Hybrid Solar and Coal-Fired Steam Power Plant with Air Preheating Using a Solid Particle Receiver

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Abstract

Fuel reduction has been achieved for coal power stations by hybridisation with solar thermal systems. Current technology uses feedwater or turbine bleed steam (TBS) heating with linear Fresnel based concentrated solar power (CSP) fields. The low temperature heat produced by these systems results in low solar to power conversion efficiency and very low annual solar shares.

In this paper the technical advantages of solarising coal fired power plants using preheated air by a novel CSP system based on a solid particle receiver (SPR) are examined. This system is compared to the current deployed state-of-the-art coal plant solarisation by modelling the systems and analysing the thermodynamic heat and mass balance of the steam cycle and coal boiler using EBSILON®Professional software. Annual performance simulation tools are also used to calculate the performance of the solarisation technologies.

Solarisation using SPR technology for preheating air in solar-coal hybrid power stations has the potential to considerably increase the solar share of the energy input by 28% points at design point and improve the annual fuel reduction from 0.7% fuel saved to 20% over the year. This is a significant reduction in fossil fuel requirements and resulting emissions. These benefits are a result of SPR solar system's higher operating temperature and integrated thermal storage, which also allow a buffered response time for handling transients in the intermittent solar resource.

Analysis indicates air-solarisation of coal plants can enable 81% higher solar to electric conversion efficiency than currently existing solar hybridisation option. Thus, the cost of the thermal energy generated by Fresnel based TBS solarisation must be up to 38% lower than thermal energy generation of secondary air preheating SPR system for economic parity between the options. Initial calculations indicate that the required thermal energy cost levels for SPR systems for this application are already achievable.

1 Introduction

As CSP has been used for hybridisation with coal plants, this paper will compare conventional TBS solarisation and the proposed more efficient air preheating option [1] using thermodynamic modelling on an annual basis. The CSP technology most appropriate for air-side solarisation must be capable of achieving high temperatures because the solar to electricity conversion efficiency increases with temperature, as does the solar share. If a large solar share is to be included in the plant design there will need to be adequate thermal storage. Thermal storage is necessary for stable plant operation and to control energy dispatchability to accommodate the variability in the solar resource and minimise combustion ramp rates within the boiler. A solar system with incorporated energy storage will significantly decrease the annual fuel requirement, as solar operation hours/capacity factor (CF) can be significantly increased. A recently developed solid particle receiver (SPR) system [2] appears to meet the requirements for good hybridisation of a coal fired power station.

A SPR mounted on a tower surrounded by a heliostat field, uses ceramic particles to directly absorb the incident solar radiation from the sun tracking mirrors. Since the particles are extremely heat resistant and robust, a particle receiver system can achieve very high efficiencies by absorbing very high solar flux densities, without the drawbacks associated with metal tube receivers (hotspots, thermal stresses and thermal fatigue). The hot particles are used directly as the thermal storage medium from which air can be heated in a cost-effective way by direct-contact heat exchange with the particles. The SPR solar system has been modelled using simulation tools and inputs from DLR experimental and numerical data.

The majority of coal plants in Australia today will soon require replacement. They are sub-critical coal plants with operating capacity greater than 300MW, and have been in operation for between 20 and 50 years [3]. Of the installed plants younger than 10 years or those which are currently being built, the overwhelming majority (83% of the power plant fleet) are composed of super- or ultra-supercritical black coal steam plants. As solarisation of a coal station is a large investment which requires decades of operation to justify, this paper will assess solarising the current generation steam power plant technology.

2 Methodology

2.1 Reference power plant description

The steam power plant investigated by the model was based on a typical supercritical steam power plant resembling the 750MW Kogan Creek black coal plant with reheat and cycle parameters 250/60bar and 540°C/560°C [4, 5]. The actual plant layout shown in Figure (1) features seven bleed steams and eight feedwater heaters (three high pressure heaters (HPHs), four low pressure heaters (LPHs) and a deaerator) and a cooling condenser.

