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NUMERICAL ANLYSIS AND VALIDATION OF HEAT TRANSFER MECHANISM OF FLAT PLATE COLLECTORS–TO TAP SOLAR ENERGY

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Abstract—*This paper presents the numerical analysis to investigate the possibilities for enhancement of thermal performance of solar flat plate collector experimentally. Effective heat energy transfer to the working fluid is analytically estimated by considering the influence of design and operating parameters such as bond conductance between absorber plate and tube, tube diameter, air gap, optical properties of glass material, number of glass covers and flow rate of working fluid are validated against numerical simulation. For study the heat transfer mechanism inside the solar flat plate collector a 3D model has been developed using ANSYS FLUENT-14 software package. The proposed model is able to predict the heat transfer rates, fluid temperature along the length of tubes, interface heat transfer with good accuracy. In this paper, an analytical study is performed on the entropy generation and heat transfer due to fluid flow in a flat plate solar collector. Water is taking as working fluid for the analysis. Entropy generation plays strategic role and also it alarms the effective heat transfer rate in collector in terms of work lost and exergy rates. Total entropy generation rate increases linearly with increase in mass flow rate and overall loss parameters but the rate of increase being higher for lower values of overall loss parameter and a decrease in fluid outlet temperature results in an appreciable increase in total entropy generation rate. This paper theoretically analyzes the heat transfer enhancement capabilities in a solar flat plate collector at steady laminar axial flow at different heat fluxes and also at different tilt angles.*

Keywords: *Flat plate collector, thermal performance, ANSYS simulation, entropy generation.*

1. INTRODUCTION

Nowadays science and technology have dramatically improved and hence requirement for energy is rushed to meet the demand. On the other hand the bearing in mind the breeding consumption of formal primary energy sources like coal, Oil natural gas and environment related concerns are demands the improving of the low temperature heat resources and its development is an expected choice to resolve problems associated with energy and environment.

Solar energy is an essentially inexhaustible source potentially capable of meet the significant portion of the world's future energy needs. Solar energy is the most promising unconventional energy sources. The solar energy collection has been the primary interests of many engineers and researchers for the last few decades due to its wide applications such as domestic water heating and commercial appliances. Solar energy collectors are special kind of heat exchangers that transform solar radiation energy to internal energy of the transport medium.Thermal performance of solar collector is considerably low due to the low value of convective heat transfer coefficient between the absorber plate and heat transfer medium. The conversion of low grade energy with improving effectiveness of these solar collectors has been carried out by many researchers for years. Mohammed Slemi et. al. [5] carried out the experiments with water as working fluid and by changing the glazing material. In current ages, some researchers have given additional concentration by the use of nano particles within the working fluids (nanofluids) to enhance the heat transfer, therefore increasing the thermal performance of the collector. In this context, Lu et al. [1] Examined the thermal performance of water based CuO nanofluid as the functioning liquid and found the mass concentration of nanofluid had amazing effects on heat transfer coefficient. Yousefi et al [2] consider the Al_2O_3 water nanofluid as absorbing fluid and found that the efficiency is improved by 28.3%with 0.2wt% nano particles. Natarajan and Sathish [8] investigated the enhancement of thermal conductivity with carbon nano tube (CNT).

The major component of any solar system is the solar collector. Among all the solar thermal collectors, the flat plate collectors though produce lower temperatures, have the advantage of being simpler in design, having lower maintenance and lower cost.

Nomenclature

A_c – Collector Area (m^2)
E – Exergy rate (W)
f – Friction factor
G_t – Total Solar radiation (W/ m^2)
h_w – wind heat transfer coefficient (W/ m^2K) (W/ m^2K)
k – Thermal conductivity (W/ m K)
L - Length of the collector (m)
N_g – Number of glass covers
Q_u - Heat absorbed by the working fluid (W)
S – Entropy (W/ m^2)
T – Temperature (K)
U_L – Overall heat loss coefficient (W/ m^2K)
U_b – Bottom heat loss coefficient (W/ m^2K)
U_e – Edge heat loss coefficient (W/ m^2K)
U_t – Top heat loss coefficient (W/ m^2K)

Greek Letters

ρ – Density (kg/m^3)
ε – Emissivity (W/K^4)
τ – Transmissivity
σ – Stefan Boltzman constant
μ – Viscosity (kg/ms)

Subscripts

a - Ambient
b – bottom
e – edge
g – Glass
i - inlet
P- Plate
t – Top

Dimensionless Numbers

Pr - Prandtl Number
Nu–Nusselt Number

Its performance can predict with different orientations on different days and can comfortably simulate it with good accuracy. Selmi et al. [5] performed a CFD simulation of flat plate solar energy collector with water flow. The flow domain consists of a flat plate absorber plate with circular absorber tube connected below the absorber. Water is supplied at the inlet of the absorber tube.

