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Estimation of heat losses from modified cavity mono-tube boiler receiver of solar parabolic dish for steam generation

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Abstract

In this article, a 3-D numerical investigation is carried out to estimate heat losses from solar parabolic dish with modified cavity receiver used for three different steam generation conditions viz. sub-cooled, saturated, superheated steam. The effect of inclination of the receiver, operating temperature, emissivity of the cavity cover, insulation thickness on the total heat loss from the modified cavity receiver has been investigated. The variable wall temperature boundary conditions and insulation thicknesses are applied to match the actual conditions. The results show that the convection heat losses are higher at 0° inclination and found to be 400 to 500 W for superheated steam generation; 300 to 425 W for saturated steam generation and 50 to 125W for sub-cooled steam. The radiation heat losses remain constant for all inclinations and vary with the temperature. Nusselt number correlations have been proposed based on the numerical analysis. The present model can be used to predict total heat losses from the modified cavity receiver under all conditions more accurately.

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Keywords: Solar parabolic dish, modified cavity receiver, heat loss estimation, Nusselt number correlation

1. Introduction

The utilization of solar energy for process heat and power generation could provide an attractive option for lesser dependency on fossil fuels. The solar thermal sources have a large amount of potential for process heat in 2050 [1]. The solar energy can be utilized in many ways, out of which concentrating solar

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Nomenclature

A	Heat transfer area (m^2)
D	Aperture diameter of the receiver (m)
d	Cavity diameter of the receiver (m)
f	Body force per unit volume (N/m^3)
Gr	Grashof number
g	Acceleration due to gravity (m/s^2)
h	Heat transfer coefficient ($W/m^2\,K$)
k	Thermal conductivity (W/m K)
Nu	Nusselt number
p	Pressure (N/m^2)
Q	Heat loss from the receiver (W)
S2S	Surface to surface
T	Temperature (K)
t	Thickness (m)
V	Velocity vectors (r, θ, z)
Greek symbols	
γ	Conductivity parameter
ε	Emissivity of the cavity cover
θ	Angle of inclination of the receiver (degrees)
ν	Kinematic viscosity of the fluid (m^2/s)
ρ	Density of the fluid (kg/m^3)
Subscripts	
conv	Convective
cu	Copper tubes
f	Fluid
h	Hot
ins	Insulation
ms	Mild steel cover
rad	Radiative
w	Receiver inner wall surface
∞	Ambient

collectors play a major role in production of high temperature heat. The solar parabolic dish collector is one of the most efficient energy conversion technologies of the four concentrated solar power technologies namely; linear Fresnel collector, parabolic trough collector, parabolic dish collector and solar power tower. The solar parabolic dish collector consists of a parabolic shaped reflector and a cavity receiver placed at focus to convert concentrated solar energy into thermal energy (Fig.1). The receiver converts the incident solar energy to thermal energy and this utilization of heat can be determined by studying the heat losses from the cavity receiver.

