Direct Steam Generation with Dish Concentrators
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ABSTRACT
This paper presents current research on Direct Steam Generation (DSG) by the Solar Thermal Group at ANU. It includes ongoing work on a transient model for a dish steam cavity receiver that is suitable for model-based control of DSG as well as reporting on the experimental set-up on a new 500m² paraboloidal dish solar concentrator.

A comparison of two transient models of a dish steam cavity receiver, one developed previously at the ANU with a new proposed model currently under development at the ANU Solar Thermal Group is presented. Both models are described and compared using a simulation using TRNSYS. A qualitative analysis is also included discussing the relative merits of each approach.

In order to assess the thermal performance of the new 500 m² dish, a steam receiver previously used in a 400m² dish steam generation system is being re-instrumented and installed on the new dish. The rest of the 400m² dish steam generation system is being assembled to work with the new 500m² dish. The results of the first test of a Direct Steam Generation system on the SG4 dish are presented in this paper.

Keywords: Direct Steam Generation, Dish Concentrators, Steam Receivers, Transient Modeling

DISSH SYSTEMS AT ANU
The ANU has pioneered the design of very large dish concentrators. Fig. 1 shows the 400 m² “SG3” system completed on the campus in 1994 and the 500 m² “SG4” system completed in mid 2009. (Lovegrove et al 2010).

The SG4 dish is currently being operated with the monotube boiler direct steam generation receiver that was previously used on the SG3 dish. In direct steam generation, water is boiled in the receiver to be fed directly to the power block, removing the need for a heat transfer fluid and a heat exchanger. The higher concentration ratio of the new dish has allowed the same receiver to be used even though the SG4 dish is larger.

The steam cavity receiver consists of a winding of steel tube coiled around a taper section and cylindrical section to form a cavity (see Fig. 2). The feed water enters the receiver at the beginning of the taper section and exits at the end of the top section. The tube is comprised of approximately 110 m of 16 mm OD mild steel pipe and 95 m of ¾
inch stainless steel pipe. The exterior of the cavity is covered with 200mm thickness mineral wool insulation and enclosed with sheet metal.

Fig. 1: The 400m² "SG3 system(left) and the new 500m² "SG4" system (right) during experimental runs

![Steam cavity receiver design used in SG3 and SG4 dish systems](image)

Fig. 2: Steam cavity receiver design used in SG3 and SG4 dish systems

This receiver has been successfully fitted to the SG4 receiver and a preliminary test has been carried out on the 6th of August 2010. In this test the outer tapered section, which is intended to capture the focal region spillage, was covered with a ceramic blanket. Feed water was pumped to the receiver, and steam was allowed to vent off to atmosphere, while the dish was tracking the sun. Table 1 summarises the operating conditions obtained at steady state for three mass flow settings.

The overall thermal efficiency is defined as the ratio of the change in thermal energy of the fluid and the dish solar radiation captured by the receiver. The calculation is expressed in equation (1), where $A_{\text{dish}}$ is the dish mirror area, $I_N$ is the incident normal radiation at the time of the experiment, $r$ is the average reflectivity of the mirrors in the dish and $f$ is a factor of interception, as calculated by Lovegrove et al (2010). The
measured reflectivity was 0.91, direct beam radiation was measured at 920 W/m² and the operable mirror area was 489 m².

\[ \eta_t = \frac{\dot{m}_t (h_u - h_l)}{A_{dish} I_{rf}} \]  

(1)

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Table 1: Summary of experimental run carried out on the 6th of August 2010

HEAT FLUX DISTRIBUTION

A ray tracing study of the cavity receiver and the SG4 dish has been carried out using OptiCAD 10. An accurate representation for the heat flux distribution in the receiver pipe is an important aspect of the cavity receiver simulations. The optical model of the dish was developed by starting with approximately known parameters such as mirror panel slope error and dish space frame positional accuracy, which were then adjusted to give a better match to experimental flux maps on a planar target (Lovegrove et al 2010).

