Trnsys Modeling of a 100 MW Combined Cycle Concentrated Solar Power Plant

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Abstract

We present further work on our Trnsys modeling of a combined cycle Concentrated Solar Thermal power plant. The model presented in the first report produced average daylight power of around 5 MW .

We have now run the model for a full year, and adjusted control strategies and thermal storage based on the winter months of weakest sunshine. We also adjusted the control strategy so that the gas cycle and combustion chamber ran for a constant time period each day, rather than switching on and off triggered by solar flux levels in the morning and evening.

We also did careful checking of both the gas and steam cycles to ensure that the model parameters and generated power were in sensible ranges, and that they matched other third party model parameters for similar steam and gas turbine power cycles.

Finally we scaled all parameters up by a factor of 20x to model a plant that had an average year round capacity of 81 MW Electric and a peak capacity of 158 MW. We also added a blower to the concrete thermal storage to account for the parasitic losses incurred when the thermal storage was being discharged.

1 The model

This plant model is based on the SUNSPOT concept proposed by Prof D Kröger. [Krö08]. It is modeled in TRNSYS [Kea05], a modeling system designed for thermal modeling of buildings, but now also widely used in the CSP industry[BMC08] , [QZY08]. The initial model on Figure 1 on page 2 consists of a field of 200 heliostats of size 100 m^2each. These focus onto an air receiver. Air is compressed to 15 bar and fed through the receiver at 75 tons/hr when the solar flux onto the receiver reaches $1x10^7$ kJ/hr or roughly 3 MW thermal. After the receiver there is a combustion chamber that raises the air temperature to 1100 C. This

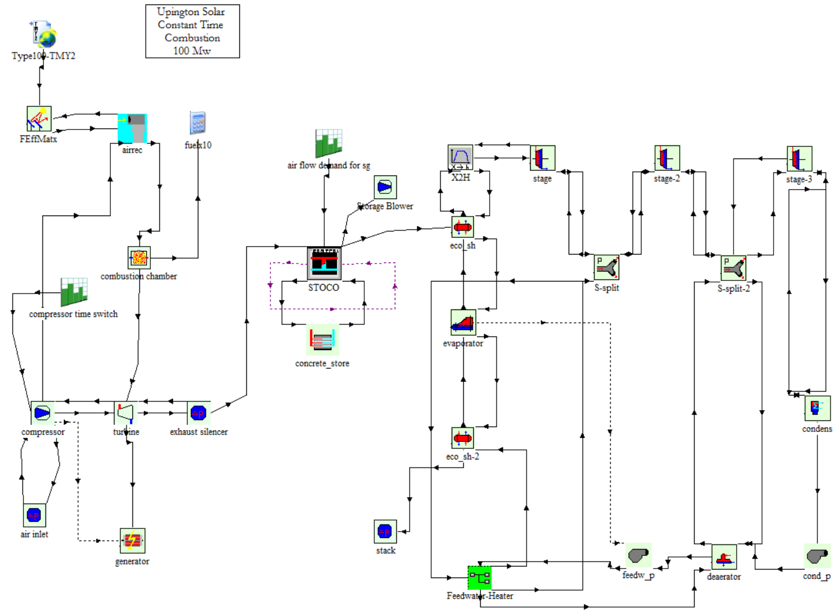

Figure 1: Plant Schematic

hot air is then fed through a turbine and exhausts from the turbine at roughly 500 C. This exhaust air is then split (depending on steam demand) between a Heat Recovery Steam Generator (HRSG) and a thermal storage consisting of 1000 tons of thermal concrete with pipes running down its length. We initially tried to use the thermal store composed of a rock bed, but problems with this model led us to use the concrete heat store. Substantial work has been done at Stellenbosch on modeling a rock bed storage [AKF09] , and it would be a useful piece of future work to incorporate a rock bed storage into the model

During the day (when we have the compressor running) the raw gas turbine power was roughly 10 MW and the turbine electrical output about 4MW. The other 6MW was used to drive the compressor, and was lost in electrical inefficiency in the generator. The dip in the electrical power output is presumably due to the slightly different thermodynamic properties of the gas (pure air at noon, more combustion products in the morning and evening).

The HRSG and thermal store is controlled by a steam demand pattern which we set empirically. We requested 20 tons/hr of hot air into the HRSG throughout the day, with a ramp up to 75 tons/hr during Eskom's peak load period. The ramp up from 20 to 75 tons/hr occurred over the hour from 15h00 to 16h00, and the ramp down back to 20 ton/hr occurred between 20h00 and 21h00. This pattern is easily configurable for any demand pattern or pricing model that a grid operator might demand. When there is hot gas flowing from the turbine exhaust, the thermal control decides whether to charge the thermal store or provide gas directly to the HRSG based on the steam demand. The thermal store can either be in charge mode or discharge mode, it cannot do both simultaneously. This model includes a water condenser, mainly because there was not an air condenser component in the STEC libraries. This might be a useful addition to the libraries to write such a component. Based on South African conditions, we would probably want to move to an air condenser[Krö03], which would save water, but at a slight decrease in efficiency.

2 Results for a full year

2.1 Air Compressor controlled by solar flux

We ran the initial 5 MW model for a full year. Initially we started the air compressor when the solar flux reached a threshold level (1x10^7 kJ/hr), and switched it off again in the evening when it fell below the threshold again. Firstly we examine performance after the system has charged up the thermal storage (1000 tons of thermal concrete) for 20 days, and examine the temperatures and flow rates of the steam cycle at dates around 20 and 21 January. In Figure 2 on page 4 and Figure 3 on page 4 we see the storage charge and discharge during summer. We see that the steam generator hot gas flow continues throughout the night, and the temperature of the hot end of the storage as shown in Figure 4 on page 5 never drops below the cut off value (which we set at 350 C). The temperature of the cold end of the thermal store is also plotted and appears at the bottom of the graph. We also note that the air cycle and combustion chamber run for about 10 hours per day (dotted line in Figure 2 on page 4).

In Figure 6 on page 6 to Figure 9 on page 7 we see the same graphs for two days in mid winter. Firstly we note that the air cycle only runs for about just under eight hours. The storage gets completely depleted and the steam generator “runs out of steam” during the early morning hours. We could try increasing the size of the storage, but there is just not enough heat input into the system to drive the steam generation during the long winter nights.

The corresponding power graphs for summer are shown in Figure 10 on page 8 and for mid winter are shown in Figure 11 on page 8. In these graphs the dotted line is the the gas turbine gross output power (before powering the compressor). The dashed line the is gas turbine electric output power and the solid line is the combined gas and steam electric output power. Note that even at the peak of power production, the electric power from the steam cycle is less than half the power from the gas turbine. In these graphs we also note the fall off in electric power around hour 4348 (a mid winter night).

