Simulation analysis of thermal losses of parabolic trough solar collector in Malaysia using computational fluid dynamics

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Abstract

The increase trend in fuel price coupled with escalating carbon dioxide concentration and energy security have encouraged the world to shift towards renewable energy sources. Parabolic trough collector is the most proven technology for indirect steam generation in solar thermal power plants. Since the annual average daily solar radiation for Malaysia are from 4.21 KWh/m² to 5.56 KWh/m² and sunshine duration is more than 2200 hours per year, parabolic trough solar collector is a very promising technology in the renewable energy field. The purpose of this paper is to simulate and describe the heat losses (radiation and convection) associated with heat collection element (HCE) of Solar Parabolic Trough Collector (PTC), the effect of different wind speeds and mass flowrate of the heat transfer fluid (HTF) on thermal losses were investigated. The receiver of the parabolic trough is modeled in CFD code ANSYS Fluent environment and the geometry is similar to that of LS-2 parabolic trough solar collector. Some assumptions have been made to ease and simplify the simulation. Solar radiation flux profile around the absorber tube is assumed uniform and the radiation flux is treated as heat flux wall boundary condition for the absorber tube. Heat loss model was simulated by employing the Surface – to – Surface (S2S) radiation model to account for the radiation exchange in an enclosure (vacuum annulus gap) of gray – diffuse surfaces. It resulted in a favorably low temperature of the glass envelope as compared to the temperature of the absorber. The convection and radiation heat loss to the surrounding was calculated by the resulting temperature of the glass envelope from the simulation model.

Keywords: Solar energy; heat losses; parabolic trough collector; System simulation; computational fluid dynamics

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1. Introduction

The application of Parabolic trough solar collector (PTC) for electricity generation is not yet competitive with fossil fuel in some countries such as Malaysia, it is however one of the economically feasible renewable energy technologies in future. In recent times, serious steps have been undertaken by the government to encourage the utilization and application of renewable resources, such as biomass, biogas, solar-hydrogen conversion technologies and mini-hydro for energy generation [1]. The most abundant renewable energy sources in Malaysia are biomass and solar [2, 3]. In view of this and couple with rising fuel price, escalating carbon dioxide concentration and energy security, further research and efficient utilization of renewable energy technologies towards replacing the conventional sources of energy should be actively exploited.

Parabolic trough collector is the most proven technology for indirect steam generation in solar thermal power plants [4]. Its capability to supply thermal energy over a wide range of temperature without serious degradation in efficiency has made it the most frequently used collector in the thermal power plant for steam generation [4]. By deployment of PTC, the solar radiation flux can be concentrated up to a value of around 80 times, and thermal fluid can be heated up to approximately 400°C [5]. Some of the solar electric power generation (SEGS) plants employing this technology include Andasol plants in southern Spain and SEGS plants in southern California [6]. A typical PTC consists of a parabolic trough reflector, which reflects the incident radiation from the sun onto the heat collection element (HCE). The HCE consists of a steel absorption pipe enveloped inside a glass tube located at the focus of a parabola. The circulating Heat Transfer Fluid (HTF), which is passed through the HCE is heated up by the radiant energy absorbed. The heat collected is used to produce superheated steam at high temperature.

The use of PTC in the electric generation system has been associated with numerous losses. Some of the losses associated with PTC including optical, thermal and geometry losses [7]. These losses lead to decrease in efficiency of the PTC. Extensive thermal losses in PTC have been reported by [8, 9]. The purpose of this paper is to simulate and describe the heat losses (radiation and convection) associated with heat collection element (HCE) of Solar Parabolic Trough Collector (PTC), the effect of different wind speeds and mass flowrate of the Heat Transfer Fluid (HTF) on thermal losses were investigated. The receiver of the parabolic trough is modelled in CFD code ANSYS Fluent environment and the geometry is similar to that of LS-2 parabolic trough solar collector.