Fig. 3: Flux map of interior of cavity receiver

Fig. 3 shows the OptiCAD model of the flux on the interior surface of the steam cavity receiver. The cylindrical and taper sections are projected onto the figure on the left, where 0mm is the end of the cylindrical section and 1700 mm is the outer edge of the...
taper section where the feed water enters the cavity receiver. The back of the cavity is shown on the right.

The receiver pipes traverse horizontally along the surface on the left, from bottom to top. After reaching the top of the cylinder, it spirals from the edge of the back disk towards the centre and then exits the cavity receiver.

Since the models previously described assume a uniformly distributed heat flux, it is necessary to adapt the transient simulations to include the ray tracing results. The receiver pipe can be broken into three segments, each of which is an instance of the transient model capturing a fraction of the total flux hitting the cavity receiver.

The first segment starts at the cavity receiver entry and comprises the entire length of tube along the taper section and along the cylindrical section before hitting the intense flux band around the 800 mm mark. The second segment is the length of pipe exposed to the intense flux in the cylindrical section. The third segment is the remaining tube in the cavity travelling to the end of the cylinder section and the disk. See Table 2.

<table>
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<tr>
<th>Receiver Segment</th>
<th>Tube length [m]</th>
<th>Tube type</th>
<th>% of Total Flux</th>
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<tr>
<td>Taper and Cylinder</td>
<td>108</td>
<td>16 mm OD mild steel</td>
<td>3</td>
</tr>
<tr>
<td>Cylinder high flux</td>
<td>35</td>
<td>3/4” stainless steel</td>
<td>76</td>
</tr>
<tr>
<td>Cylinder end and disk</td>
<td>59</td>
<td>3/4” stainless steel</td>
<td>21</td>
</tr>
</tbody>
</table>

Table 2: Proposed subdivision of cavity receiver transient simulation

Flux percentages are relative to the total flux hitting the cavity and are linked to the area inside the cavity covered by each segment of pipe.

The amount of flux per unit of depth, represented as 1 horizontal pixel line in the cylinder/taper flux map is added and plotted in Fig. 4 (left). This is then integrated over the depth of the cavity to obtain the cumulative flux on the right. The slope on the curve of cumulative flux is used in the demarcation of cavity into segments of relatively homogeneous flux.

About 8% of the total flux hits the back of the cavity. This is accounted for in the cumulative flux at the beginning of the integration. This flux is considered approximately homogeneous over the segment of pipe covering it and of similar intensity than the cylindrical section of pipe that feeds it.
DESCRIPTION OF MODELS

Both models presented in this paper use a modelling method called moving-boundary method (Tummescheit 2002, Jensen 2003). The moving-boundary method is seen as a middle ground between a lumped parameter model and a finite element model, with the advantage of being more accurate than a lumped parameter model and less computationally expensive than a finite element model. Both models in this paper follow the moving-boundary approach where a series of time-varying sized regions contain the different phases of water as it traverses through the boiler.

Siangsukone Model

A detailed transient model and simulation of the Australian National University’s 400 m$^2$ Solar Generation 3 (SG3) system was developed by Piya Louis Siangsukone (2005). This model is derived from first principles for the steam cavity receiver of the SG3 system and validated by comparing experimental data with TRNSYS simulations.

It can handle sub-cooled water, saturated mixture and superheated steam coexisting in the evaporator and the dynamic disappearance/appearance of any of the regions. It also incorporates a pressure drop correlation to compensate for the extended length of the cavity receiver pipe.

The properties of the fluid are averaged within all regions. For the single phase regions a temperature gradient is derived to predict the dynamic behaviour of sub cooled water and super heated vapour. For the two phase region a specific volume gradient is derived to estimate the dynamic behaviour of the saturated water/steam mixture. This model is implemented in the TRNSYS simulation environment and it is possible to create
multiple instances connected in series to allow greater spatial resolution of conditions such as incident flux.