We did attempt to double the size of the thermal store to avoid this power fall off, and this did give us an extra three hours of running in the mid winter nights, as seen in Figure 12 on page 9 but it was still not enough to prevent

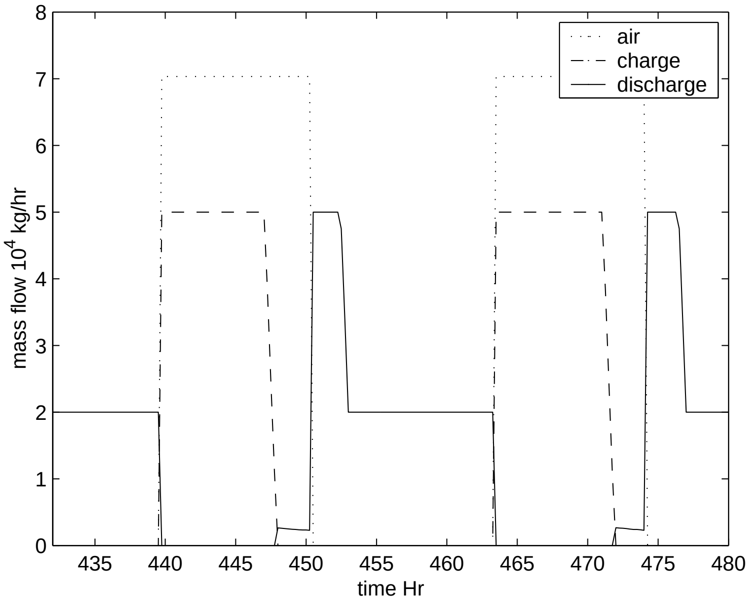

Figure 2: Jan Air flow rates after 20 days. Solar Switch

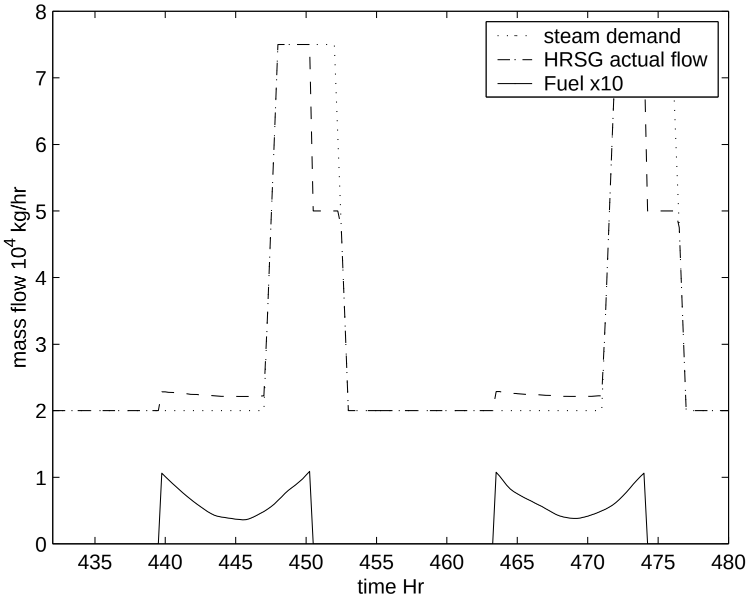

Figure 3: Jan Steam generator demand and flow. Solar Switch

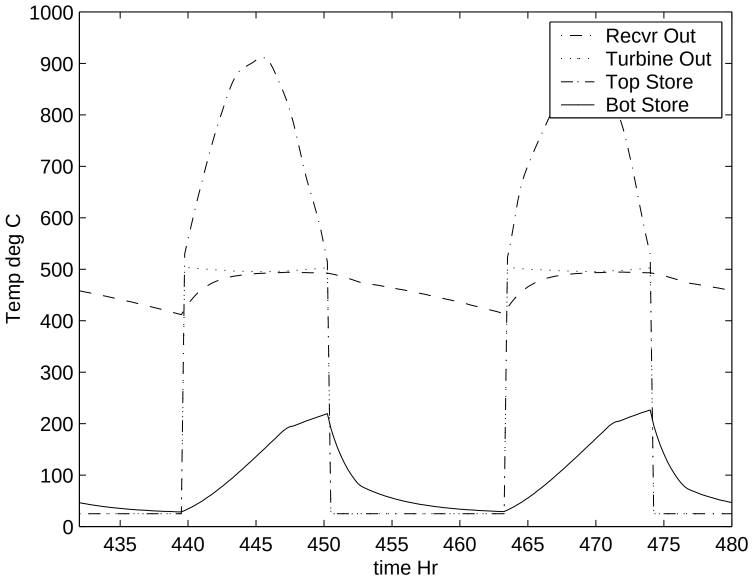

Figure 4: Jan Temperatures of Turbine and Storage. Solar Switch

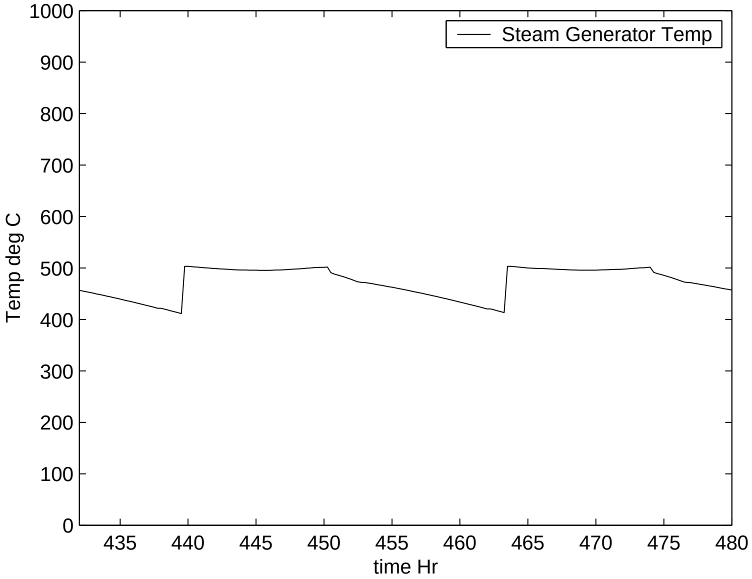

Figure 5: Jan Temperature of Steam Generator air. Solar Switch

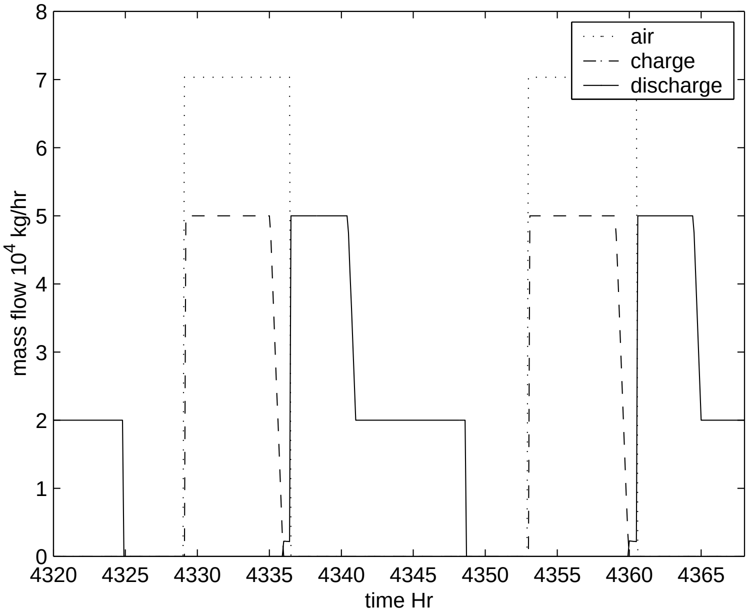

Figure 6: Flow rates from Storage in midwinter. Solar Switch

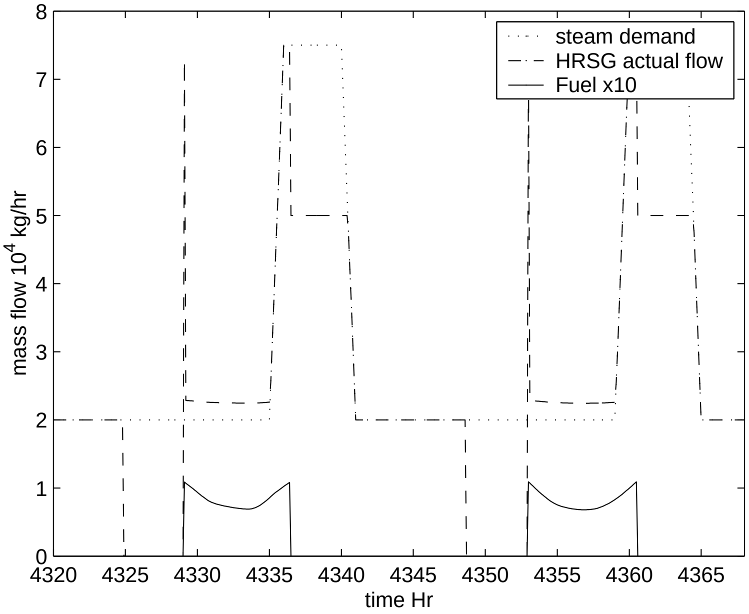

Figure 7: Midwinter Steam generator demand and Flow. Solar Switch

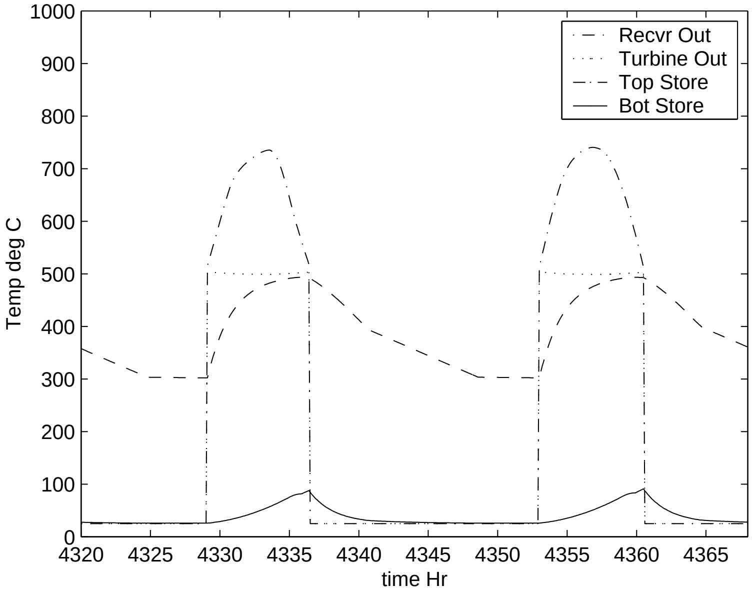

Figure 8: Midwinter Temperatures of Turbine and Storage. Solar Switch

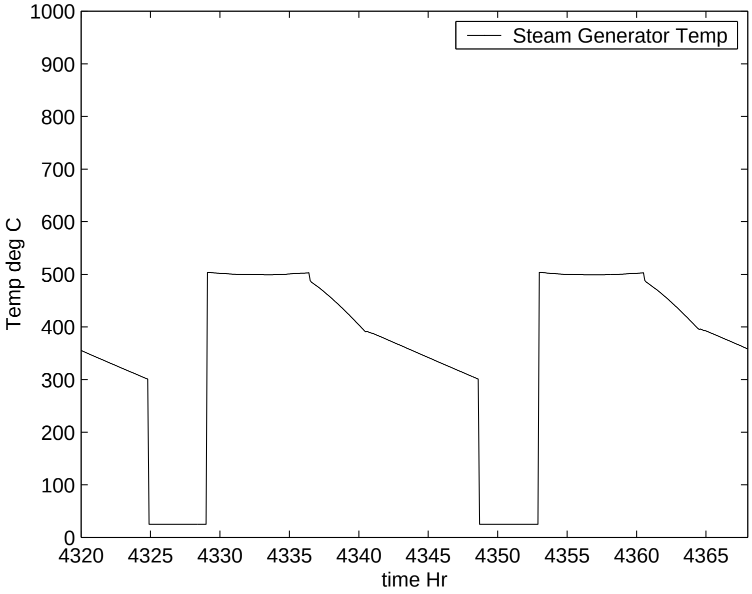

Figure 9: Midwinter Temperature of Steam Generator air. Solar Switch

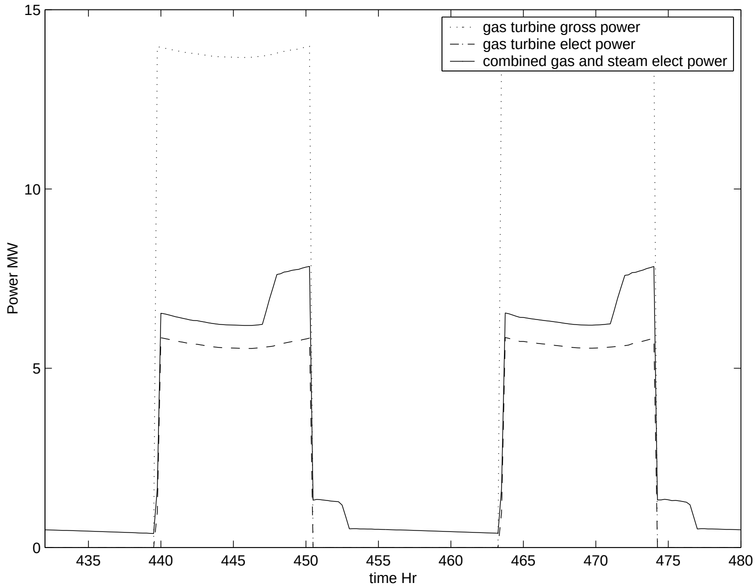

Figure 10: Gas Turbine and combined Electric Power Mid Summer. Solar Switch

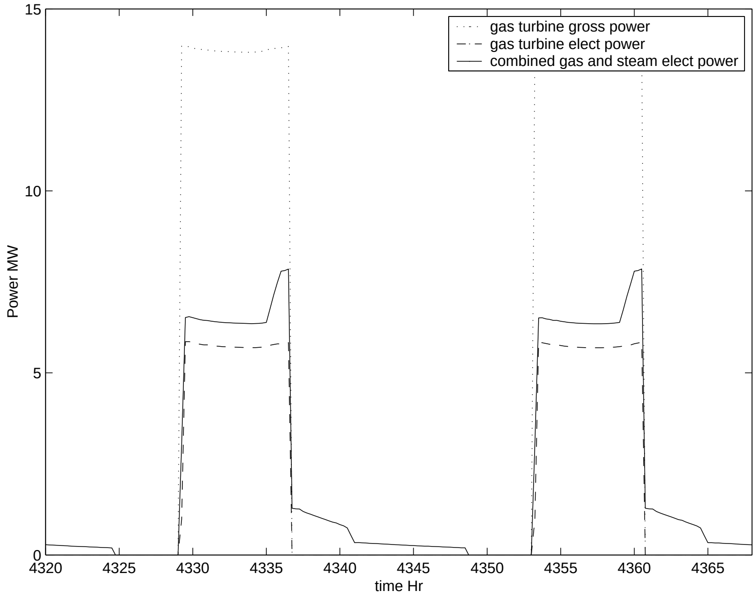