| Nomenclature | Description |
|--------------|-------------|
| $h$ | convection heat transfer coefficient of air (W/m²-K) |
| $D_g$ | glass envelope outer diameter (m) |
| $T_g$ | glass envelope outer surface temperature (K) |
| $D_g$ | outer diameter of glass envelope (m) |
| $v_w$ | wind velocity (m/s) |
| $\sigma$ | Stephan–Boltzmann constant ($5.67 \times 10^{-8}$ W/m²-K⁴) |
| $\rho$ | density (kg/m³) |
| $C_p$ | specific heat capacity (W/kg-K) |
| $K$ | Thermal conductivity (W/m-K) |
| $\mu$ | kinematic viscosity (kg/m-s) |
| $e_g$ | glass envelope emissivity |
| $T_{sky}$ | temperature of the sky (K) |
| $k_{in}$ | turbulence intensity (%) |
| $D_h$ | hydraulic diameter (mm) |
| $Re_{Dh}$ | Reynolds number based on hydraulic diameter |
| $\dot{q}_{conv}$ | Heat loss by convection per unit length, (W/m) |
| $\dot{q}_{rad}$ | Heat loss by radiation per unit length, (W/m) |
2. Modeling of the parabolic trough solar collector

2.1. Physical Model

This paper seeks to analyze the heat losses associated with the parabolic trough collector. The Heat Collection Element (HCE) consists of absorber tube surrounded by a glass envelope as shown in Fig 1. The steel absorption pipe is coated with selective coating so as to acquire the desired properties. The thermal radiation loss from the steel pipe is reduced by coating with high absorbance in solar energy spectrum, and low emittance in the long wave energy spectrum. The glass is coated with anti-reflective coating to reduce reflective loss. The annular gap between glass envelope and steel absorber tube is separated with a vacuum pressure in order to reduce heat losses as well as protecting the selective coating from degradation. The heat transfer model is based on the energy balance between heat transfer fluid and the surrounding as shown in Fig.1.

2.2. Governing equations

Heat is dissipated to the ambient by means of convection and radiation losses. The convection heat transfer can be of natural or forced convection. The mode of convection depends on the wind condition at the collector surrounding [8].

\[ \dot{q}_{\text{conv}}' = h_a \pi D_g \left(T_g - T_a\right) \left(\frac{W}{m}\right) \]  

(1)

The convection heat transfer coefficient of air can be calculated by using Mullick & Nanda correlation [8] which is given by:

\[ h_a = 4D_g^{-0.42} \cdot \nu_w^{0.5} \]  

(2)

Heat transferred by radiation from the glass envelope to the ambient can be calculated by [8]:

\[ \dot{q}_{\text{rad}}' = \sigma \varepsilon_g \pi D_g \left(T_g^4 - T_{\text{sky}}^4\right) \left(\frac{W}{m}\right) \]  

(3)

The total heat loss is given by:

\[ \dot{q}_{\text{thermal loss}} = \dot{q}_{\text{conv-glass-sky}}' + \dot{q}_{\text{rad-glass-sky}}' \left(\frac{W}{m}\right) \]  

(4)

For turbulent water flow inside the absorber, the partial differential equations governing the fluid flow and heat transfer in the enclosure include the continuity, Navier–Stokes energy equations and RNG k-ε model have been
adopted [4, 9, 10]. The turbulence intensity is calculated based on hydraulic diameter of $D_h$ (=66 mm) by the relation [2]:

$$k_{in} = 0.16(Re_{Dh})^{-1/8} \times 100 \%$$

(5)

2.3. PTC Geometry and grid generation for PTC

The data used in the present modelling is shown in Table 1. The receiver is modelled with its geometry similar to that of LS-2 parabolic trough receiver [4]. Once the receiver has been modeled, the model is then exported for meshing. Meshing is a process whereby the model domain is divided or discretized into a finite number of smaller elements. After the meshing is completed, the domains are defined as fluid or solid according to their respective condition. The inner and outer diameters have been selected such that they correspond to those of commercial parabolic trough receivers. Fig. 2a shows the grid for the plain tube configuration using a quadrilateral face mesh over the volume of the cylinder. The boundary conditions for the geometry were defined for inlet, outlet, and cylinder wall. The continuum volume of fluid region was defined as water (see Fig.2b). Fig.2c-d shows the Isometric and front view respectively of meshed vacuum annulus gap that was configured in CFD code ANSYS.