The design limit of the actual Kogan Creek turbine is stated to be 781MW gross, while standard operation is 744MW gross [4, 6]. The net output of the plant in nominal operation is 95% of the gross power [5]. While it is stated in public documentation that the Kogan Creek solar booster provides a 44MW net solar-boost [7, 8], other sources suggest that only a 37MW gross boost is realised [6]. This can be explained as the parasitic energy demand of the high pressure feedwater pump may be reduced in operation if the solar field installation is larger than required for the turbine boost and re-circulating steam flow to the feedwater pump/turbine. Installation of a solar field to accomplish this parasitic reduction can be done anytime and in combination with any solar augment option, therefore we ignored this option in the study. In the actual installation the maximum solar-boost power is limited by the design limit of the turbine. The solarisation options presented in this paper are designed for fuel saving mode (same output with less fuel) rather than solar booster (increasing output) mode so as to not restrain the solar share by the design limits of the turbine and thereby provide a fair comparison between solarising methods.

2.2 Power cycle modelling

In order to model accurately the effect of solarisation on the power cycle and boiler of a coal fired power station, a powerful thermodynamic cycle analysis tool was needed. EBSILON®Professional was chosen due to its flexibility and level of detail. The first step was to create a representative model of the plant. Two available heat balance diagrams (HBDs) of the solar-boosted (more output with same fuel use) Kogan Creek plant were utilised to provide steam conditions at critical points in the cycle as well as give the efficiencies of the turbine stages during solar-boosting [8, 9]. Then to adjust the plant to its unboosted state, the mass flow to the existing solar field was shut off, and the model parameters were calibrated so that the available performance data of the non-boosted plant and those of the boosted plant were met. The fuel composition of Queensland coal with heating value 30MJ/kg was provided by Ebsilon. Variation of the ambient conditions was neglected, and maintained at 101.3 kPa and 25°C. The final modelled power cycle HBD is presented in Figure (1), including the modelled boiler.

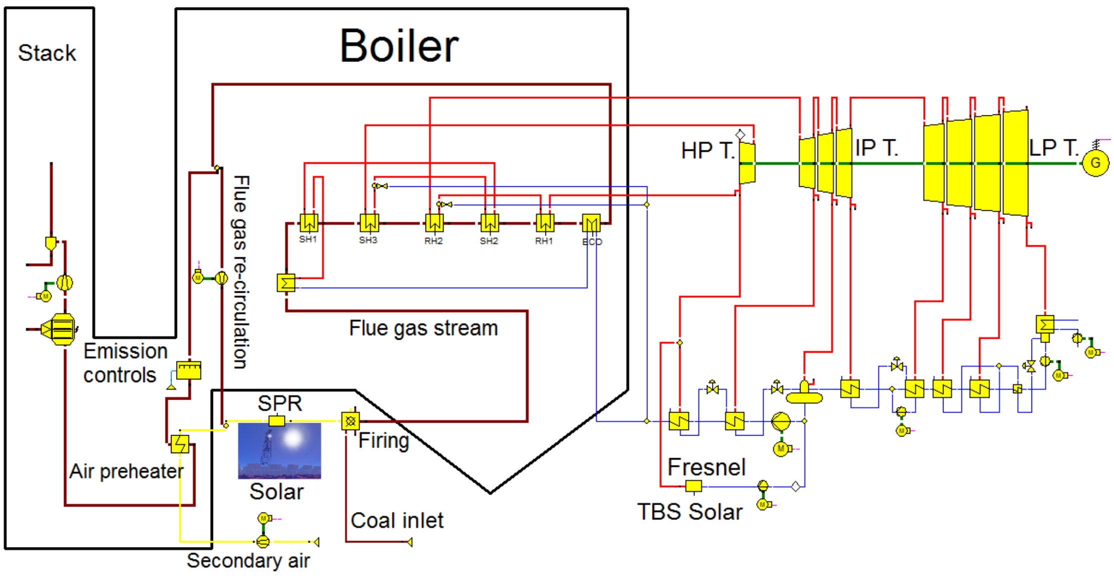

Figure. 1 Heat balance diagram of supercritical coal fired power plant showing solar SPR air-preheating and TBS solarisation points.

2.2.1 Boiler model

A model to represent a typical Benson once-through supercritical boiler was created to enable the full power plant simulation in combination with the steam cycle. The model was calibrated for a typical arrangement of superheat and reheat surfaces [10] to match realistic conditions for a boiler operating with live steam conditions of 540°C and 250bar [12], with feedwater and spray attemperation. The layout was adjusted to approximate design temperatures of the flue gas and water/steam of an existing boiler at inlet and outlet of the superheaters, re-heaters and economiser [13].

