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Design of Tubular Pass Solar Air Heaters

Disha Dewangan[1*], Jasinta Poonam Ekka[2]

[1,2] Department of Mechanical Engineering, Guru Ghasidas Vishwavidyalaya, Bilaspur, Koni 495009, India.

*Corresponding author email: *dewangan.disha@gmail.com, Tel. :

ABSTRACT

Solar energy is inexhaustible and has a strong affinity to give surplus power among other sources of renewable energy which is received on the surface of Earth. The study focused on the design of tubular-type solar air collectors where the tubes are placed adjacent to each other at an equidistance of 10.08 mm and are made from a conductive material like aluminium. The parameters of design namely size and volume flow rate from an air heater are estimated by assuming that it gives a maximum efficiency of 35% with a flow rate of 0.0125 kg/s. The other parameters like the equivalent diameter of the duct, number of tubes, Reynold number, frictional effect, pressure drop, and power consumption by fan used in tubular type solar air heater are evaluated for an airflow rate of 0.0125 kg/s. The diameter of tubes used in the tubular solar heater is selected based on the heat removal factor and it is available in standard size here it is considered as 38.1 mm for analysis. The newly designed solar air heater maximizes the usage of solar energy in the tubular absorber surface and gives a maximum temperature of the air at the exit.

Keywords - Absorber plate, equivalent diameter, heat removal factor, pressure drop, tubular solar air heater.

1. INTRODUCTION

Solar energy is the most promising source of non-conventional energy as it is easily available during the period, ecologically balanced, and is received everywhere throughout the globe. With the increase in energy dependency and environmental alarms, it provides a viable solution for industrial and domestic applications. Effective utilization of solar energy helps in nation-building and minimizes the dependency on fossil fuels. Thermal energy conversion systems (TECS) which utilize solar energy and chiefly employ solar air heaters (SAHs) have a good potential to harness an ample amount of the sun's energy [1]. SAHs extensively found their application in energy transformations processes like space heating, crop drying, and process heating, etc. due to their simple design, low cost and easier maintenance of the system, and working temperature ranges of 30-80^oC (Satcunanathan & Deonarine, 1973). Most absorber surfaces of SAH are off the flat-type plate, which is further applied for medium and low-temperature applications [3].

SAHs are mainly encountered with the problem of low thermal efficiency, as the air flowing through the collector has low heat capacity and, thermal conductivity in comparison to the liquid-type collector, these two factors majorly affect the performance of SAH [4]. The thermal performance parameters of SAH improve the heat transfer coefficient between the absorbing air and medium will enhance its thermal efficiency. The performance of SAH is strongly affected by its design and which has attracted researchers toward design modification techniques [5]. The design methodologies overcome this situation by quoting several works based on the research carried out by numerous researchers which aims to improve the SAH design. A lot of work is carried out in design modification by employing the obstacles (Vivekanandan et al., 2020; Sharma et al., 2019) like various designs of baffles [8], modifications in rib shape (Sureandhar et al., 2021), and modifications in the number of passes (Hussein et al., 2023; Satcunanathan & Deonarine, 1973; Wijeysundera et al., 1982), etc. Several research papers have been carried out by researchers on the improved design of absorber plate surfaces viz. Hassan et al., (2020) designed and compared the performance of tubular-type solar heaters (TTSAH) with flat-type solar air heaters (FTSAH) for three air flow rates of 0.075, 0.05, and 0.025 kg/s respectively. It was found that the outlet temperature of air increases by 13.8^oC, 10^oC, and 3.6^oC for flow rates of 0.075, 0.0,5, and 0.025 kg/s in TTSAH in comparison to FTSAH. The daily average efficiency of the TSAH is found to be 83.6%, 76.3%, and 59.8% for an airflow rate of 0.075, 0.05, and 0.025 kg/s, with a corresponding increment of 132.6%, 58.6%, and 43.5% in average efficiency, to that of FTSAH.