Fig. 2. Grid and boundary conditions used in numerical simulation (a) Isometric and (b) Front view of meshed absorber tube and fluid region (c) Isometric and (d) front view of meshed vacuum annulus gap.

| Geometric of PTC                  | Value(m) |
|----------------------------------|----------|
| Collector aperture area          | 39       |
| Collector aperture width         | 5        |
| Collector aperture length        | 7.8      |
| Glass envelope outer diameter    | 0.115    |
| Glass envelope inner diameter    | 0.109    |
| Absorber tube outer diameter     | 0.07     |
| Absorber tube inner diameter     | 0.065    |

2.4. CFD modeling

A proper model has to be selected in order for ANSYS to solve for appropriate equations. The energy equation is turned on so that ANSYS will solve for heat transfer equation. The $\kappa$-$\varepsilon$ model is turned on as well to account for
turbulent flow in the absorber tube. Heat loss simulation is then modeled by employing the Surface – to – Surface (S2S) radiation model to account for the radiation exchange in an enclosure (vacuum annulus gap) of gray – diffuse surfaces. In this work, Water was selected as the working fluid (heat transfer fluid HTF), and the thermo physical properties were assumed to be temperature independent. The thermo physical properties of water and other materials used for simulation are listed in Table 2. The properties of the materials such as the density and thermal conductivity are defined explicitly and it can be referred in Table 2. The materials created are assigned to their respective cell zone.

| Material       | \( \rho \) (kg/m\(^3\)) | \( C_p \) (J/kg-K) | \( k \) (W/m-K) | \( \mu \) (kg/m-s) |
|----------------|---------------------------|---------------------|-----------------|---------------------|
| Water - liquid | 998.2                     | 4182                | 0.6             | 0.001003            |
| Air            | 0.0001245                 | 1009                | 0.3095          | 0.00002181          |
| Pyrex          | 2225                      | 835                 | 1.4             |                     |
| Copper         | 8978                      | 381                 | 387.6           |                     |

2.5. boundary conditions

Prior to solving of the equations, boundary conditions have to be specified. The inner surface of the absorber tube is modeled as stationary with no – slip condition. The two surfaces of side walls of the absorber tube are modeled as adiabatic or zero heat flux condition. The outer surface of the absorber tube is modeled as non – zero heat flux condition. The effective incoming solar radiation is modeled as heat flux at the outer absorber tube. Heat flux around the circumference of the outer absorber tube is assumed constant. For the S2S radiation model, the following boundary conditions are applied at its respective named selection. The outer surface of the absorber tube is modeled as stationary with no – slip condition. Its internal emissivity is set to 0.27. The surface of the glass tube is assumed to undergo convection and radiation from the background. Internal emissivity of the surface of the glass is set to 0.86. Prior to simulating the model, monitors are created for variables of interest. Monitor for outlet temperature and glass tube surface temperature for instance are created to provide better observation of these variables of interest. Calculations are run until convergence is achieved.

3. Result and discussion

The resulting simulation model is represented graphically to provide a clearer view of temperature distribution along the length of the absorber tube as well as around the circumference of the HCE. For the specified absorber tube and collector specification, axial temperature distribution of the absorber tube is determined (refer to Fig. 3a). For the given HTF flowrate, the inlet and outlet temperatures are 153°C and 164 °C respectively, thus the difference in temperature gives 11 °C, this result is expected due to the fact that, the system under study is only a single PTC whose length is 7.8m.

![Fig.3. Temperature distribution](image-url)
This result is in agreement with the experimental result from Fabian [11], in his work, he used 8 PTC whose inlet and outlet temperatures are 155 °C and 290 °C respectively. Under the current boundary condition Solar radiation flux profile around the absorber tube is assumed as uniform and the radiation flux is treated as heat flux wall boundary condition for the absorber tube. Figs 3b-c presents the cross sectional temperature contours around the circumference of the absorber tube and of S2S radiation model. The circumferential temperature (see Fig. 3b) decreases radially towards the centre.

3.1. Glass envelope temperature distribution

The effect of mass flow rate on temperature with respect to glass envelope, the absorber and HTF outlet temperature were studied as shown in Fig. 4, the output temperature was found to be inversely proportional to the mass flow rate. The temperature distribution for the glass envelope, the absorber and HTF were at their lowest value for flow rate of about 1kg/s. As the flow rate reduces, the temperature of the absorber and the outlet temperature of the HTF increase considerably. For example at 0.02kg/s, the HTF outlet temperature is 614 °C. This suggests that mass flow rate has a significant influence on resulting absorber, HTF and glass envelope temperature which eventually affects the thermal losses of the system.

