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Advanced power cycles for concentrated solar power

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ABSTRACT

This paper provides a review of advanced power cycles under consideration for CSP. As variable renewables make rapid commercial progress, CSP with thermal energy storage is in an excellent position to provide low cost stability and reliability to the grid, however higher efficiency and lower costs are critical. Steam turbines provide a robust commercial option for today but more advanced power cycles offering greater agility and flexibility are needed. Supercritical steam turbines are attractive at large scale but presently commercial products are too large for today's solar towers, unless multiple towers with an aggregating heat transfer fluid is used. CSP/PV hybrids combine benefits of PV's and low cost thermal storage. Supercritical CO_2 closed loop Brayton cycles are early in their development but promise high efficiency at reasonable temperatures across a range of capacities, with the prospect of significantly lowering costs. The next few years building knowledge on materials and components cost and performance along with demonstration is crucial. Gas turbine combined cycles driven by CSP are one of the highest efficiency options available, though other bottoming and topping cycle configurations should be progressed also. Again, component demonstration at the required high temperatures is critical.

1. Introduction

This paper reviews some of the most promising power cycle options for concentrating solar power (CSP) in the present to medium term future. The heat engine is one of the most critical components in a CSP plant. The heat engine, usually some form of turbine cycle when generating electricity, dictates the temperature that the solar concentrator, receiver and storage must provide. It is the component that has the single most impact on overall system efficiency and thus a significant effect on cost of electricity. Heat engines are the generic thermodynamic conversion device for thermoelectric or thermochemical systems. This paper considers CSP applications for electricity production and thus turbine cycles will be assumed unless otherwise mentioned. The discussion concentrates on cycles beyond conventional subcritical steam Rankine cycles.

CSP plants have always built upon heat engines used in other sectors of the power industry. All commercial CSP plants in operation today use a sub-critical steam turbine Rankine cycle, common to the coal-fired power industry, with some minor refinements to suit the diurnal nature of solar operation. Thus the heat engine operating conditions are usually well-established and the front-end of the CSP system (concentrator, receiver and storage) must meet these conditions. This is both an advantage (in that part of the system is already commercially available) and a drawback in that efficiency limits are already in place and an even greater cost reduction burden is then placed on the collector and storage components. Thus the possibility of more advanced thermodynamic cycles and machinery is of immense potential benefit to the CSP industry. This is particularly so as, even though the solar photons are free, the cost of generating heat from these photons is relatively expensive compared to fossil fuels today – the more expensive the heat source the greater the importance of efficiency in the cost of electricity calculation (Fig. 1).

Cycle efficiency increases with turbine inlet temperature, all other parameters remaining constant. Thus one of the key technological targets for CSP is in the production of higher temperatures to match the most efficient cycles. The solar concentrator itself (ie the mirrors) can, in principle, produce the flux necessary to generate the highest temperatures used by the power industry today.

Abbreviations: BCSS, Bottoming Cycle Storage Systems; CC, Combined Cycle system; CIT, compressor inlet temperature; ISCCS, Integrated Solar Combined Cycle Systems; LCOE, levelized cost of electricity; ORC, organic Rankine cycle; PCHE, Printed circuit heat exchangers; sCO_2, supercritical carbon dioxide; SGT, solar gas turbine; TES, Thermal energy storage; TET, turbine exit temperature; TIT, turbine inlet temperature.

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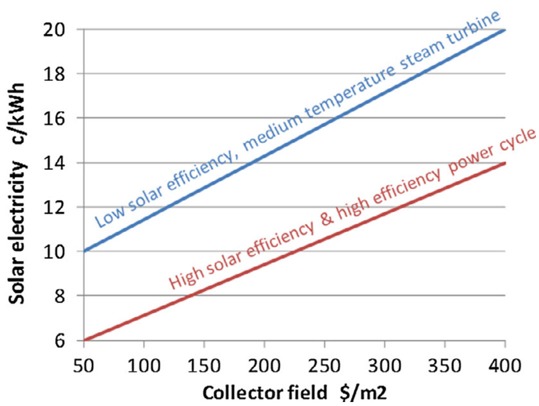

Fig. 1. The effect of efficiency and collector cost on LCOE. The blue line shows a conventional CSP system with 38% efficiency steam turbine. The red line shows an advanced CSP system using a sCO_2 cycle with TIT of 700 °C. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

For example at a solar concentration ratio of 1000 times which is relatively easily achieved by dish and tower concentrators, stagnation temperature is >1500 °C (Winter et al., 1991). The upper temperature limits are usually imposed by the properties of available materials and heat transfer fluids in the receiver and/or storage that can deliver suitable life expectancy. Temperatures of about 1000 °C out of the receiver are reasonably achievable today in pilot plants, though not yet for commercial scale.

Fig. 2 shows the relationship between turbine cycle efficiency and turbine inlet temperature for some of the main power cycles that have or are being considered for application in CSP systems.

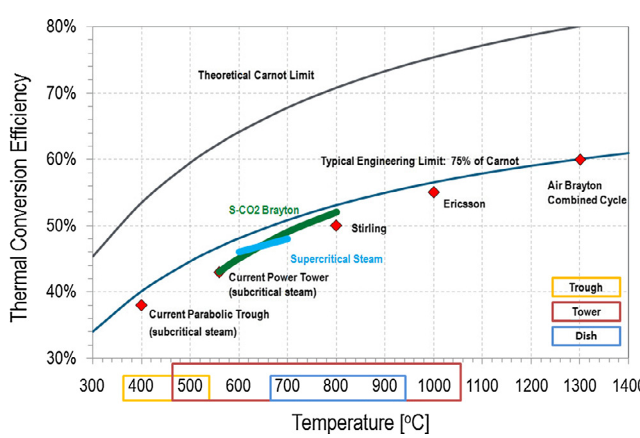

Fig. 2. Turbine cycles with potential application for CSP (Siegel et al., 2014).

The current parabolic trough plants, based on oil as the receiver heat transfer fluid with a maximum oil temperature of 395 °C and a sub-critical steam turbine with 380 °C/100 bar steam (single reheat) have a net efficiency of about 37.5% (39% gross) (Hirsch and Khenissi, 2014). The solar towers today using molten salt or water/steam as the receiver heat transfer fluid are able to achieve higher steam turbine inlet conditions (550 °C/120 bar) and have a net turbine efficiency of about 41.5% (43% gross). These efficiencies are simply a reflection of the steam turbine – troughs or linear Fresnel using salt or direct steam generation and producing the same temperature as the molten salt towers would still benefit from these higher turbine efficiencies. These efficiencies will change slightly depending on factors such as condenser pressure (dictated by ambient temperature and cooling technology), level of regenerative feedwater heating and pressure. The use of reheat in a steam Rankine cycle allows higher turbine inlet pressures to effect higher efficiency without the detrimental effect of lower steam quality (higher moisture) on the last stage blades.

The efficiency of rotating machines typically improves with increasing capacity, due to both leakage (between the blade periphery and the turbine casing) and windage losses. The specific capital cost ($/kWe) also tends to improve with increasing capacity as rotating machines are volumetric. Typically the greater the capacity the higher the efficiency and/or the lower the specific capital cost (Fig. 3). This is a well-documented phenomena in steam turbines and has had a significant effect on where the commercial CSP industry is today, seen in the push for larger plant capacities by commercial developers. Though the commercial reasons for this are understandable, the downside is that it has been a struggle for small steam turbine-based systems and new developers to enter the market.

Another important consideration is the turbine cycle part load performance. Steam turbines typically exhibit a relatively flat part load performance curve compared to the much peakier part load curves of open cycle combustion gas turbines. Fig. 4 shows an operating range (turn-down) of 75% and only a 5% increase in heat rate at 50% load (Denholm and Mehos, 2011). Part of the value of CSP to the grid in the future is likely to be the flexibility it affords. Flexibility, that is the ability to respond quickly and "in-fill" for variable renewables or when ramp rates demand means that, storage notwithstanding, there will be times when the turbine needs to operate at part load, and thus any new thermodynamic cycles should also offer good part load performance and turn-down ratio.

2. Power cycles for CSP applications

There are a number of technically feasible options for converting concentrated solar energy to electricity. This paper will discuss two that are receiving the most development attention at present –

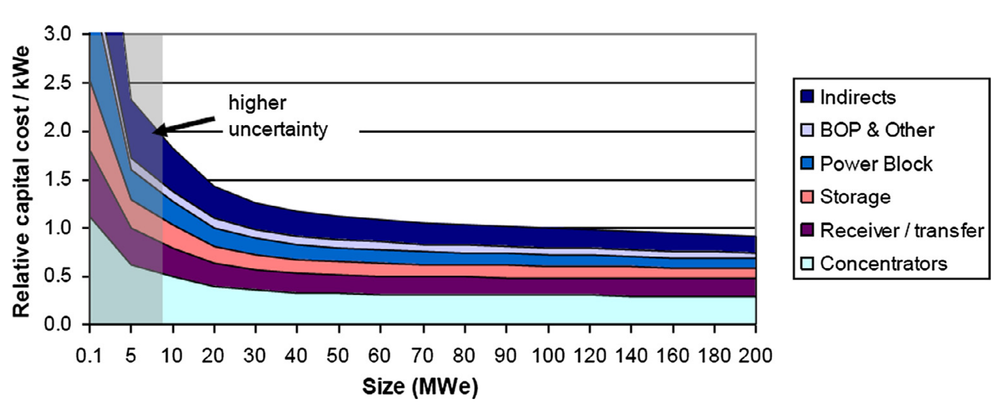

Fig. 3. The effect of plant capacity on capital cost. Note this is for a complete CSP plant with 5hrs storage Component break-ups are shown (Lovegrove et al., 2012).