Sozen et al., (2020) modify the design of tube-type (TTSAH) with an iron mesh configuration to enhance the thermal performance of an indirect-type solar dryer (ITSD). Observed that the thermal efficiency was found to be in the range of 59.94–67.69% in modified heater design, while it was obtained in the range of 51.19–56.54% in hollow TTSAH. Similarly Abo-Elfadl et al., (2020) carried out a performance investigation on tubular absorber surfaces of a double-pass solar air heater (DPSAH). The novel design leads to improving the maximum efficiency of TSAH by 90.1% in comparison to FSAH which reaches 75.37% for an airflow rate of 0.075 kg/s. The average efficiency of TSAH is estimated as 86.03%, 76.3%, and 59.8%, with an increment of 19.4%, 21%, and 40.3% as compared to FSAH for an inlet mass flow rate of 0.075 kg/s, 0.05 kg/s, and 0.025 kg/s respectively. The design modifications in TPSAH using a porous material and to investigate its energy, exergy, and economic study is also performed by (Abo-Elfadl, Yousef, et al., 2021a; Abo-Elfadl, S. Yousef, et al., 2021b). It was observed that porous material in conjunction with to double pass arrangement raises the outlet temperature of the air in TSAH by 3.4◦C and FSAH by 17.7◦C respectively for a flow rate of 0.025 kg/s. It was found that an airflow rate of 0.075 kg/s in TSAH in conjunction with porous material enhances the energy efficiency from 3.5% to 8.9% and degrades the exergy efficiency from 8.2% to 0.9% respectively for single and double pass modes.

The design modifications carried in the shape of the absorber plate by making it corrugated, V-shape, and triangular, fins done by Kavoosi et al., (2015) investigate the design and performance of a triangle channel air collector (TCAC) which is obtained by joining two pieces in V-shape on a flat absorber surface for different flow rates. It was observed that the design of TCAC is 22% more efficient than FPC and 10% more than V type collector, with a maximum output temperature reached 102ºC obtained in natural convection mode. It was observed that the appropriate flow rate for drying agricultural crops is 0.04 kg/s. Karim & Hawlader, (2006) investigated the factors which majorly influence flow and performance parameters of a flat plate, V-corrugated and finned SAHs which can be employed in dryers. Developed and design a SAH which is further used in drying agricultural products and investigates the performance of different configurations collectors in single and double pass modes by replacing the simple SAH with fin and V-corrugated configuration. Sureandhar et al., (2021) developed and analyzed the variation of ribbed fin on the absorber plate of length 1 and 2 m respectively for an airflow rate of 0.02 kg/s to 0.06 kg/s. Observed that the maximum air outlet temperature of 318 K is achieved in a variable arc rib fin of 2 m length configuration for air flow rate of 0.02 kg/s.

Hussein et al., (2023) designed and performed numerical analysis on double pass SAH with three arrangements in first configuration the double pass solar air heater (DPSAH) is used with a flat plate solar absorber (DPSAHWFP), in second arrangement the tubes are fitted perpendicularly to the flow direction of a DPSAH with a tubular absorber surface (DPSAHWT-1), and in third arrangement the tubes are fitted along axially to the flow direction viz. a DPSAH with a tubular absorber (DPSAHWT-2) for varied flow rates of 0.01-0.03 kg/s. It was observed that DPSAHWT-2 gives maximum thermal performance in comparison to other configurations and the maximum effective efficiency reached to 80.9 % for an air flow rate of 0.03 kg/s. The effective efficiency raises from 4.2 % to 8.8 % in comparing DPSAHWFP and DPSAHWT-2. Vivekanandan et al., 2020 modified the design of SAH and compared its performance to that conventional SAH. Observed that the performance of the modified SAH which is obtained by attaching fins along the cross-sectional area increases the volume flow rate of heated air, the, and enhance the heat transfer area. Satcunanathan & Deonarine, (1973) designed a two pass SAH while Wijeysundera et al., 1982, performed and carried out a detailed study on the double pass SAH experimentally and analytically which is designed by Satcunanathan & Deonarine, (1973). The performance analysis was carried out by developing a thermal model and its results are validated from an experimental data.

In usual practice the SAH is design on the basis of a constant mass flow rate ($\dot{m}$) and it is based on the assumptions of steady-state where the radiant solar energy, air flow rate, ambient temperature, and the velocity of wind entering the system are considered as constant. Based on the first law of thermodynamics the thermal efficiency of the SAH is estimated from the following relation (Hegazy, 1999) [18]. The schematic diagram of tubular type solar air heater is shown in Fig. 1

$$\eta_{th} = \frac{\dot{m}C_p(T_{out} - T_{in})}{A_c I} \tag{1}$$

Where C_p specific heat of air in kJ/kgs, T_{out} and T_{in} are outlet and inlet temperature of air from SAH in K, A_c is the collector area of in m^2 and I is the intensity of solar radiation in W/m^2. The above equation gives a practical value of thermal efficiency (η_{th}) of SAH based on the characteristics of the construction materials. But we mainly focus on the design of tubular pass solar air heater which will enhance the heat transfer rate and thermal efficiency.