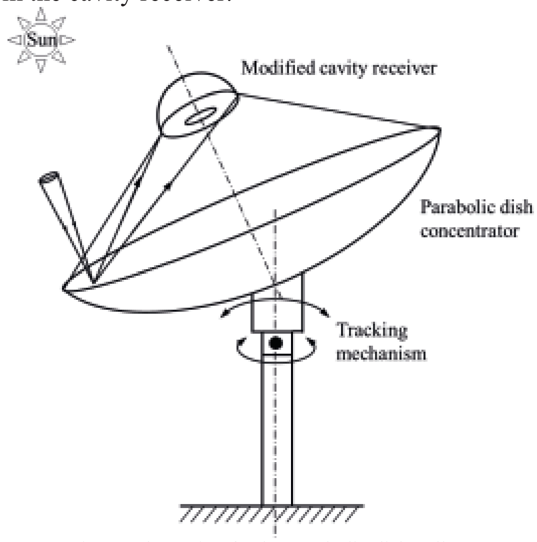


Fig. 1 Schematic of solar parabolic dish collector

Various cavity studies related to electronic cooling and buildings have also been presented by Wu et al., [2] apart from those related to solar receiver design. Reddy and Kumar [3] performed combined laminar convection and radiation heat loss analysis from a modified cavity receiver. The receiver wall temperature is considered isothermal and the outer cavity surface is considered to be adiabatic. Based on the analysis, it is found that the cavity receiver with area ratio of 8 has better performance. Prakash et al. [4] numerically investigated the natural convective heat loss from open cavities viz., cubical, spherical and hemispherical cavities for equal heat transfer area with constant wall temperatures of 100, 200, 300°C and proposed a Nusselt number correlation with the involving effect of cavity shape, Rayleigh number, inclination angle and opening ratio. Balaji and Venkateshan [5] reported combined conductive, convective and radiative heat loss from a vertical slot. The parametric studies have been made to study the effect of various parameters with the conjugate heat transfer along with nature and coupling between various modes of heat transfer.

Various researchers considered isothermal receiver cavity surface and adiabatic cavity walls with or without wind conditions. But in practical, the cavity wall surface temperature depends on the type of application it is used for and the cavity walls are not adiabatic. The varying temperature boundary condition along the walls of the cavity receiver is considered for three different steam generation conditions say, superheated (500°C, 45 bar) [6], saturated (250°C, 40 bar) and sub-cooled (180°C, 10 bar) conditions [7]. The total heat loss from solar parabolic dish collector with modified cavity receiver is investigated for different operating temperatures, different inclinations of the receiver, insulation thickness and cavity cover emissivities.

2. Modelling of Cavity Receiver for Solar Parabolic Dish Collector

2.1. Mathematical model

The modified cavity receiver consists of tubes wound in a hemispherical shape and covered with insulation to reduce the heat loss from the receiver. Covering provided over the insulation keeps the insulation intact. In the present analysis, the tubes are considered as a strip equal to half the radius of the actual tube. The surface temperature of the receiver is considered as that of the temperature of the fluid flowing in the receiver coil. In the present study, the modified cavity receiver of diameter 30.5 cm has been considered with area ratio of 8.

The flow and heat transfer simulations were based on the simultaneous solution of the system of equations describing the conservation of mass, momentum and energy. In cylindrical coordinates, these equations can be represented as [8]:

Continuity equation

$$\frac{1}{r}\frac{\partial V_r}{\partial r}+\frac{1}{r}\frac{\partial V_\theta}{\partial \theta}+\frac{\partial V_z}{\partial z}=0 \tag{1}$$

Momentum Equation

$$\frac{DV_r}{Dt}-\frac{V_\theta^2}{r}=f_r-\frac{1}{\rho}\frac{\partial p}{\partial r}+\nu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial V_r}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 V_r}{\partial \theta^2}+\frac{\partial^2 V_z}{\partial z^2}-\frac{V_r}{r^2}-\frac{2}{r^2}\frac{\partial V_\theta}{\partial \theta}\right] \tag{2a}$$

$$\frac{DV_\theta}{Dt}-\frac{V_r V_\theta}{r}=f_\theta-\frac{1}{\rho r}\frac{\partial p}{\partial \theta}+\nu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial V_\theta}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 V_\theta}{\partial \theta^2}+\frac{\partial^2 V_\theta}{\partial z^2}-\frac{V_\theta}{r^2}-\frac{2}{r^2}\frac{\partial V_r}{\partial \theta}\right] \tag{2b}$$

$$\frac{DV_z}{Dt}=f_z-\frac{1}{\rho}\frac{\partial p}{\partial z}+\nu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial V_z}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 V_z}{\partial \theta^2}+\frac{\partial^2 V_z}{\partial z^2}\right] \tag{2c}$$

