (T_{pm}-T_{bm})$$

The Finite Difference equation is (FDE)

$$I\ \tau_b\alpha_b = U_l\ (T_{pm}[i]-T_a) + h_{fp}\ (T_{pm}[i]-T_f[i]) + h_r\ (T_{pm}[i]-T_{bm}[i])$$

On solving the above equation we get,

$$T_{pm}[i] = \frac{I\ \tau_b\alpha_b + U_l T_a + h_{fp} T_f[i] + h_r T_{bm}[i]}{U_l + h_{fp} + h_r}$$

2. For Air Stream:

Governing Differential Equation is

$$m.c_p \frac{dT_f}{dx} = h_{fp}\ (T_{pm}-T_f) + h_{fb}\ (T_{bm}-T_f)$$

Solving FDE using Central Difference formulation

$$m.c_p.\frac{T_f[i+1]-T_f[i-1]}{2\Delta x} = h_{fp}\ (T_{pm}[i]-T_f\ [i-1]) + h_{fb}\ (T_{bm}[i]-T_f\ [i+1])$$

On solving we get

$$T_f[i+1] = \frac{\frac{2\Delta x}{mc_p} h_{fp} T_{pm}[i] + \frac{2\Delta x}{mc_p} h_{fb} T_{bm}[i] + T_f[i-1]\left[1-\frac{2\Delta x}{mc_p} h_{fp}\right]}{1+h_{fb}}$$

Associated Boundary condition $T_f\ [-i] = T_1$

3. For Bottom Plate:

Governing Differential Equation is

$$h_r\ (T_{pm}-T_{bm}) = h_{fb}\ (T_{bm}-T_f)$$

The FDM equation is

$$h_r\ (T_{pm}[i]-T_{bm}[i]) = h_{fb}\ (T_{bm}[i]-T_f[i])$$

On solving we get,

$$T_{pm}[i] = \frac{T_{bm}[i][h_{fb}-h_r]-h_{fb}T_f[i]}{h_r}$$

For convenience heat transfer coefficient between absorber plate and bottom plate are equal.

Therefore $h_{fp} = h_{fb} = h_f$

V .Results

A program in MathCAD has been developed to simulate the conventional FPSAH. Further FDM has been employed to solve differential equations obtained by heat balance across each component (glass cover, air stream and absorber plate).

FPSAH is simulated for standard Solar Radiation Data for Chennai. Average solar insolation for Chennai is 5.5 kW/m^2 for 7.5 sunshine hours i.e. @ 734 W/m^2. Intensity of solar radiation is also calculated for different incident angles. Collector selected for simulation has single glass cover, selective coated aluminum plate absorber, glass wool insulation enclosed in a metallic frame. Outlet temperature of air was predicted for specific mass flow rate of air for entire day. Various losses were calculated. Useful heat gain of the collector was estimated. Finally thermal efficiency of the collector was calculated. Mass flow rate of the air flowing through a collector is one of the important parameter as far as collector performance in concern. Outlet temperature of air from the collector at various flow rates was also calculated using the developed model. Effect of change in mass flow rate, intensity of solar radiation, absorber material on collector performance was predicted using the developed model and then compared with experimental data available in literature.

5.1 Important Outcomes for Conventional FPSAH:

1. Effect of changing parameter on Thermal efficiency: Fig 5.1 shows the variation in thermal efficiency with mass flow rate. First the efficiency increases rapidly for mass flow rate 0.01kg/s to 0.1 kg/s then it rise is almost constant. This is because the useful heat gain is directly proportional to the mass flow rate and the thermal efficiency is the ratio of useful gain to the total solar radiation incident on it.

2. Effect of changing parameters on $\Delta t/I_o$: Fig 5.2 depicts the variation of $\Delta t/I_O$ with mass flow rate. Initially $\Delta t/I_O$ decreases rapidly then becomes almost constant after the mass flow rate of 0.1 kg/s. This may be because the solar heat is converted into thermal mass of the system initially.

3. Effect of changing parameters on air outlet temperature ($t_{a.o}$): Fig 5.3 depicts the variation of air outlet temperature with mass flow rate. Air outlet air temperature $t_{a.o}$ is comparatively high at lower mass flow rates. The drop in outlet temperature is significant in early stages of rise in mass flow rate but becomes almost constant after mass flow rate of 0.1 kg/s.

