Thermal Resistance Model For CSP Central Receivers

O.A.J. de Meyer\textsuperscript{1,2, a)}, F. Dinter\textsuperscript{2, b)} and S. Govender\textsuperscript{1, c)}

\textsuperscript{1} Author Affiliation) Eskom, Eskom Enterprises Park, Cnr Leeukop & Simba road, Sandton, South Africa
\textsuperscript{2}Solar Thermal Energy Research Group (STERG), Stellenbosch University, Stellenbosch, South Africa
\textsuperscript{a)}Corresponding author: dmeyeroe@eskom.co.za
\textsuperscript{b)}frankdinter@sun.ac.za
\textsuperscript{c)}govends@eskom.co.za

Abstract. The receiver design and heliostat field aiming strategy play a vital role in the heat transfer efficiency of the receiver. In molten salt external receivers, the common operating temperature of the heat transfer fluid or molten salt ranges between 285°C to 565°C. The optimum output temperature of 565°C is achieved by adjusting the mass flow rate of the molten salt through the receiver. The reflected solar radiation onto the receiver contributes to the temperature rise in the molten salt by means of heat transfer. By investigating published work on molten salt external receiver operating temperatures, corresponding receiver tube surface temperatures and heat losses, a model has been developed to obtain a detailed thermographic representation of the receiver. The steady state model uses a receiver flux map as input to determine: i) heat transfer fluid mass flow rate through the receiver to obtain the desired molten salt output temperature of 565°C, ii) receiver surface temperatures iii) receiver tube temperatures iv) receiver efficiency v) pressure drop across the receiver and vi) corresponding tube strain per panel.

INTRODUCTION

The thermal resistance concept \cite{1} was utilized to derive a thermal resistance network representing the receiver. This steady state one-dimensional heat transfer model is used to determine the receiver temperatures from known variables such as ambient conditions, incident flux onto the receiver and heat transfer fluid mass flow rate with an inlet temperature. It is evident from the thermal resistance network model that the equations are not solvable without an iterative process and initial guess values for the surface temperature. This paper further elaborates on the methodology used for the iterative process to obtain the final surface temperatures, inner tube temperatures, heat losses and heat transfer fluid mass flow rate. From these results the receiver efficiency is determined with the corresponding pressure drop across the receiver and tube strain per panel.

Receiver Design

The theoretical receiver used in this paper is utilized in a 100 MW\textsubscript{e} plant with 12 hours of storage, situated in Upington, South Africa. The external receiver consists out of 16 panels with tube diameters and thickness of 50 mm and 1.5 mm respectively. Two molten salt flow regimes flow in series through the panels in a serpentine configuration \cite{3}, where a cross over flow commences halfway. The molten salt enters the receiver from the south and exits at the north. The heliostat design, field layout, aiming strategy and incident thermal power onto the receiver at design point (equinox at noon) correlates with a 652 MW\textsubscript{th} receiver with an optical height of 208 m, diameter of 16.32 m and height of 19.24 m. The flux maps from the heliostat field design are obtained from DELSOL3 \cite{2}.
MATHEMATICAL MODEL

The receiver thermal resistance network model was developed using the thermal resistance concept, commonly used in heat transfer evaluations [1]. This model is illustrated in Figure 1, from which the following energy balance is applicable;

![Figure 1. Receiver thermal resistance network applicable represented at the receiver tube cross section](image)

\[
\dot{Q}_{\text{field}} - \dot{Q}_{\text{ref}} = \dot{Q}_{\text{field}} (1 - \rho) = \dot{Q}_{\text{in}} + \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} \tag{1}
\]

The incident flux onto the receiver (\( \dot{Q}_{\text{field}} \)) is resulting from the flux map obtained from DELSOL3 [2]. The reflected radiation (\( \dot{Q}_{\text{ref}} \)) from the receiver is influenced by the receiver reflectivity (\( \rho \)). Thus the energy transferred to the heat transfer fluid (\( \dot{Q}_{\text{in}} \)) is subjected to the incident flux onto the receiver, reflected radiation, external thermal radiation loss (\( \dot{Q}_{\text{rad}} \)) and external convective heat loss (\( \dot{Q}_{\text{conv}} \)). The convection losses, radiation losses and resulting thermal energy to heat transfer fluid are expressed as;

\[
\dot{Q}_{\text{conv}} = \frac{(T_s - T_{\text{amb}})}{R_{\text{conv}(\text{ext})}} \tag{2}
\]

\[
\dot{Q}_{\text{rad(\text{ext})}} = \frac{(T_s - T_{\text{surr}})}{R_{\text{rad(\text{ext})}}} \tag{3}
\]

\[
\dot{Q}_{\text{in}} = \frac{(T_s - T_{\text{in}})}{R_{\text{cond}}} \quad \frac{(T_{\text{in}} - T_{\text{out}})}{R_{\text{conv}(\text{in})}} = \frac{(T_s - T_{\text{m}})}{R_{\text{cond}} + R_{\text{conv}(\text{in})}} = \dot{m}C_p(T_{\text{out}} - T_{\text{in}}) \tag{4}
\]

