Steam-circuit Model for the Compact Linear Fresnel Reflector Prototype

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Abstract

The Compact Linear Fresnel Reflector (CLFR) is a linear-concentrating solar thermal energy system currently in the prototype stage at the Liddell power station in the Hunter Valley, NSW. A system-level thermo-fluidic model for the CLFR has been developed in order to predict steady-state operating conditions, required pumping power and pressure, mass transfer around the system and likely start-up, shutdown and cloud transient effects. Flow regimes within the absorber at different irradiance level are found. The interaction of the absorber pressure, irradiance and pump speed is discussed. A simple model for steady-state system pressure is presented, built around an assumed controller that maintains the absorber steam exit quality as 80\%. Finally, a more complex steady-state model, in which the total mass of fluid in the system is used to drive fluctuations in system pressure, is presented.

1. INTRODUCTION

1.1. Background

The Compact Linear Fresnel Reflector (CLFR) was first conceived of in 1992-1993 and was patented in 1995 (Mills & Morrison 1996). It is intended to be a low-cost solar thermal energy system for medium- to large-scale thermal energy capture. Parallel mirrors at ground level focus light onto a linear radiation-absorber positioned above them. The array is ‘compact’ because when a field of parallel mirrors has more than one absorber line, alternate mirrors in the centre of the mirror field between the absorber lines point in alternating directions in such a way as to reduce shading between adjacent mirror lines, thereby allowing denser packing of mirrors, as shown in Figure 1.

![Figure 1: one 'module' of a CLFR system. Inset shows the denser packing of mirrors made possible by pointing alternate mirrors at different absorbers.](image-url)
The system is low-cost by virtue of the off-the-shelf materials and components used throughout. Mirror focus is achieved by elastic deformation onto a corrugated roofing material substrate. Mills et al (2003, 2004) have given further details of the structure, optics, and construction.

Heat transfer to a working fluid takes place to a parallel array of small diameter pipes at the focal plane of the Fresnel mirror array. It is expected that in most cases, the flow would be in the two-phase regime. Pipes are closely packed side-by-side in order to absorb as much of the direct incident radiation as possible. A sealed air cavity is located below the pipes, to reduce convective losses from the inverted hot surface of the absorber pipes. The pipes are coated with a low-emissivity paint.

In a fully solar power generation facility based on the CLFR, the steam outlet from the absorber array ideally would be run directly through a turbine, however depending on the ability of the system to be controlled to produce constant-quality steam at varying radiation levels, a heat exchanger may be necessary, even though undesirable for its expense.

1.2. CLFR Prototype

A prototype CLFR system is in the final stages of construction at the Liddell power station near Singleton, in the Hunter Valley, in New South Wales. A photograph is shown in Figure 2.

Figure 2: Photograph of the CLFR prototype system at Liddell power station. There are 12 mirror rows focusing radiation at a single elevated absorber line. The height of the absorber is 8m. Mirror tracking is achieved by driving the ‘hoops’ via an independent motor mounted at each mirror row.

The prototype constructed is the first stage of a three-stage process that will eventually see Macquarie Generation equipped with a large array giving approximately 100MW thermal of solar energy collection, which will be used to provide final-stage boiler feed water heating. Hu et al (2003) studied the power station thermodynamics and it is argued that the electrical generation equivalence of the array thus installed is approximately 35 MWe.

1.3. Similar systems elsewhere

The CLFR system is similar to the SolarMundo Fresnel concentrator, which reached the prototype stage in Belgium (de Lalaing et al, 2001).

The Direct Solar Steam (DISS) project at the Plataforma Solar de Almería is another related project. This is a trough concentrator based on the SEGS system but replacing the heat transfer oil of the SEGS system with direct steam generation in the absorber tube (Eck et al 2003, Zarza et al 2004).
The control system for the DISS system is interesting. This system uses a series of parabolic troughs approximately 500 m long. Initially it was designed to produce superheated steam in a once-through-the-collector fashion. This was found to be unstable, and there was a risk of thermal fatigue damage to the collector tubes as the boundary between the constant-temperature saturated-steam stage and increasing-temperature superheated-steam stage moved up and down the length of the collector.

For this reason, recirculating flow and water injection along the absorber were added to the DISS system. Such an approach to system control may be necessary in the CLFR project, although it is hoped that with sufficiently high flow rates, the outlet steam will remain within the saturated region, and these thermal transients will not be a concern.

It is also important to ensure that a steady two-phase flow pattern occurs at the outlet. Intermittent flow patterns, such as slug flow, will lead to instability and would put components such as pumps and heat exchangers at risk of damage.