Figure 11: Gas Turbine and combined Electric Power Mid Winter. Solar Switch

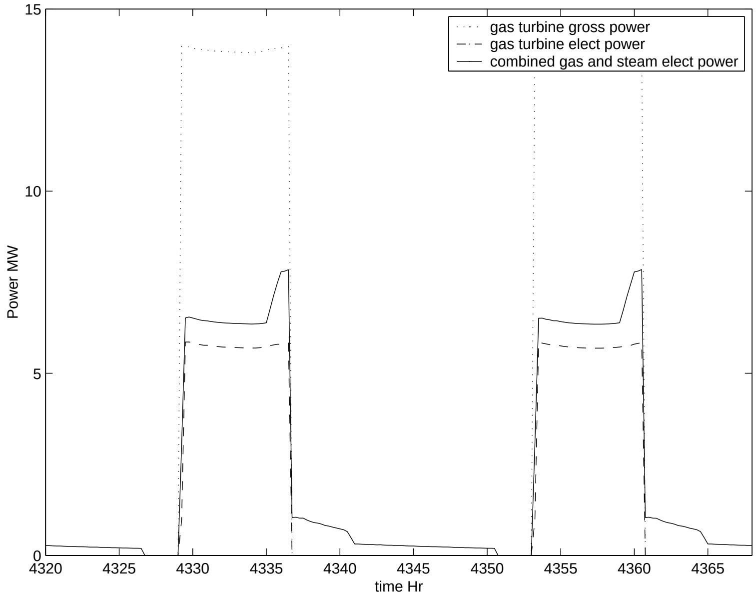

Figure 12: Gas Turbine and combined Electric Power Mid Winter Double Storage. Solar Switch

Period	Whole year	Jan	July
Average Elect Power MW	2.82	3.25	2.36
Average Efficiency %	41	40	41
Solar fraction %	49	54	40
Steam Fraction %	24	24	24

Table 1: Efficiencies for system switched on solar flux

power loss in the early morning hours. We then concluded that the total input power into the system in winter (solar and fuel for the combustion chamber) was insufficient to power the plant as required for a whole day. After this we went to a model where the combustion chamber and air compressor were switched on at a constant time each day, and the power input to the system stayed roughly constant through summer and winter, the only thing that varied between the seasons was the solar fraction.

The steam generator hot gas temperatures for a full year are shown in Figure 13 on page 10 , where we can clearly see the bottoming out during the winter months. In fact bottoming out starts around hour 2500 (mid April) and continues till hour 6000 (August).