From citing various literature, it was found that several works were performed on SAH to increase its thermal performance, but a limited work has been reported in the distribution of flow area inside the dividing header and among the tubes of a solar air heater in turbulence zones. The effective method for improving the performance of SAH is to modify the design of absorber plate surface, which is obtained by maintaining the uniform air flow distribution from header section to individual tubes. The studies shows that minimum pressure variation is required for uniform flow distribution of air in tubular tubes. The objective of the study is to design an TSAH which effectively maximizes the useful energy from its surface due to longer contact area within the tubes, it is designed to heat air for a fixed mass flow rate of air. With an aim that the present system will have a potential towards clean and heated air generation.

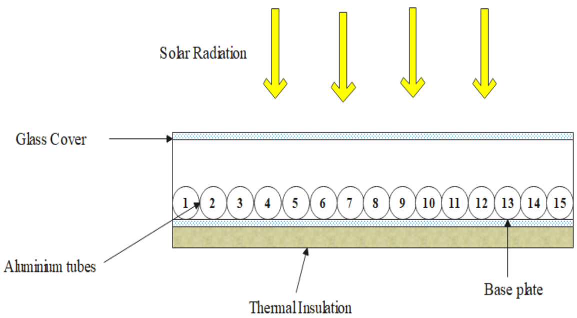

Fig. 1 Schematic diagram of a tubular pass solar air heater.

2. DESIGN METHODOLOGY

2.1. Design parameters of solar air heater: A TSAH finds its intense application in the field of removing the moisture from the given agricultural product. The following points were also taken into consideration for designing a tubular solar air heater.

1. Daily sunshine hours for the location.
2. Quantity of air required for heating.
3. Daily solar radiation to estimate energy received by air heater per day.

By assuming that the solar air heater will give a maximum efficiency of 40%, a mass flow rate of 0.0125 kg/s [19] a temperature difference of 45ºC exist between outlet and inlet air with an average intensity of 700 W/m^2 are considered in order to determine the collector area as mentioned in Eq. 1. The collector dimensions are evaluated from Eq. 1 the area of solar air heater is found as 2.02 m^2 by taking an aspect ratio of 2:1 the length, width and height are estimated to be 2.01 m, 1.01 m, and 0.12 m respectively as shown in Fig.2.

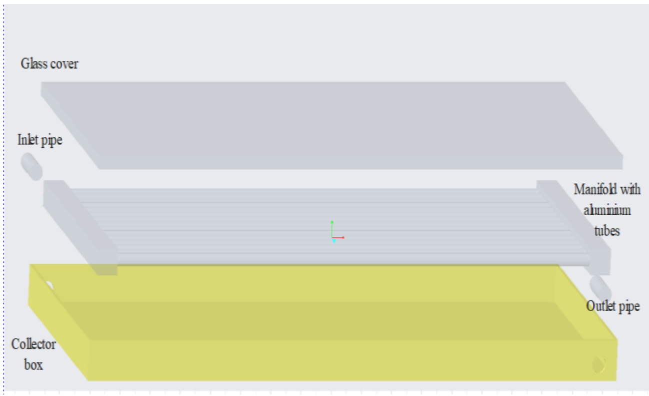

Fig. 2 Schematic of the solar air heater with tubular type absorber surface.

While designing the tubular surface of absorber tubes its diameter is based on the fact that neither it should be too narrow, nor it should be too wide. The circular tubes are chosen because they can withstand a large pressure difference between inside outside without undergoing significant distortion (Bhatti and Shah, 1987). The diameter of tubes is decided on the basis of its heat removal capacity from SAH and the diameter which gives higher value of heat removal factor is preferred (Sukhatme, Nayak, 2015). The diameter can also be decided by analysing the pressure drop and pumping power which is directly proportional to the length of the pipe, square of the velocity, but it is inversely proportional to the pipe diameter. Thus, the pumping power requirement of the system is reduced by doubling the pipe diameter (Bhatti and Shah, 1987). The tubular tubes of SAH are available in standard sizes (diameter and lengths), the diameter of tubular SAH is selected from the standard charts [20], in which it is available in customized sizes. Therefore, in designing a TSAH an internal diameter of 38.1 mm with a thickness of 1 mm is considered here for present work.

2.1.1 Number of tubes: It is required to estimate tube number for design of tubular pass SAH system it is basically determined from flow through a heat exchanger relation. The required surface area for the SAH can be given as [20].

$$A_c = \pi\, D\, L N_{tubes} \qquad (2)$$

Where Ac = area of SAH in m^2, D = tube diameter in m, L = length of tube in m, N_{tubes} = number of tubes.