2. STEADY-STATE SYSTEM MODEL

The focus of the present work is to determine the operating points of the final system when connected to the power station. In particular, the required pumping power and surge tank capacity need to be determined. Also, the behaviour of the system during start-up and shutdown, as well as during cloud transients, must be simulated, in order that a suitable control strategy can be devised.

A steady-state system model has been built with C++ code, and is depicted in Figure 3. An overview of each of the components being modelled follows.

Figure 3: System model for the CLFR steam circuit
2.1. Absorber

The absorber in the prototype model consists of 16 parallel DN 25 pipes, each 60m long, made of 304 stainless steel, and mounted side-by-side, for a total absorber width of 575 mm. Absorber pipes in the prototype are connected via a set of manifolds into a four-pass configuration, so that the total length of a flow path is approximately 240m.

About 300mm beneath the absorber surface, a plastic film will enclose an air cavity, acting to limit convective losses from the hot absorber. Pye et al (2003), Reynolds et al (2002) and Reynolds & Jance et al (2003) show modelling results for the air cavity. The cavity model from Pye et al (2003) has been used here.

Inside the absorber pipes, wet steam at approximately 60 bar at 270°C with a final quality of 0.8 to 1.0 will be produced, with a pressure drop of 2 to 10 bar as it passes through the absorber.

Modelling of the forced-convection boiling process has been performed using the method of Odeh (1999) and Reynolds (2002). This entails dividing the flow path into a series of one-dimensional elements, and then solving the pressure drop and enthalpy rise for each element in sequence, using local flow properties based on the outlet of the previous segment. For the pressure drops, the Colebrook equation is used in single-phase elements, and the Martinelli-Nelson correlation (Martinelli & Nelson, 1948) is used in two-phase elements. Likewise, for the heat transfer coefficients, the Dittus-Boelter and Gnielinski correlations (Incropera & DeWitt, 1996, p. 444 ff.), and the Gungor-Winterton method (1987) are used.

Solution of the absorber model in isolation shows that under normal steady radiation, unsteady two-phase flow will not be expected at the absorber outlet (Figure 4, and Mills et al, 2004). However, this does not prove that during transient conditions unsteady flow patterns will not occur, so further analysis is required, necessitating a full transient system model.

![Figure 4: Flow regimes expected along the absorber pipes in the full-scale CLFR. For each irradiation level, the flow rate has been adjusted to give outlet quality of 0.8.](image)

2.2. Heat exchanger

A heat exchanger is required for the transfer of energy from the solar array to the power station. This is required to protect the reliability of the power station, and ensure that contaminated water doesn’t
enter the turbines. It is hoped that a stand-alone CLFR system would not need a heat exchanger.

The heat exchanger is effectively a sub-cooling steam condenser. The component is modelled in two parts: firstly the latent heat transfer that occurs as the steam on the hot-side is reduced to saturated water, and secondly the sensible heat transfer that occurs as the saturated hot-side water is sub-cooled (Figure 5). This calculation requires an iterative procedure, since the hot-side outlet temperature must be guessed and an error based on the heat transfer rate in the sensible stage minimised.

![Figure 5: Heat transfer in the heat exchanger](image)

A sizing calculation is performed based on expected hot-side inlet conditions, design hot-side outlet sub-cooling, and required total heat transfer. For the prototype system, taking overall heat transfer coefficients to be $U_f = 4000 \text{ W/m}^2\text{K}$ and $U_l = 1200 \text{ W/m}^2\text{K}$ (Incropera & DeWitt, p. 586), the approximate required heat transfer area is $12 \text{ m}^2$. For the full-scale system, the heat exchanger will be large, and a major cost item.

### 2.3. Pump

The pump is modelled by an ideal quadratic performance curve. A reference pump curve defined the speed $N = N_{\text{ref}}$ is taken to be a quadratic $H = f(Q)$, with a maximum head at $Q = 0$ of $H = H_{\text{max}}$, and a specified flow rate $Q_{\text{ref}}$ at $H = 0.9 H_{\text{max}}$.

The isentropic efficiency for the pump $\eta_i$ is also specified, allowing all of the pump outlet conditions to be calculated when the pump speed is specified along with the inlet conditions and flow rate.

The above constraints allow the pump to be solved for any chosen speed using Equation 1:

$$H = C_0 \frac{N}{N_{\text{ref}}} + C_1 \frac{N}{N_{\text{ref}}} Q + C_2 Q^2$$

### 2.4. Other Components

A simple model for the steam separator is used: if the inlet is saturated, the liquid component goes to the throttle, and the gas component goes to the heat exchanger; if the inlet is superheated, then all flow goes to the heat exchanger; if the inlet is sub-cooled, then all flow goes to the throttle. Obviously in a real separator, these assumptions for sub-cooled and superheated flow are not correct, and will need to be corrected for the transient model.