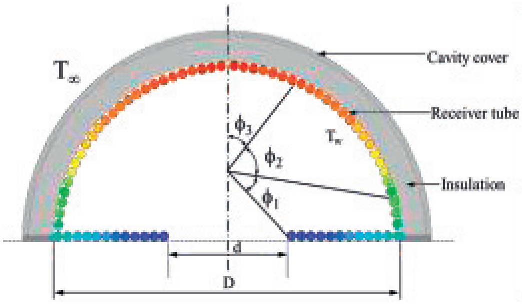
Energy Equation

$$\rho\frac{DH}{Dt}-\frac{DP}{Dt}=k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}\right] \tag{3}$$

The laminar, steady state and 3-D governing equations are solved in FLUENT 6.3.26 [9] using an implicit solver. Non Boussinesq approximation is considered while solving the momentum equation. Along with the above-mentioned conditions, surface-to-surface radiation model is incorporated to account for the radiation exchange between the surfaces of the modified cavity receiver. The variation of thermo-physical properties are considered as the function of temperature [10] and are incorporated while solving the model. The internal surfaces of the receiver were assumed as gray and diffuse. The energy exchange between two surfaces depends in part on their size, separation distance, and orientation. These parameters are accounted by using a geometric function called a view factor. The main assumption of the S2S model is that any absorption, emission, or scattering of radiation can be ignored; therefore, only "surface-to-surface" radiation is considered for the analysis.

2.2. Boundary Conditions

The internal receiver surface is continuously exposed to concentrated solar flux. The helical tubes carry the working fluid which is directly or indirectly used for process heat or power generation applications. In the present study, water is considered as the working fluid that flows through the tubes. The heat input to the receiver is considered to be same in all the three configurations. The mass flow rate of the fluid passing through the receiver coils varies with steam conditions say superheated steam, saturated steam and hot water generation. The receiver wall is subjected to temperature boundary condition. The boundary conditions considered are shown in Fig 2.



Receiver configuration	Zone	Boundary condition, T_w
Sub-cooled	$\phi_1 + \phi_2 + \phi_3$	30 to 180°C
Saturated	ϕ_1	30 to 250°C
	$\phi_2 + \phi_3$	250°C
Superheated	ϕ_1	30 to 257°C
	ϕ_2	257°C
	ϕ_3	257 to 500°C

Fig. 2 Boundary conditions of modified cavity receiver (θ = 90°) for three configurations (sub-cooled, saturated, superheated)

The emissivity of the internal cavity (receiver) surface is kept constant as 0.1 since the inner cavity surface is coated with selective absorber coating. The outer surface of the receiver is covered with insulating material of ceramic wool. The emissivity of the cavity cover is varied from 0 to 1. The receiver is exposed to ambient air and it can enter into the receiver from all the directions. Hence, pressure inlet boundary condition is applied to the outer domain. Coupled wall boundary conditions are applied at the interfaces of the tubes, insulation, outer cover and the ambient to consider the conduction.

2.3. Numerical Procedure

The numerical model is created using GAMBIT 2.4.6. The atmosphere surrounding the cavity receiver is modelled such that the cavity receiver is placed inside large spherical enclosure. The size of the enclosure is increased until it had an insignificant effect on the working fluid and heat flows from the receiver. The fine mesh is considered for the receiver and coarse mesh is considered for outer atmosphere/domain. For pressure velocity coupling, SIMPLE algorithm has been used, with first order upwinding scheme for the discretization of equations. A convergence criterion of 10^{-3} was imposed on the residuals of the continuity equation, momentum equation. A convergence criterion of 10^{-6} is considered for energy equation. The present numerical procedure is validated with Kumar and Eswaran [11] model. The model is validated for pure convection and combined convection and radiation. The maximum deviation for pure convection is 0.22% and for combined convection and radiation, it is 7.29%. The present numerical study is also validated with Reddy and Kumar model [12]. The Nusselt number and heat loss obtained using the present numerical procedure was in good agreement and had the deviation within ±10%.