![Fig. 5: Control volume for Cavity receiver (Siangsukone 2005)](image)

**Model Derivation**

A simplified energy balance equation of a straight cylindrical pipe carrying a moving heat transfer fluid and subjected to an incident heat flux can be derived:

\[
\frac{dQ_s}{dt} = \dot{m}_h h_l - \dot{m}_h h_u = m_{rec} \frac{du_{rec}}{dt} + m_{fl} \frac{du_{fl}}{dt} + u_{fl} \frac{dm_{fl}}{dt}
\]

(2)

Where \(Q_s\) is the incident solar flux and \(Q_l\) are the radiation and convection losses from the receiver and the subscripts \(rec\) and \(fl\) pertain to the receiver and fluid respectively.

Radiation and convection losses can be derived from:

\[
\dot{Q}_l = G e A_{tube} (T_{cv}^4 - T_a^4) + U A_{tube} (T_{cv} - T_a)
\]

(3)

Heat losses are modelled using a geometrical correlation factor \(G\) for the radiative portion and a heat transfer factor \(U\) for the convective portion that have been determined experimentally. In equation (3) the subscript \(cv\) refers to the combined receiver and fluid control volume, as it is assumed that the receiver tube will be at the same average temperature as the fluid.

For the single phase regions, it is established that the rate of change of fluid mass is negligible in comparison with the other terms of the energy equation for a single time step, eliminating the need for the third term of the right hand side of equation (2). Additionally, single phase fluids are considered at constant pressure, which means that the internal energy of the tube and fluid only depend on the temperature of the control volume. This implies that the transient energy balance in a single phase region is linked to a rate of change in temperature. It is therefore possible to rewrite the energy balance in terms of a transient temperature change. Substituting (3) into (2) and rearranging for the rate of change of temperature of the fluid and receiver, equation (4) is obtained.
When the fluid is a mixture of saturated liquid and vapour, the rate of change of internal energy cannot only depend on temperature, as the average specific volume of the mixture changes with the fraction of vapour mass to total mass in the control volume. In addition to this, when the pressure is held constant, the transient temperature change is zero, due to the fluid being saturated. The energy balance of equation (2) is rearranged into equation (5), which links the change of energy in the two-phase fluid to the rate of change of its specific volume.

\[
\frac{dT_{cv}}{dt} = \frac{\dot{Q}_s - A_{tube}[G \varepsilon (T_{cv}^4 - T_a^4) + U(T_{cv} - T_a)]}{m_{rec}c_{rec} + m_{fi} \frac{A u}{AT}} + \dot{m}_i(h_i - h_{s})
\]

\[
\frac{dv}{dt} = \frac{\dot{Q}_s - A_{tube}[G \varepsilon (T_{cv}^4 - T_a^4) + U(T_{cv} - T_a)]}{m_{fi} \frac{A u}{AV} + (h_f - u_f) \frac{V_{fi}}{v^2}} + \dot{m}_i(h_i - h_{s})
\]

**Calculation of energy changes and phase changes**

The fluid properties are calculated at each time step of the simulation, starting at the inlet and progressing through the pipe. If the incoming fluid is sub-cooled liquid, then expression (4) is used. If the inlet fluid conditions are of a saturated mixture, then expression (5) is used. If receiver is subject to incoming flux, feed water passing through the cavity receiver will transition from single phase liquid, to saturated mixture and into super-heated steam. The outlet temperature is calculated from the inlet fluid temperature and the average control volume temperature:

\[
T_o = 2T_{cv} - T_i
\]

If the outlet temperature is greater or equal to the saturation temperature of the fluid at the control volume pressure, then a liquid region length is calculated. This means that before exiting the tube, the fluid changes phase and equation (5) has to be used to calculate the change in energy for the remainder of the tube. The fluid conditions at the exit of the liquid region, i.e. mass flow, temperature and pressure are the conditions at the inlet of the saturated region. It is important to note that the heat flux and receiver tube mass quantities in the transient equations (4) and (5) are linked to these calculated region lengths.

