Numerical investigation of performance analysis of Triangular Solar air heater using Computational Fluid Dynamics (CFD)

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Abstract. Computational fluid dynamics based research is performed to investigate the thermal efficiency of triangular solar air heater (TSAH) passage. A systematic analysis of the device output parameters was carried out controlled under all test conditions including solar intensity, mass flow rate and air temperature at the inlet section. The glazing side of the triangular section is subjected to the solar intensity 746 W/m². The three-dimensional TSAH model is developed and simulated using the FVM method with CFD code. The flow control equations have been solved with commercial ANSYS 14.5 software. A uniform triangular grid is generated according to the configuration using tetrahedron. The k-ε, RNG turbulence model with enhanced wall treatment function is used to solve the energy transport equations for turbulent flow. The optimal values of the geometric parameters are obtained on the basis of the performance index in the Reynolds number range from 2263 to 18103. It is observed the value of thermal efficiency increases with respect to velocity of air. It also increases the absorber and glass plate temperature.

Keywords: Triangular solar heater, Heat transfer enhancement, CFD, performance parameter.

1. Introduction
 Generally, the conventional solar air heater consists of an absorber plate with a parallel plate underneath which forms a small passage.[1] [2] The air is to be heated between the glazing cover and the absorber plate. This hot air is used in various applications like dairy, textile, paper, chemical, beverages and timber by-products. [3] TSAHs were widely used at low temperature ranges 300 K to 338 K.

In the current scenario, energy supply becomes a big problem in everyday life. A quantitative approach is needed to predict available energy resources Conventional energy sources and environmental risks.[4] Solar energy is a cost-effective renewable energy source. That can meet the ever-increasing energy consumption.[5] To get the in depth knowledge of the flow field developed in the duct above absorber plate,[3] analysis using computational fluid dynamics (CFD) is done by different researchers. CFD analysis of solar air heater with arc shaped ribs was done by Kumar and Saini[6] by making use of ANSYS FLUENT taking RNG k-ε turbulence model.

Singh et al.[7] did CFD as well as experimental study of SAH using roughness plates one having multiple broken transverse ribs and the other with square wave shaped ribs. It was found that for
square shaped rib roughened absorber plate the maximum thermo hydraulic efficiency was more than twice of that obtained from plane absorber plate but at the cost of three times the pumping power. For SAH with multiple broken ribs absorber plate efficiency as well as pumping power was thrice the normal value. Pitch to height ratio and height to diameter ratio of 10 and 0.043 respectively for ribs were taken for study.

A. Boulemtafes et al.[8], In the CFD study of SAH with transverse rectangular rib roughened absorber plate four closure models k-ε RNG, k-w standard, k-ε RZ and k-w SST were taken. Reynolds number was taken between 3000 to 15000.Important parameters like Nusselt number, friction factor; thermo hydraulic parameters were analysed in the study.

Manjunath et al.[9] used spherical shaped rib roughness to develop turbulent flow of air over absorber plate. Reynolds numbers between 4000to25000 were taken. Diameter of spherical ribs were taken between 5 to 25mm Nusselt number was found to be twice higher than the base model. It was concluded that spherical ribs produce higher thermo hydraulic performance.

Nidhul et al.[10] performed the CFD as well as exergy analysis of SAH with v shaped ribbed absorber plate and triangular cross sectional duct. It was found that combining ribbed absorber plate with triangular shaped duct gives higher efficiency as compared to using v ribbed plate in rectangular duct.

Singh et al.[11],analysed thermo hydraulic performance by experiment using triangular projections as roughness geometry on absorber plate of solar air heater. The roughness geometry resulted in increase in heat transfer, Nusselt number but also at the cost of some increase in friction factor value. Correlations were developed between friction factor and Reynolds number.

Singh et al.[12] observed the Nusselt number and friction factor characteristics of a SAH having rib roughness of non uniform cross section in its absorber plate. Computational fluid dynamics with renormalization group (RNG) and k-ε model were used in the investigation. Friction factor and Nusselt number were found maximum for roughness pitch value of 16.

