Combined-cycle solarised gas turbine with steam, organic and CO_2 bottoming cycles

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Abstract

Dish concentrators permit the collection of very high-temperature heat, at temperatures limited only by the properties of the containment vessel material. Using this heat efficiently is a major challenge in the design of energy conversion systems for dish concentrators: the heat must be used either immediately on the dish, or else transported, with losses, to a remote energy conversion device.

This paper presents a study of the technical feasibility a solarised combined-cycle gas turbines with a dish concentrator, with several different possible Rankine-cycle working fluids considered: carbon dioxide, toluene, ammonia and water. Carbon dioxide is found to be unsuitable for the bottoming cycle, but options toluene, water and ammonia are all found to be similar at ~46% efficiency. The toluene cycle shows promise because of its suitability for thermal storage.

1 Introduction

The ANU 'SG4' Big Dish is a 500 m² dish concentrator constructed at the Australian National University (ANU) as the result of a collaboration between ANU and Wizard Power Pty Ltd, an Australian startup that aims to commercialise the design. Since construction in 2009, the optical performance of the new dish has been characterised, and, more recently, on-sun thermal tests have been performed with a thermal receiver taken from the earlier 400 m² 'SG3' dish. Work is underway to design a new steam receiver, optimised for the new SG4 collector.

This study aims to determine whether a combined-cycle gas turbine (CCGT) cycle could be viable with the SG4 collector. The intention would be to have a Brayton cycle engine mounted at the focus of every dish, together with a heat exchanger transferring the waste heat from the turbine to a working fluid that would circulate through the whole dish field, then be returned to a central bottoming Rankine cycle.

Gas turbines, it is claimed [1], are cheaper, simpler engines that would be used a great deal more often were it not for the fact that natural gas, usually used to drive these engines, is so expensive relative to coal. Solar heat can be used to replace (or work in parallel with) the combustors of gas turbines, potentially providing a cheaper overall system. Gas turbines also permit much higher cycle temperatures, thereby offering the promise of overall higher cycle efficiencies. The aim is to determine whether a CCGT SG4 is viable.

2 Solar combined-cycle systems: background

Previous studies of solar combined-cycle systems have been performed. Dersch et al, 2004 [2] studied how parabolic troughs could provide the latent-heat part of the input for a Rankine cycle, with a topping Brayton cycle providing the other part. That approach is relevant for trough systems, but not appropriate in the case of point-focus concentrators, because it does not take advantage of the 1000 °C-or-above temperatures possible from these concentrators. Goswami [3] mentions a somewhat similar configuration due to Washam et al, 1993, and also notes the benefit of other bottoming cycle configurations, such as the Kalina cycle, to reduce irreversibilities in the combined-cycle heat exchanger.

The study by Kribus et al, 1998 [4] involved a solar-fired Brayton cycle topping a steam Rankine cycle. As we see in this study, there may be compelling reasons to consider alternative fluids in the bottoming cycle, because of pinch-point issues in the steam generator, and because of irreversibilities in thermal storage.

There do not appear to have been studies of non-steam bottoming cycles connected to solarised gas turbines at this time.

3 Organic rankine cycles

The Organic Rankine cycle (ORC) is a well-known option for conversion of low grade heat to mechanical work. These systems have been used with solar ponds in Israel [5] and low-temperature parabolic troughs in Australia [6] but neither of these projects claim to have achieved thermal-to-electric conversion efficiencies of more than 6%, due mostly the the low (<96°C and <180°C, respectively) temperatures of the heat source.

Drescher and Brüggemann [7] performed an thorough study of fluids suitable for use in regenerative ORCs at temperatures up to 630 K (357 °C, a limitation imposed due to chemical decomposition of many of the fluids they studied). They used Peng-Robinson equation-of-state data with materials from the DIPPR database and found 700 fluids meeting a set of criteria oriented towards biomass cogeneration applications. Of those, after allowing for toxicity, leading fluids were found to be several alkylbenzene fluids, followed closely by toluene. The peak efficiencies were of the order of 25%. The optimisation of that study allowed for heat rejection at down to 363 K (90 °C), which is higher than we would like for the present application, as no cogeneration output is proposed here.