The receiver temperatures are represented by surface temperature (\( T_s \)), ambient temperature (\( T_{\text{amb}} \)), surrounding or sky temperature (\( T_{\text{surr}} \)), inner tube temperature (\( T_{\text{in}} \)), heat transfer fluid mean temperature (\( T_{\text{m}} \)) and heat transfer fluid inlet (\( T_{\text{in}} \)) and outlet (\( T_{\text{out}} \)) temperatures. The heat transfer fluid heat capacitance and mass flow rate is expressed as \( C_p \) and \( \dot{m} \) respectively. The thermal resistance (\( R \)) for each of the abovementioned expressions; [1]
\[
R_{\text{rad}}(\text{ext}) = \frac{1}{h_{\text{rad}}(\text{ext})A_s} \tag{5}
\]
\[
R_{\text{conv}}(\text{ext}) = \frac{1}{h_{\text{conv}}(\text{ext})A_s} \tag{6}
\]
\[
R_{\text{cond}} = \frac{L_1}{k_1A_t} \tag{7}
\]
\[
R_{\text{conv}}(\text{in}) = \frac{1}{h_{\text{conv}}(\text{in})A_{(\text{in})}} \tag{8}
\]

With the corresponding exposed surface area \((A_s)\) used for radiation and convection losses, whereas the cross sectional area \((A_t)\) is used for conduction of heat from the outer tube area to the inner tube area. The tube cross section length \((L_s)\) is equivalent to the tube thickness \((t)\). The thermal conductivity of the tube \((k_1)\) is applicable to the tube’s conductive resistance \((R_{\text{cond}})\). Heat transfer coefficients \((h)\) for the external radiative resistance \((R_{\text{rad}}(\text{ext}))\), external convection resistance \((R_{\text{conv}}(\text{ext}))\) and internal convection resistance \((R_{\text{conv}}(\text{in}))\) respectively is;

\[
h_{\text{rad}}(\text{ext}) = \varepsilon\sigma\left(T_s^2 + T_{\text{surr}}^2\right)(T_s + T_{\text{surr}}) \tag{9}
\]
\[
h_{\text{conv}}(\text{ext}) = \left(h_{\text{nat}}^{3.2} + h_{\text{wind}}^{3.2}\right)^{\frac{1}{3.2}} \tag{10}
\]
\[
h_{\text{conv}}(\text{in}) = \frac{k_{\text{HTF}}\mu_{\text{HTF}}}{r_i} \tag{11}
\]

with,

\[
h_{\text{nat}} = \frac{k_{\text{film}}\mu_{\text{nat}}}{H_{\text{rec}}} \tag{12}
\]
\[
h_{\text{wind}} = \frac{k_{\text{film}}\mu_{\text{wind}}}{D_{\text{rec}}} \tag{13}
\]

Where \(\varepsilon\) and \(\sigma\) the receiver emissivity and Stefan Boltzmann’s constant respectively represents. The mixed external convection heat transfer coefficient \((h_{\text{conv}}(\text{ext}))\) is determined as in [3] and [4]. The heat transfer coefficients for natural convection \((h_{\text{nat}})\) and forced convection or wind \((h_{\text{wind}})\) are obtained by determining the Nusselt numbers for each. Nusselt numbers for natural convection \((\mu_{\text{nat}})\), forced convection \((\mu_{\text{wind}})\) and internal forced convection \((\mu_{\text{HTF}})\) is determined as per [1] or [4].

The receiver dimensions are height \((H_{\text{rec}})\), external diameter \((D_{\text{rec}})\) and receiver inner tube radius \((r_i)\). Where thermal conductivity for air at film temperature \((k_{\text{film}})\) and heat transfer fluid bulk fluid temperature \((k_{\text{HTF}})\) is applicable.

**METHODOLOGY**

This section elaborates on the method to be used for utilizing the receiver thermal resistance model to determine; i) mass flow rate through the receiver to obtain the desired molten salt output temperature of 565°C, ii) receiver surface temperatures iii) receiver tube temperatures and iv) receiver efficiency.