Menni et al.[13] used four baffle plates in the passage of flow of air through the duct of rectangular section and performed its CFD analysis Ansys FLUENT software version 6.3.Aspect ratio and width to height ratio of 1.31 and 5.137 were taken respectively. The ratio of spacing between baffles and its height were taken as 0.972. The research resulted into a finding that thermal enhancement factor TEF increases with increase in Reynolds number. Its maximum value was obtained when Reynolds number of 32000 was taken.

Xiao et al.[14] performed the exergy analysis of solar air heater using exergy destruction minimization principle.SST k-w turbulence model and Reynolds number of 12000 was taken. Detailed study of velocity and temperature distribution was done. The Nusselt number and friction factor values were increased by 1.81 and 3.13 times over smooth and plain duct respectively. It was concluded that optimization approach is an effective tool in increasing the performance of solar air heater.

Qader et al.[15] employed the response surface methodology as well as CFD simulation in SAH with inclined fins in absorber plate. ANSYS fluent version 16.1 is used it the study. The results obtained in the study were compared with that of some conventional roughness geometry set up. Nusselt number and friction factor values of 2.53 and 2.22 respectively were found.

R. Kumar et al.[16], Investigate the thermal efficiency using triangular duct and describe the design and construction of triangular section. It was observed in result, the thermal efficiency increases with increases the mass flow rate.

R. Kumar et al.[17], [18] investigated the solar air heater performance using absorber plate selective coating. The selective coating was formed by nanomaterial embedded in black paint. The result obtained, the system's maximum efficiency at 1 m/sec air velocity was found to be 48.23%.

Potgieter et al. [19] combined the parallel and counter flow passage for air circulation through the solar air heater by incorporating baffles and hence studied its hybridized model in computational fluid dynamics. The results obtained were compared with the experimental values and came to conclusion that hybrid approach in SAH improves overall efficiency of SAH. The mean error between the results predicted by CFD analysis and experimental results was found to lie near 9.165K.
Dezan et al.[20], made the use of longitudinal vortex flow generator which gives better mixing of cold and warm fluid and performed its CFD analysis. High channel performance was found to be achieved when vortex generators are not periodically distributed along its length.

2. Numerical Analysis
The numerical analysis of triangular solar air heater is carried out by computational fluid dynamics based on the ANSYS 14.5 software. The computational model is developed in the Solidworks software and the mesh is generated in A fluid flow (fluent)-meshing [Ansys CFD]. The flow governing equations are solved using Fluid Flow fluent to investigate the outlet temperature of the triangular passage. This parameter is used for the performance analysis of TSAH.

The schematic diagram of a triangular heater is shown in Figure 1 below. The triangular duct has a one side of 60 cm, height 52 cm and apex angle of 60. Based on the hydraulic diameter of duct, the value of the Reynolds number varies 2263 to 18103. The glazing plate of the triangular section is heated with the constant heat flux of 746 w/m².[21] The computational analysis is performed using the following assumption is given below:
- There is no slip condition at the wall i.e. at the wall fluid velocity gradient is zero.
- Fluid properties are constant at the inlet section of the TSAH.
- Flow inside the duct is fully turbulent, steady and incompressible.
- The working fluid has constant properties throughout the length of the section.

![Diagram of a triangular heater Passage](image)

**Figure 1: Diagram of a triangular heater Passage**

| S.No. | Physical Properties             | Value             |
|-------|---------------------------------|-------------------|
| 1     | Thermal Conductivity            | 0.0242 W m⁻¹K⁻¹  |
| 2     | Constant pressure Specific heat  | 1006 J kg⁻¹ K⁻¹   |
| 3     | Constant volume Specific heat    | 718 J kg⁻¹ K⁻¹    |
| 4     | Density                         | 1.2256 kg m⁻³     |
| 5     | Viscosity                       | 1.7894x 10⁻⁵ Nm²  |
| 6     | Gas constant                    | 287 J kg⁻¹ K⁻¹    |

**Governing equations**
The flow of fluid is described by the continuity, momentum and energy equations. The ANSYS 14.5 fluid flow fluent implements finite element method for solving the PDEs. These can be expressed as,
Equation of Continuity,
\[ \frac{\partial (\rho u_j)}{\partial x_j} = 0 \] 2.1