2.4 Estimation of Heat Loss from the Cavity Receiver

The combined natural convection and surface radiation heat loss from the modified cavity receiver is calculated from the numerical analysis and total Nusselt number is calculated from the equations given below.
The total Nusselt number is given by:

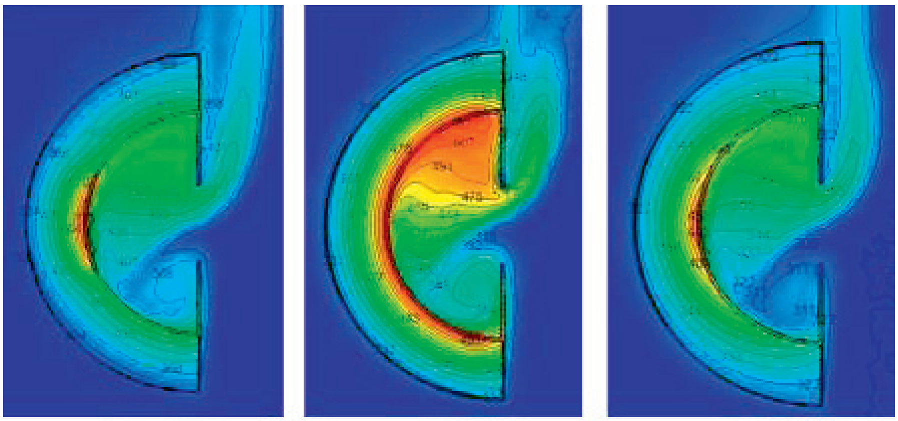
$$Nu_{total} = \frac{Q_{total} \times D}{A_w \times (T_w - T_\infty) \times k_f} \tag{4}$$

3. Results and Discussion

The combined convection and radiation heat losses from the modified cavity receiver were estimated for different inclinations, cavity cover emissivities, different insulation thicknesses and three different temperature profiles using 3-D numerical analysis.

3.1 Effect of inclination of the receiver

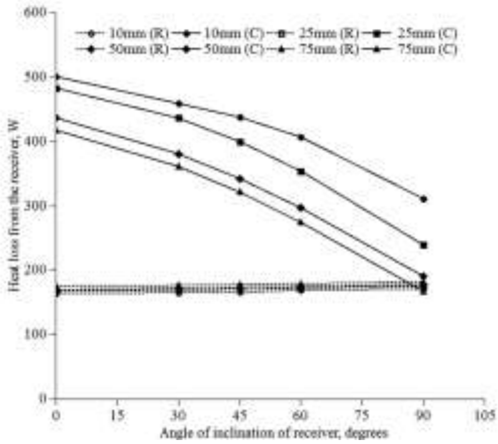
A 3-D numerical analysis of modified receiver was carried out for various inclination (θ) angles ranging from 0° (cavity aperture facing right side of viewer) to 90°(cavity aperture facing down). The temperature contours for superheated steam conditions of the cavity receiver for various inclinations from 0° to 90° is shown in Fig.3 (a). It is observed that for 0° inclination, the hot air occupy the top part of the receiver and convection currents escape out of the receiver. The hot air recirculation at the bottom inner corner of the receiver decreases as the inclination varies from 0° to 90°. The convective heat loss ranges between 400 to 500 W and 150 to 325W for 0° inclination. The radiation heat loss remains between 150 to 200W irrespective of the inclination of the receiver. Also, the amount of hot air leaving the cavity decreases with increase in the inclination from 0°. The temperature contours for saturated steam conditions of the cavity receiver for various inclinations from 0° to 90° is shown in Fig.3 (b). The convective heat loss varies between 300 to 420W and 125 to 250W for 0° inclination. The radiation heat loss varies between 150 to 175W and remains more or less same irrespective of the receiver inclination. The temperature contours for sub-cooled steam condition of the cavity receiver for various inclinations from 0° to 90° is shown in Fig.3 (c).



