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Effect of selective coating on the top heat loss characteristics of trapezoidal cavity: a computational approach

Subhankar Ray[1, 3], Arun K Tripathy[1], Sudhansu S Sahoo[1] and Swarup K Mahapatra[2]

[1]Dept. of Mechanical Engineering, College of Engineering & Technology, Bhubaneswar, Odisha, India.
[2]School of Mechanical Sciences, Indian Institute of Technology, Bhubaneswar, Odisha, India.

Email: [3]ray02.subhankar@gmail.com

Abstract. In the present work the effect of selective coating on the top heat loss characteristics of a trapezoidal cavity has been presented. Selective coatings are multi-layered coatings which enhances absorption of radiation on a surface while inhibiting emission from it. In the present study selective coating is applied on the bottom absorber plate of the trapezoidal cavity. Computational domain of the trapezoidal cavity is developed and analysed. Finite Volume Method (FVM) is employed to calculate natural convection in the domain, radiation from the bottom absorber plate to glass cover, wind induced forced convection on the glass cover and radiation from glass cover to ambient. Parametric study has been carried out by varying the input heat flux, the absorptivity of the selective coating and emissivity of the selective coating.

1. Introduction

Rectangular and trapezoidal cavities are widely employed as receivers for solar thermal applications. The inside wall of the solar cavity receivers are panted black to improve absorption of solar rays. Insulation is provided on the side and underneath the absorber plate to minimize heat losses. Heat flux travelling through the glass cover strikes the aluminum absorber plate which gets heated up. Trapezoidal cavities are generally found in solar box cookers. Current investigation focuses on studying the effects of coating the absorber plate with selective coating, on the heat loss from the glass cover of the trapezoidal cavity and the temperature attained by the absorber plate and glass cover.

Heat loss from the trapezoidal cavity has been presented by several authors [1-4]. Top heat loss factor (U_t) for a double glazed box type solar cooker was found out for different wind velocities and ambient temperature through indoor experiments [1]. A correlation was developed between the above parameters with the heat loss factors between the absorber plate and inner glass cover, in between the two glass covers/ glazing and from the outer glass cover to the ambient [1]. Commercially available box type solar cooker was used for the above experiment. Natural convection heat transfer in trapezoidal cavity of a box type solar cooker is presented [2]. Experimental approach has been followed to obtain the temperatures of the absorber plate, inner and outer glass cover. A simplified correlation too was developed [2] to relate the external parameters with heat loss. It was observed that correlations developed for rectangular enclosures cannot be used for trapezoidal enclosures as they under predict the heat loss. Analytical modeling methodology for heat loss characteristics of box type solar cooker was presented [3]. Parametric study was carried out by varying the absorber plate

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emissivity, absorber plate temperature and wind velocity over the outer glass cover. Correlation for wind induced heat losses from outer cover of solar collectors has been developed [4]. Indoor experiments were carried out by blowing air over horizontal flat surfaces. Solar radiation flux on the absorber surface was considered uniform and the solar flux was modelled as heat flux wall boundary condition [5]. Similar conditioning is used in this investigation.

CFD simulation of trapezoidal cavity is a rarity among the literature surveyed. Selective coating which enhances the solar flux absorption on a surface while reducing the heat loss by radiation from the surface, thus enhancing the absorber plate temperature for the same input heat flux has not been studied till now.

2. Mathematical Modelling

2.1. Computational Domain

In the present investigation, a trapezoidal cavity of a solar receiver has been modelled. The outer casing and the side and bottom insulation of the receiver has not been modelled. For the simplicity of calculations, single glazing/ glass cover has been modelled, further inclined solid side walls have also not been modelled. The dimensions of the domain have been kept similar to that presented in [3] with the exception of glass cover whose dimensions have been kept similar to the upper surface of the trapezoidal cavity. Dimensions are presented in Table 1. Schematic diagram of the computational domain has been presented in Figure 1.

Table 1: Physical dimensions of the domain.