2.2.2 Boiler operation with solar air-preheating

Secondary air enters the boiler from ambient, and is heated in the air pre-heater, by heat exchange with exhaust flue gasses to around 280°C. This air is further heated by heat exchange with heated particles to the SPR outlet temperature. The heated secondary air then joins with a pulverised coal/primary air mixture; and is combusted. Since approximately 80% of the air combusted comes from the secondary air stream, raising the temperature of the secondary air can dramatically reduce fuel requirements. After combustion the flue gas stream is cooled by feedwater and steam at the evaporator, superheaters and economiser (a portion can be re-circulated to the pre-solarisation/combustion air stream). The major air stream then undergoes de-noxification, cooling at the air pre-heater, de-sulphurisation and filtering, and finally venting to the atmosphere via the stack at temperatures exceeding the sulphur dew point (typically at around 150°C).

2.2.3 TBS solarisation

For TBS solarisation, the efficiency of solarisation is limited by the steam conditions (temperature and pressure) at the solar steam injection point. According to the available HBDs the solarisation option for the Kogan Creek solar booster power station supplies bleed steam at roughly 335°C to the final HPH. It does this using steam generated by water flow from the feedwater tank situated before the final two HPHs. For TBS solarisation, these are the most efficient heaters to solarise [14] as higher performance is achieved by substituting TBS to high pressure heaters (HPHs) than to low pressure heaters (LPHs). In this paper the TBS solarisation option used in the Kogan Creek plant will be compared to a solar air preheating solarisation option.

2.3 Solar system modelling

The CSP plants were simulated at the same location as the Kogan Creek Power station (Chinchilla QLD 26.8°S and 150.6°E), using an hourly DNI time series approximating an average year in terms of solar resource [15] at the site (2006.3 kWh/m^{2}a).

2.3.1 Fresnel system model

The US NREL-SAM software was selected to predict the output from a general purpose model of a CSP system and this was used to calculate the annual efficiency of a typical Linear Fresnel solar collector. This model was taken from the public resource of system files specifically created for Australian conditions [15] under a project supported by ARENA. Calculations were based on Novatec Solar’s guaranteed key performance indicators which are used as a basis for contractual agreements and altered to approximate the parameters of the Kogan Creek HBD bleed steam generation conditions (once through boiler model, with 335°C output, operating with a solar multiple (SM) of 1 to ignore solar dumping loss).

2.3.2 Particle receiver system model

A quasi-dynamic simulation environment for annual simulation of heliostat-tower CSP systems based on a particle receiver was developed. The simulation environment includes the ability to economically optimise the system for lowest energy cost. The optical performance of the solar field was calculated using the DLR software HFLCAL [16] while thermodynamic relationships and supplier information were used to model thermal and other performance losses at every macro-component of the system. The highest temperature a SPR can currently achieve is 1000°C. If a particle-air heat exchanger can provide a driving temperature difference of 50K, the maximum solarisation temperature would be 950°C, which suits this application as preheated air entering a boiler may be limited to 1000°C [17].

3 Figures of merit and definitions

The solar incremental electricity is the annual amount of net electricity produced by the solar-hybrid plant compared to the electricity produced by the same power plant without solarisation and using the same amount of fuel. This can be calculated using Eq. (1) [18], where *E* is electrical energy, *Q* is thermal energy and subscripts a, c, hybrid, ref and s refer to annual, coal, hybrid, reference plant and solar respectively.

$$\Delta E_{s,a} = E_{hybrid,a} - \frac{E_{net,ref,a}}{Q_{c,ref,a}} . Q_{c,hybrid,a} = E_{hybrid,a} - \eta_{ref} . Q_{c,hybrid,a} \quad (1)$$

All drawbacks and benefits of solarisation are assigned to the solar part of the hybrid plant so that a fair comparison between systems with different solar shares can be made.