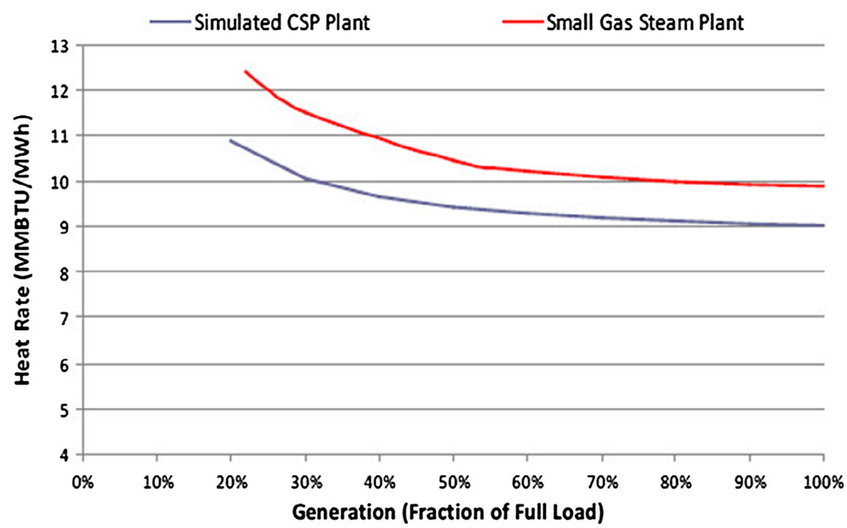

Fig. 4. Part load efficiency (defined as heat rate here) for steam turbines typically used in CSP plants and in gas-fired steam turbines. Note heat rate is inversely proportional to efficiency.

Brayton cycles based on closed loop supercritical carbon dioxide and air turbine combined cycles. There are other specific concentrating solar power cycles of interest which are not discussed here, including concentrating photovoltaics (Horne, 2012) and solar thermoelectric (Glatzmaier et al., 2016). Simple hybrid arrangements such as solar steam integration into coal-fired power stations (sub critical and supercritical turbines with solar integrated either at HP inlet, reheat or through feedwater heating, or through solar preheating of the secondary air) should be pursued where opportunities afford as in principle they can be very cost-effective though the solar contributions are relatively low (Prosin et al., 2015; Deng, 2013). As turbine suppliers for the fossil industry seek to lower CO_2 emissions through higher efficiency, it may also be valuable for them to ensure their new designs make later retrofit, for example of solar thermal steam, technically easier to accommodate. Similarly, the integration of solar steam and sub-critical steam turbines as the bottoming cycle with gas turbines (commonly called integrated solar combined cycle systems) (Kelly et al., 2001) is essentially a matter of engineering and is in commercial use today (NREL, 2017), though again the solar contribution is low, of the order of 1–4% of annual energy (Pihl et al., 2013), up to double this if the solar multiple is increased and some storage is included.

A number of more advanced power cycles are under active consideration by the CSP industry today with a view to commercial application in the short to medium term future, and are discussed below.

2.1. Supercritical steam

In the family of steam turbine Rankine cycles, the next most advanced option is supercritical steam turbines. The critical point for water is 374 °C/22.1 MPa. Beyond this there is no nucleate boiling thus steam generation is once-through without the need for separation in a drum. The coal-fired generation industry has progressed these state-of-the-art steam turbines enormously, with improved materials leading to ultra-supercritical turbines with double reheat now being deployed with turbine inlet conditions of 29 MPa/600 °C/620 °C/620 °C and net plant efficiency (LHV) of 47.94% [IEA]. As this efficiency includes the coal boiler, turbine cycle efficiency would be of the order of 50%. Designs are also being progressed for advanced ultra supercritical with inlet conditions of 35 MPa/700/720/720 °C (Fig. 5). However steam turbines operating above supercritical conditions and exhibiting these efficiencies are much larger than the largest CSP plant today (Peterseim and Veeraragavan, 2015) and at these capacities would only be feasible if multiple solar towers were employed using an intermediate heat transfer fluid such as a liquid metal or bulk particle heat transport to a central location. Studies have shown that if smaller turbines, of the order of 200 MW, were to become available, there could be modest improvements in LCOE (<10%) whilst still needing relatively advanced direct absorption receivers (Kolb, 2011; Singer et al., 2013).

2.2. CSP PV hybrids

Though the benefits of locating stand-alone PV and CSP-TES (thermal energy storage) plants in close proximity are well understood for their commercial benefits, the CSP PV hybrid technology aims to make lowest cost use of the entire solar spectrum by using the short wavelengths (blue end) of the spectrum for PV's and the longer wavelengths for use in a thermal application (Orosz, 2015; Imenes et al., 2007; Lasich, 2009). The "conventional" arrangement involves interposing a dichroic filter between the primary reflector and the receiver where the filter's band edge matches the band gap of the PV device, generally around 1.4 eV – for such a device about 30% of the full spectrum energy is not utilised by the PV. The longer wavelengths pass through to the receiver, and the shorter wavelengths are reflected to the PV device. Thus the one collection device can be optimised for electron production and heat generation for low cost thermal storage, to be subsequently used for either dispatchable electricity or some other thermal application.

An alternative arrangement involves high temperature PV's operating at 350–400 °C on top of the thermal absorption surface, with unused energy from the PV being used for low cost thermal storage and dispatchable solar (Fig. 6).

2.3. Stirling engines

Of all CSP technologies, the dish Stirling engine system still holds the efficiency record for demonstrated conversion of solar energy to 3-phase power at 31.25% by Stirling Energy Systems (Andraka, 2013) though there is a more recent report of 32% by Ripasso Energy in Sweden. In any case the Stirling engine, with its close approximation of the Carnot ideal and relatively small capacity, and the dish, with its excellent optical performance but

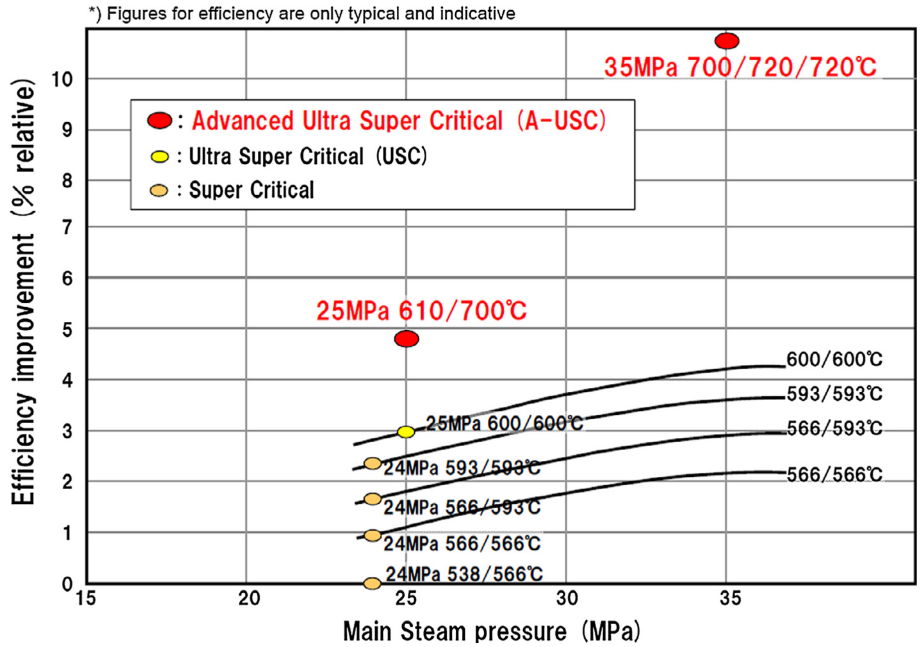

Fig. 5. The future for supercritical steam turbine cycles (Toshiba, 2015).