The surge tank can be ignored for the purpose of this steady-state system model. However, when the mass transfer from component to component is included in the system, this component will be required for an accurate representation of the system. The surge tank will exist in reality too. Its purpose is to contain during normal operation the water that will be present in the absorber and associated pipe work when the system is starting up. It will also act as a time-lag component, which may aid controllability when trying to achieve a constant temperature at the heat exchanger cold-side outlet.
The **throttle** is required to bring the pressure at water outlet of the steam separator in line with the pressure at the pump inlet. The throttle is isenthalpic. There will be a pressure drop here since a small pressure drop has been allowed for in the heat exchanger model.

Where the throttle outlet joins the pump inlet, a **mixer** is required in the model. The mixed outlet flow is chosen to have the mass-weighted average of the specific internal energy of the two inlet streams, and inlet pressure of the surge tank outlet.

### 3. MODEL RESULTS

The overall model has been solved using the approach shown in Figure 6. There are a number of model parameters which can be used as inputs into the calculation. These are shown in Table 1. Other system parameters which were taken as constants in this study are shown in Table 2. The method of solution was to take an assumed operating point at the absorber inlet, then to move around the cycle, solving each component in turn, based in each component’s inlet conditions. Where components had variable parameters as yet unspecified, such as pump pressure rise, these values were guessed. Then, when the whole cycle had been solved in this way, the Brent solver algorithm (Press et al, 2002) was used to adjust the parameters such as pump speed so that the steam pressure and enthalpy at the end of the cycle match those at the start of the cycle.

![Figure 6: Calculation procedure for the steady-state model. Bold boxes show input/outputs that can be specified. The dotted lines show predictor/corrector value-pairs that can be used along with the circuit calculation to determine the remaining unknowns. Therefore, two operating parameters are required in order to find a valid operating point using the above method.](image)

The parameters shown in Table 2 were initially chosen since it is relatively simple to solve the system with these values fixed.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$</td>
<td>Irradiation ($\text{W/m}^2$)</td>
<td>500 – 1100 $\text{W/m}^2$</td>
</tr>
<tr>
<td>$p_i$</td>
<td>Pressure at absorber inlet (bar)</td>
<td>50 – 90 bar</td>
</tr>
<tr>
<td>$\dot{m}_p$</td>
<td>Pump mass flow rate (kg/s)</td>
<td>0.4 – 1.0 kg/s</td>
</tr>
<tr>
<td>$x$</td>
<td>Absorber outlet steam quality</td>
<td>Aim for 80%</td>
</tr>
</tbody>
</table>
An absorber inlet temperature of 240 °C is used as the first value for the iterations.

Figure 7 shows a series of results for operating conditions where the inlet pressure of the absorber is 60 bar. As the pump flow rate is increased, the required pumping pressure increases. Higher radiation also requires a higher pumping pressure. Note that these are artificial conditions since the absorber inlet pressure would naturally 'float' to a different pressure if the pump flow rate were altered. In effect, by fixing the absorber inlet pressure, a variation in the mass of fluid in the system is being permitted.

The upper half of Figure 7 shows the steam quality at absorber outlet for the same range of conditions. As the pump flow rate increases, the outlet quality at the absorber decreases. At high radiation, the outlet is superheated at low flow rates. For low radiation and higher flow rates, the exergy in the hot-side flow was not sufficient to achieve the desired cold-side outlet temperature, so no solution can be shown.

The following diagram, Figure 8, shows an interesting effect: as the pump outlet pressure increases, the pressure drop over the pump decreases. One might expect the pressure drop in such a situation to decrease, however, it appears that a two-phase flow phenomenon is occurring. At higher pressure, the steam is more highly compressed and this is resulting in a lower average friction factor for the flow in the absorber. Figure 7 is for a constant pump flow-rate and varying outlet quality.