The average efficiency and solar fraction for this solar switched configuration was 41% and 49% respectively, as shown in Table 1 on page 9

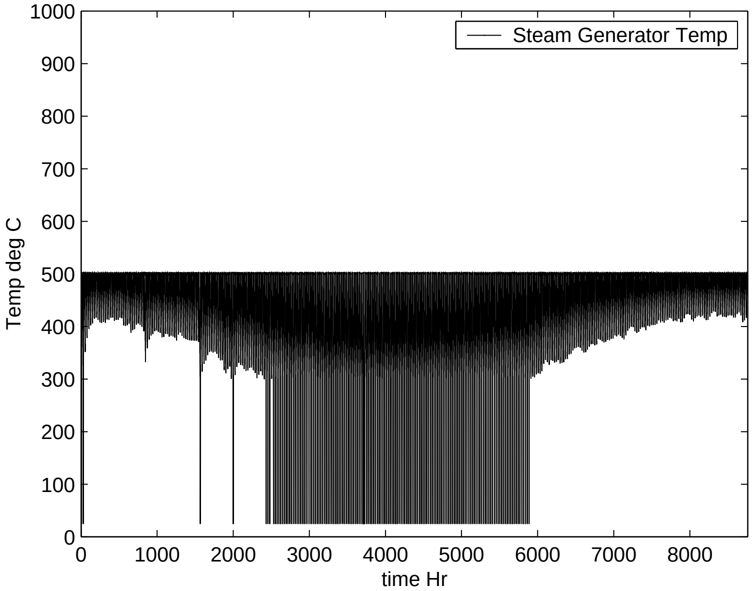

Figure 13: Steam generator Hot Gas Temperatures Whole Year. Solar Switch

2.2 Air Compressor controlled by time switch

In this configuration, the air compressor and combustion chamber were switched on at the same time each day, summer and winter. This enables other components in the plant to be correctly sized, as the total energy into the system day by day remains the same throughout the year. The air flow ramps up during 5h45 to 6h00 from 0 to 75 tons/hr, and ramps down again from 18h45 to 19h00. The initial charging of the thermal storage in shown in Figure 14 on page 11 and Figure 15 on page 11. From this graph we can see that when charging a completely cold thermal store, only on the first night do we loose steam capability, and steady state is reached after roughly 5 days.

Graphs of the flow rates for this constant time system in midsummer and midwinter are shown in Figure 16 on page 12 to Figure 23 on page 15 respectively. These should be compared to Figure 2 on page 4 and Figure 6 on page 6which show the same variables, but the system switched on and off by solar flux.