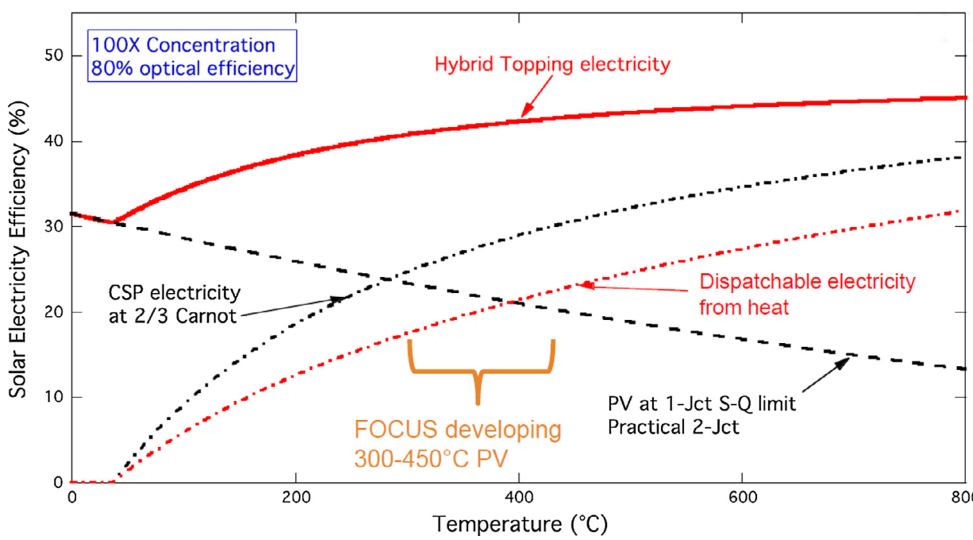

Fig. 6. Topping arrangement of PV/CSP hybrid for high efficiency dispatchable solar (Branz, 2014).

limits to aperture area, offer a highly efficient combination, with Stirling engine inlet temperatures of the order of 700–800 °C (Schiel and Keck, 2012). Dish Stirling engines received considerable attention for their modularity and ability to be clustered to suit larger outputs, until some years ago when PV's, with no moving parts (usually) and ability to work in a wide variety of sites and situations became a dominant market player. Part of the issue was perhaps a non-solar one in that Stirling engines have not been able to penetrate enough markets to allow them to become low cost and reliable. More recently attention has moved to integrating storage so they are no longer a variable renewable but a dispatchable one. Andraka (2013) has reported on development of integrating phase change material storage combined with heat pipes to allow 6hrs of storage to be contained within 1 m^3 on the back of a 100 m^2 dish providing 1.25 solar multiple for a 25 kWe engine.

3. Supercritical carbon dioxide closed loop Brayton cycle

Power cycle efficiencies of the order of 40% are unlikely to be sufficient to allow CSP to be competitive in the future. The US DoE Sunshot Initiative for example has recently revised its cost target for utility scale PV's from 6c/kWh in 2020 (now largely achieved) to 3c/kWh by 2030 (US DOE Sunshot, 2016). Though, unlike CSP, this does not incorporate the inevitable additional cost of storage, there is no doubt CSP needs to make some considerable technological leaps to remain competitive. A high efficiency power cycle receiving considerable interest at present by all sectors of the power industry is the supercritical CO_2 (sCO_2) closed loop Brayton cycle. Steam Rankine cycles suffer from the significant loss of low grade thermal energy at the condenser because the cycle relies on condensation in order to be able to pump water. Since water is essentially incompressible, the pumping power required is low, of the order of 2%, but approximately 50% of low temperature energy is lost to cooling water. Open cycle gas turbines on the other hand have no cooling loss but use approximately 45% of the power produced by the high temperature expansion in compressing air, along with considerable heat loss in the exhaust. Thus the efficiency of industrial gas turbines is usually slightly less than steam turbines, with aeroderivative gas turbines being similar to steam turbines.

The attraction of a cycle using a supercritical fluid in a closed loop is that it combines the best attributes of both Rankine and standard air Brayton cycles by recovering exhaust heat but needing only to compress a supercritical fluid which ideally will have very low compressibility. However for the benefits of this characteristic to be realised, compressibility must be low at a temperature compatible with likely sites for CSP, ie high dry bulb temperatures. Fig. 7 shows the significant change in compressibility of CO_2 over a small temperature range around its critical point (31 °C). It is noteworthy, therefore, that Turchi (2013) assumed a compressor inlet temperature of 56.5 °C in modelling of a sCO_2 cycle for Daggett, California, some way removed from the critical temperature, but representative of peak CIT's likely in many CSP locations.

There are other candidate fluids, with a small selection shown in Table 1. Though none of the fluids are "perfect", carbon dioxide has been progressed, originally for the nuclear industry where cooling temperatures match well, but also due to its chemical stability in the range of interest, industry handling knowledge (eg carbon capture and sequestration), and turbine dimensions that are an order of magnitude smaller than steam turbines (Dostal et al., 2004). It is noted that supercritical ethane has been investigated as an alternative fluid of interest (Enriquez, 2015) though for temperatures below 400 °C due to chemical decomposition. Closed loop Brayton cycles based on helium or nitrogen have also been developed (Forsberg et al., 2006; Wright et al., 2006; Dunham and Iverson, 2014) though inlet temperatures typically need to be higher than sCO_2 to achieve the same efficiency.

Though turbine physical size is often noted as one of the key attributes due to the possibility of lower turbine capital costs, materials notwithstanding, it is important to recognise that the cycle only performs well because of significant recuperation. Thus heat exchangers are a fundamental component when considering benefits and optimising design. The quantum of thermal energy handled by the recuperators is much greater than in the turbine itself, and the total cost of recuperators is a high proportion of the power block, so compact, low cost, high effectiveness designs are essential. A range of heat exchanger effectiveness from 0.9 to 0.98 is commonly assumed for sCO_2 power cycles (Carlson et al., 2014), though there is still much more work required on complete system performance before a smaller ideal effectiveness range can be determined. Printed circuit heat exchangers (PCHE) based on diffusion bonded plates with small hydraulic diameters are used for the recuperators in most of the experimental applications to date (Pasch et al., 2016; Stein et al., 2016), with cast metal heat exchangers (CMHE) also under development. It is also important that all components in this cycle are modelled based on real gas properties rather than ideal, due to the significant property variations that occur around the critical point. One particular issue is specific heat, which spikes in this region such that the surface area required for the cooler can change enormously over a very small temperature difference, though the effect is much less for dry cooling further from the critical point. The other effect is that the specific heat of the cold side of the recuperator can be two to three times higher than the hot side (Ahn et al., 2015). This limits the maximum temperature that the high pressure CO_2 can be raised to. The solution for this is to use a recompression cycle, such that part of the flow is split to match heat exchange properties on both sides of the recuperator and increase effectiveness.

Table 1
Critical point for some candidate fluids.

Fluid	Formula	Critical temperature (°C)	Critical pressure (MPa)
Ammonia	NH_3	132.89	11.28
Carbon dioxide	CO_2	30.98	7.38
Sulfur hexafluoride	SF_6	45.56	3.76
Water	H_2O	373.89	22.10
Xenon	Xe	16.61	5.88

A number of cycle configurations have been proposed for various heat sources and there is a certain amount of optimisation of different configurations to different applications. In the case of CSP, several drivers must be considered:

Turbine capacity: CSP turbine unit capacities to date have either been large (preferred >100 MW due to steam turbine

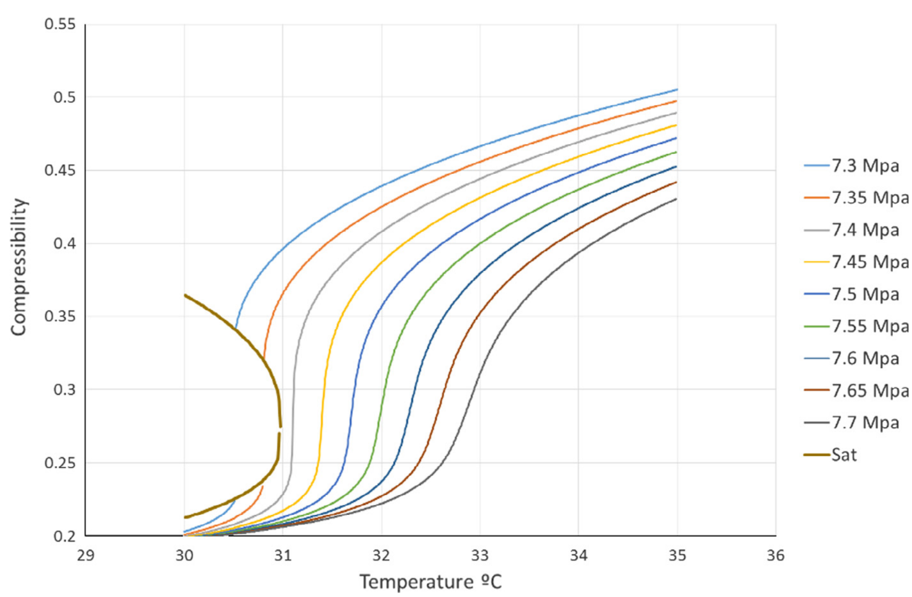

Fig. 7. Compressibility of CO_2 near its critical point, data generated using REFPROP.

characteristics) or small (dish Stirling <100 kW) whereas there is an emerging market for dispatchable solar technologies in between these capacities, particularly as electricity growth slows in some markets and the share of variable renewables grows, placing greater emphasis on agile technology responses. One of the potential attractions of sCO_2 is that it could provide high efficiencies at smaller capacities where steam turbines would not be suitable. An example is the 10MWe plant as shown in Fig. 8 which is relatively complex, but shows the opportunity for high efficiency.

Storage integration: Low cost storage forms an essential part of the value proposition that CSP offers and so the cycle must be amenable to storage integration. It may even be sensible in some cases to reduce the turbine inlet temperature and accept the negative effect on efficiency, but allow use of a lower cost storage solution to yield greater net present value (NPV) overall. The lower ΔT across the primary heat exchanger for a recuperated sCO_2 Brayton cycle than for a Rankine cycle also impacts the choice of storage technology.

Turbine temperature and power block cost: An area of uncertainty at present is the effect of increased turbine temperature on power block (ie turbine, recuperators, high temperature piping) capital cost due to more exotic materials. A study (AghaeiMeybodi et al., 2016) investigated TIT's of 560, 610, 700 and 1000 °C based on a recuperated partial recompression configuration (Fig. 9). The results showed that even though the efficiency increased strongly, the cost of storage (which was a variable in the analysis) needed to maintain an annual capacity factor of 46.9% and meet an LCOE cost target of approximately US9c/kWh was similar for the 560 and 610 °C cases and slightly lower for the 700 °C case, meaning that slightly lower cost storage was needed to keep the same LCOE as the other cases (see Table 2).