The following assumptions are made in the analysis [14].

- Water is a continuous medium and incompressible.
- The flow in solar collector is fully developed flow and steady state and posse's laminar flow characteristics.
- The contribution of the size of headers in the collector surfaces are neglected
- The fluid flows in all risers are uniform.
- Heat flow through a cover is one dimensional and temperature drop across the cover in neglected.
- The thermal-physical properties of the absorber plate, water and the bsorber tube are independent of temperature. Loss through front and back are to the same ambient temperature.

2. METHODOLOGY

A. Collector configurations for the analysis

The project model consists of a collector with 915mm long, 810mm wide and 95 mm and it is covered with transparent borosilicate glass. The collector contains a copper plate used as an absorber with nichrome black coating with high absorbivity and low emissivity. Absorber plate length and width for all the design is taken as 0.84m and 0.8m respectively. The absorber tube diameter is 0.0127m with a thickness 5.6X10^{-4}m with 0.115m central distance. The diametrically placed planar is placed all along the length of absorber tube and has a thickness of 0.00012m.

B. Experimental method: Experiments are conducted on test rig with water as working fluid for evaluating heat transfer rate and entropy generation. The copper material used for both absorber plate and the water tube. Borosicate glass is taken as the glazing material and air gap is taken into account at the atmospheric pressure. The efficiencies can be attributed to the use of low iron, tempered glass for glazing (low-iron glass allows the transmission of more solar energy than conventional glass), improved insulation, and the development of durable selective coatings. A number of thermocouples are attached to the absorber plate, pipe at some selected points, and outside of the solar collector, to measure the collector inside, ambient, water inlet and outlet temperatures. The model pipe is connected to water source through manual control valve at one end, and a drain hose at the other. The water flow is controlled by means of a manual control valve.

The total radiation fall on the collector plate is not aloneuseful to rises the temperature of medium which useful further use but some energy is losses through various portions like heat lost from the top, bottom and edges of the collector system. It is evaluated in terms of heat loss coefficient and hence calculates the collector efficiency factor. And also evaluate the effects of device tilt angle, amount of radiation and mass flow rate on Nusselt number and entropy generation. For analysis the ambient temperature, wind velocity transmissivity, emissivity of absorber plate and wind heat transfer coefficient are considered.

C. Thermal analysis

The solar radiation after passing the glass cover, the major amount of radiation is absorbed by the working fluid (Qu), but a portion of it, dissipates from top, bottom, and edges to the ambience with temperature of Ta. Then, the heat gain by the working fluid is equal to

$$Q_u = A_c[\eta_0 G_t - U_L(T_p - T_a)] \quad (1)$$

Where η_0 would be the optical efficiency of the cover and it is assumed to be 0.84. U_L, Overall heat loss coefficient, sum of top, bottom and edge loss coefficients.

$$U_L = U_t + U_b + U_e \quad (2)$$

in which U_t is the heat loss coefficient from the top, U_b is the heat loss coefficient from the bottom, and U_e is the heat loss coefficient from the edges of collector.
To obtain U_t, can use the following correlation

$$U_t = \frac{1}{\dfrac{N_g}{\dfrac{C}{T_p}\left[\dfrac{T_p - T_a}{N_g + \tau}\right]^{0.33} + \dfrac{1}{h_w}}} + \frac{\sigma\left(T_p^2 - T_a^2\right)\left(T_p - T_a\right)}{\dfrac{1}{\varepsilon_p + 0.05N_g\left(1 - \varepsilon_p\right)} + \dfrac{2N_g + \tau - 1}{\varepsilon_g} - N_g}$$

Where N_g is the number of glass covers, σ is Stefan–Boltzmann constant, ε_p is the emissivity of the plate, ε_g is the emissivity of the glass and h_w is wind heat transfer coefficient and it reads as

$$h_w = \frac{8.6 v_w^{0.6}}{L^{0.4}} \quad (4)$$

Where v_w is the wind velocity and L is the length of collector. It should be noted that the heat transfer coefficient of still air is about 5 W/m^2 K, therefore if the value of h_w becomes less than this value, the value of h_w is assumed to be 5 W/m^2[6]