Therefore, from the above equation, the required number of tubes for the SAH can be determined.

2.1.2 Tube pitch: It is based on tube diameter it is estimated as mentioned by [20]

$$p_t = 1.25 \times D \qquad (3)$$

2.1.3 Area ratio: A nondimensional and crucial parameter which gives the information about the flow rate in individuals tubes through header. The diameter of inlet header is 75 mm with fifteen tubular namely 1,2,3,…..15 tubes of 38.1 mm are attached through it as shown in Fig. 1 [21].

$$AR = \frac{\text{Crosssectional area of header } (A_H)}{\text{Crosssectional area of a tubular tubes } (A_T)} \qquad (4)$$

2.1.4 Continuity equation: It is the equation of conservation of mass when the fluid is in motion. It provides the important information of fluid transport behavior during its flow through a pipe, ducts, tubes, and hose. The velocity of air is determined from the continuity equation [22]

$$\dot{m} = \rho A_c V \qquad (5)$$

Where Density of air

$$\rho = 1.1774 - 0.00359(T - 27) \text{ in kg/m}^3 \qquad (6)$$

V = velocity of air in m/s.

2.1.5 Equivalent diameter of the duct/hydraulic diameter: It is generally used term during flow distribution through non-circular tubes and channels [23].

$$D_e = \frac{4LH}{2(L+H)} \qquad (7)$$

Where L is length of collector in m, H distance between absorber plate and tube in m.

2.1.6 Collector efficiency and heat removal factor: It is important parameter in design of SAH are given in Eq. (8) and (9) respectively [9]

$$F_C = \frac{1}{W U_L \left[\frac{1}{U_L[(W-D_o)\emptyset + D_o]} + \frac{\delta_a}{k_a D_o} + \frac{1}{\pi D_i h_f}\right]} \qquad (8)$$

$$F_R = = \frac{\dot{m} C_p}{U_L A_p}\left[1 - \exp\left(\frac{-U_L A_p F_C}{\dot{m} C_p}\right)\right] \qquad (9)$$

W = Pitch; D_o = Outer diameter of tube; φ = plate effectiveness

$$\varphi = \frac{\tanh\left(\frac{m(W-D_o)}{2}\right)}{m\left(\frac{W-D_o}{2}\right)} \qquad (10)$$

$$m = \sqrt{\frac{U_L}{k_p \delta_p}} \qquad (11)$$

k_p = thermal conductivity of absorber plate; δ_p = thickness of absorber plate; δ_a = average thickness of adhesive; k_a = thermal conductivity of the adhesive material; D_i = inner diameter of tube; h_f = heat transfer coefficient on the inside surface of tube.

U_L= overall heat loss coefficient coefficient.

$$U_L = U_t + U_b + U_e$$

The top loss coefficient is given as, *(Sukhatme, Nayak, 2015).*

$$U_t = \left[\frac{M}{\left(\frac{c}{T_{pm}}\right)\left(\frac{T_{pm}-T_a}{M+f}\right)^{0.252}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma(T_{pm}{}^2 + T_a{}^2)(T_{pm}+T_a)}{\frac{1}{\epsilon_p + 0.0425M(1-\epsilon_p)} + \frac{2M+f-1}{\epsilon_g} - M}\right] \qquad (13)$$

Where,

$$f = \left(\frac{9}{h_w} - \frac{30}{h_w{}^2}\right)\left(\frac{T_a}{316.9}\right)(1 + 0.091M) \qquad (14)$$

$$c = \frac{204.429(\cos\beta)^{0.252}}{L_c{}^{0.24}} \qquad (15)$$

L_c = Spacing between the plate and the cover; M = number of glass cover; T_{pm} = average temperature of absorber plate.

Where U_t is top heat loss coefficient and is mainly ranges from 6 to 9 W/m^{2}K

The Convective heat transfer coefficients due to wind velocity is considered as *(Sukhatme, Nayak, 2015)*:

$$h_w = 2.8 + 3.3V \qquad (16)$$

Where V is the wind velocity.

U_b is bottom heat loss coefficient is given as;

$$U_b = \frac{k_i}{x_i} \qquad (17)$$

U_e is edge heat loss coefficient,

$$U_e = \frac{(L+w)L_e k_i}{LWx_i} \qquad (18)$$

the physical properties of air like specific heat, density, viscosity etc., vary linearly with temperature (ºC)

Specific heat of air

$$C_p = 1.0057 + 0.000066(T - 27) \qquad (19)$$

The operating parameters like Re and f is determined from Eqs. 20 and 21 [24].