Though much more work is to be conducted by the turbine industry before there is any certainty on sCO_2 power block costs, the point to be made is that when higher temperatures and more exotic materials are involved, increased costs can sometimes outweigh the efficiency benefits in the search for an optimised cycle. Steiner et al. (2016) for example suggests that temperatures of >650 °C may be needed before sCO_2 cycles provide greater performance than steam turbines. Cheang et al. (2015)) studies subcritical steam, supercritical steam, and supercritical CO_2 for CSP systems in a technoeconomic analysis and concludes that subcritical steam provides the best outcome. The sCO_2 efficiencies appear low, and $/kW for the power block high, compared to other literature (White et al., 2015) - in particular the sCO_2 turbine cost is based on a learning rate derived from a 2nd or 10th turbine built which, along with the low efficiencies, would account for the high $/kW. Given that steam turbines have had over a century of development, it would be interesting to see an "Nth-of-a-kind" sCO_2 plant analysis. Nonetheless the instructive point of this paper is the influence of cost in an optimisation process. The Quadrennial Technology Review (US DoE, 2015) conducted a comprehensive analysis of steam turbine vs sCO2 performance and shows that the sCO_2 recompression cycle has higher performance than steam above 425 °C (Fig. 10). It is noted that there is a broad range of efficiencies (cycle and isentropic) quoted in the literature. In addition the modelling approach for heat exchange in the recuperators and cooler is often based on the conventional industry method for ideal gases of assuming an effectiveness, whereas the non-ideal and very non-linear properties of sCO_2 near the critical point mandate a more detailed analysis based on discretised heat exchangers and an enthalpy-based heat transfer calculation (Turchi et al., 2013). Even though CSP systems will most likely include storage, modelling also needs to include part load and off-design performance (Dyreby et al., 2014).

Another characteristic of the closed loop sCO_2 cycle that will impact the selection of operating conditions and configurations is the relatively small temperature difference across the primary heat source to the power block, of the order of 150 °C, compared to a steam turbine of approximately 300 °C and an unrecuperated closed loop Brayton cycle with around 500 °C. As the primary heat source will usually be thermal storage in a CSP plant, a lower ΔT would require an increased inventory of bulk storage material

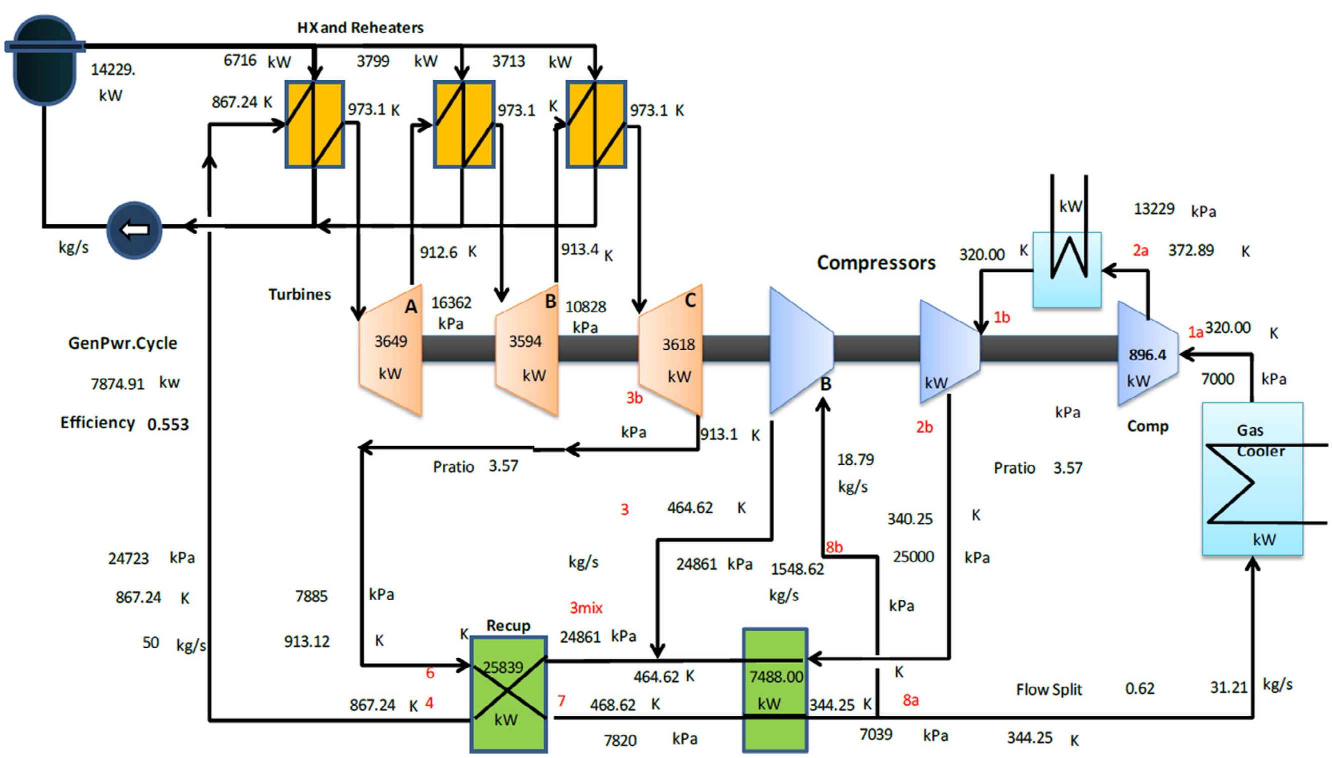

Fig. 8. 10MWe sCO_2 cycle designed for high efficiency, 650degC TIT and 47 °C CIT (Wright et al., 2011).

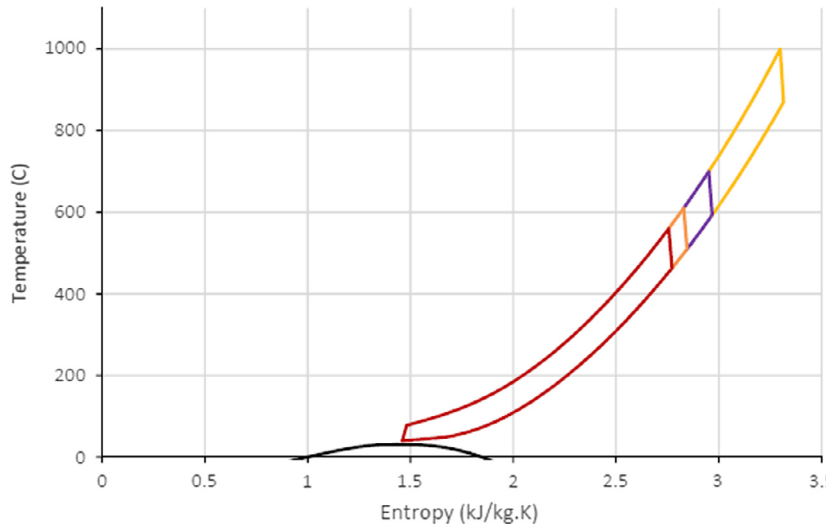

Fig. 9. Temperature entropy diagram for recuperated sCO_2 cycle with TIT's of 560, 610, 700 and 1000 °C.

Table 2
Analysis of increased TIT on overall system benefits (from (AghaeiMeybodi, 2016)). Other sources suggest lower power block costs (Ho et al., 2016).

TIT, °C	Modelled cycle efficiency	Estimated sCO_2 power block cost $/kW	Cost of storage ($/kWhth) needed to break even at 9c/kWh (range due to probabilistic analysis)
560	45.7	1110	18.26–38.78
610	48	1080	19.31–46.93
700	51.4	1590	17.41–29.71

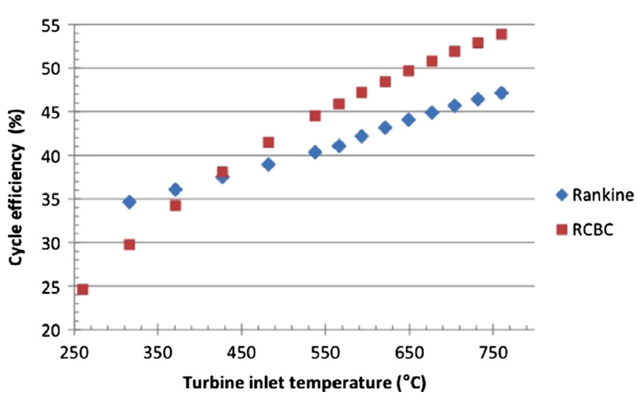

Fig. 10. Cycle efficiency of subcritical steam Rankine vs recompression sCO_2 Brayton. Courtesy US DoE.

and increased flow rate in the case of sensible heat exchange. Integration with high temperature phase change materials promises a valuable thermodynamic match in the case of a low ΔT (Liu et al., 2015). Ho et al. (2016) provides a good analysis of the conflict between wishing to increase the primary heat source temperature difference in order to reduce storage costs, but at the expense of lower thermal efficiency. They conclude that, for the assumptions made, there is a minima for the mass flow rate of the storage heat transfer media at a ΔT of approximately 400 °C, with the best cycles being the recuperated simple closed Brayton cycle, recompression closed Brayton cycle and a combination bifurcation with intercooling (Garg et al., 2014). This latter cycle considers the possibility of operating across the CO_2 transcritical region such that a pump is then included in the cycle. Given that pure CO_2 in a CSP application with dry cooling is unlikely to reach condensation conditions, a mix of CO_2 and propane (with a critical temperature of ≈63 °C) is considered, though stability is noted as being an issue. It is however just one example of investigations into the marked influence of fluid properties on the sCO_2 cycle and the possibility of fluid mixtures, tuned to particular operating climates and applications, in order to maximise annual performance, particularly so for CSP and its higher ambient temperatures. Other molecules that have been considered for CO_2 mixtures include argon, xenon, nitrogen and even oxygen (Jeong et al., 2011). There is still much work to be done on matching different cycle configurations to fluid mixtures.