Values of solar irradiance on collector, ambient temperature, and wind velocity for 10:00 AM and 1:00 PM in a typical day for Warangal on December 21(*Winter Solstice*) are as follows

Table 1: Solar Irradiation on winter Solstice day of Northern Hemisphere at NIT Warangal

Time	Solar Irradiance (W/m²)	Ambient Temp (K)	Wind velocity (m/s)
10:00AM	803.4	295	1.25
1.00PM	951.2	299	1.4

The constants τ and C are determined by:

$$\tau = \left(1 - 0.04 h_w + 0.0005 h_w^2\right)\left(1 + 0.091 N_g\right) \quad (5)$$

$$C = 365.9\left(1 - 0.00883\beta + 0.0001298\beta^2\right) \quad (6)$$

In which β is the collector slope in degrees. The major amount of heat loss always from the top only for the collectors. Specially, for the flat plate collectors' heat receiving area and lost area are almost same as top loss. Neverthless the heat loss from the bottom and edges are calculated by the following relations:

Heat loss coefficient from bottom:

$$U_b = \frac{1}{\frac{t_b}{k_b} + \frac{1}{h_{ba}}} \quad (7)$$

and heat loss coefficient from edges:

$$U_e = \frac{1}{\frac{t_e}{k_e} + \frac{1}{h_{ba}}} \frac{A_e}{A_c} \quad (8)$$

Where t_b and t_e are the thicknesses of insulators in bottom and edges, respectively, k_b and k_e are the thermal conductivities of insulators in bottom and edges, respectively, and A_e is the surface area of edges. In addition, h_{ba} and h_{ea} are convection heat transfer coefficients in the bottom and edges, and their values are assumed the same and equal to the heat transfer coefficient of still air.

Nusselt number can be calculated by the Gnielinski correlation Nusselt number,

$$Nu = \frac{(f/8)(\mathrm{Re} - 1000)\,\mathrm{Pr}}{1 + 12.7(f/8)^{0.5}(\mathrm{Pr}^{0.66} - 1)}$$

$$f = (0.79\ln(\mathrm{Re}) - 1.64)^{-2}$$

This correlation is valid for a wide range of Reynolds number and Prandtl number, i.e. 3000 $\leq$ Re $\leq$ 5X10^5 and 0:5 $\leq$ Pr $\leq$ 2000 where f is the Darcy friction factor and usually it is estimated for smooth tubes through Petukhov correlation as follows

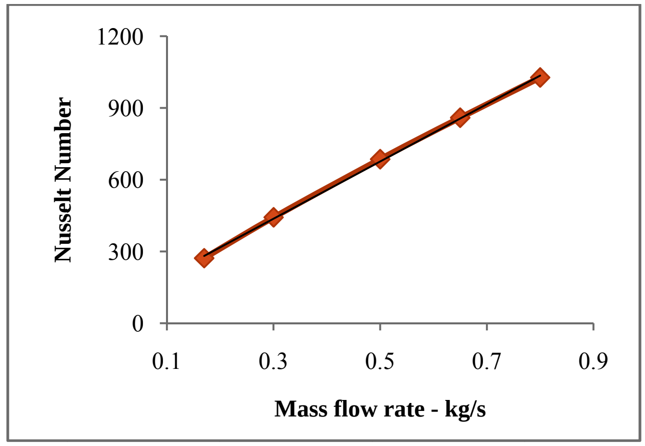

Fig. 1: Variation of Nusselt Number Vs Mass flow rate

D. Numerical scheme: To validate the experimental results of the flat plate collector, numerical simulation is carried out using ANSYS FLUENT-14. The collector is modelled using geometric modelling module as shown in fig.2 and fig.3. The computational domain is meshed into 5,79,552 elements as in fig.4. In pre-processing, the simulation was carried out using steady state implicit pressure based absolute velocity solver. The governing partial differential equations, for mass and momentum are solved for the steady incompressible flow. The velocity-pressure coupling has been effected through SIMPLE least squared cell based gradient for discretization.