With this extra energy input, we noticed that the thermal storage was not getting depleted during the evenings, and increased the off peak steam demand, first to 30 tons/hr, then to 40 tons/hour to try to extract maximum electrical energy from the system. These efficiencies are shown in Table 2 on page 17. Although there was a marginal increase in power for an off peak steam demand of 40 ton/hr, we noticed that at this demand, the thermal storage was depleted each night, so we again ran out of steam during the night. Thus it

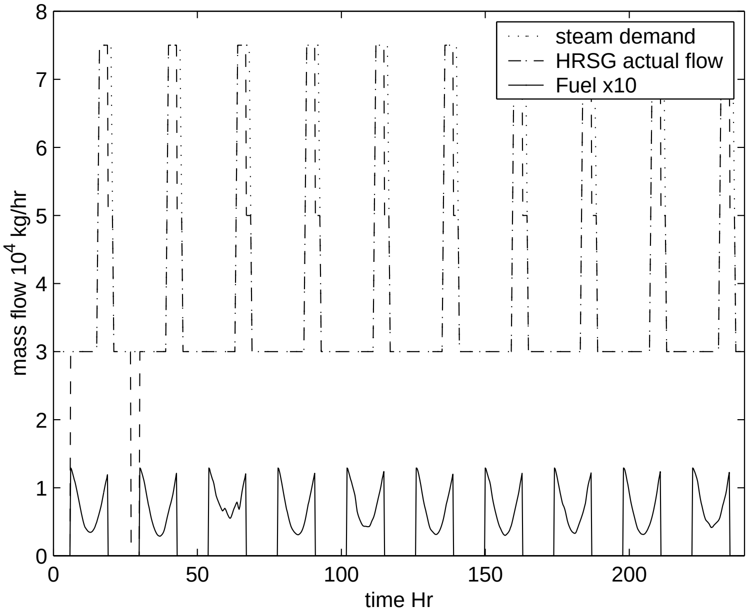

Figure 14: Initial Charging of Thermal storage. Steam Flow. Time Switch

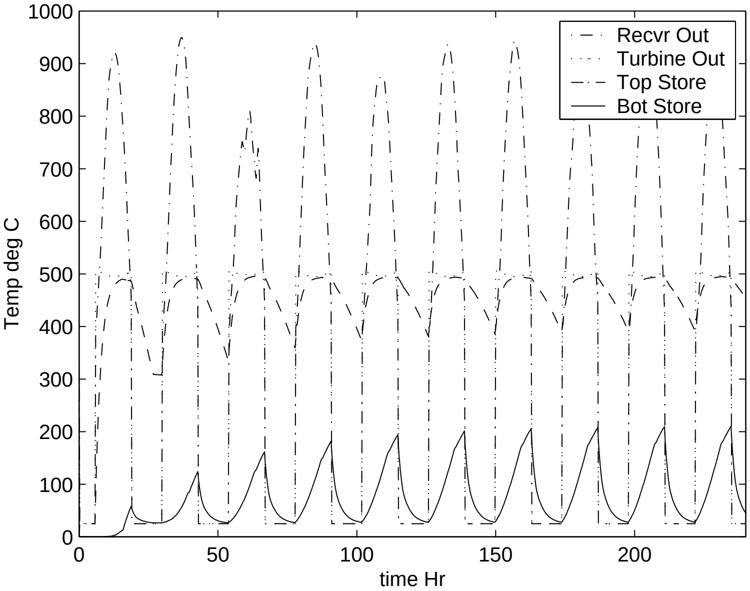

Figure 15: Initial Charging of Termal Storage. Temperatures. Time Switch

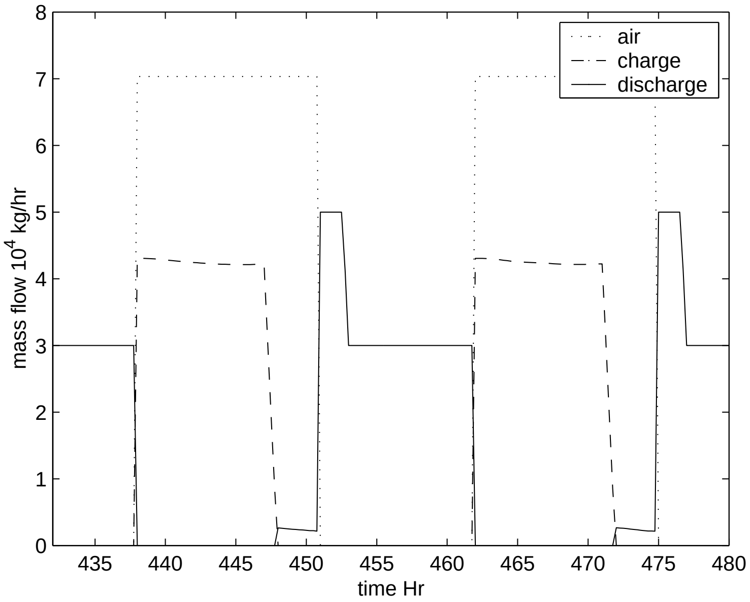

Figure 16: Air flow rates from Storage in midsummer. Time Switch

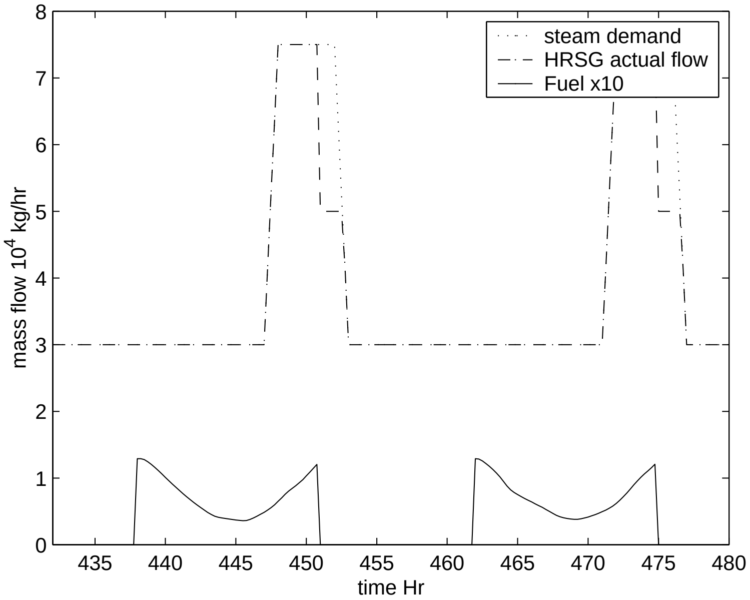

Figure 17: Steam Generator demand and flow rates midsummer. Time Switch

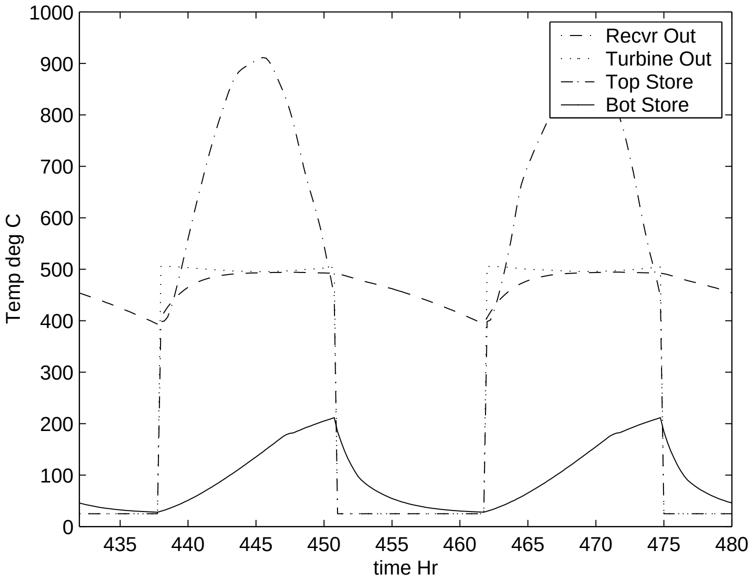

Figure 18: Temperature of turbine and storage midsummer. Time Switch

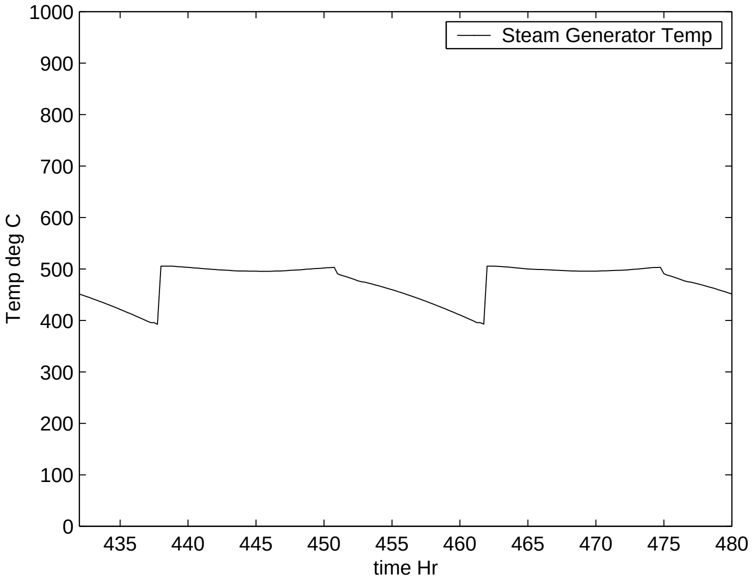

Figure 19: Steam Generator Temp midsummer. Time Switch

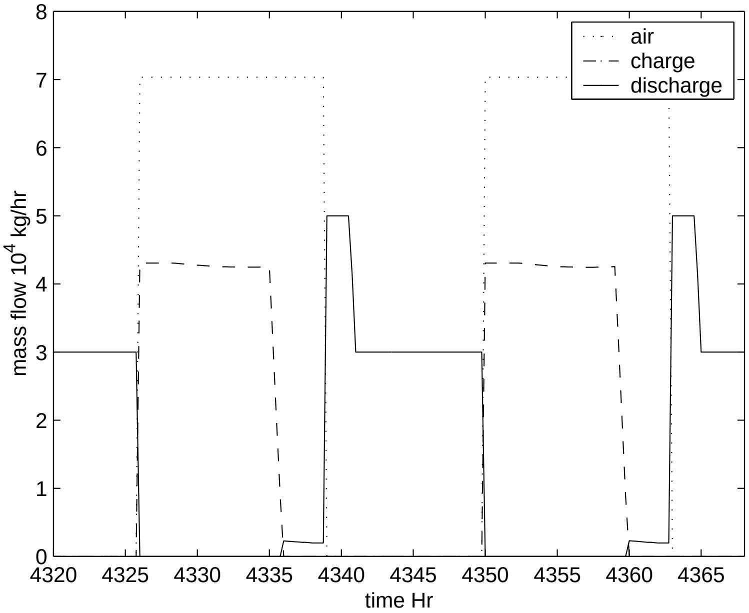

Figure 20: Air flow rates from Storage in midwinter. Time Switch