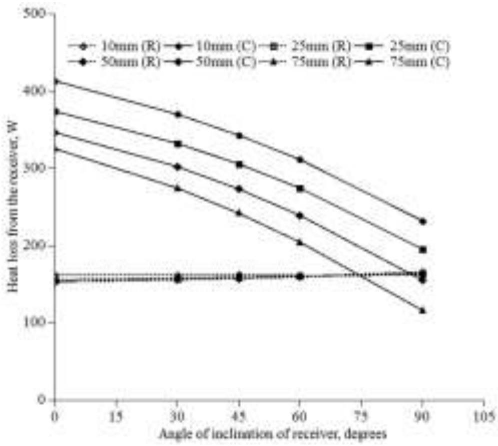
(a) (b) (c)

Fig. 3 Temperature contours for three different steam generation condition at θ = 0°
(a) superheated steam (b) saturated steam (c) sub-cooled steam

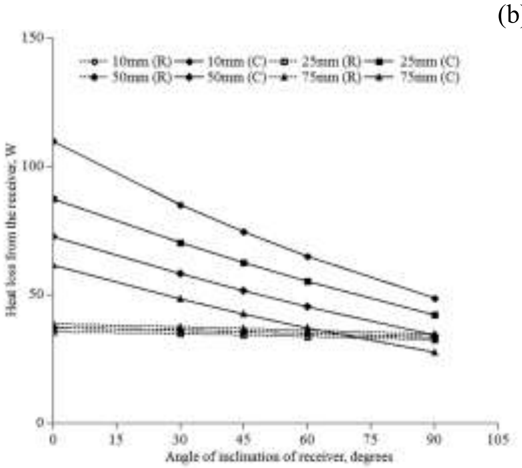
The heat loss from the receiver aperture is less as the temperature in this region is within 100°C. The convective and radiative heat loss varies between 60 to 110W for 0° inclination and less than 50W for 90° inclination. The radiation heat loss ranges between 30 to 40W. The hot air rising from the cavity surface depends on the vertical exposure area of over the cavity receiver and the temperature of the hot air. When the temperature of the hot air leaving the surface is at higher, the plumes move without the recirculation. The variation of convective and radiative heat losses from the modified cavity receiver for superheat, saturated and sub-cooled temperature boundary condition at ε = 0.5 is shown in Fig. 4 It is clear that receiver inclination is an important parameter influencing the convective heat losses from the modified cavity receiver and on the other hand, the variation of radiative heat losses from the modified cavity receiver with inclination of the receiver is negligible. Higher heat loss is observed for receiver inclination of 0° and reduces as it approaches 90°.



(a)



(b)



(c)

Fig. 4 Variation of heat loss from modified cavity receiver with inclination
(a) Superheat condition (b) saturated condition (c) sub-cooled condition

3.2 Effect of insulation thickness

The effect of insulation thickness on heat loss from the cavity receiver has been studied. The variation of heat loss for three different temperature conditions with different insulation thicknesses are shown in Fig. 5. The heat loss from the receiver depends on the type of application and the temperature range. Based on the numerical analysis, it is observed that there is a 10% reduction in total heat loss from the cavity receiver for superheat condition from 10mm insulation thickness to 25mm, whereas there is only about 8% and 3% reduction for subsequent increase in insulation thickness. For saturated steam condition, there is a reduction of 16%, 7% and 4%. But the reduction in heat loss of 30% when the insulation thickness in changed from 10mm to 25mm and 15% reduction when it is changed from 25mm to 50mm and 11% reduction, when it is changed from 50mm to 75mm.

3.3 Effect of emissivity of cavity cover

The variation of total, convective and radiative heat loss from the receiver with emissivity of the cavity cover is shown in Fig.6. The variation is plotted for 0° inclination of the receiver with 25 mm insulation thickness covered over the receiver. It is understood that emissivity of the cavity cover affects the radiative heat loss from the modified cavity receiver greatly than the convective heat loss. The variation of radiative heat loss is about 70% from changing the emissivity from 1 to 0, whereas, convective heat loss varies only 20%.