Subsequently, if the outlet specific volume of the mixture region is equal or greater than the specific volume of saturated vapour, then a liquid mixture region length is determined, and a temperature rate of change using (4) is calculated for the super-heated steam. The length of the super-heated region is the length of the tube minus the combined length of the liquid and saturated regions.
Pressure drops in the control volume

It has been mentioned that for each region, the pressure of the fluid is assumed constant and furthermore equal to the pressure at the inlet of such region. It is however needed to account for pressure drops in the cavity receiver due to the length of the tube. The Darcy-Weisbach static correlation (Potter 1991) is applied with a factor $K$ to fit the model with measured experimental data.

$$\Delta P = K \left[ \psi L_{\text{region}} \frac{\rho U_m^2}{4r} \right]$$  \hfill (7)

The friction factor $\psi$ is calculated from the formulation of Swamee and Jain (1976). Therefore the pressure at the outlet of each region will be the pressure at the inlet minus this calculated pressure drop. The recalculation of these pressure drops at each time steps allows a more refined determination of the properties of the fluid travelling through the receiver.

Proposed Model Description

The present study has been developed with the purpose of producing a model-based predictive control scheme for a dish steam cavity receiver plant. It combines a moving-boundary approach as used by Tummescheit (2002) and Jensen (2003) with a novel equation-switching strategy by McKinley (2008) and incorporates a pressure drop calculation by Yebra (2005).

![Control volume for cavity receiver in moving-boundary formulation](image)

Fig. 6: Control volume for cavity receiver in moving-boundary formulation

One of the aims of this model is to match the benefits of the Siangskone approach, i.e. dynamically switching the number of regions and accounting for pressure drops in the evaporator. This approach also provides explicit time derivatives of all the variables in the model, which is currently not possible in the existing model, providing greater insight into the dynamic behaviour of the receiver. This structure is also useful for devising control strategies for the operation of the system.

Derivation of Proposed Model

Fig. 6 represents the control volume of the steam cavity receiver. It is subdivided in three regions of sub-cooled water, saturated mixture and super-heated steam. Each region occupies a length in the pipe, $L_1$, $L_2$ and $L_3$ respectively. An unsteady balance of
fluid mass, fluid energy, fluid momentum and wall energy is derived for each region forming a system of ordinary differential equations of the form

$$Z \dot{x} = f(x, u, t)$$  \hspace{1cm} (8)

Where the state vector $x$ is expressed in (9), the matrix $Z$ contains the equation coefficients and $f$ is the forcing function of the system.

$$x = [L_1 \ L_2 \ P_1 \ \dot{m}_1 \ P_2 \ \dot{m}_2 \ P_3 \ \dot{m}_3 \ h_{\text{out}} \ T_{w1} \ T_{w2} \ T_{w1} \ \gamma]$$  \hspace{1cm} (9)

The state vector denotes that there are 13 variables simultaneously calculated in the system, including the lengths of the regions, mass flows, pressures and wall temperatures. The time derivative of all the state variables is explicitly declared in the set of equations, meaning that not only the state of the system is known but also the rate of change of all the states. Note that the void fraction $\gamma$ is considered time-varying, and therefore a state of the system. This is incorporated to allow the switching of equations when the control volume changes the number of regions present.

**Switching Criteria**

Analogously to Siangsukone's approach, it is desired that the model handles all operation modes of the system. It is therefore required that the system models the appearance/disappearance of each region. In this work a set of conservation equations for each case depicted in Fig. 7 has been developed. Furthermore, the diagram shows that only certain transitions are allowed between states, and these are consistent with a progressive heating or cooling of the steam cavity receiver.

Fig. 7: Possible switching combinations for steam cavity receiver model

The cases where no sub-cooled region is present are required when multiple instances of the model are connected in series in a simulation, for the purpose of better capturing the incoming flux hitting the receiver.
In order to transition from one set of equations to the next, a set of rules that examines the inlet and outlet conditions of the control volume is applied. For example, consider a receiver which is currently being heated by an incident heat flux, and contains regions of types 1 and 2. If the fluid continues to heat, eventually a super-heated region will appear. Therefore it is of interest to know if there exists complete evaporation in the saturated region. Since the specific enthalpy of the fluid depends on the void fraction $\gamma$, it is examined against the void fraction at full evaporation:

$$L_2 (\gamma_{sat} - \gamma) < L_{min} \quad \text{and} \quad \frac{d\gamma}{dt} > 0$$

In expression (10) $L_{min}$ is an arbitrarily small value and the void fraction is multiplied by $L_2$ in order to ensure the comparison is performed at a representatively sized saturated region. If the void fraction of the saturated region is arbitrarily close to the void fraction at complete evaporation, and continues to increase, then the system is switched to a 3 region boiler.