Name	Dimension	Thickness	Material
Absorber Plate	0.366 m x 0.366 m	0.002 m	Aluminum
Air Domain	0.366 m x 0.366 m (bottom) 0.475 m x 0.475 m (top)	0.0944 m	Air
Glass Cover	0.475 m x 0.475 m	0.004 m	Glass

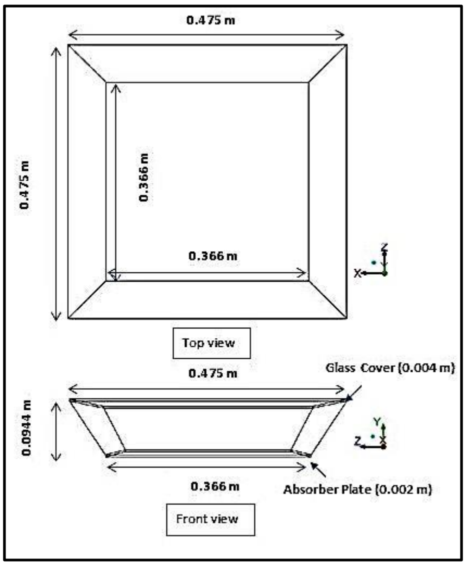

Figure 1. Schematic diagram of the computational domain.

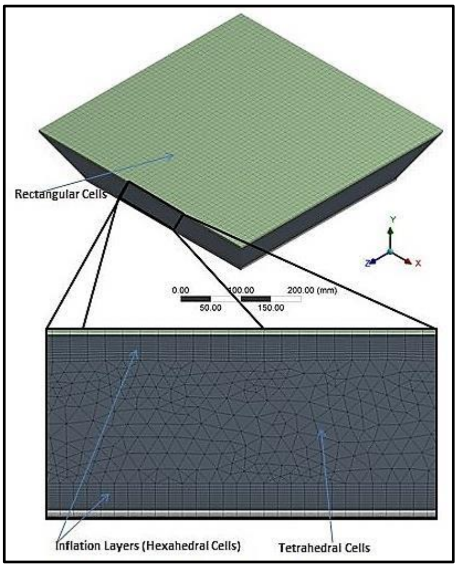

Figure 2. Meshing scheme followed for the domain.

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2.2. Meshing
Hybrid meshing scheme has been adopted for the computational domain. Absorber plate and glass cover has been meshed with hexahedral cells. Solid surfaces have been meshed with rectangular cells. Trapezoidal air domain has been meshed with tetrahedral cells. Inflation layer of 13 cell width comprising of hexahedral cells have been added at solid fluid interface zone (near absorber plate and glass cover) to capture the heat transfer from the absorber plate and to the glass cover well. Advance size function, particularly proximity and curvature has been used for mesh refinement and controlling the number of cells. Mesh generated has been presented in Figure 2.

2.3. Assumptions
The following assumptions have been considered.

a. Empty trapezoidal cavity has been used for the simulations.
b. Air is considered Newtonian and air properties are taken to be piecewise linear function of temperature only.
c. No slip boundary condition at the walls has been used.
d. Radiative heat losses from the side walls of the trapezoidal domain have been neglected.
e. Heat flux is applied normal to the absorber plate.
f. Input heat flux on side walls has been neglected.
g. Selective coating has been applied only on the surface of the bottom absorber plate.

2.4. Material properties
Table 2 lists the properties of the materials used. Temperature dependent properties of air have been sourced from [6].

Table 2: List of material properties used.