As noted above, one of the largest unknowns affecting the temperatures and thus efficiencies that sCO_2 cycles can reach is materials. Materials are constrained not only by creep and fatigue (Gardner et al., 2016) but also corrosion-related issues. Eastland et al. (2014)) notes turbine inlet temperatures for materials commonly used in indirectly-heated cycles today is 593 °C (1100F), with 704 °C (1300F) being recently certified for use in Advanced Ultra Supercritical steam. The CO_2 environment (and subsequent increased carburisation potential compared to steam) poses specific material challenges and significant work is under way to analyse this issue (Olivares et al., 2015; Pint et al., 2016; Dheeradhada et al., 2015; Subbaraman et al., 2016) particularly at temperatures above 600 °C where Ni-based alloys and/or protective coatings are likely to be required. The robustness of any protective scale formed is critical as spalling and dislodgement would result in damage to turbine blades or recuperator blockage. There is also an important need for investigation of material life under rotating conditions.

3.1. Options for CSP-based sCO_2 power cycles with other power cycles

Combined cycles, in their generic definition, are well-known. Those based on air cycles are presented in the next section. It is useful to note that sCO_2 cycles can offer promising performance advantages either as a topping or bottoming cycle. Besarati (2013) analyses a sCO_2 cycle as a topping cycle with an ORC bottoming cycle and shows a 3–7% points improvement in efficiency, though cost benefits need to be assessed whenever high temperature heat is traded.

Supercritical CO_2 cycles could, on the other hand be used as the bottoming cycle for some higher temperature fuel cells such as Solid Oxide (Badwal et al., 2014) or Molten Carbonate Fuel Cells

(Chacartegui et al., 2011). A particular application would involve CSP driving a reforming reaction to produce syngas for the fuel cell topping cycle, and then additionally providing heat to an sCO_2 bottoming cycle.

Using a gas turbine as a topping cycle and sCO_2 as the bottoming cycle also holds promise, particularly if smaller total capacities are desired. Huck (Huck et al., 2016) concludes that an industrial gas turbine with well-designed 3 pressure reheat steam cycle would outperform a sCO_2 bottoming cycle with all but very high operating pressures and component efficiencies, but an aeroderivative gas turbine topping cycle with a sCO_2 bottoming cycle would outperform a 2 pressure reheat steam turbine.

In summary there are a number of issues to be addressed if sCO_2 cycles are to reach their potential. These include materials with proven commercial life above 600 °C, low cost compact heat exchangers with ability to handle a high ΔP and material compatibility with different fluids to minimise the possible occurrence of a fluid solidifying in the passages, cost sensitivities for all components, and on the turbine itself seals, bearings and generators (Iverson et al., 2013; Milone, 2011; Brun, 2016; Qin et al., 2016).

There is very limited commercial industrial expertise with sCO_2 turbines at present. The US DoE is looking to accelerate sCO_2 technology by addressing the issues noted above through their Supercritical Transformational Electric Power (STEP) Program (US DoE, 2016) which will build a 10MWe sCO_2 turbine as a test platform for these critical issues. The company Echogen are concentrating upon waste heat recovery as the initial path to market (Held, 2016). Another company, NetPower, have broken ground for a 50MWth demonstration of an advanced sCO_2 cycle called the Allam cycle, based on natural gas combustion by oxygen at 1150 °C/30 MPa (Netpower, 2016).

4. Solar gas turbine systems

4.1. Introduction

Modern conventional gas turbine systems achieve in Combined Cycle (CC) configuration thermal conversion efficiencies in the range of 60%, at reasonable investment cost. This is a nearly 50% higher conversion efficiency than power blocks used currently in CSP systems. For this reason the application of solar gas turbine systems looks very attractive, as the high thermal conversion efficiency results in a significant reduction of solar field size at a given power level. Since the solar field is the main cost contribution in a CSP system, this promises a direct reduction of solar LCOE.

Integrating solar energy into a gas turbine system is achieved by heating the pressurized air partially or fully by solar energy. The conventional fuel combustor is either supported or fully replaced by a solar receiver. Fig. 11 shows a scheme of solar gas turbine system, with a bottoming steam cycle (CC configuration).

Development of solar gas turbine systems has been on-going for several decades, but has not entered commercial application yet. A detailed review of solar gas turbine system R&D can be found in Blanco and Santigosa (2017).

4.2. Solar gas turbine cycle concepts

Modern high efficiency gas turbines operate at very high turbine inlet temperatures (TIT), in the range of 1500 °C. Except for lab-scale devices, it is not presently possible to provide such high temperatures directly with CSP systems (an indirect arrangement involves the use of a solar-driven thermochemical process to produce a fuel such as syngas that can then be combusted in the gas turbine combined cycle). Current point focus technologies like parabolic dishes or solar tower can nowadays provide temperatures up to around 1000 °C. Two options exist to overcome this mismatch: (a) using the solar system as preheater, then heating the air to the required TIT with fuel combustion, or (b) developing "solar-specific" gas turbine models that operate at a TIT of about 1000 °C. While option (a) results in a significantly reduced solar share, option (b) means the development of gas turbines with different design strategies than modern gas turbines.

For solar heat input into a gas turbine it must be possible to preheat the compressed air before entering the turbine section. Conventional gas turbines are not designed for solar heating, but must be modified in several aspects:

- air path: compressed air extraction and preheated air re-introduction
- "solarized" combustor, in parallel or serial connection
- optimization for solar operation (e.g. reduced TIT)
- adapted safety and emergency measures
- adapted control system

Several solar gas turbine system concepts were developed, the most important ones are:

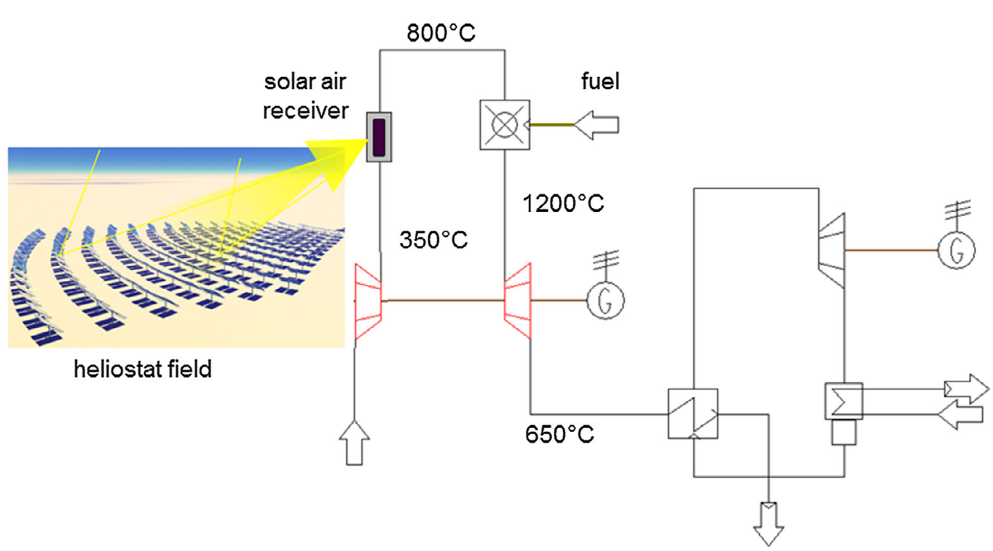

Fig. 11. Schematic of a solar gas turbine system with bottoming cycle.

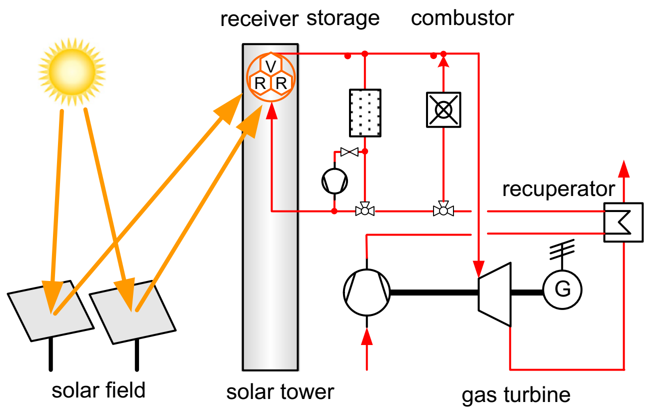

Fig. 12. Scheme of a recuperated solar gas turbine system with storage.