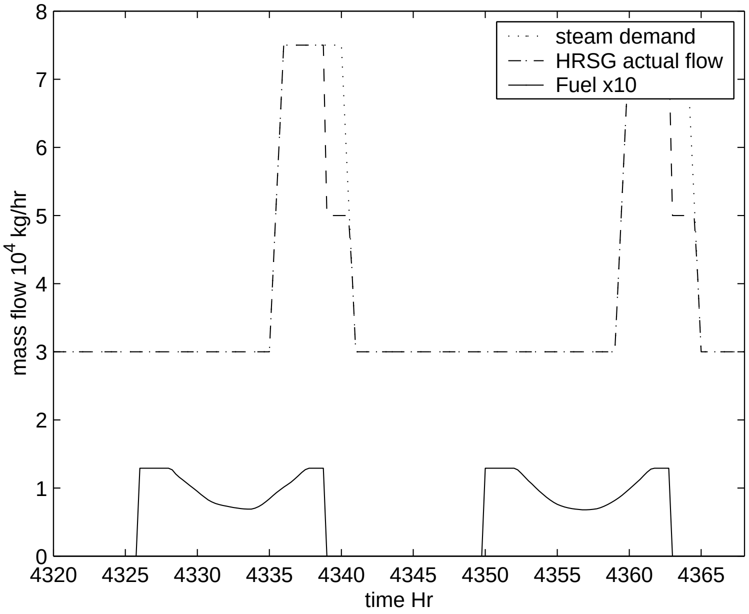

Figure 21: Steam demand and flow midwinter. Time Switch

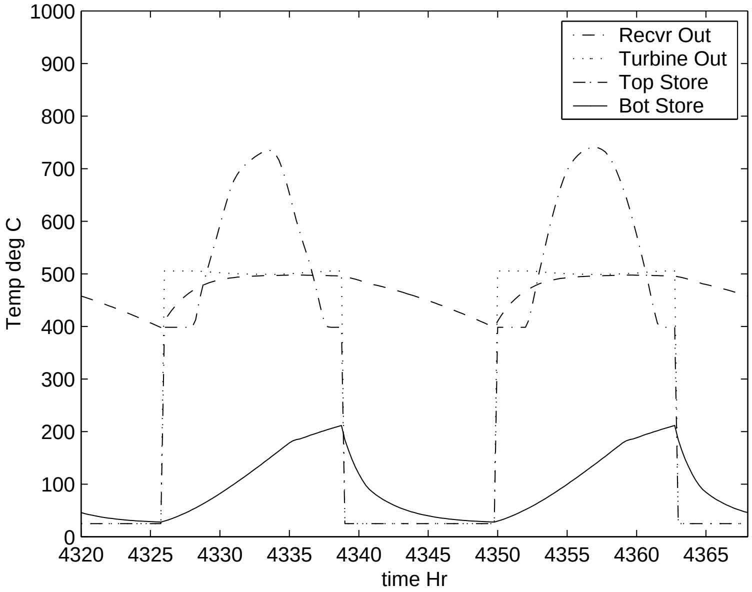

Figure 22: Temperature of turbine and storage midwinter. Time Switch

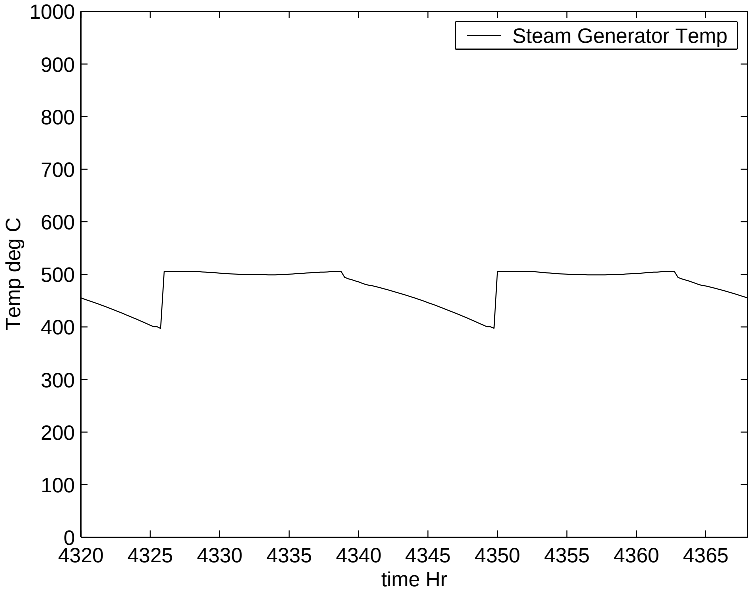

Figure 23: Steam Generator Temp midwinter. Time Switch

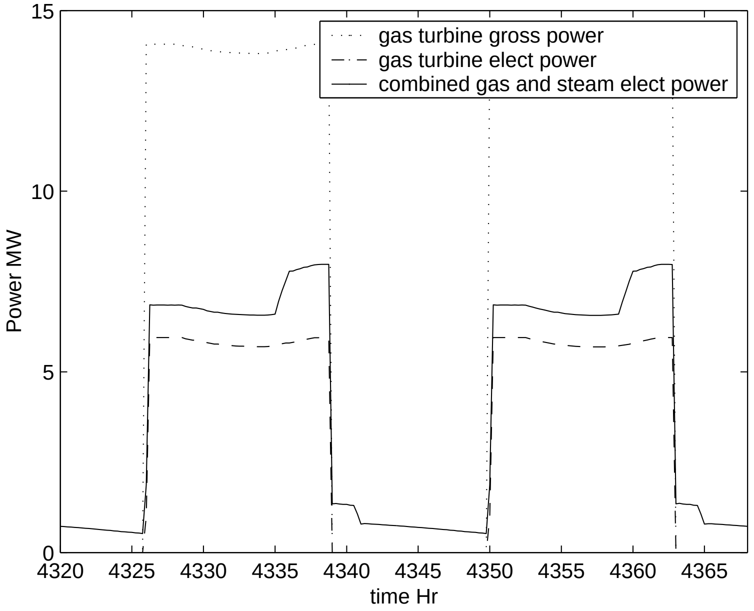

Figure 24: Gas Turbine and combined Electric Power Mid Winter.Time Switch

seems that an off peak steam demand of 30 ton/hr is roughly optimum for this plant size and control strategy. The remainder of this discussion and modeling was done with an off peak steam demand of 30 ton/hr.

The power graphs for this constant time case for midwinter are shown in Figure 24 on page 16. The graph for midsummer is very nearly identical. This similarity between summer and winter is clearly seen in the whole year plot of Figure 25 on page 17 , especially when it is compared to Figure 13 on page 10 (switched on solar flux).

The steam fraction (percentage of output power generated by the steam plant) was constant throughout the year and varied from 22% for the 20 Ton/Hr case to 24% for the 40 ton/hr case.

3 Checking of model against known 3rd party models

3.1 Heliostat - air receiver and turbine