Equation of momentum,
\[ \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] \] 2.2

Equation of energy,
\[ \frac{\partial (\rho u_i T)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \Gamma + \Gamma_t \right) \frac{\partial T}{\partial x_j} \right] = 0 \] 2.3

The flow inside the triangular section is turbulent, so k-\( \varepsilon \) model is used for the analysis. This model is based on the Navier-Stokes equations, averaged by Reynolds. The transport equations are used in the RNG, the k-\( \varepsilon \) model is shown below.
\[ \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \nu_{eff} \frac{\partial k}{\partial x_j} \right) + G_k - \varepsilon_p \] 2.4

Mesh generation
The 3-D model of the TSAH passage is taken into consideration in the present CFD analysis. In order to better fit into the cross section of the duct, the triangular grids are generated within the computational domain. To examine the heat transfer in the inside region and flow behaviour of the fluid fine mesh have been formed at the inlet and outlet section and also near the wall in the test section. Constant mesh size with number of nodes and number of element as 17662 and 92571 respectively is taken. Relevance is taken as 100 while fine sizing is selected.

Boundary Conditions
The pressure and velocity boundary conditions of the inlet and outlet section are used in the computational domain. In order to calculate the mean inlet velocity of the fluid flow, the Reynolds number is used. Air enters in the duct from the inlet section with ambient condition and it has inlet pressure and temperature of 1bar and 300K respectively. At the glazing plate side of the duct with the heat flux intensity of 746 w/m², the constant heat flux boundary condition is maintained. This solar
intensity is calculated with the help solar calculator by using University of GLA Mathura (Latitude and longitude coordinates are 27.49, 77.67). The no-slip boundary condition between the absorber plate and the fluid flow through the passage of the duct is applied.

3. Data Reduction:

The glass plate loss coefficients $U_t$ is calculated by the formula [22],[23]

$$U_t = \left( \frac{N}{N_c} \left( \frac{T_p - T_a}{(N+f)} \right) \right)^{-1} + \frac{\sigma(T_p - T_a)(T_p^2 + T_a^2)}{1 + \frac{2N+f}{1+e(0.00591 Nh_a)} + \frac{e}{e_p}}$$

$$f = (1 + 0.089h_a - 0.01166h_a e_p)(1 + 0.07866N)$$

$$e = 0.43 \left( 1 - \frac{100}{T_p} \right)$$

$$C = 520[1 - (0.000051\beta^2)] \text{ For } 0^0 < \beta < 70^0$$

The absorber plate and edge loss of heat coefficients are fixed for a particular setup and is given by mathematical relation,[24]

$$U_b = \frac{k_b}{X_b}$$

$$U_e = U_b \left( \frac{A_e}{A_c} \right)$$

The overall heat coefficients $U_L$ is the sum of glass plate loss, absorber plate and edge loss coefficients,

$$U_L = U_t + U_b + U_e$$

The heat transfer from convection between the ambient and glass cover was calculated by the correlation [25]

$$h_a = 5.7 + 3.8V$$

$$\dot{m} = \rho_o A_d V$$

$$R_e = \frac{\rho_o V d}{\mu}$$

$$P_r = \dot{m} C_{pair} k_a$$

Nusselt number can be determined by equation[26],

$$N_u = \frac{3.66 + 0.0668(D_h/L)R_e P_r}{1 + 0.04[(D_h/L)R_e P_r]^{2/3}}$$

The convective heat transfer coefficient by using the correlation of Nusselt number is determined by,

$$h = \frac{N_u k_a}{D_h}$$

$$F' = \frac{h}{h + U_L}$$
Heat removal factor can be determined from by using the equation,[27]

\[ F_R = \frac{\dot{m}C_{\text{pair}}}{A_cU_L} \left[ 1 - \exp \left( \frac{-F_UA_p}{\dot{m}C_{\text{pair}}} \right) \right] \quad 3.16 \]