A key consideration with organic fluids for use in Rankine cycles is thermal stability. The solvent toluene is reported to be stable up to around 350 °C or even 400 °C, but above this temperature, it starts to chemically break down [8, 9]. The exact temperature limits appear not to be very well understood, and seem to be affected by oxygen and other impurities. In this study, we assume an upper limit of 375 °C for toluene.

It is clear that a detailed material selection study, including cycle modelling as well as a large set of considered fluids could lead to a substantial improvement in cycle efficiency and/or cost.

4 Modelling system

4.1 ASCEND

The free, open-source modelling environment ASCEND has been used for the modelling in this paper. ASCEND provides an equation-based object-oriented modelling language that allows system models to be built up hierarchically from simpler sub-models. Models can be entirely described in ASCEND code, or can optionally make use of subroutines in external C/C++ or Fortran code.

ASCEND is primarily a solver for non-linear systems of equations, but has some degree of support for dynamic modelling (ODEs and DAEs), boundary value problems, optimisation, and mixed integer/non-linear programming (MINLP). The software is an open collaborative effort, first started at Carnegie-Mellon university in the 1980s. New users and contributors are very welcome.

The main advantage of ASCEND over spreadsheets or Matlab scripts is the fact that one can easily reuse model components, and one can reconfigure the model (fixing one value, and making another value an unknown) very easily, without having to change any model code.

4.2 FPROPS

FPROPS is a new thermodynamic properties library for pure components, developed by the author in collaboration with other members of the ASCEND community[1]. The library makes use of Helmholtz fundamental equation correlations to determine fluid properties from the highest-accuracy published data available in the literature. These are the same publications used for property calculations in commercial software packages such as *REFPROP*, *PROPATH* and *FLUIDCAL*.

Features implemented as of the time of writing were those required for the present simulation, namely:

1. evaluation of reduced Helmholtz function (real and ideal components) and all first and second derivatives as functions of (T, ρ),
2. property functions for pressure, entropy, enthalpy, Gibbs energy and Helmholtz energy in terms of 'native' variables (T, ρ),
3. property derivatives for arbitrary combinations $(\partial a/\partial b)_c$ as function of (T, ρ),
4. saturation curve solution from Maxwell phase criterion, iteration converges reliably for all

1 Detailed information is available at http://ascendwiki.cheme.cmu.edu/FPROPS

temperatures from triple point to critical point, and

5. iterative solution of fluid state for desired values of (p, h)

FPROPS has been validated using data sourced from the original publications, where available, together with data evaluated from REFPROP 8.0 software [10] . Currently it includes correlation data for a set of 36 pure substances including water [11] , CO2 [12], ammonia [13] and toluene [14].

5 Design and optimisation

All the models presented in this section have been uploaded to the web and may be accessed at http://ascendwiki.cheme.cmu.edu/CCGT.

5.1 Simple Rankine cycle model

An initial investigation of cycle design we performed to assess the relative merits of the fluids water, toluene, carbon-dioxide and ammonia. For this initial investigation, a simple Rankine cycle was modelled, with the following constraints:

- T_{max} = 580 °C in boiler (unless working fluid thermal stability limits are lower)
- p_{max} = 150 bar
- x_{min} = 0.9 within the turbine
- T_{min} = 40 °C in condenser (allowing for an ambient temperature of 30 °C)
- Turbine efficiency η_t = 0.85, pump efficiency 0.8.

The results for the simple Rankine models are shown in Figure 1.