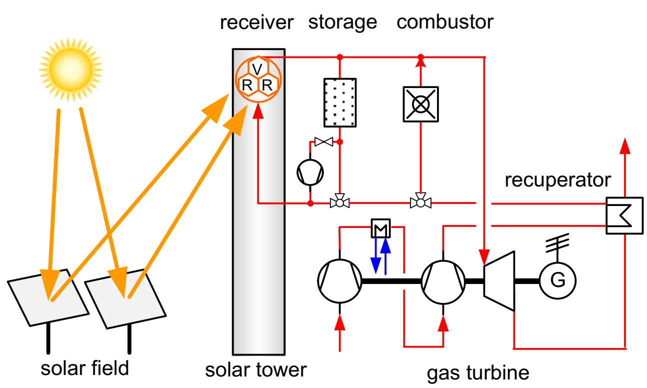

Fig. 13. Solar gas turbine cycle with intercooling.

- Combined Cycle with steam or organic Rankine bottoming cycle
- recuperated gas turbine systems
- both with or without intercooling

A solar gas turbine in CC configuration is shown in Fig. 11. The CC configuration is common in conventional power blocks at higher power levels. Fig. 12 is a scheme of a recuperated solar gas turbine cycle, where the compressed air exiting the compressor is preheated by the (still hot) air leaving the turbine. In this configuration the inlet air of the receiver is at significantly higher temperature (typically up to 600 °C). Note that Fig. 12 also includes a storage system, as described later. Conventional recuperated cycles are typical for smaller power levels, commercially available from microturbine level up to about 20 MWe.

In both configurations intercooling between compressor stages can improve thermodynamic performance of the gas turbine cycle, as the power consumed for compression is reduced. Fig. 13 shows a scheme of an intercooled solar-hybrid system. Due to the increased complexity of the compressor section with additional heat exchangers and air ducting, only a few commercial gas turbines use intercooling.

Further concepts include the so-called Integrated Solar Combined Cycle Systems (ISCCS) and the Bottoming Cycle Storage Systems (BCSS). In the ISCCS concept as mentioned in Section 2, the topping gas turbine cycle is only run on conventional fuel, and solar energy is introduced into the bottoming steam cycle, usually by adding solar-generated steam. This concept results in low solar shares of less than 10% on an annual energy basis. In the BCSS system (Fig. 14), the gas turbine is operated in solar-only or solar-hybrid mode, intended mainly for operation during sunshine hours. The hot exhaust gas of the gas turbine is then powering a bottoming cycle and charging a medium temperature storage system, e.g. a regenerator (Agalit et al., 2015). The bottoming cycle can then be operated directly from gas turbine exhaust, from storage or in mixed mode. Reducing the power level of the bottoming cycle extends the daily operation time. Thus, high power delivery is possible during sunshine hours (from gas turbine and bottoming cycle), and lower power (from the bottoming cycle only) during the remaining hours.

In (Puppe et al., 2015) several configurations were compared with respect to efficiency and cost. The comparison was based on the assumption of a TIT of 970 °C, enabling very high solar shares in combination with storage systems, and two operation modes:

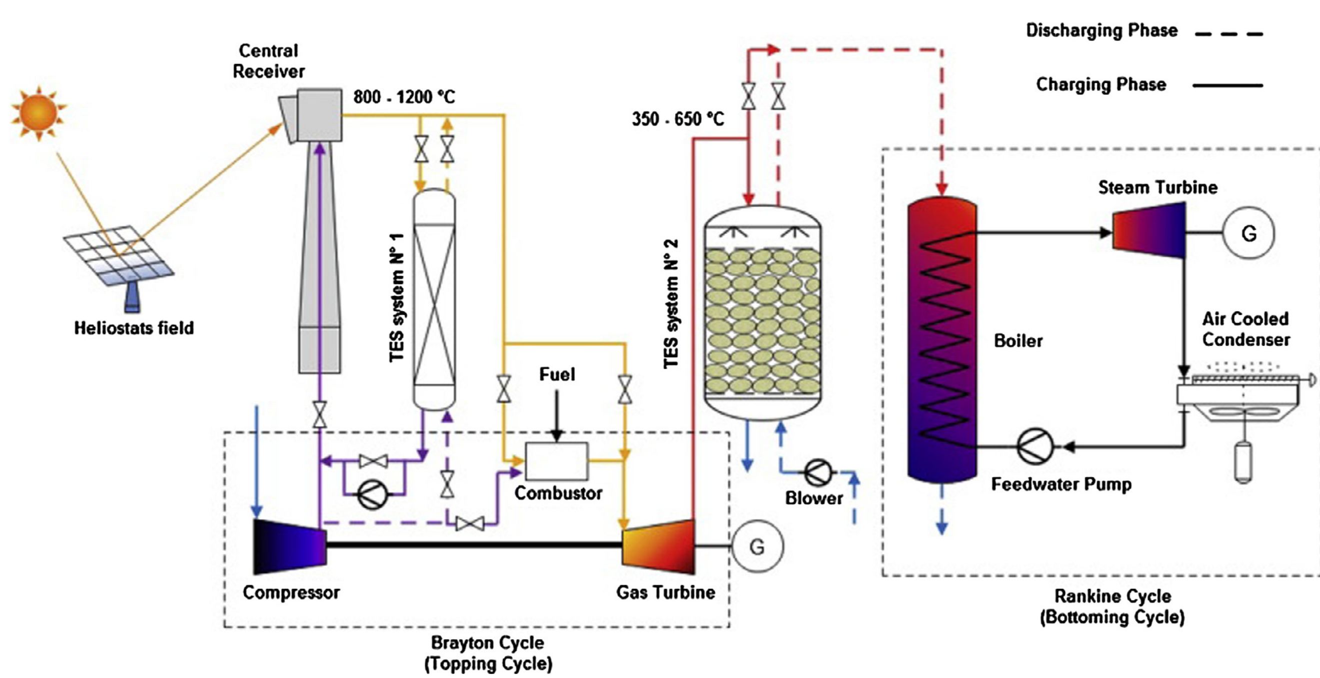

Fig. 14. Gas turbine with bottoming cycle storage system (Agalit et al., 2015).

operation during sunshine hours or baseload operation. Due to the reduced TIT, the gas turbine power block efficiencies ranged from 42% to 46%. Combining efficiency with cost data, the intercooled cycle looked most promising.

4.3. System components

4.3.1. Gas turbine

External air heating of the compressed air requires interface ducting between the gas turbine and the external components of the solar subsystem. Gas turbines with external can combustors allow easy adaptation with little modification to the gas turbine casing. Several gas turbine manufacturers have units with external can combustors in their portfolio. Also, most recuperated gas turbine models already have the interfaces for external air heating, as the recuperator itself is an external air preheater. An example of such a recuperated gas turbine is shown in Fig. 15, where the air ducting between compressor, recuperator and combustor can be identified.

However, most of the modern gas turbine units use annular combustor systems that are highly integrated with all other components. In this case a major redesign of the casing and the air flow path would be required to allow external air preheating.

External air heating can be performed via direct or indirect heating. Direct heating is the preferred solution, with the solar receiver directly integrated into the compressed air flow path. Indirect heating using a high temperature heat exchanger is another option that allows use of a different heat transfer medium in the solar subsystem. Due to the high temperatures, such heat exchangers are quite expensive and are currently not commercially available.

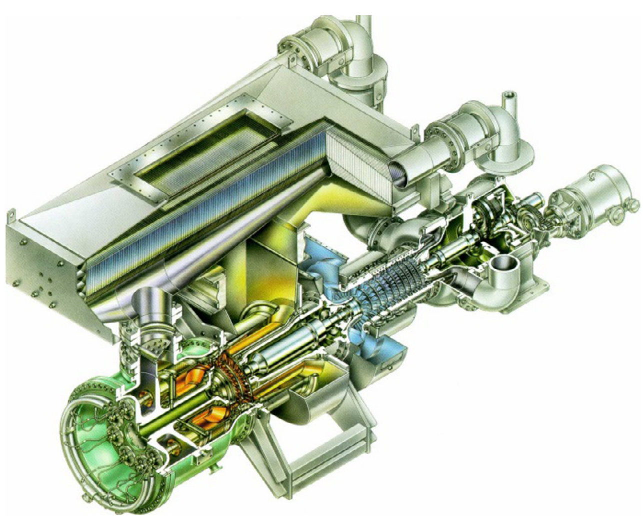

Fig. 15. Standard Mercury-50 gas turbine from Solar Turbines (©Solar Turbines).

In a solar gas turbine system, a number of external components are included in the flow path between compressor and turbine, mainly the solar receiver, storage system (if used), connection piping and flow control equipment. These components result in additional pressure drop, volume and thermal inertia, affecting gas turbine performance and control.

A combustor can be integrated in any solar gas turbine system, even if solar-only operation is foreseen at design point conditions. This allows stable operation during transients and ensures system availability even without solar energy. The combustor can be integrated in serial or parallel connection (see Fig. 16).

In serial connection, the combustor can receive high air inlet temperature (up to 1000 °C), varying dependent on the actual solar power input and resulting air preheating. Due to the serial connection, the pressure drop between receiver and turbine section is increased. Fuel flow is controlled according to the air inlet temperature, resulting in different combustion regimes and flame stability issues. The fuel/air mixture also tends to self-ignition at higher air inlet temperatures, with the risk of component damage. Development work on a combustor for air inlet temperatures up to 1000 °C is described in (Coogan et al., 2014).