4. Effect of changing parameters on Useful heat gain (Q_U): Fig 5.4 depicts the variation of Useful heat gain with mass flow rate. At higher mass flow rates useful heat gain is more. More heat is carried away by flowing air this lowers the collector temperature. Lower collector temperature results in reduced losses. This in turn increases useful heat gain.

Results obtained from the simulation study of Conventional Flat plate Solar Air Heater under consideration is tabulated below:

Mass flow rate is varied (mflow) from 1 kg/hr till 570 kg/hr in steps of 10: [At 12.30pm at Chennai,(13.0810° N, 80.2740° E)]: Similar values can be calculated from 9.30 am till 4.30pm anywhere within India.(Conventional FPSAH:I=734 W/m², t(a,i)=48°C

Mass Flow Rate(m_{flow}),kg/hr	Mass Flow Rate (m_{flow}),kg/s	Air inlet temperature($t_{a,i}$), K	Air Outlet temperature($t_{a,o}$),K	($t_{a,o}$-$t_{a,i}$) or Δt	$\frac{\Delta t}{I_O}$	Effective heat transfer coefficient(h_e), W/m.K	Collector Efficiency (FCEF)	Collector Heat removal factor(F_R)	Useful heat gain(Q_U),W	Pressure drop across collector (P.D.),N/m²	Instantaneous Efficiency of collector (η), %
1	0.000	321	441.615	120.615	0.1519	0.384	0.071	0.031	33.672	0.000879	1.9
10	0.003	321	401.725	80.725	0.1017	2.269	0.31	0.209	225.356	0.049	12.9
20	0.006	321	383.487	62.487	0.0787	3.763	0.427	0.323	348.887	0.165	20
30	0.008	321	372.588	51.588	0.0650	5.013	0.498	0.4	432.051	0.336	24.8
40	0.011	321	365.167	44.167	0.0556	6.12	0.548	0.457	495.193	0.557	28.3
50	0.014	321	359.734	38.734	0.0488	7.129	0.585	0.5	540.657	0.823	31
60	0.017	321	355.562	34.562	0.0435	8.065	0.615	0.536	578.917	1.132	33.2
70	0.019	321	352.247	31.247	0.0394	8.945	0.639	0.565	610.623	1.482	35
80	0.022	321	349.543	28.543	0.0359	9.781	0.659	0.59	637.463	1.872	36.6
90	0.025	321	347.291	26.291	0.0331	10.579	0.677	0.611	660.573	2.301	37.9
100	0.028	321	345.385	24.385	0.0307	11.346	0.692	0.63	680.745	2.767	39.1
110	0.031	321	343.748	22.748	0.0286	12.087	0.705	0.647	698.556	3.269	40.1
120	0.033	321	342.326	21.326	0.0269	12.805	0.717	0.661	714.434	3.806	41
130	0.036	321	341.079	20.079	0.0253	13.502	0.728	0.675	728.706	4.379	41.8
140	0.039	321	339.975	18.975	0.0239	14.182	0.737	0.687	741.624	4.985	42.6
150	0.042	321	338.991	17.991	0.0227	14.846	0.746	0.697	753.391	5.625	43.2
160	0.044	321	338.108	17.108	0.0215	15.496	0.754	0.707	764.168	6.297	43.9
170	0.047	321	337.311	16.311	0.0205	16.132	0.762	0.717	774.085	7.002	44.4
180	0.050	321	336.587	15.587	0.0196	16.757	0.768	0.725	783.252	7.739	45
190	0.053	321	335.927	14.927	0.0188	17.371	0.775	0.733	791.757	8.507	45.4
200	0.056	321	335.323	14.323	0.0180	17.976	0.781	0.74	799.676	9.306	45.9
210	0.058	321	334.767	13.767	0.0173	18.571	0.786	0.747	807.073	10.135	46.3
220	0.061	321	334.254	13.254	0.0167	19.157	0.791	0.754	814.002	10.995	46.7
230	0.064	321	333.779	12.779	0.0161	19.736	0.796	0.76	820.511	11.884	47.1
240	0.067	321	333.338	12.338	0.0155	20.307	0.801	0.765	826.64	12.803	47.4
490	0.136	321	327.686	6.686	0.0084	33.047	0.867	0.847	914.645	44.647	52.5
500	0.139	321	327.568	6.568	0.0083	33.515	0.869	0.849	916.742	46.253	52.6
510	0.142	321	327.453	6.453	0.0081	33.98	0.871	0.85	918.778	47.884	52.7
520	0.144	321	327.343	6.343	0.0080	34.443	0.872	0.852	920.755	49.539	52.8
530	0.147	321	327.236	6.236	0.0079	34.904	0.874	0.854	922.677	51.219	53
540	0.150	321	327.133	6.133	0.0077	35.363	0.875	0.856	924.544	52.922	53.1
550	0.153	321	327.033	6.033	0.0076	35.82	0.876	0.858	926.36	54.649	53.2
560	0.156	321	326.937	5.937	0.0075	36.275	0.878	0.859	928.128	56.399	53.3
570	0.158	321	326.844	5.844	0.0074	36.728	0.879	0.861	929.848	58.174	53.4
580	0.161	321	R_e>20000 (flow is not smooth)			Hence equations	Not valid	Giving error values.			