Incident solar energy ($Q_{s,inc}$) is the energy incident on the total mirror area of the solar plant, before solar field and solar receiver optical and thermal losses, and is defined by Eq. (2), where Q_s is the solar thermal energy actually delivered to the power cycle, and η_{s-th} is the solar to thermal conversion efficiency.

$$Q_{s,inc} = \frac{Q_s}{\eta_{s-th}} \quad (2)$$

Different solarisation options will have varying *solar to power efficiency* (sometimes known as the net incremental solar to electric efficiency) due to the thermodynamic nature of how heat is utilised in the power cycle. Eq. (3) gives the efficiency at which incident solar energy is converted to electricity, where ΔE_{hybrid} is the electricity generated from the solar energy [18].

$$\Delta \eta_{s,a} = \frac{\Delta E_{hybrid,a}}{Q_{s,inc,a}} \quad (3)$$

The solar share illustrated by fuel (coal) usage represents the portion of fuel saved due to solarisation during design conditions and is described in Eq.

(4) by the fuel mass flow rates. Solar share calculated using this method shows the same effect as change in heat rate. This is considered more valuable than solar share calculated by instantaneous energy input, Eq. (5), as solar share by fuel flow takes into account the power use efficiency of the solarisation (including the influence solar modifications have on the cycle and boiler performance). Results of Eq. (4) do not often equal those of Eq. (5).

$$X_{s,\dot{m}} = 100\left(\frac{\dot{m}_{c,ref} - \dot{m}_{c,hybrid}}{\dot{m}_{c,ref}}\right) \tag{4}$$

$$X_{s,Q} = 100\left(\frac{Q_s}{Q_c + Q_s}\right) \tag{5}$$

Annual solar share, Eq. (6) is the best figure of merit to show proportional fuel savings due to solarisation. It can be derived by the solar incremental electricity or for this study's simplified assumptions (including full load operation of the plant for the complete year) by Eq. (4) and solar CF.

$$X_{s,a} = \frac{\Delta E_{s,a}}{E_{hybrid,a}} = 100\left(\frac{\dot{m}_{c,ref} - \left(\dot{m}_{c,ref} - \dot{m}_{c,hybrid}\right)CF_{th,s}}{\dot{m}_{c,ref}}\right) \tag{6}$$

4 Results

Solarisation by air-preheating was simulated at 540°C and 950°C, to give a range of reasonable operation temperatures supplied by a SPR system. Feedwater supplied to the superheated and reheated steam paths in the boiler is used to keep steam temperatures during air-preheat solarisation constant. Cycle efficiency can be notably improved, up to 0.8% points for the air solarisation options presented, if steam temperatures may increase with solarisation (to a maximum temperature of 580°C).

4.1.1 Air-cycle

Isolated (air ratio unchanged) air solarisation increases the adiabatic flame and flue gas temperatures and therefore increases the temperature difference between flue gas and heat extracting fluid; which can lead to higher steam temperature and cycle efficiency. As energy is input from solar to generate the same amount of electricity, less fuel is required to reduce flue gas flow. With lower flow to the existing air pre-heater, heat exchange between hot flue gas and ambient secondary air over the constant heat exchange area is more efficient, reducing the exhaust stack temperature and improving boiler efficiency.

However, to ensure that temperatures at the convective heat exchange surfaces remained constant, the boiler duty was kept constant by increasing secondary air flow and thus air ratio (for equal boiler exhaust gas flow in all cases). While this decreases the boiler efficiency due to lowering the flame temperature (by diluting the stoichiometric flue gas mixture with non-reacting gases) and increasing the stack losses, an overall positive effect to the boiler efficiency from the reference case is noted. This is because stack temperature is lower than the reference case since there is more ambient air flow in the air pre-heater. Additionally solar share can be increased as more mass flow (preheated air) can be heated by the solar system, allowing for a larger solar system to be installed (the solar system is sized by inlet and outlet conditions and the flow rate, given a design outlet temperature). TBS fuel saving solarisation only results in an excess air increase without increasing preheat temperature, lowering boiler efficiency compared to the reference case.

4.1.2 Re-circulation of flue gasses

Re-circulation is utilised in conventional operation of coal plants [19] to improve performance and reduce emissions. While around 49 coal plants worldwide have up to 30% flue gas re-circulation [20], air solarisation enables even higher portions as combustion can take place at higher air ratios, as described in section 4.1.1. If the boiler system does not already include a re-circulation loop, a cost-benefit analysis of the blower and air ducting costs against performance improvement must be made before such a re-circulation loop would be retrofitted.