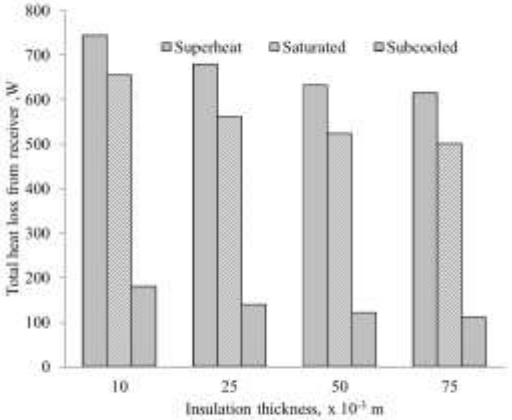


Fig. 5 Variation of total heat loss with insulation thickness

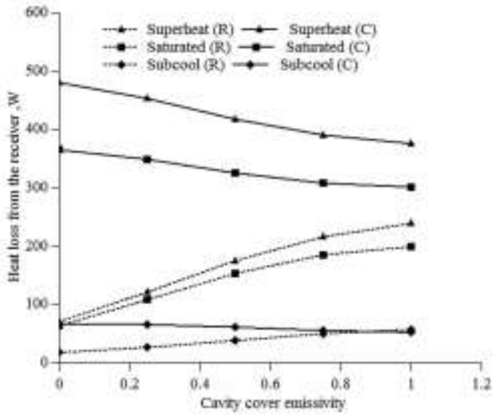


Fig. 6 Variation of heat loss with emissivity

3.4 Nusselt Number Correlations

From the 3-D numerical analysis, convective, radiative and total heat losses from the cavity have been estimated. The average heat transfer coefficients and Nusselt number were determined based on large set of numerical data that were obtained. Correlations for convective, radiative and total Nusselt numbers have been proposed incorporating the influencing parameters like Grashof number, angle of inclination of the receiver, cavity cover emissivity, temperature ratio and conductance parameter. Two total Nusselt number correlations have been proposed based on the temperature conditions.

The total Nusselt number (Nu_{total}) for superheat and saturated steam condition is given by

$$Nu_{total} = 0.0399 Gr^{0.33} (1+\cos\theta)^{0.57} \varepsilon^{0.1} \left(1-\left(\frac{T_\infty}{T_w}\right)^4\right)^{-2.56} \left(\frac{1}{1+\gamma}\right)^{7.99} \quad (5)$$

The number of data points used for generating the correlation is 156 and the correlation coefficient is found to be 0.98 with ±10% deviation.

The total Nusselt number for sub-cooled steam condition is given by

$$Nu_{total} = 0.189 Gr^{0.33} (1+\cos\theta)^{0.76} \varepsilon^{0.17} \left(1-\left(\frac{T_\infty}{T_w}\right)^4\right)^{3.59} \left(\frac{1}{1+\gamma}\right)^{10.95} \quad (6)$$

The number of data points used for generating the correlation is 81 and the correlation coefficient is found to be 0.93 with ±10% deviation.
Where, Grashof number (Gr) is given by:

$$Gr = \frac{g(1/T_w)(T_w - T_\infty)D^3}{\nu^2} \tag{7}$$

In Eq. (9), T_w is calculated as the average temperature of maximum surface temperature and bulk temperature.
The conductance parameter is ratio of effective thermal resistance offered by the solid to the thermal resistance offered by the fluid and is given by:

$$\gamma = \left[\left(\frac{t_{cu}}{k_{cu}}\right) + \left(\frac{t_{ins}}{k_{ins}}\right) + \left(\frac{t_{ms}}{k_{ms}}\right)\right] \Big/ \left(\frac{D}{k_f}\right) \tag{8}$$