Material	Density (kg/m^3)	Specific Heat (j/kg-K)	Thermal Conductivity (W/m-K)	Viscosity (kg/m-s)	Refractive Index
Aluminium	2719	871	202.4	-	1
Glass	2225	835	1.1	-	1.478
Air	*f*(t)	*f*(t)	*f*(t)	*f*(t)	1

2.5. Boundary conditions & solver inputs
Conservation equations for mass, momentum and energy are solved iteratively with appropriate boundary conditions using ANSYS® Fluent®. Details of equations are omitted for the sake of brevity. Following boundary conditions has been used. Heat flux is applied directly on the absorber plate surface after multiplying the factors for glass cover transmissivity and absorptivity of the surface, thus a flux boundary condition is used. Emissivity of the surface has been defined by the type of coating used. Side walls have been defined with emissivity value zero, thus no radiative heat exchange takes place from it. Outer surface of glass cover has been defined as mixed boundary condition, wherein both convective and radiative losses take place from the glass surface. Emissivity of glass is taken as 0.9 and ambient temperature (T_a) as 308 K.

Steady state pressure based solver has been used. Turbulence has been modelled with RNG k-ϵ with scalable wall functions. Radiation has been modelled with Discrete-Ordinate method. SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithm has been used for pressure-velocity coupling. Discretization of momentum, turbulent kinetic energy, turbulent dissipation rate and energy has been done with power law formulation. For pressure body force weighted is used while discrete ordinates equation has been discretized with 2nd order upwind scheme.

2.6. Grid independence test

A grid independence test has been conducted by varying the grid sizes. Grid has been varied from 147,041 to 245,130 elements. Grid independence test has been carried out for uncoated absorber plate with emissivity 0.3, receiving a flux of 600 W/m^2. A grid with 224,624 elements has been chosen for further study as the change observed in heat loss from glass cover was 0.00036% from the next higher grid. The findings of the grid independence test are presented in Figure 3.

2.7. Validation

Validation of the current numerical methodology has been carried out with the correlation developed from experiment conducted by [1] on the commercially available solar cooker. As the experimental model [1] has double glazing, while numerical has single glazing, validation is carried out by comparing the heat loss in the trapezoidal cavity, after fixing the temperature of absorber plate, side walls and inner glass similar for both numerical method (current study) and co-relation [1]. Validation has been carried out with emissivity of absorber plate and side walls at 0.9. The findings of the validation are presented in Figure 4. As the numerical results are in good agreement with that of the experimental one, further parametric study has been done on this computational model.

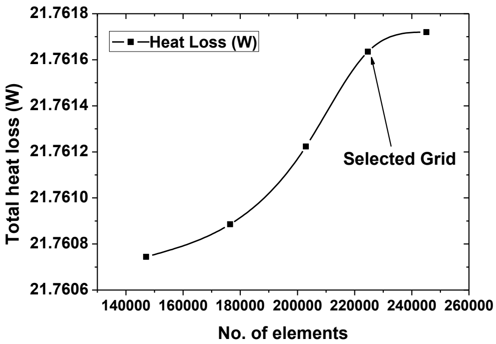

Figure 3. Grid independence test.

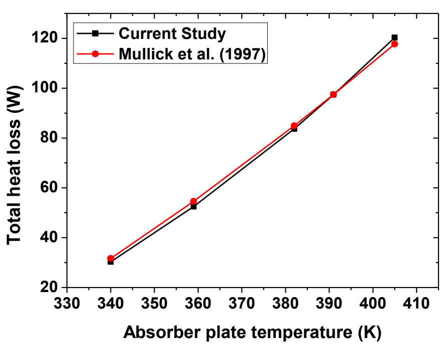

Figure 4. Validation of heat loss values with *Mullick et al*. (1997).

3. Results and Discussion

3.1. Comparison of heat loss from glass cover of trapezoidal cavity for non-selective coated and selective coated absorber plates.