**MODEL COMPARISON**

There are many similarities between Siangsukone's model and the proposed model, mainly due to the fact that the proposed model has to, at least, match all the features of Siangsukone's model. These include: A transient response heat transfer with multiple phase flow, incorporation of radiative and convective losses in the receiver, pressure drop correlations and the ability to connect multiple instances of the model in series to better capture the incoming flux distribution.

The proposed model is an improvement on Siangsukone's model for a number of reasons. Firstly, as it is expressed as a set of differential equations, control theory can be applied on the model to derive optimum mass flow profiles. The proposed model is also more detailed in calculating the fluid properties and heat transfer mechanisms, for instance, Siangsukone's model assumes the pipe wall and the fluid have the same temperature. There is more versatility in the model calibration as many parameters are considered for each region (e.g. heat transfer coefficients) as opposed to system wide. This however means that there is potential for more complexity in the calibration of the simulation, when validating with experimental runs.

**SIMULATION OF THE CAVITY RECEIVER IN TRNSYS**

A simulation of the SG4 system during the experimental run on the 6th of August 2010, using Siangsukone's model in TRNSYS 16 is presented in Fig. 8. The simulation takes the measured feed water mass flow, insolation, ambient temperature and other measurements to calculate the transient response of the receiver. In the experimental run, mass flow rates are maintained until the fluid temperature of the receiver reaches steady state.
In this experiment, the steam is blown off to ambient after passing through the receiver, which affects pressure drop over the receiver length, in comparison to a receiver connected to a steam generation system. The graphs show that the mass flow and inlet pressure are closely related. The feed water mass flow exhibits small variations around each set point, which are also reflected and amplified in the inlet pressure. This creates a complex pressure drop profile in the receiver. The simulation assumes less drastic changes and variations in pressure and this may cause the fluid properties and temperatures to be estimated incorrectly.

There is also a time delay in the simulated temperature in response to changes in mass flow. This indicates that the time constants of the system may not have been accurately reproduced in the simulation, or that there is a time lag in the response due to other reasons.

It is apparent that the experimental results indicate a system with a much longer time constant than the model predicts. Siangsukone's model achieved good agreement between TRNSYS simulations and the SG3 dish using the same receiver, after careful calibration of the system parameters. Those experiments were carried out with steam fed to a reciprocating steam engine that maintained back-pressures up to 4 MPa. It is suspected that the low pressure exit conditions are the reason for the lack of agreement for this first test.

CONCLUSION

This paper reports on the ongoing research by the ANU on direct steam generation with dish concentrators. Two models for the transient simulation of steam cavity receivers for dish concentrators have been presented. The second model is currently under development and it is intended to be an improvement on Siangsukone's model, for
system simulations and automatic control. The Siangsukone model has previously been successfully applied to predicting the behaviour of a monotube boiler receiver on the ANU 400m² dish.

The first experimental run of the 500m² SG4 system with the same monotube steam system shows receiver efficiencies of 84% at 500°C and up to 94% at lower temperatures. This is encouraging, particularly given that the operating configuration is not yet optimised.

There is currently only fair agreement between experimentally measured steam receiver outlet temperature and simulations, using a model developed for the SG3 dish. Ongoing work includes fine tuning of the model by adjusting input parameters to match SG4 and obtaining successful simulations with the proposed model.

REFERENCES


Jose Zapata is a PhD student at the Solar Thermal Group, Australian National University. His research interests include dynamic modelling, control and optimisation of solar thermal energy systems.