Re-circulation decreases the air ratio, as less ambient air is required to meet the boiler duty flow rate condition. However as flue gas flow through the air pre-heater with fixed heat transfer area is still lower (also compared to the case with 20% excess air without re-circulation), the heat exchange is more efficient and the stack temperature is reduced, thereby significantly increasing boiler efficiency. This also means hot flue gas at approximately 375°C is combined with preheated air before solarisation, reducing the solar system size as there is a smaller temperature range for solar heating; yet solar share calculated by mass flow ($X_{s,\dot{m}}$) increases (since solar to power efficiency is increased so dramatically). Table 1 summarises key calculated design point results. For TBS solarisation, power block efficiency is calculated taking into account the solar energy input.

Table 1. Design point performance results

Result / Case	Unit	Ref. case	Air 950°C	Air 950°C re-circ.	Air 540°C	Air 540°C re-circ.	TBS 335°C	TBS 335°C re-circ.
$Solar_{in}$ (air/H_2O)	°C	-	276	328	278	299	279	287
$Solar_{out}$ (air/H_2O)	°C	-	950	950	540	540	335	335
Solar system size	MW_{th}	-	547	520	201	187	95	95
Fuel flow	kg/s	58.1	39.8	39.4	51.4	51.3	55.7	55.5
Excess air ratio	%	20	79.5	20	36.9	20	25.5	20
Air re-circulation	%	0	0	36	0	13.5	0	5.2
Stack temp.	°C	155.8	151.1	137.8	154.1	149.9	155.2	153.6
Boiler eff. (LHV)	%	94.4	94.6	96.8	94.5	95.3	94.2	94.6
Cycle eff. (gross)	%	45.2	45.2	45.2	45.2	45.2	44.5	44.5
System efficiency	%	40.4	40.5	41.6	40.5	40.9	39.8	39.9
Solar share ($X_{s,Q}$)	%	-	31.4	30.5	11.5	10.8	5.4	5.4
Solar share ($X_{s,\dot{m}}$)	%	-	31.5	32.2	11.6	11.7	4.1	4.4

*Boiler eff. (HHV) may be approximately 5 percent lower than the boiler eff. (LHV)

4.1.3 Performance figures

When comparing the solar share calculated by fuel mass flow, Eq. (4), and solar share calculated by energy input, Eq. (5), it can be seen that the efficiency of the power plant is increased due to the solarisation, when solar share ($X_{s,\dot{m}}$) is greater than solar share ($X_{s,Q}$), and vice versa. Solarisation by TBS decreases the power cycle efficiency due to proportional bleed steam reduction, while solarisation by air-preheating increases the power system efficiency (product of boiler and net cycle efficiencies).

4.1.4 Annual results

While design point results are useful in understanding a system, figures of merits for energy system comparisons are only valid on an annual basis. Therefore the annual solar performance which is affected by the operating temperature of the solar receiver, and respectively the thermal storage if applicable, must be known. Annual solar to thermal efficiencies ($\eta_{s\text{-}th,a}$) for the assessed technology options are shown in Table 2.

Table 2. Approximate annual solar to thermal efficiency of compared technologies at varying conditions in Chinchilla, Queensland

Technology / Outlet Temperature	335°C	540°C	950°C
Solar tower particle receiver with storage (SM=3.2)	52.6%	52.3%	50.3%
Linear Fresnel DSG (SM=1)	40%	N/A	N/A

The optimisation of the SPR tower system using a performance and cost model results in a SM of 3.2, and small scalable module sizes with very high peak solar concentration ratios (~2.5MW/m^2 averaged over the aperture). This results in high receiver efficiency and minor variability in annual performance efficiency with varying system outlet temperature. Table 3 outlines a range of CFs for systems assessed in this study.

Table 3. Approximate annual solar to thermal energy capacity factors of compared technologies located in Chinchilla QLD.

Technology / Outlet Temperature	335°C	540°C	950°C
Solar tower particle receiver with storage (SM=3.2)	64.7%	64.3%	61.9%
Linear Fresnel DSG (SM=1)	15.3%	N/A	N/A

The quoted thermal CF of the actual Fresnel Kogan Creek solar booster system is 11.5% [8, 21]. The higher simulated result suggest that in reality there may be availability issues due to difficulties in regulating steam conditions without storage (given a fluctuating solar resource), as solar steam for TBS solarisation must precisely match the cycle TBS conditions.