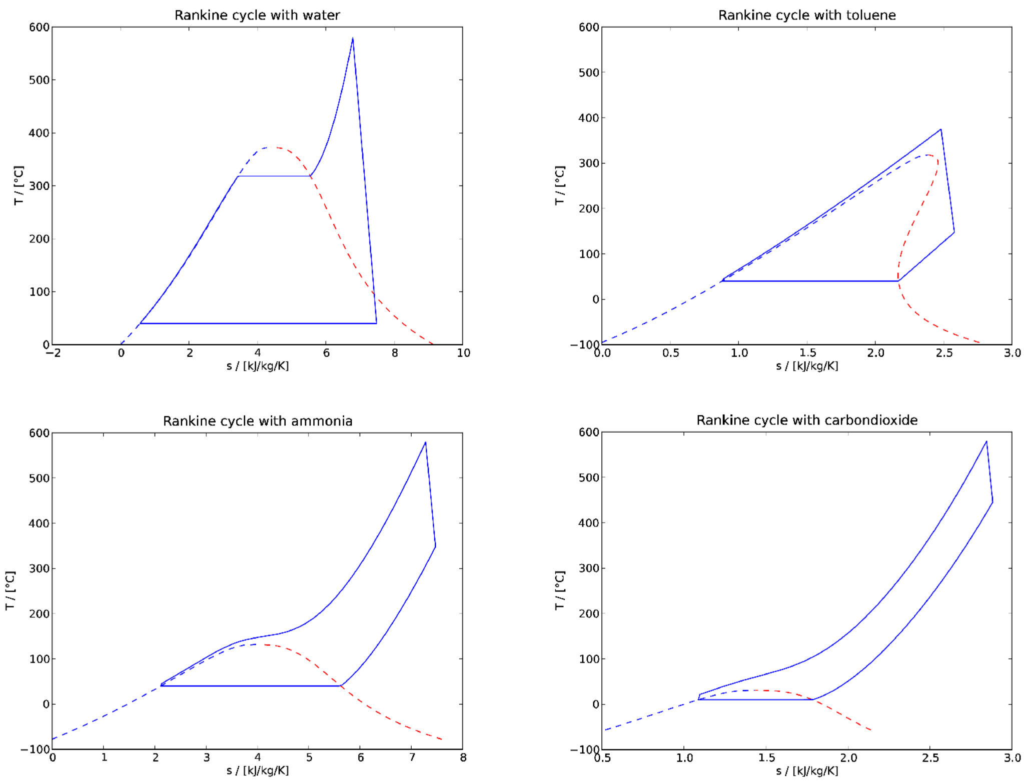

Figure 1. Simple Rankine cycles for water, toluene, ammonia, and carbon dioxide. The cycle temperature limits are 580 °C and 40 °C; the upper pressure limit is 150 bar, and the vapour mass fraction at the turbine outlet is not allowed below 90%.

Carbon dioxide could not be used within these process limit: the specified heat-rejection temperature of 40

°C is above the critical point temperature for carbon dioxide. The cycle shown is for a heat rejection temperature of 10 °C, but this is not going to be a practical renewable energy cycle for places where dry cooling towers must be used used. The condenser pressure has also had to increase up to 45 bar for this case. Zhang, Yamaguchi, et al [15, 16] proposed a CO2 Rankine cycle in which heat rejection was made at 10 °C via water cooling, but the very low heat-rejection temperature makes the cycle unsuitable for large-scale applications.

The cycle efficiencies for the above analysis were found to be as shown in Table 1.

Working fluid	**Cycle efficiency**
water	36.00%
toluene	26.92% (with lowered T_{max} = 375 °C)
ammonia	24.63%
carbon dioxide	16.26% (with lowered T_{min} = 10 °C)

Table 1. Cycle efficiencies for the simple Rankine cycle models

For a simple Rankine cycle, we can see that water is the obvious choice for working fluid, easily outperforming the others in terms of cycle efficiency.