In parallel connection, the compressor mass flow is split between receiver and combustor (Fig. 16, middle), with the split ratio depending on the actual solar power input. The combustor always receives air at relatively low temperature, but with a large range of mass flow. Appropriate control valves equalize the resulting pressure drop in the parallel flow paths of receiver and combustor. Typically the resulting pressure drop between compressor and turbine section is lower than in serial connection, enabling slightly higher cycle efficiencies.

Component cooling, especially for the combustor walls, and material issues are critical in both configurations. Further development is required to provide proven and cost-effective solutions for those components.

4.3.2. Receiver

Several receiver types for SGT systems have been developed, mainly for direct heating of the compressed air. Tube receivers and volumetric receivers are the most advanced technologies. Fig. 17 shows a tube receiver that was developed in the SOLUGAS project (Korzynietz et al., 2016). This cavity receiver was successfully tested at temperatures up to 800 °C, together with a Mercury-50 gas turbine.

Another tube receiver concept was proposed for the French PEGASE project. This concept uses multiple tubes embedded in a copper body that is enclosed in a nickel based super alloy (Grange, 2001). The copper body effectively redistributes the absorbed power to multiple layers of tubes, thus increasing the convective heat transfer surface. Simulation studies predicted a receiver efficiency in the range of 80%.

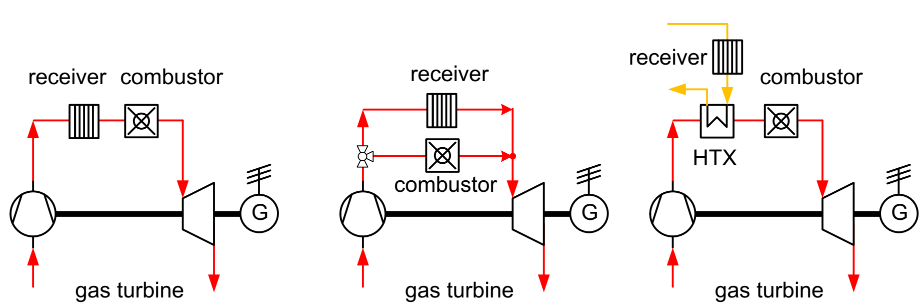

Fig. 16. Combustor integration options (left: serial, middle: parallel, right: serial, indirect heating).

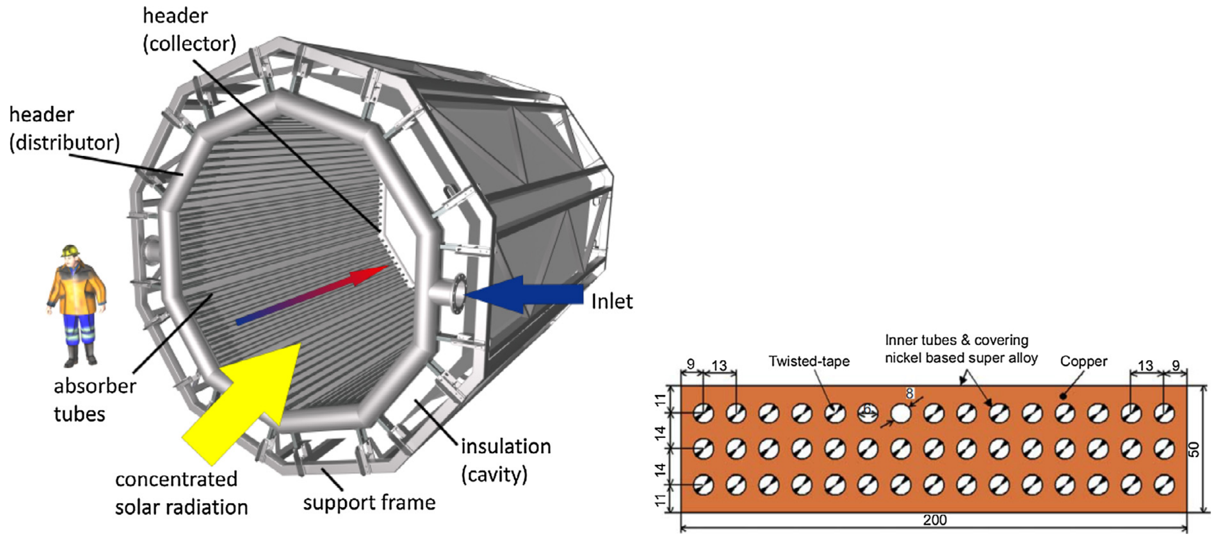

Fig. 17. SOLUGAS tube receiver (left); PEGASE embedded tube receiver (right).

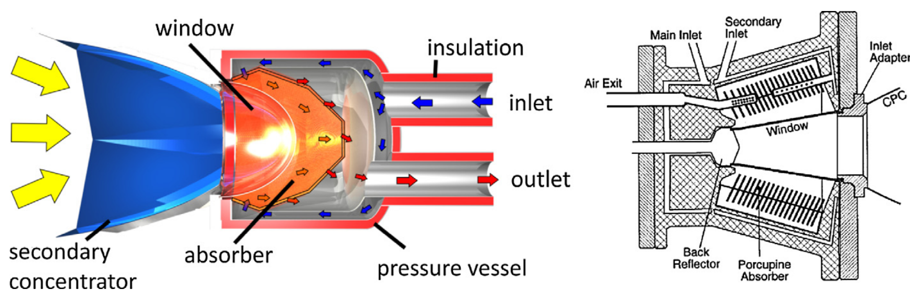

Fig. 18. Pressurized volumetric receivers for high temperatures (left: SOLGATE; right: DIAPR).

Pressurized volumetric receivers are the second important development line. Such receivers use highly porous structures for solar absorption and heat transfer. Effective heat transfer is realised by the large internal surface area, as the solar radiation can penetrate into the porous structure. The air is passing through the structure and is convectively heated by the solar energy absorbed in the structure. To operate a volumetric receiver under pressure, a quartz window must be used to cover the receiver aperture and form an enclosed containment. Several receiver versions were successfully demonstrated. A modular receiver using a SiC foam absorber (Fig. 18, left) was designed for a power of 400 kWt and was later tested up to 1000 °C within the SOLGATE project, the first full system demonstration using a commercial gas turbine (Buck et al., 2002; Heller et al., 2006).

The Directly Irradiated Annular Pressurized Receiver (DIAPR), designed for air at up to 30 bar and 1300 °C, uses multiple ceramic pins as volumetric absorber (Fig. 18, right). A prototype of this receiver was tested reaching outlet temperatures of 1200 °C with receiver efficiencies of 70–90% at a pressure level of up to 20 bar (Kribus et al., 2001).

4.3.3. Storage

To achieve high solar capacity factors, the use of thermal storage systems is necessary. Regenerative storage systems are seen as a feasible solution. Commercial regenerators for high temperatures exist as so-called Cowper stoves used with blast furnaces. For the storage inventory a fixed bed with stacked material or a packed bed with beads or other filler material are possible. Because of the high temperature, ceramic materials must be used.

The storage system must be installed in parallel to the receiver, as shown in Fig. 12. The solar field and receiver of a CSP system with storage is oversized to enable storage charging during daily sunshine hours. A blower at the cold exit of the storage delivers a controlled additional air mass flow through the receiver and the storage in a closed loop, not altering the mass flow through the compressor and the turbine section. System operation with storage is as follows:

- Charge mode: when the receiver delivers more power than the gas turbine can take, the excess power is taken by an increased mass flow, obtained from the compressor and an additional mass flow introduced by a blower downstream of the storage unit. Thus, a certain fraction of the receiver outlet air is passing through the storage. By controlling the blower, the receiver outlet temperature is maintained at the desired outlet temperature.
- Discharge mode: when the receiver delivers less power than the gas turbine requires, the mass flow from the compressor is split into a fraction towards the receiver and another through the storage (with flow then reversed compared to charge mode).
- in both modes fluctuations in the receiver/storage outlet temperature can be compensated by a combustor connected in parallel or serial.

4.3.4. Control

In a solar gas turbine system, the external solar components (receiver, piping, storage) represent an additional pressure drop showing high thermal inertia and a large additional pressurized volume. Special measures need to be taken to safely operate such a system. During normal operation, modifications include mainly the settings of the control parameters to reflect the changed transient response. The fast response of the combustor enables stable operation as long as the combustor is within its safe operation regime. However, also hardware changes must be implemented for emergency situations (e.g. load shedding). In such situations overspeed in the turbine section can occur within sub-second time periods. Several measures were proposed or tested to safely handle emergency situations:

- addition of blow-off valves and/or shut-off valves, both with fast reaction time
- mechanical brakes
- shunt resistors loading the generator

In the SOLGATE project (SOLGATE, 2005) a blow-off valve was added between the receiver and the combustor. Before the valve, a water cooler was installed in the flow path to limit the air temperature at the valve. This allowed the selection of an inexpensive low-temperature valve. An orifice plate was installed downstream of the valve to limit the blow-off mass flow when the valve is open. Thus, the mass flow through the turbine was reduced while also limiting the pressure gradient in the piping. This was especially important since the internal insulation of the piping could have been damaged by too sharp pressure gradients.

Felsmann et al. (2015) developed a dynamic simulation model and investigated the response to critical operation conditions. He concluded that appropriate and fast-reacting hardware is necessary to avoid damage to the solar gas turbine system. A scheme of the model including the additional emergency hardware is shown in Fig. 19.