Comparison of heat losses from glass cover top surface of a trapezoidal receiver with non-selective (α, ε=0.95) coated and selective coated absorber plate (α=0.95, ε=0.08) has been presented in figure 5. As can be seen, even though the emissivity is very low for selective coating, heat loss is higher for the fact that absorber plate temperature is very high (Figure 6) leading to greater convection losses from the absorber pate. This higher heat loss should not be viewed as a disadvantage as, for the same input solar heat flux, for the selective coated absorber plate the temperature attained is very high in comparison with uncoated or black coated absorber plates. Temperature performance of absorber plate for a particular input heat flux can be defined by a parameter (P_t) where,

$$P_t = \left[\left(\frac{T_p - T_a}{T_a}\right)\frac{1}{q_{loss}}\right] \times q_{in} \tag{1}$$

P_t is non-dimensional parameter which compares the performance of trapezoidal cavity for different coatings when input heat flux is kept fixed. Thus for selectively coated (α=0.95, ε=0.08) absorber plate P_t is 7.606 and for non-selectively (α, ε=0.95) coated absorber plate P_t is 2.357 for 600 W/m^2 input heat flux.

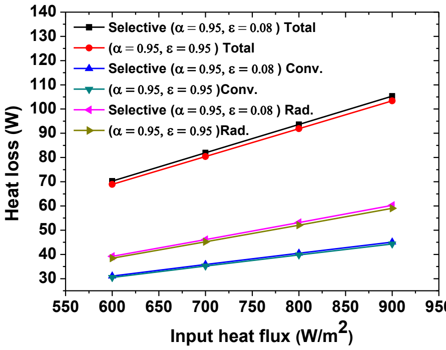

Figure 5. Comparative plot between total, convective, radiative heat loss from glass cover for absorber plate coated with selective and non-selective coating.

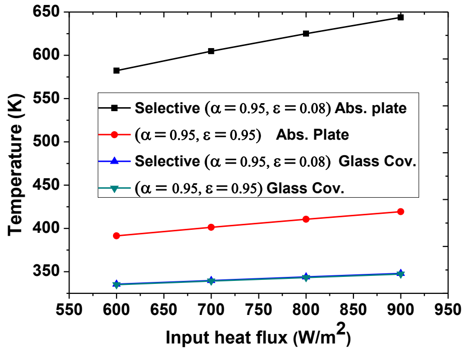

Figure 6. Comparative plot of absorber temperature and glass cover temperature for different input heat flux for absorber plate coated with selective and non-selective coating.

3.2 Effect of absorptivity of selective coating.

Effect of absorptivity of selective coating on the top heat loss and absorber plate and glass cover temperatures has been presented in Figure 7 and Figure 8 respectively. As the absorptivity of the selective coating reduces the top heat loss (total, convective, radiative) reduces, temperature of absorber plate also reduces. Glass cover temperature too reduces but marginally. This is observed for all input solar heat flux investigated. Thus, a selective coating with higher absorptivity value is desired even though heat losses increases for reasons discussed in section 3.1.

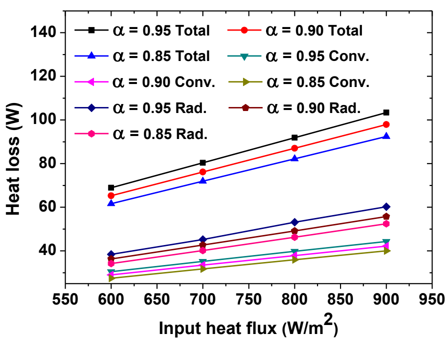

Figure 7. Plot between total, convective, radiative heat loss form glass cover for different input heat flux for selective coating varying α. (0.95, 0.9 and 0.85).

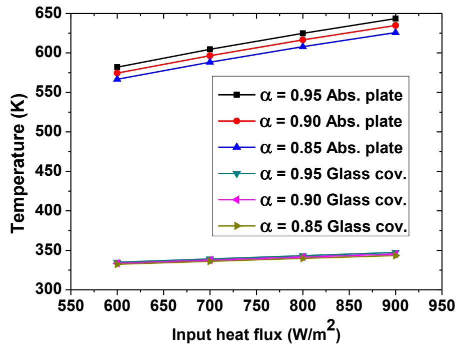

Figure 8. Plot between absorber temperature, glass cover temperature for different input heat flux for selective coating varying α. (0.95, 0.9 and 0.85).