However, the most interesting differences between these cycles is their shape. The toluene cycle has an almost linear temperature-entropy line in the boiler, whereas water has the familiar 'stepped' curve resulting from its passing through sub-critical pressures. The near-linear shape of this curve makes it an interesting prospect because in a sensible heat exchanger, this fluid will allow heat transfer with much reduced exergy loss. This is especially important when coupling a cycle with a topping cycle or with thermal storage.

5.2 Regenerative Rankine cycle model

At this point we reject carbon dioxide as a fluid for this investigation. Firstly, it requires a high-pressure condenser because its triple point pressure is 5.1 bar, but secondly, and more importantly, its critical point temperature is below the cycle lower temperature limit we have set, which means that we will no be able to use a conventional condenser to return the fluid to pumpable liquid state.

For the water Rankine cycle, we can not make use of regeneration without bleeding steam from the turbine, hence reducing the turbine output. But for the other fluids, regeneration can be done without reducing the turbine power, because the fluids are still superheated at the turbine outlet. We would therefore anticipate a significantly greater improvement in cycle efficiency for the other cycles if regeneration is used.

In order to model this cycle, we require to add a heat exchanger to the cycle. For simplicity, the heat exchanger is not modelled in detail. Instead, temperature limits are adjusted until the cooling curves are safely spaced apart with a pinch point of not less than 12 °C. Sizing calculations have not been performed.

As we see in Figure 2, the straight-line heating and cooling curves in the toluene cycle will have resulted in significantly reduced exergy destruction in the heat exchanger, as compared to the open feedwater heater of the Rankine cycle. The result is a significantly better gain with regeneration for toluene than for either ammonia or water. Cycle efficiencies are summarised in Table 2.

This section has identified efficient regenerative Rankie cycles that will work within the proposed cycle limits. Water is still the best option on the basis of efficiency, though the difference between the options has reduced. We now turn attention to the design of a suitable topping Brayton cycle.