Table 5.1 Results for Chennai Region

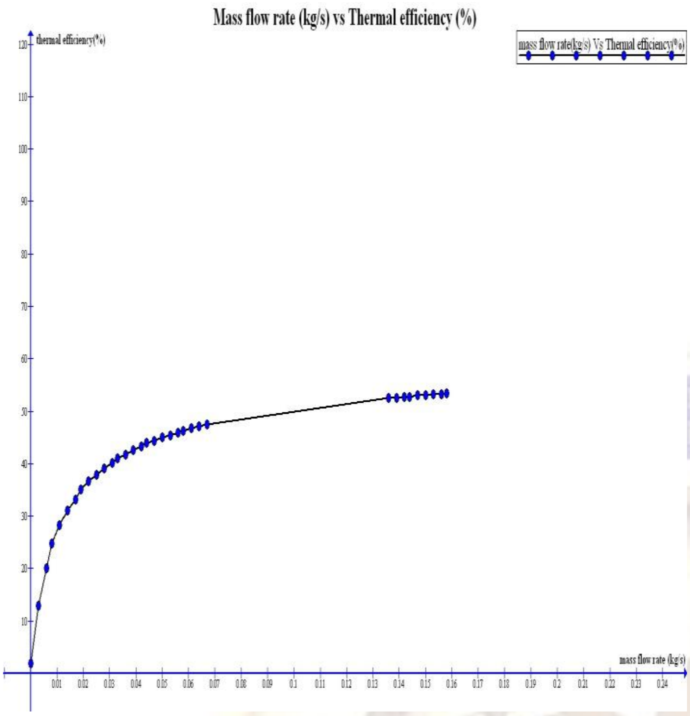

Fig.5.1 Variation of Mass Flow Rate (m) vs. Thermal Efficiency (η)