3.3 Effect of emissivity of selective coating.

Effect of emissivity of selective coating on heat loss and absorber plate and glass cover temperature is presented in Figure 9 and Figure 10 respectively. As the emissivity of the selective coating increases, the absorber plate temperature decrease for the same input solar flux. The convective losses decrease due to decrease in temperature gradient between the absorber plate and glass cover. So the increase in heat loss due to increase in emissivity is balanced by decrease in convective losses in the trapezoidal cavity and the total heat loss remains constant as seen in Figure 9. The decrease in absorber plate temperature with increase in emissivity can be seen in Figure 10.

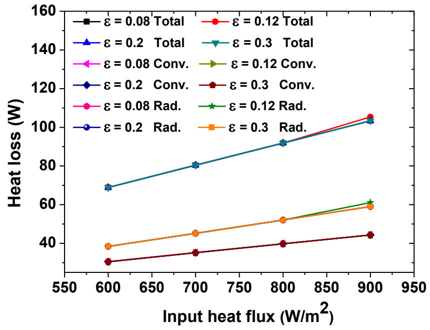

Figure 9. Plot between total, convective, radiative heat loss form glass cover for different input heat flux for selective coating varying ε. (0.08,0.12,0.2,0.3).

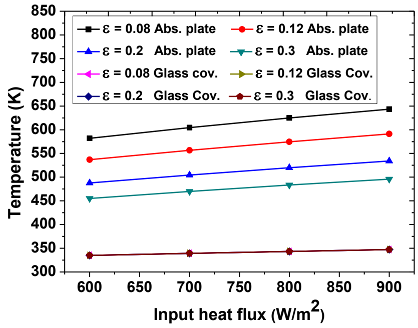

Figure 10. Plot between absorber temperature, glass cover temperature for different input heat flux for selective coating varying ε. (0.08,0.12,0.2,0.3).

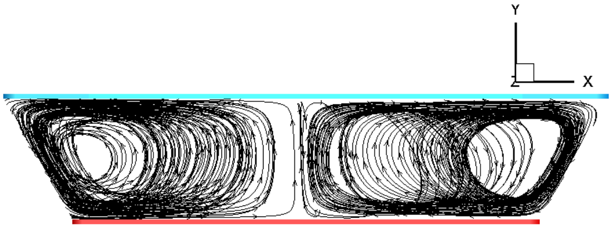

Figure 11. Streamlines showing the formation of natural convection 'cells' in the trapezoidal cavity for selective coated ($\alpha = 0.95, \varepsilon = 0.08$) cooker at z=0.09/L for 700 W/m^2 flux.

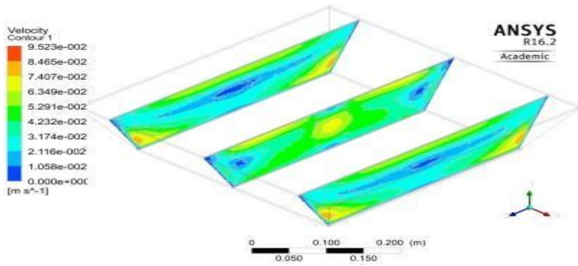

Figure 12. Velocity contour for selectively coated ($\alpha = 0.95, \varepsilon = 0.08$) cooker at front (x=0), mid (x=L/2) and end (x=L) planes for 700 W/m^2 flux.

4. Conclusion

Selective coating on the absorber plate of the trapezoidal cavity receiver has been presented in this paper. Selective coating increases the absorber pate temperature substantially, while the heat loss increases marginally. The temperature performance parameter (P_t) is 7.606 for selective coating and 2.357 for non-selective coating for 600 W/m^2 input heat flux. From the parametric study it can be concluded that a selective coating with higher absorptivity (α) and lower emissivity (ε) should be preferred. This work can be extended to study the improvement in the performance of box type solar cookers due to the application of selective coating.

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