The useful heat rate \( Q_u \) determined for the triangular solar air heater can be presented in form of heat energy absorbed by the system and energy lost by the system,[28]

\[ Q_u = A_p F_R \left[ I(\tau \alpha) - U_L(T_{\text{pm}} - T_i) \right] \quad 3.17 \]

The first law efficiency of system is given by the ratio of heat gain to input heat energy,

\[ \eta_{\text{thermal}} = \frac{Q_u}{IA_p} \quad 3.18 \]

4. Results and Discussion

**Grid Independence Study**

To obtain a solution independent of grid number and sizes, the effect of grid size on the friction factor and Nusselt number is ordered. The friction factor and Nusselt number is determined for three set of different grid model including fine, medium and course. The cell number is varied from the course, medium and fine mesh and it is observed that the percentage difference in friction factor and Nusselt number after fine mesh is less than other type of mesh. Therefore, the simulation is done by fine mesh for all instances.

The figure below illustrates the total temperature contour of the TSAH. We noticed that there are regions with the temperature difference at the inlet and outlet. At the constant solar intensity 746 w/m² and (Re=4526 and Ti=300 K) as represented in Figure 2.
Figure 2: The contour of total temperature variation in TSAH

Figure 3 describe the distribution of velocity inside the triangular passage. The velocity variation is shown by the streamline pattern. We see a high vortex is generated at the middle area and a small vertex at the outlet section side. These eddies are responsible for the energy losses and the destruction near the wall of the absorber and glass plate of the laminar sub layer. It therefore produces heat transfer enhancement.
Figure 3: Velocity distribution by stream lines

Figure 4 determines the variation in efficiency of solar air heater with respect to the velocity of air. The thermal efficiency value is increases with the increase of the velocity of air. The values of thermal efficiency for triangular passage are 45.4%, 54.6%, 59.21%, 62.2%, 64.21%, 65.45%, 66.67% and 67.23%. For each observation set, the constant solar intensity of $I=746 \text{ w/m}^2$ and $T_i=300 \text{ K}$ inlet temperature are taken. Thus, thermal energy is transferred to air by increasing the effect of turbulence in the test section at higher mass flow rates.

Figure 4: Variation of Thermal efficiency and velocity of air
From Figure 5 gives the variation of different type of temperature and the mass flow rate of air. If mass flow rate value is increase it increases the absorber and glass plate temperature but the outlet section temperature is decreases. The inlet section temperature is constant throughout the process. Due to the absence of any obstruction within TSAH passage, the exposure time between the air and the absorber plate surface is lower due to less air contact, thereby decreasing heat transfer rate. Finally, the outlet temperature is decreases.

![Figure 5: Variation of temperature and mass flow rate of air](image)

In Figure 6 shows the variation of friction factor and Nusselt number with Reynolds Number. We observe in the graph the value of Reynolds number increases; it also decreases the friction factor and increases the Nusselt number. If Reynolds number increases, it produces air turbulence inside the triangular passage, which improves heat transfer rate. The friction factor is inversely proportional to the Reynolds number, thereby reducing the friction factor.
Figure 6: Variation of friction factor and Nusselt number vs Reynolds Number

The Figure 7, illustrate the contour of the total pressure inside the triangular passage and inlet and outlet section. We observed the pressure of the outlet section is less compare to the inlet section.

Figure 7: The contour of total pressure at inlet and outlet

5. Conclusions
In this paper we presented the results obtained from the CFD analysis of the triangular passage at constant solar intensity $I=746 \text{ w/m}^2$ and inlet section temperature $T_i=300\text{K}$. The study was based on
fluent Numerical simulation using the ANSYS 14.5 code. Based on the CFD analysis, the following are notable conclusions drawn:

- The efficiency of the TSAH is increases with respect to air velocity at the inlet.
- Reynolds number and Nusselt number of air increase with respect to mass flow rate and the friction factor is observed to be affected by the variation of the number of Reynolds.
- Absorber and glass plate temperature is increases with increases the mass flow rate of air.
- CFD review reveals analytical results and the scope of thermal efficiency enhancement using TSAH.
- CFD analysis provided a 12 K temperature difference between the inlet and outlet section.

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