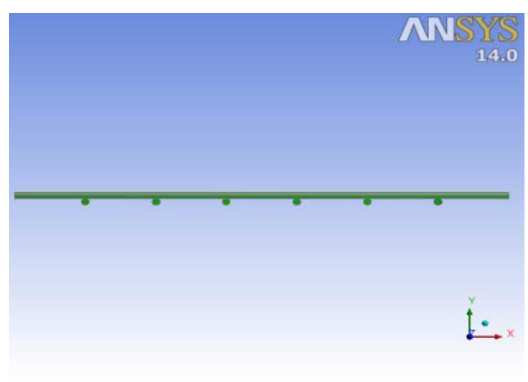

Fig. 2: Geometric model of the collector

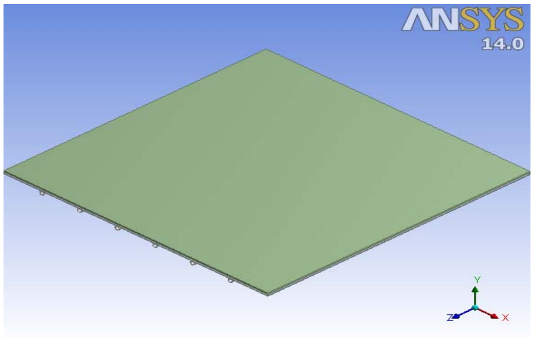

Fig. 3: Geometric model of experimental test collector with six tubes.

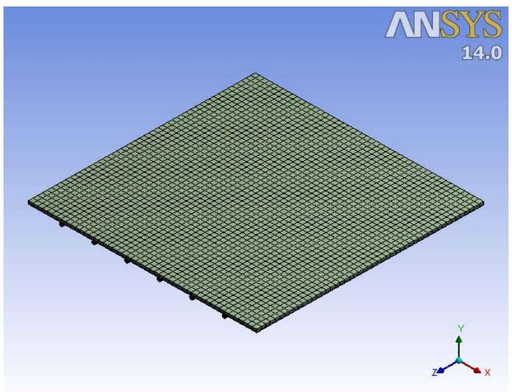

Fig. 4: Mesh model of the solar collector

First order upwind schemes were chosen for the solution. Laminar flow condition was used. The boundaries should have both convection and radiation heat transfer mechanisms

E. Boundary conditions and Operating parameters

Appropriate boundary conditions were impressed on the computational domain, as per the physics of the problem. Inlet boundary condition was specified as velocity inlet condition. Outflow boundary condition was applied at the outlet. Wall boundary conditions were used to bound fluid and solid regions. In viscous flow models, at the wall, velocity components were set to zero in accordance with the No-slip and impermeability conditions that exist there. The interface between the glass-air, air-absorber and water with the absorber tube is defined as wall with coupled condition to effect conjugate heat transfer from absorber tube to the water. The modes of mixed convection and radiation heat transfer absorber plate and glass cover. A constant heat flux equivalent to the solar insolation is applied at the top surface of the absorber plate. The mixed convective heat transfer in the circulating water inside the tube and conduction between the base and tube material. The bottom and side surfaces of the absorber plate and the outer surface of the absorber tube are defined as wall with zero heat flux condition to effect insulated conditions. Convergence was effected when all the residuals fell below 1.0e-6 in the computational domain.

3. RESULTS AND DISCUSSION

In the analytical formulation the experiments are conducted with water as working fluid. Experiments are carried at different radiation levels of the same day and also at different days. Different orientations of collector and at different wind speeds effects are considered. From the experimentation it shows that almost all values are linearly varying with respect to mass flow rate.

In the present work the solution is carried out with conjugate heat transfer with coupled interfaces in the pipes and fluid, and the natural convection flow is confined air in the cavity between the absorber and the glass cover as in Fig. 6. It is observed that there is a decrement in the plate temperature as shown in Fig. 5. with a consequent increase in the mass flow rate for the condition of a simulated constant solar heat flux of 803.4W/m^2. This is because, as the mass flow rate increases, the heat is carried away from the plate due to convection at a faster rate within the tube.

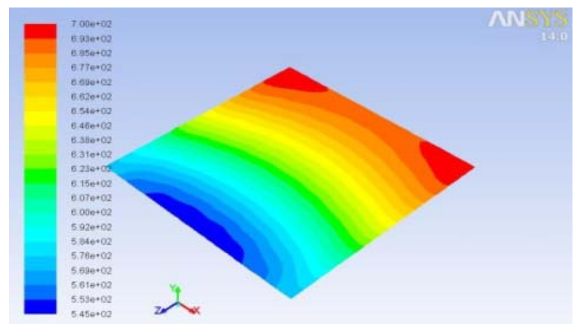

Fig. 5: Contours of static temperatures at Top surface (Glass Top)