Table 2: System properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Absorber length</td>
<td>60 m</td>
</tr>
<tr>
<td>Pipe configuration</td>
<td>16 x DN 25 SS304 pipes</td>
</tr>
<tr>
<td>Optical concentration</td>
<td>40</td>
</tr>
<tr>
<td>Optical efficiency</td>
<td>0.8</td>
</tr>
<tr>
<td>Absorber inlet temperature</td>
<td>240 °C</td>
</tr>
<tr>
<td>Heat exchanger contact area</td>
<td>12 m²</td>
</tr>
<tr>
<td>Overall heat transfer coefficient in latent heat transfer</td>
<td>1000 W/m²K</td>
</tr>
<tr>
<td>Overall heat transfer coefficient in sensible heat transfer</td>
<td>50 W/m²K</td>
</tr>
<tr>
<td>Heat exchanger cold-side inlet temperature</td>
<td>200 °C</td>
</tr>
<tr>
<td>Heat exchanger cold-side outlet temperature</td>
<td>242 °C</td>
</tr>
<tr>
<td>Pump isentropic efficiency</td>
<td>0.65</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>20 °C</td>
</tr>
<tr>
<td>Pump speed</td>
<td></td>
</tr>
</tbody>
</table>

(Pump design) At a reference speed of 2000 RPM, the maximum pump pressure is 10 bar at zero flow and at this same speed but a flow rate of 2 kg/s, the pump generates 9 bar pressure. The density at reference conditions was taken as 704 kg/m³.
Figure 7: Pump pressure rise and absorber outlet quality for a range of irradiation levels, and a series of pump flow rates. The absorber inlet pressure as set at a constant 60 bar for all of these cases.

Figure 8: Pressure rise across the pump for a fixed pump flow rate of 0.6kg/s, with varying pump outlet pressure and radiation.
3.1. Effect of fixed total mass in the system

A preliminary study of the effect of a fixed total mass of water in the system has been performed by considering only the volume of the absorber, a pipe between the absorber and the heat exchanger, and a pipe from the heat exchanger to the surge tank. This corresponds to points (3) and (6) on Figure 3. The approach used was to take an operating point as before, then to adjust the (previously arbitrarily fixed) absorber inlet pressure until the mass of water in the system is the desired (fixed) amount.

Using 50 m-long, DN 50, schedule 10S, 304 stainless steel pipe in each of these locations, with an external heat transfer coefficient of 0.5 W/m², and then setting a fixed mass of 185 kg of water in the system, the operating pressure for fixed fluid mass and for fixed absorber outlet quality $x=0.80$ are found as shown in Figure 9. This shows the expected increase in the pressure in the system following from a greater amount of heat being transferred by steam of the same quality but in the same sized space; the steam must be compressed in order for the required steady-state heat transfer to occur.

![Figure 9: Variation of operating pressure and system flow rate with irradiance](image)

More detail will be added to the modeling to allow for the volume of the heat exchanger and surge tank as well as other pipe-work. Under the fixed-total-mass-of-water constraint, the system pressure was found to be quite sensitive to pump flow rate.

4. FURTHER MODELLING

The above steady-state system modelling does not give as 'natural' a picture of the behaviour of the system as would be liked. In order to determine the pressure at which the system will operate, given a certain fixed mass of water inside the system, the behaviour of the system must first be solved for some arbitrary variables (absorber inlet pressure) and then a root solving process applied to find the operating pressure at which the system would operate for a given fixed mass of water. There is no obvious way to fix this situation without moving to a full transient model for each component in the system. This would allow the system to move between varying operating points, all the time observing mass conservation through each flow segment.
Another reason for wanting to move to a transient model is the need to predict cloud transient effects and startup and shutdown behaviour for the system. A steady state model is unsuitable for accurately predicting behaviour in these scenarios.

At present the system model does not include calculation of two-phase flow regimes. This will need to be added before a transient model is developed since two-phase flow regimes at the absorber outlet during low-radiation periods could be a concern.

Line losses will be included as there will be a significant length of pipe connecting the full-scale CLFR to the power plant, and these will add pressure and enthalpy losses to the system.

Mass transfer around the system needs to be included throughout the model, so that surge tank sizing can be performed and all of the model components should be able to robustly handle supersaturated flow, so that dry-out effects can be simulated.

5. CONCLUSIONS

A steady state steam-circuit model for the compact linear Fresnel reflector was presented and the interaction of the system operating pressure, flow rates, solar radiation intensity and pumping head were studied. At higher operating pressure, there is a two-phase flow effect predicted by the model that results in pump head being reduced. When the total mass of water in the system was used to drive the operating pressure of the system in the steady state, and the outlet steam quality was fixed at 80%, the operating pressure was shown to increase linearly with irradiance.

The modelling shows plausible behaviour for the system as a whole, however the development of the system model is ongoing, since the choice of system parameters does not yet allow ‘natural’ behaviour of the model to occur.

The steady model will be of assistance in sizing some components, however in order to tackle the question of a good control strategy for the steam circuit, and to investigate the behaviour of the system during system startup, shutdown, and during ‘cloud transients’, a detailed transient model will be developed.

REFERENCES