We initially checked this against the individual STEC models for a heliostat field with an air receiver with a gas turbine, and the separate STEC model for a steam cycle.

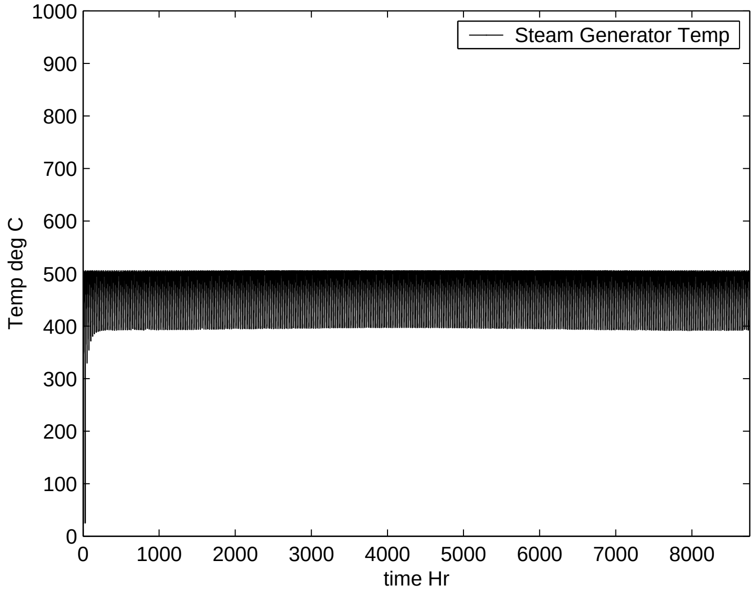

Figure 25: Steam generator Hot Gas Temperatures Whole Year. Time Switch

		Year		Jan		Jul		Peak
solar fraction %		37%		47%		25%		
steam off peak air		Pow	Eff	Pow	Eff	Pow	Eff	Pow
20T/hr		3.94	41%	3.90	41%	3.98	42%	6.60
30T/hr		4.06	43%	4.02	42%	4.10	44%	6.80
40T/hr		4.07	43%	4.03	42%	4.11	44%	7.12
x20 600T/hr		81	44%	80	44%	82	45%	158

Table 2: Power (MW) and efficiencies for constant time system

Parameter	STEC Model	SUNSPOT Model
Heliostats number	150	200
Heliostats area	100	100
Weather model	BARST84.DAT	Upington.DAT
Peak power onto receiver (noon)	$3.5 * 10^7$kJ/hr	$5 * 10^7$kJ/hr
electrical Power (noon)	$2.05 * 10^7$kJ/hr	$2.0 * 10^7$kJ/hr
Compression ratio	15	15
Compressor mass flow	75 ton/hr	75 ton/hr
Combustion chamber exit temp	1100	1100
Turbine exhaust temp after silencer	498	492

Table 3: Comparison of air receiver model

3.1.1 Conclusion

The air receiver match was a good one as shown in Table 3 on page 17, the main difference being that there was more incident solar power available in Upington, and so less fuel was burnt in the combustion chamber.