The correlation coefficient of this fit is 0.98 with error band of ± 20 % with 68 data points. Fig. 7 shows the parity plot for combined Nusselt number.
Range of parameters:
$6.71 \times 10^7 \le Gr \le 1.01 \times 10^8$ for (5) and $Gr = 1.03 \times 10^8$ for (6); $0° \le \theta \le 90°$; $0.25 \le \varepsilon \le 1$; $0.559 \le T_R \le 0.797$; $0.011 \le \gamma \le 0.088$
The Nusselt number is function of Grashof number (Gr), angle of inclination of the receiver (θ), surface emissivity of the cavity cover (ε), temperature ratio (T_R) and conductance parameter (γ). The exponent of Grashof number has been considered as 0.33 based on the flow conditions, also the variation in Grashof number is very less and hence this value has been fixed. The angle of inclination of the receiver has been considered in the form ($1+\cos\theta$) to take care the effect of the inclination i.e., even at 90° the term would not become zero. The form ($1- T_R^4$) was chosen because the radiative flux is always proportional to (T_w^4-T_R^4) and hence the total Nusselt number. For the parameter γ, the present form has been chosen so that as γ tends to zero, this term reduces to unity. The effect of insulation thickness on heat loss is represented by this term. The conductivity parameter has greater effect on combined Nusselt number as seen in the above equation. Hence conductivity parameter plays a major part in the combined Nusselt number equation. The present model is compared with Reddy and Kumar [12] model for three different temperature conditions. The Nusselt number is on the higher side as isothermal and adiabatic conditions are considered. The comparison for different temperature conditions is shown in Fig. 8.

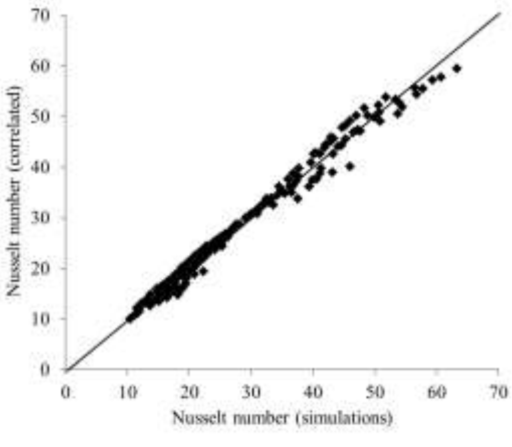


Fig. 7 Parity plot for total Nusselt number

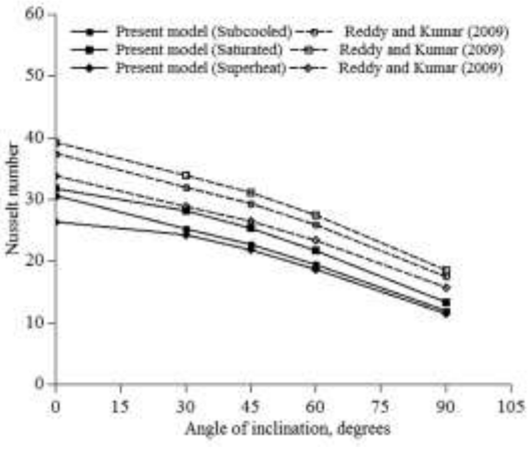


Fig. 8 Comparison of Nusselt number with other model [12]

4. Conclusions

A 3-D numerical investigation has been carried out to study the effect of various parameters on the heat losses from the modified cavity receiver of solar parabolic dish collector. The effect of inclination of the receiver, insulation thickness, operating temperature and cavity cover emissivity has been studied. The convection heat loss is dominated by the inclination of the receiver. The radiative heat loss varies about 70% when cavity cover emissivity is changed from 0 to 1. It is also found that the effect of insulation on the heat loss for medium and high temperature is found to be less significant whereas for low temperature, its effect is significant. Nusselt number correlation has been proposed to estimate the total heat loss from the modified cavity. The present model can be used to predict the total heat loss more accurately than the existing models.

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