2.1.7 Reynolds number: It is a dimensionless quantity which is used to predict the fluid flow behavior as laminar or turbulent in conduits.

$$Re = \frac{\rho \dot{V} D_e}{\mu} \qquad (20)$$

2.1.8 Friction Factor: It indicates the resistance offer by a fluid in pipe or ducts it is mainly categorizing as basic friction factor & fanning friction factor. It plays a major role in determination of the pressure drop along the duct. It varies with the Reynolds number, using fanning equation (Eq. 18)

$$f = 0.079\, R_e^{-0.25} \qquad (21)$$

2.1.9 Pressure drop: It is the difference in total pressure among two points in a fluid transport system. It occurs when frictional forces offered by the flow resistance, acts on the fluid when it flows through a conduit like channel, pipe, or tube [25].

$$\Delta P = \frac{2f\rho V^2 L}{D_e} \qquad (22)$$

2.1.10 Fan power Consumption: As the air passes through the tubular SAH there is a pressure drop induced to propel the air inside the tubular section extra pumping power is required. Fan power consumption is directly related to the pressure drop along the flow channels and the amount of flowing air. The expression has been derived to be found as the following [26]

$$P_{fan} = \dot{V}\, \Delta P \qquad (23)$$

Where P_{fan} the fan power required in Watts, and ΔP is the pressure drop along the inlet and outlet of SAH.

Table 1 Design parameters of solar air heater

	Parameters	Symbol	Value
1	Mass flow rate, m/s	$\dot{m}$	0.0125
2	Equivalent diameter of the duct, m	De	0.0502
3	Reynolds number	Re	27756
4	Air velocity, m/s	V	9.254
5	Volume flow rate, m^3/s	$\tilde{V}$	0.0106
6	Friction Factor	f	0.00612
7	Pressure drop, N/m^2	ΔP	49.7
8	Fan power consumption, W	Pm	0.524
9	Dimensions of the collector, m	L, W, H	2.01×1.01×.012
10	Absorber tilt angle	β	22º
11	Effective length of the Aluminium tube, m	L_e	1.8
12	Effective diameter of the Aluminium tube, m	d_e	0.0381
13	No. of tubes	N_t	10
14	Density of air, kg/m^3	ρ	1.1848
15	Dynamic viscosity of air, Ns/m^2	μ	1.95×10^{-5}
16	Blower Power, HP	P	0.25
17	Emissivity of transparent glass sheet	ε_g	0.88
18	Emissivity of absorber plate	ε_p	0.9
19	Absorptivity of plate	α_p	0.95
20	Transmissivity of glass cover	τ_g	0.88

21	Solar radiation intensity, W/m^2	I	300-1000
22	Ambient temperature, K	T_a	300

3. RESULT AND DISCUSSIONS

The paper gives a progressive design methodology in the development of TTSAH. The origin of the new design lies in the fact to improve the efficiency and performance of SAH. The main aim of the present work is to provide a detailed design terminology which will be helpful in developing a tubular single pass or multiple pass solar air heater with minimum input it gives maximum output. The present work covers all design elements which are essential in designing a solar air heater which will improve system efficiency and its overall performance.

It is further necessary to acclaim that abundance amount of solar energy is available however is presided by inconsistency of its availability and dilution. Present work explains the implication and provides the in-depth knowledge of system design and associates the understanding of parameters which improves the overall system efficiency. Although various design of solar air heaters has been mentioned earlier prior to this, in present work attempts to incorporates a new viewpoint in design of efficient SAH which enhance the performance of TSAH by minimizing the thermal losses.

5. CONCLUSION

The main purpose of the paper is to focus on crucial design modifications for improving the thermal efficiency of SAH. Based on the theoretical evaluation related to design modifications in TSAH, following implications can be drawn:

- The TSAH design modifications is incorporated in collector tube configuration which modifies thermal efficiency of air heaters.
- The selective coating increases the temperature of air heaters which ultimately improves the thermal efficiency.
- The design modifications mainly related to geometry of tubular absorber which mainly tends to raise the surface area of absorption.
- Inclusion of inserts like baffles tends to increase fluid turbulence in the direction of flow thus increases momentum transfer within the adjacent layers of fluid flow and thus increases the thickness of thermal boundary layer for high energy transfer rate among absorbing surface and heat transfer fluid.
- The newer design modification and changes if suitably adopted, will results in significant gain efficiency in terms of thermal performance and exergy efficiency.
- For the mass flow rate of 0.0125 kg/s the proposed design achieves the average thermal efficiency of 48.7%.

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