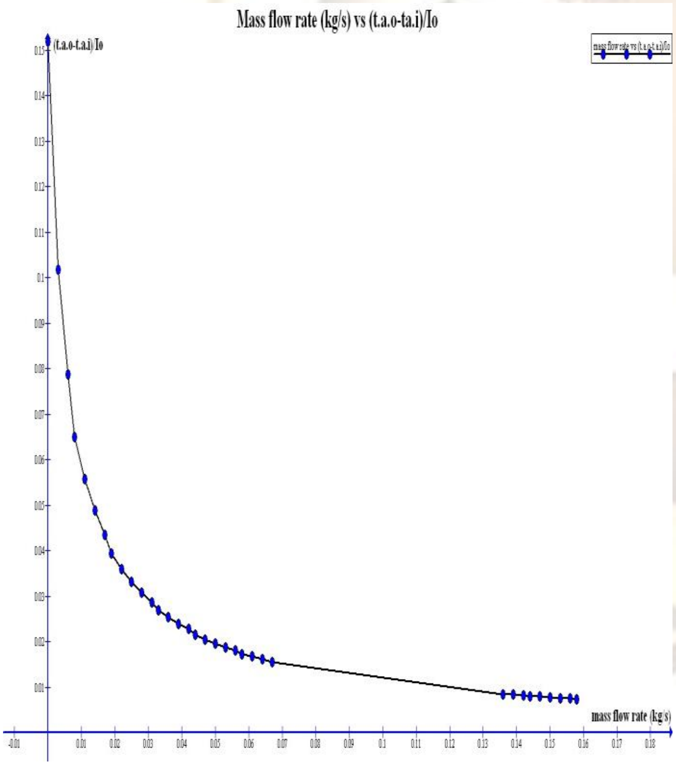

Fig.5.2 Variation of Mass Flow Rate (m) with $\Delta t/I_o$

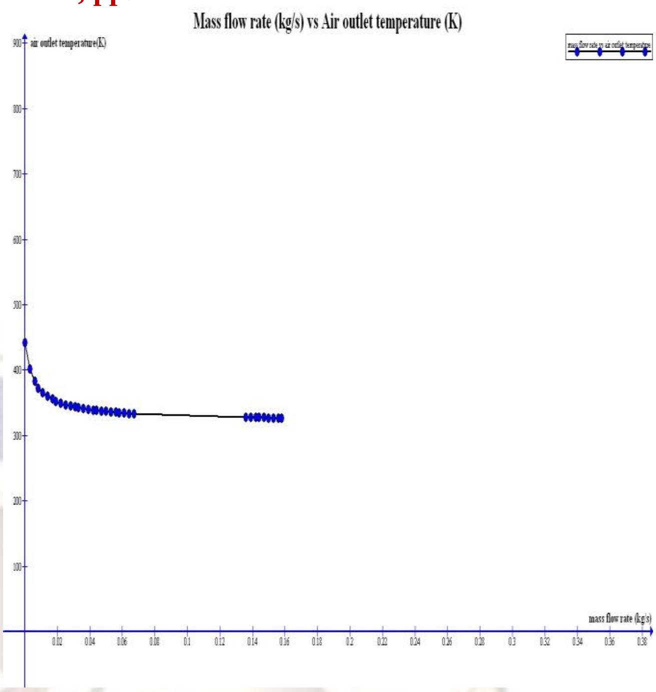

Fig.5.3 Variation of Mass Flow Rate (m) with Air Outlet Temperature ($T_{a.o}$)

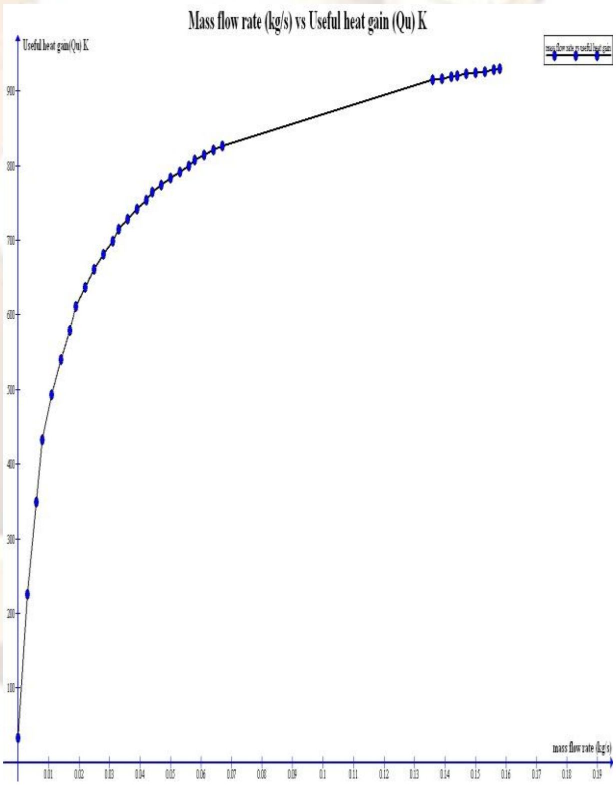

Fig.5.4 Variation of Mass Flow Rate (m) with Useful Heat Gain (Q_u)

VI.Conclusions and Recommendations for future work

Results from the developed model such as efficiency of the collector for specific flow rate, inlet and outlet temperature of air or temperature lift of the collector are compared with experimental data available in the literature. It is found that, model as well as simulation program are predicting closely matching results with experimental data. This model may be a suitable model for analyzing conventional FPSAH.

Simulation program was run for location specific condition, Chennai (13.0810° N, 80.2740° E). It is revealed from the simulation program that, at solar insolation of 734 W/m^2 and, mass flow rate of 440 kg/h (0.122 kg/s) gave efficiency about 51.8%. This result is closely matching with results recorded in literature for the same type of collector.

Recommendations for future work

Use of computational fluid dynamics along with developed model may lead to exhaustive thermal analysis of flat plate solar collectors. Temperature distribution along the absorber surface as well as temperature distribution in the glass cover sheets can be predicted in a better way using CFD. Knowledge of heat distribution in each component and thermal analysis of the collector lead towards better ways to further minimize various losses from the collector. This will certainly lead towards enhanced efficiency of the collector. Further Work can also be extended for varying solar radiation instead of constant solar radiation as in present work.

Use of Finite Volume Method along with Implicit modeling technique may reduce error in predicting thermal behavior and performance of the collector.

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