4.4. Technology status

Due to the need for high receiver temperatures, solar gas turbines can only be implemented with parabolic dish or solar tower systems. Besides several associated receiver development programs, only a few complete system projects have been demonstrated or are under development.

In 2001, the SOLGATE project started with the goal to demonstrate a first solar-hybrid gas turbine system on a solar tower (SOLGATE, 2005). A modified ALLISON 250 helicopter engine was used, driving a generator. The combustor was designed for air inlet temperatures up to 800 °C, using kerosene to provide the remaining heat input to achieve the nominal TIT. The system was installed at the solar tower test facility PSA (Plataforma Solar de Almería, Spain). Solar testing at up to 230kWe was demonstrated. In a following test campaign, receiver air exit temperatures up to 1030 °C were achieved using an air bypass with a modified receiver unit (Buck, 2005).

In the SOLHYCO project a first step towards commercialization was made (SOLHYCO, 2011). A modified 100kWe industrial microturbine system was integrated with a 200kWt tube receiver. Solar-hybrid system tests were again performed at the PSA, demonstrating system operation with receiver outlet temperatures up to 803 °C (Amsbeck et al., 2010).

In the SOLUGAS project (Korzynietz et al., 2016) upscaling of SGT technology was the main goal. During this project a complete solar-hybrid gas turbine system was built near Seville, Spain, including heliostat field and tower. A modified Solar Turbines Mercury-50 gas turbine was installed, accepting solar preheating up to 650 °C at the combustor inlet. An air bypass around the receiver allowed operation of the receiver at temperatures up to 800 °C. In total, about 1000 solar test hours were accumulated in different load conditions.

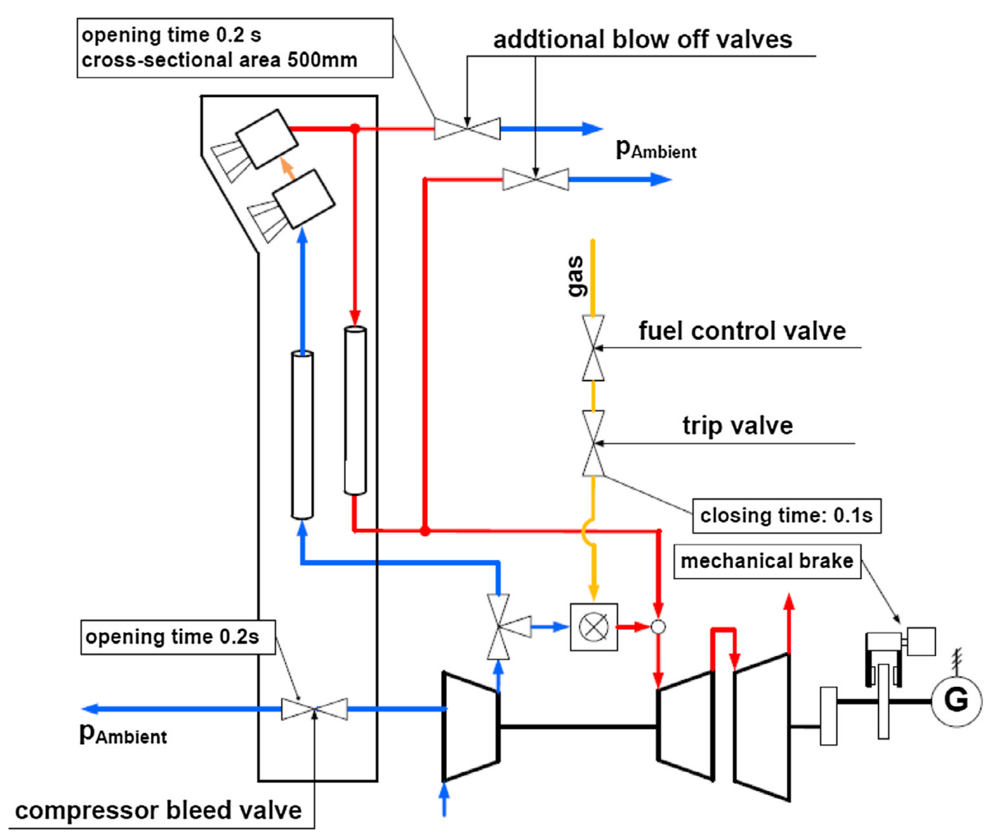

Fig. 19. Measures for control of emergency situations (Felsmann et al., 2015).

Under the acronym PEGASE a number of R&D projects regarding SGT systems were initiated (Grange et al., 2014). As next step a SGT system with 1.4MWe will be installed and tested. Several receiver concepts for high temperatures were developed and tested in the solar tower THEMIS in France. Experimental work is accompanied by system simulation and optimization including thermal storage.

Currently, the company AORA (2017) is the only company offering a small SGT system commercially. The Tulip™ system is based on a 100 kWe microturbine adapted for solar operation. The microturbine and the solar receiver are mounted on a tower, with a small heliostat field providing the concentrated solar radiation. Besides electricity production, up to 170 kW of process heat can be supplied from the hot turbine exhaust gas. Two prototype systems are operational in 2017. Within the OMSOP project (OMSOP, 2017), dish-based microturbine systems were developed, but little information is available on the results.

The company Wilson Solarpower (Wilson, 2017) is planning for mass production of modular SGT systems. The intended module power level is 400 kWe and shall include storage.

Additional to the hardware development, numerous system studies were made to assess the performance and cost reduction potential of SGT systems. Among the most recent ones is the HYGATE study assessing the integration of solar energy and thermal energy storage into gas turbine systems operating at a TIT of 950 °C. In this configuration the solar receiver can provide the necessary temperature for solar-only operation of the plant at design point. Compared to the defined reference molten salt solar tower the solar-hybrid gas turbine plants have higher plant efficiencies, but have a slightly lower potential for CO_2 reduction. The LCoE are comparable and therefore the SGT can be considered as an alternative to molten salt tower plants (Puppe et al., 2015).

A multi-objective optimization was used by (Spelling et al., 2014) to identify Pareto-optimal designs and highlight the trade-offs between minimizing investment cost and minimizing specific CO_2 emissions of SGT in CC configuration with thermal energy storage. The authors concluded that advanced SGT systems can provide up to 60% LCoE reduction, compared to parabolic trough power plants.

4.5. Perspective

Several studies have shown that SGT systems have the potential to significantly reduce the cost of solar electricity, offering full dispatchability, reduced cooling water consumption and simple operation. Although a few demonstration systems were built and operated, SGT systems still need further development before market introduction. The main development tasks are:

- *Gas turbines for solar operation:* Highly efficient gas turbines adapted to solar conditions are not readily available. The required modifications are associated with interfacing to external air heating, modification of the combustor system, component cooling and system control. Currently the gas turbine manufacturers hesitate to do these developments due to an unclear market perspective.
- *High temperature receivers:* The required receiver technology is not mature yet. Several concepts are developed to different levels of technical readiness, but upscaling and long-term operation experience are lacking.
- *General aspects:* Further development effort is required to bring SGT systems to the market. Modular system designs might be appropriate to limit the development cost on the gas turbine and receiver side, distributing costs over a larger number of identical units. Selecting and developing a suitable modular configuration needs to be done.

5. Conclusions and recommendations

As variable renewables make rapid commercial progress, CSP with thermal energy storage is in an excellent position to provide low cost stability and reliability to the grid. However it needs to continue to make technological advances in order to fulfil this promise. One of the most important ways is by increasing efficiency, particularly through advanced power cycles. Such power cycles must not only offer greater efficiency, but do so without exorbitant additional cost and at temperatures well within the limits of collector technologies. Sub-critical steam turbines are commercially robust and offer reasonable efficiency and cost at large scale but do not provide the technical or performance agility needed to respond to rapid market and consumer-led demand changes. Supercritical steam turbines continue to push efficiency upwards with improved materials but the smallest capacity at present is at the upper end of the largest trough and double the size of the largest tower that would be needed for these temperatures. CSP/PV hybrids could provide a way to combine the best of PV and CSP in a single technology, and dish Stirling is being developed with a storage option.

Supercritical closed loop Brayton cycles using CO_2 as the working fluid are at an early stage of development but promise high efficiency at reasonable temperatures across a range of capacities. They are in a position to build on much of the power industry expertise in other turbines – high pressures from supercritical steam, high temperature materials from gas turbines and heat exchangers with novel heat transfer fluids such as liquid metals on one side from the nuclear industry. It is clear from the literature that there is still a wide spread of modelled efficiencies and it is important that analyses incorporate costs and sensitivities, particularly as experimental programs provide more data on material life expectancy and turbine developers better understand isentropic efficiency, component life and scaling issues. It is also crucial that the thermal components – receiver, storage and turbine, are developed together so that operating conditions are optimised jointly. Demonstration projects will play a critical role.

Gas turbines in a combined cycle configuration offer the highest efficiencies of today's commercial power cycles. First technical solutions for the integration of solar energy into gas turbine cycles were developed and tested. In feasibility studies promising approaches were identified, and a coordinated development effort is now required to advance the technological status of the components, particularly receiver and associated high temperature storage, and the adapted gas turbines. It requires the support of gas turbine companies which need to consider ways to technically accommodate high temperature air from a CSP source in their gas turbines.

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