**Determine Heat Transfer Fluid Mass Flow Rate**

An initial average surface temperature is assigned to each panel (450°C to 650°C). From the incident flux map and equations (2) and (3), the thermal losses for each panel are calculated to obtain the total thermal loss of the receiver. The resulting energy transferred to the heat transfer fluid \((Q_{\text{in}})\) is obtained from equation (1). With equation (4), the heat transfer fluid mass flow rate is calculated for each flow regime in the receiver.
Determine Surface And Tube Temperature

In the second step, each panel is divided into the number of nodes corresponding to the incident flux map. Starting with the first node in the first panel of each flow regime, see Figure 2(a), the thermal losses for the node are calculated and resulting energy transferred to the heat transfer fluid is obtained. With the heat transfer input temperature known, the output temperature is calculated using equation (4) using the calculated heat transfer fluid mass flow rate. The bulk fluid temperature can thus be obtained to determine the corresponding tube (T_t) and surface (T_s) temperature respectively using equation (4).

Receiver Efficiency

As illustrated in Figure 2(b), from the newly calculated surface temperature the resulting heat losses are recalculated for the node. The aggregated heat transferred from all the nodes in each flow regime to the heat transfer fluid is re-evaluated and mass flow rate recalculated accordingly. From equation (4) it is evident that any deviation in the amount of heat transferred to the heat transfer fluid will induce a corresponding mass flow rate. The process is reiterated until final surface temperatures are obtained. The receiver efficiency can thus be calculated utilizing the following equation;

\[ \eta_{rec} = \frac{\dot{Q}_{in}}{\dot{Q}_{tot}} = \frac{\dot{Q}_{in}}{\dot{Q}_{in} + \dot{Q}_{rad(\text{ext})} + \dot{Q}_{conv(\text{ext})} + \dot{Q}_{\text{ref}}} \tag{14} \]

Receiver Pressure Drop

The receiver pump is sized according to the pressure drop in the tower and across the receiver. It is therefore important to analyze the pressure drop in the receiver. The pressure drop is influenced by factors such as the heat transfer fluid mass flow rate, receiver height, number of panels, tube diameter and tube thickness. This is evident in the total system loss equation for losses in pipe systems [5], with the pressure drop correlation being \( \Delta P = \rho gh \)

\[ \Delta h_{tot} = h_f + \sum h_m = \frac{V^2}{2g} \left( \frac{fL}{d} + \sum k \right) \tag{15} \]
Head loss due to friction is represented by $h_f$. Minor losses ($h_m$) in the receiver tube are accounted for in 2x45° and 2x90° elbows, each with a resistance coefficient factor of 16 and 30 respectively [6]. Pipe entrance and exit resistance coefficients are 0.78 and 1 respectively. Interconnecting piping between receiver panels and pressure drop in the tower has not been included in this paper. The friction factor for turbulent flow in rough pipes can be calculated as [7]

$$f = 0.184 \times Re^{-0.2}$$  \hspace{1cm} (16)$$

**Receiver Tube Strain**

The receiver is subjected to daily varying temperature operating conditions, imposing thermal stresses within the material. Furthermore, the incident flux onto the receiver results in a temperature gradient across the material. This phenomenon imposes a great risk to the receiver material during cloud transients as thermal shock is induced onto the receiver material. It is thus also important that the heliostat field aiming strategy accounts for evenly distributed flux among the receiver. For mechanical strain resulting due to a temperature gradient, the following equations are applicable [8] and [9].

$$\varepsilon = \frac{1}{\alpha} \left( \frac{T_s - T_i}{2(1-\nu)} + \frac{T_s + T_i}{2} - T_{avg} \right)$$  \hspace{1cm} (17)$$

where

$$T_{avg} = T_m + \frac{1}{\pi} \left( \frac{T_i + T_s}{2} - T_m \right)$$  \hspace{1cm} (18)$$

with $\varepsilon$ representing the strain and $T_{avg}$ the average temperature of the cross section. The coefficient of thermal expansion and Poisson’s ratio is represented by $\alpha$ and $\nu$ respectively. From the receiver pressure drop calculations, the internal pressure per tube can be obtained. For stresses in the tube wall resulting from the internal pressure ($P_i$) and thermal gradient ($\Delta T$) across the tube wall, further analysis can be done. These stresses comprises of radial ($r$), tangential ($\theta$) and longitudinal ($l$) stresses. Thick wall pressure vessel calculations are considered ($t/d > 1/20$). The following literature [10]-[12] elaborate on stresses resulting from internal pressure and the thermal gradient across the tube wall.