Table 4. Figures of merit (annual)

Annual result / Case	Unit	Ref. case	Air 950°C	Air 950°C re-circ.	Air 540°C	Air 540°C re-circ.	TBS 335°C	TBS 335°C re-circ.
Solar share ($X_{s,\dot{m}}$)	%	-	19.5	20.0	7.4	7.6	0.66	0.72
Solar to el. eff.	%	-	20.4	22.1	21.3	23.2	12.8	13.9
Heat rate$_a$	kJ/kWh	8899	7162	7121	8236	8226	8840	8835

a) all annual results are given for net total plant (LHV), combining boiler and turbine/cycle

Table 4 shows that solarisation by air preheating with a SPR system can potentially be significantly more effective than the current state-of-the-art.

4.1.5 Issues to be addressed for solarisation by air preheating

While the advantages are clear, there are a few technical issues regarding implementation of secondary air solarisation which should not be ignored for a specific project. Most notably;

- Can existing burner and secondary air system hardware be used, or are modifications or custom designed burners that run on higher pre-heat temperatures required.
- As stack temperature reduces, particularly at 950°C with flue gas re-circulation, it can get very close to the sulphur dew point (~138°C). In order to create a safety buffer either a slightly lower solarisation temperature or less re-circulation could be implemented.
- There may be complications with ash contained within flue gasses coming into contact with the SPR particles and causing particle soiling. A simple flue gas dust carry over experiment would clarify if this is an issue.

5 Conclusion/summary and suggested future work

Solarisation by TBS led to reduced power cycle efficiency, while solarisation by air-preheating increases the power cycle performance. An improvement in the annual heat rate of around 20% was calculated for a 950°C SPR air preheating system, which considerably reduces yearly fossil fuel requirements. A very low annual heat rate for a coal plant through solarisation (design heat rate is meaningless for describing real world operational fuel reduction) is only achievable with high temperature air solarisation and a storage capable solar technology, such as SPR based solar towers. Annual performance is often overlooked when assessing options for hybridising fossil plants [22], despite being the only indicator suitable to evaluate the actual fuel saving opportunity. While maximum solar system size was calculated to be up to 547MW_{th}, a SPR can be installed with much lower power ratings (as small as ~1MW_{th}) providing equivalent performance. This could also allow deployment of a small demonstration system at minimal cost.

Air solarisation of a coal power plant by SPR technology enables notably higher annual solar to electricity conversion efficiency, by up to 81%. It also allows significantly greater annual fuel savings, from 0.7% saved using state-of-the-art TBS solarisation to 20% saved using SPR air solarisation. Further work to quantify and evaluate the commercial opportunity with respect coal pricing for a positive business case for SPR air-preheat solarisation of coal power stations in the current Australian energy market is needed.

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Nomenclature and Abbreviations

Δ	Net incremental
η	Efficiency
C	Celsius
CF	Capacity factor
CSP	Concentrated solar power
DLR	German Aerospace Centre
DSG	Direct steam generation
E	Electrical energy
H_2O	Water
HBD	Heat balance diagram
HFLCAL	Heliostat field layout calculation
HPH	High pressure heater
LHV	Lower heating value
LPH	Low pressure heater
$\dot{m}$	Mass flow
MJ	Megajoule
MW	Megawatt
Pa	Pascal
Q	Thermal energy
SM	Solar multiple
SPR	Solid particle receiver
TBS	Turbine bleed steam
X	Solar share

Subscripts

a	annual
c	coal
inc	incident
ref	reference plant configuration
s	solar
th	thermal

References

[1] Deng, S., “Hybrid Solar and Coal-Fired Steam Power Plant Based on Air Preheating”, Journal of Solar Energy Engineering, 136, (2014).

[2] Wu, W., Amsbeck, L., Buck, R., Uhlig, R., Pitz-Paal, R., Proof of concept test of a Centrifugal Particle Receiver, Proc. SolarPACES 2013, Las Vegas, USA, September 2013.

[3] Finkenrath, M., Smith, J., Volk, D., “Analysis of the Globally Installed Coal-Fired Power Plant Fleet”, International Energy Agency, (2012).