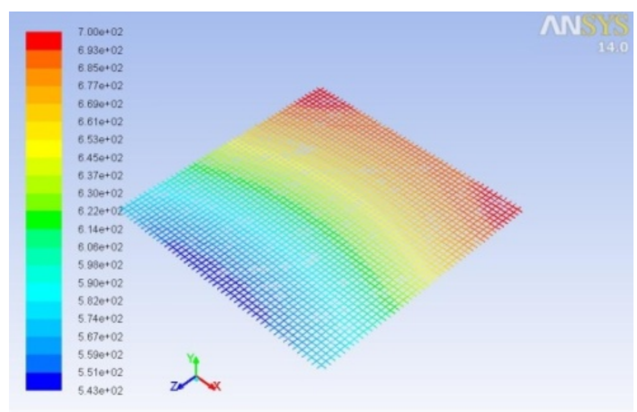

Fig. 6: Contours of static temperature at Glass air interface.

Above is corroborated by the temperature gradient plots for the water passing through the absorber tubes. The static temperature distribution is as shown in Fig. 7 that there is a gradual increase in the temperature of water along the length of the tube.

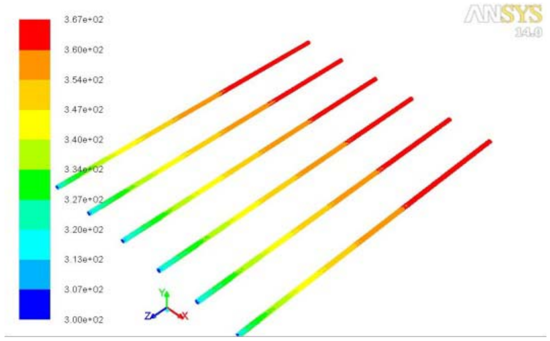

Fig. 7: Temperature profiles of water after converged

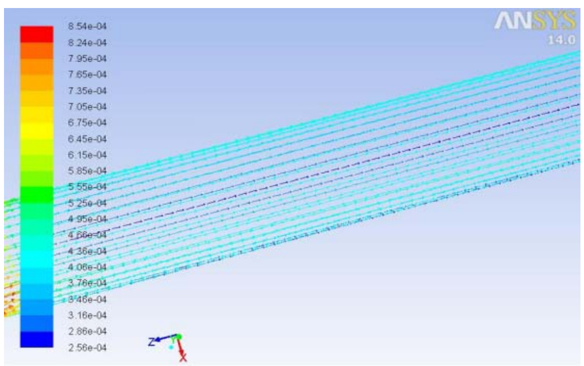

Fig. 8: Velocity vectors in single tube after converged

As explained earlier, the temperature increase is due to the heat transfer at the absorber plate all along its length due to conjugate heat transfer.

This can be explained by the fact that as the velocity is increased due to increase in the mass flow rate, the fluid particles are carried forward at a faster rate thereby a perceptible lower temperature rise is observed for the water. In the foregoing analysis, different mass flow rates as shown in the graph 9.

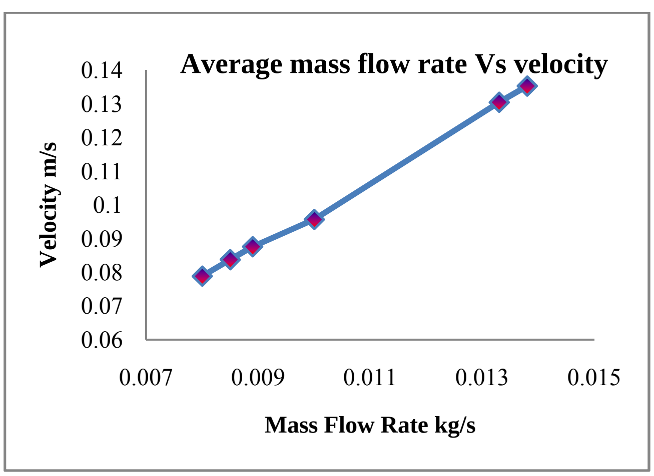

Fig. 9: Velocity variation Vs Mass flow rate

4. CONCLUSION

Solar energy is one of the alternative for conventional sources. My present work is aims to enhance the efficiency and performance of the solar thermal systems by using water as working medium. There is a good agreement between the experimental and simulated results for outlet water temperatures. The analysis of convection heat transfer mechanism inside the pipes shows a higher Nusselt number in the fully developed region for the fluid. The buoyancy effects in conjugate with heat flux are accurately implemented in the process of absorption-transmission of solar load.

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