**RECEIVER THERMOGRAPHIC REPRESENTATION**

A thermographic representation of the receiver is obtained from the thermal resistance model’s results. The model’s results various in representations of the receiver based on the incident flux map applied, ambient conditions and heat transfer fluid inlet temperature at the receiver. Through an iterative process, the heat transfer fluid mass flow rate, inner tube and surface temperatures are calculated. The corresponding thermal losses are obtained to determine the receiver efficiency. The receiver thermographic representation furthermore provides a useful tool in tube strain analysis and receiver design considerations. For illustrative purposes, the following results have been obtained for the theoretical receiver discussed in this paper operating at design point. The following figures are the resulting thermographic representation of the receiver; Figure 3(a) the applied flux maps, Figure 3(b) the receiver efficiency, Figure 3(c) the resulting surface temperatures and Figure 3(d) the corresponding inner tube temperatures. Radiation and convective heat losses are represented in Figure 3(e) and Figure 3(f) respectively.

Similarly, results have been obtained for the same day at time 08h00 and 16h00, presented in Figure 4(a) and Figure 4(b) respectively. It is interesting to note the receiver efficiency drop to below 70% where low incident flux and relatively high surface temperatures are present. Furthermore, the effect of the cross-over flow regimes to mitigate high surface temperatures is clearly noticeable in Panels 4-5 and Panels 12-13 in Figure 5. The peaks in both scenarios are reduced by introducing the heat transfer fluid coming from lower intensity flux panels, thus having a lower fluid temperature. Thus the cross over flow assists with the overall receiver heat transfer efficiency.
FIGURE 3. Receiver thermographic representation at design point (a) DELSOL3 applied flux map (b) receiver efficiency (c) inner tube temperatures (d) receiver surface temperatures (e) thermal radiation losses (f) thermal convective losses
The axial strain due to the thermal gradient across the tube wall is presented in Figure 6 at times 08h00, 12h00 and 16h00 respectively. It is observed that there is a relationship between the tube strain and the incident flux map. This further supports the importance of the heliostat field aiming strategy and flux distribution effects during cloud transients. Although higher surface temperatures are observed at the receiver outlet panels, the temperature gradient is higher at lower heat transfer fluid temperatures with a high incident flux.
The following table presents an overview for the results obtained from the thermal resistance model. With higher incident flux onto the receiver, the greater the mass flow rate of the heat transfer fluid to achieve the desired output temperature of 565 °C. This correlates with a higher pressure drop observed across the receiver. Convection losses in the morning and evening are higher due to a drop in ambient temperatures, thus impacting the receiver efficiency.

<table>
<thead>
<tr>
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<th>08h00</th>
<th>12h00</th>
<th>16h00</th>
</tr>
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<tbody>
<tr>
<td>Ambient temperature [°C]</td>
<td>25.0</td>
<td>33.4</td>
<td>33.3</td>
</tr>
<tr>
<td>Wind Speed [m/s]</td>
<td>1.45</td>
<td>4.4</td>
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<td>Direct normal irradiation (DNI) [W/m²]</td>
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<td>957</td>
<td>502</td>
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<td>Receiver incident flux [MWth]</td>
<td>546</td>
<td>652</td>
<td>296</td>
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<tr>
<td>Receiver radiation losses [MWth]</td>
<td>22.5</td>
<td>23.8</td>
<td>20.7</td>
</tr>
<tr>
<td>Receiver convection losses [MWth]</td>
<td>11.0</td>
<td>12.1</td>
<td>15.1</td>
</tr>
<tr>
<td>Heat transferred to heat transfer fluid [MWth]</td>
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<td>570</td>
<td>240.4</td>
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<tr>
<td>Receiver efficiency [%]</td>
<td>86.9</td>
<td>87.5</td>
<td>81.3</td>
</tr>
<tr>
<td>Heat transfer fluid mass flow rate [kg/s]</td>
<td>1134</td>
<td>1363</td>
<td>574</td>
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<tr>
<td>Pressure drop across receiver [kPa]</td>
<td>570</td>
<td>791</td>
<td>167</td>
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</table>

**CONCLUSION**

A receiver thermal resistance model has been developed utilizing the thermal resistance concept. The results obtained from this model present a detailed thermographic representation of the receiver. The receiver pressure drop plays an important role in the auxiliary consumption during operation. Furthermore, the importance of the heliostat field aiming strategy and effects of cloud transients on strain induced by thermal gradients across the tube wall has been highlighted. This paper emphasized the relationship between the receiver design, configuration and incident flux map onto the receiver through the results. Various interpretations of the receiver design and configuration can be made regarding these results. This could furthermore assist in optimizing the design of the receiver by changing parameters such as the receiver height, number of panels, tube diameter, tube thickness, receiver material and heat transfer fluid flow regimes.

**REFERENCES**