![Graph showing temperature variation with mass flow rate](image)

**Fig. 4. Effect of mass flow rate Temperature variation.**

![Graph showing effect of wind speed on glass temperature](image)

**Fig. 5. Effect of wind speed on glass temperature.**

The wind speed was manipulated in the range of 0.5 m/s to 4 m/s in order to investigate the effect of wind speed on the glass envelope temperature. It can be observed from Fig. 5 that as the wind speed is increased from 0.5 m/s to 4 m/s, the glass envelope temperature decrease correspondingly for all the mass flow rate cases. The decrease in this temperature is attributed to the fact that the convection heat loss across the glass envelope increases with induced wind speed. An increased in convection coefficient has resulted in decrease in glass envelope temperature (see Table 3) and this confirms that thermal loss due to convection has been found to be the greatest contributor to heat loss especially with the presence of wind [6].
Table 3. Effect of wind speed and convection heat transfer coefficient on glass envelope temperature.

| Wind velocity (m/s) | Convection coefficient, $h$ (W/m²-K) | Glass Envelope Temperature (K) | HTF @ flow rate 0.8 kg.s⁻¹ | Glass Envelope Temperature (K) | HTF @ flow rate 0.4 kg.s⁻¹ | Glass Envelope Temperature (K) | HTF @ flow rate 0.08 kg.s⁻¹ |
|---------------------|--------------------------------------|--------------------------------|-----------------------------|--------------------------------|-----------------------------|--------------------------------|-----------------------------|
| 0.5                 | 7.01                                 | 327.13                         | 328.78                      | 334.28                         |                             |                                |                             |
| 1                   | 9.92                                 | 324.09                         | 325.52                      | 330.31                         |                             |                                |                             |
| 1.5                 | 12.15                                | 322.28                         | 323.57                      | 327.93                         |                             |                                |                             |
| 2                   | 14.03                                | 321.00                         | 322.21                      | 326.24                         |                             |                                |                             |
| 2.5                 | 15.68                                | 320.03                         | 321.16                      | 324.95                         |                             |                                |                             |
| 3                   | 17.18                                | 319.26                         | 320.33                      | 323.92                         |                             |                                |                             |
| 3.5                 | 18.56                                | 318.62                         | 319.63                      | 323.06                         |                             |                                |                             |
| 4                   | 19.84                                | 318.07                         | 319.05                      | 322.33                         |                             |                                |                             |

3.2. Thermal Loss Analysis

The main thermal loss from the absorber tube outer wall to the evacuated glass tube (surrounding the absorber) occurs by radiation. The heat loss from the glass cover tube occurs by radiation to the sky and by convection to the surrounding air by wind or natural convection. The convective heat transfer coefficient is calculated from Mullick & Nanda correlation [12]. In this study the effect of different wind speeds on thermal losses were investigated. In Fig. 6, it can be observed that as the wind speed increases the total heat losses (convection and radiation) also increase but by small amount. For example at 0.04kg/s the total heat loss increases from 198 W/m to about 2010 W/m whereas at 0.8kg/s the total heat loss increases by about 4.8 %. It can be concluded from Fig. 6 that the combine effect of radiation and convection heat losses are not significantly affected by changes in wind speed.

![Fig. 6. Effect of wind speed on heat loss.](image1)

![Fig. 7. Components of heat losses (a) wind velocity =2m/s (b) wind velocity =4m/s.](image2)
Fig. 7 (a),(b) shows the components of heat losses across the glass envelope. Fig. 7 (a) illustrates the effect of parameters such as wind and mass flow rate of HTF on heat losses. It can be observed that radiation loss is less affected by both parameters, whereas convection heat loss contributes large percentage of heat loss. In summary, referring to Fig. 7 (a), convection heat loss contributes 64% of total heat loss across the glass envelope whereas radiation contributes 36% of the total heat loss; these results are for wind speed of 2 m/s. The same trend can be observed in Fig. 7 (b) where the convection heat loss increase to 71% of the total heat loss. In conclusion, heat loss is consistently higher in the convection case primarily due to effect of wind speed.

4. Conclusion

A detailed model of heat loss in parabolic trough receivers has been developed using commercial heat transfer software that includes radiation as well as conduction and convection heat transfer. It is observed from numerical analysis that the combine effect of radiation and convection heat losses are not significantly affected by changes in wind speed. How the components of the heat losses shows that convection heat loss contributes 64% of total heat loss across the glass envelope whereas radiation contributes 36% of the total heat loss; these results are for wind speed of 2 m/s.

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