3.2 Steam Cycle

The original STEC model for steam generation had different input parameters as shown in Table 4 on page 19 to those available from the gas turbine exhaust in our model, so we iteratively adjusted parameters till we had a good working set. Initially we just added the existing exhaust gas from our air turbine model to the existing STEC steam turbine model. This model was designed for a higher hot air flow rate. (Compare column 2 and Column 3 in Table 4 on page 19 .) This had the effect of producing a reasonable output power, but with steam pressure only half of what was expected.

These combined cycle plants often run at a steam pressure of 100 bar [KRHS09] , and this initial model only produced a steam pressure of 47 bar. This was caused by the turbines being too large for the heat input, so steam pressure could never build up properly. Another factor indicated a mismatched HRSG was the fact that the pinch point was nearly zero. Pinch point is defined as the hot gas exhaust temperature leaving the evaporator, and the steam temperature leaving the evaporator. For most steam plants this is between 8 and 15 C.

In column 4 of the table we halved the steam flow rates through the turbines and halved the heat heat transfer capacity of the heat exchangers and this had the effect of bringing the steam pressure up to expected levels, although with a slight decrease in power output. (Buying a turbine of twice the needed capacity is probably not the most economical way to increase power output). The pinch point here was 6.5 C which is more reasonable.

3.2.1 Conclusion

After adjusting the model to the maximum available heat flow, we got good agreement, and power output in the expected range. We also checked the steam cycle against a model in a combined cycle book,[KRHS09] and it was within expected ranges.

4 Scale up to 100 MW Plant size

Next we scaled all flow rates in the plant up by a factor of 20 to get a plant in the 100 MW range. At this stage we also added a blower to model the parasitic losses caused by the discharge of the thermal store. The results of this run are also summarized in Table 2 on page 17 . The mean power over the whole year

Parameter	STEC Model (steady state)	Sunspot model (4pm peak) identical params to STEC model	Sunspot model (4pm peak) steam flow rates halved
Hot side input flow	130 ton/hr	75 ton/hr	75 ton/hr
Hot side input temp	574	497	497
Condensing Temp	35	35	35
T De-aerator water out	151	126	147
T preheat cold side out	207	178	204
T econ cold side out	306	261.3	304
T evap cold side out	311	261.2	303.6
T evap hot side out	316	261.6	310.2
Pinch Point	5	0.4 (no pinch point)	6.5
T superh cold side out	500	437	438
Steam Press superh bar	100	47	90
Total steam power out 10^6kJ/hr/MW	17.03 / 4.73	7.90 / 2.19	7.17 / 1.99 (pressure better but power worse)
economizer HTC (kJ/hr.K) *1000	390	390	195
evaporator HTC (kJ/hr.K) *1000	495	495	248

Table 4: Steam model comparison

Number heliostats	4000
Heliostat area each	100 m2
Combustion chamber exit temp	1100
Combustion chamber air flow	1500 ton/hr
compressor pressure	15 bar
Peak power electric	117 MW
Peak turbine shaft power	280 MW

Table 5: Solar Field and gas turbine Plant Parameters for 100 MW Plant

Mass thermal concrete	20,000 ton
Length	100 m
Total cross section area pipes	20 m^2

Table 6: Thermal Storage Plant Parameters for 100 MW Plant

was 81 MW, and the peak power 158 MW. Average efficiency (output power divided by combined solar and fuel input) during the whole year was 44%. For a dedicated combined cycle plant, efficiencies are of the order of 57% [KRHS09]. The difference between our Sunspot model and a dedicated plant can be explained by the fact that our steam cycle is running at a sub-optimum (keep alive) mode for much of the day, and is only running at full design capacity during peak hours. (16h00 to 20h00). On the other hand we are consuming no fuel at all during the night time hours. Our efficiency on just the fossil fuel burned is 70%, so the solar input allows us to exceed the efficiency of any pure fossil fuel plant.

The control and design strategy that will result in the most economical levelised cost of electricity will depend on the ratios of capital cost for equipment, interest rates, expected plant lifetimes and fuel costs.

The average power used by the storage blower over the whole year was 0.6 MW and the peak blower power consumption was 1.9 MW

5 Plant Parameters

The parameters for the 100 MW nominal plant are shown in Table 5 on page 20, Table 6 on page 20 and Table 7 on page 21. The number of receivers (4000) might be too large for a single tower, and a distributed tower system might be needed. There are other approaches[Fre09] , notably the Google funded company eSolar, which use smaller (1m^2) mirrors and multiple towers which might lead to lower capital cost.

Hot side flow rate	1500 ton/hr (peak 16h to 20h) 600 ton/hr (other)
Steam/water flow rate (peak)	180 ton/hr
Preheater (heated with steam) HTC	1860 MJ/hr.K
Economizer HTC	4000 MJ/hr.K
Evaporator HTC	5000 MJ/hr.K
Super heater HTC	1860 MJ/hr.K
Steam turbine 1	Pressure drop 100 bar-20 bar peak Elect 15.4 MW
Steam turbine 2	Pressure drop 20 bar-5 bar peak Elect 9.5 MW
Steam turbine 3	Pressure drop 5 bar-0.05 bar peak Elect 15.8 MW
Condenser	cool water inlet 20 ton/hr

Table 7: Heat recovery steam generator and steam turbines for 100 MW Plant

6 Problems

We tried several times to generate a steam model from scratch, but abandoned due to the time taken, and many small undocumented gotchas in the model components. For example the steam bypass indicator on the turbines must be initially set to 1 (not its default setting) for the turbines to generate steam pressure and output power. Another anomaly we found was the variable used to indicate the dryness of the steam. We expected this to vary between 0 (saturated or wet steam) and 1 (fully superheated steam). However unless this variable is set to 2 for fully superheated steam, the turbines do not generate power as expected.

7 Conclusions and Suggestions for Further Work

We have constructed a usable model in TRNSYS of a combined cycle solar power plant with a central receiver. We have tested it with different control strategies over a whole year. We concluded that it was necessary to run the auxiliary combustion chamber for a constant time each day throughout the year to enable us to properly size the thermal storage and steam cycle.

It would be useful to have more work done on two components in the model, and one additional control strategy:

- Rock bed storage component
- Air Condenser component
- An additional control strategy that could be tried is to just run the steam cycle at night (ramping it up as the air cycle ramps down), instead of switching it on at 16h00, and getting the peak steam sitting on top of the peak air cycle power (see for example Figure 24 on page 16)

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