[4] Cziesla, F., Kremer, H., Much, U., Riemschneider, J., Quinkertz, R., Advanced 800+ MW Steam Power Plants and Future CCS Options, Proc. Coal-Gen Europe 2009, Katowice, Poland, September 1-4, 2009.

[5] Modern Power Systems, (2006), Kogan Creek enters the commissioning phase, retrieved 2014, from <http://www.modernpowersystems.com/features/featurekogan-creek-enters-the-commissioning-phase/>.

[6] Australian Energy Market Operator, (2014), Electricity Registration and Exemption List, retrieved January 2014, from <http://www.aemo.com.au/Electricity/Registration>.

[7] CS Energy 2013, CS Energy Annual Report 2012/2013, CS Energy Limited, Brisbane.

[8] CS Energy, (2014), Kogan Creek Solar Boost Project, retrieved November 2013, from <http://kogansolarboost.com.au/about/>.

[9] Richardson, I., Kerr, B., Rawlings, I., Doucet, D., Chapman, L., Joy, G., CS Energy’s Kogan Creek Power Station Chemistry Experience, (2012), retrieved 2013, from <http://acc-usersgroup.org/wp-content/uploads/2011/02/KoganCreek-2010-ACC.pdf>.

[10] Venetos, M., Conlon, W., Solar Steam Boosters for Coal-fired Power Plants, Power Engineering, volume 114, issue 11, November (2010), retrieved October 11, 2013, from <http://www.power-eng.com/articles/print/volume-114/issue-11/features/solar-steam-boosters-for-coal-fired-power-plants.html>.

[11] Teir, S., Jokivuori, A., Steam Boiler Technology: Thermal Design of Heat Exchangers (2002), Helsinki University of Technology Department of Mechanical Engineering, Energy Engineering and Environmental Protection Publications.

[12] Kumar, R., Talukdar, P., Subbarao, P., CFD Analysis of Flue Gas Ducts of 500 MW Pulverized coal Boiler, Proc. ISHMT-ASME, January 4-6, (2010), IIT Mumbai, India.

[13] Subbarao, P., Final Audit: Utilization of Flue Gas Energy, Mechanical Engineering Department Indian institute of technology Dehli (2011), retrieved December 2013, from <http://web.iitd.ac.in/~pmvs/index.php?q=mel725>.

[14] Siros, F., Philibert, C., Le Moullec, Y., Tusseau, M., Bonnelle, D., The Value of Hybridising CSP, Proc. SolarPACES 2012, Marrakesh, Morocco, September 2012.

[15] Lovegrove, K., Franklin, S., Elliston, B., Australian Companion Guide to SAM for Concentrating Solar Power, ACT Australia, (2013).

[16] Schwarzboezl, P., Pitz-Paal, R., Schmitz, M., Visual HFLCAL - A Software Tool for Layout and Optimisation of Heliostat Fields, Proc. SolarPACES 2009, 15.-18.September (2009), Berlin, Germany.

[17] Woudstra, N., van der Stelt, T., Exergy analysis of combustion systems, Proc. ECOS 2003, June 30 – July 2 (2003), Copenhagen, Denmark.

[18] Schwarzboezl, P., Buck, R., Sugarmen, C., Ring, A., Crespo, J.M., Altwegg, P., Enrile, J., “Solar gas turbine systems: Design, cost and perspectives”, Solar Energy, 80, 1231–1240 (2006).

[19] IEA Clean Coal Centre, Flue gas recirculation for NOx control, Retrieved January 2014, from <http://www.iea-coal.org.uk/site/ieacoal/databases/ccts/flue-gas-recirculation-for-nox-control>.

[20] Li, J., Zhang, X., Yang, W., Blasiak, W., “Effects of Flue Gas Internal Recirculation on NOx and SOx Emissions in a Co-Firing Boiler,” Journal of Clean Coal and Energy, 2, 13-21, (2013).

[21] Clean Energy Acton Project, Kogan Creek Power Station: Kogan Creek Solar Boost Project Case Study, (2012), retrieved January 2014, from <http://www.cleanenergyactionproject.com/CleanEnergyActionProject/CS.Kogan_Creek_Solar_Boost_Project___Hybrid_Renewable_Energy_Systems_Case_Studies.html>.

[22] Meehan, R., Rudge, M., Hybridisation of Fossil Fuel Energy Generation in Australia, Public Report, November 20, (2013).