Working fluid	**Cycle efficiency**
water	38.68%
toluene	32.42%
ammonia	35.25%

Table 2. Cycle efficiencies for the regenerative Rankine cycle models

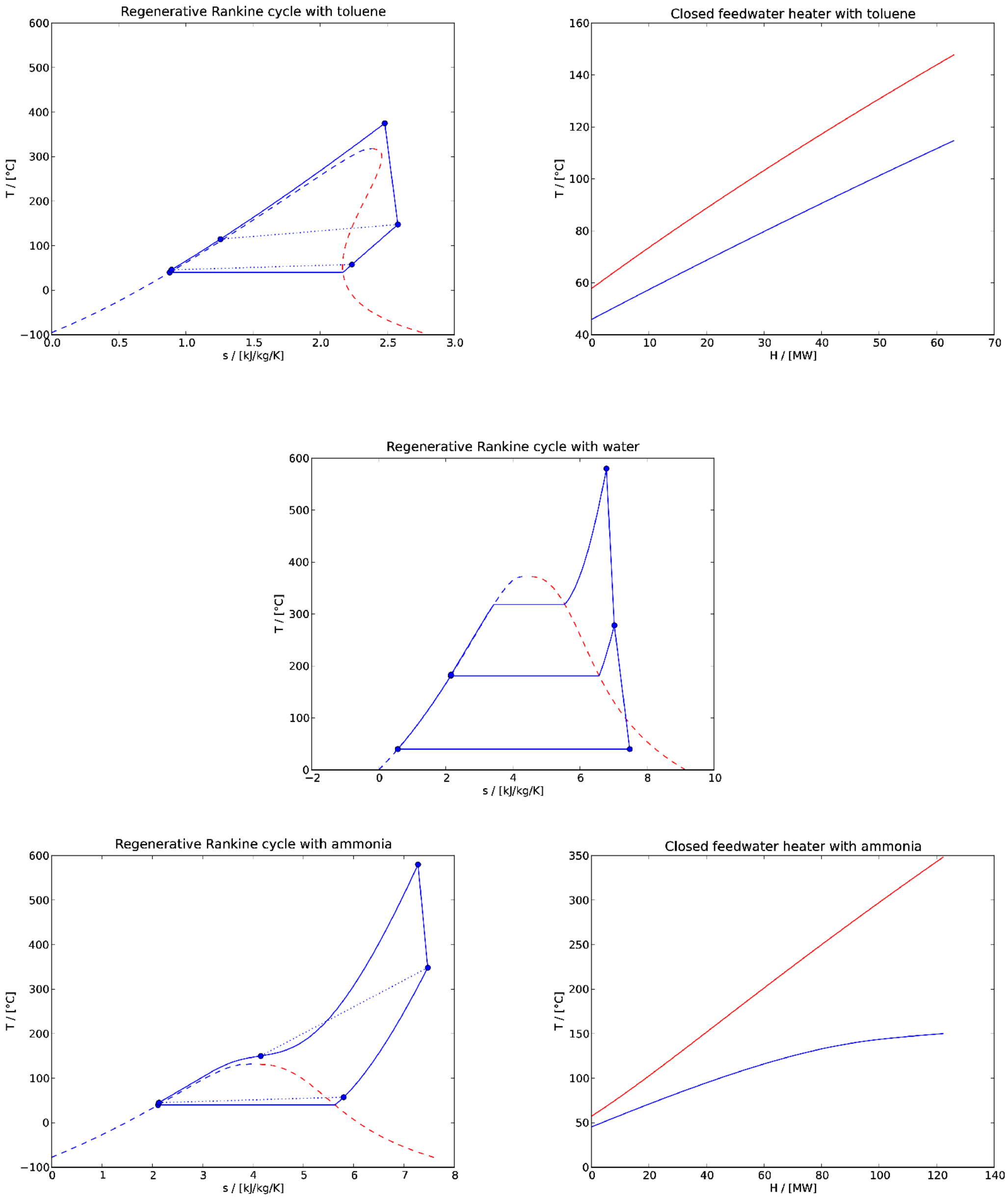

Figure 2. Regenerative Rankine cycle models with the same cycle limits as earlier. The feedwater heater gives a greater improvement for the non-water cycles. Heat exchanger profiles are shown for the toluene and ammonia cycles. In the water cycle, an open feedwater heater is assumed.

5.3 Brayton cycle

Workable limits on the temperatures of gas turbine blades have been increasing steadily since the 1940s [17]. Without expert advice, it seems likely that an upper bound on cycle temperature of 970 °C is necessary to avoid extremely expensive manufacturing methods for the proposed dish/Brayton system. This corresponds to the upper temperatures achieved in the 1970s with cast Nickel-alloy blades. No upper limit on pressure has been imposed.

The other important limit on Brayton cycle operation, for open Brayton cycles, is the inlet temperature. We will assume that the ambient temperature is 30 °C (the earlier assumption of 40 °C for the Rankine cycle allows for a small temperature difference across the condenser). We assume fixed isentropic efficiencies of 88% for the compressor and 95% for the turbine. Ideal air is assumed, but with variable $c_p(T)$ defined by a

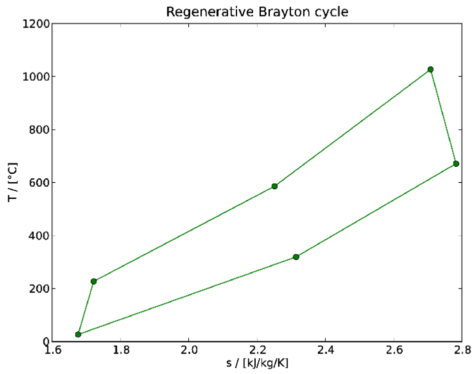

Figure 3. Regenerative air Brayton cycle with regenerator effectiveness of 80%. The cycle efficiency is 40.88 %. For plotting, curved lines have been approxiated as straight.

quartic polynomial [18]. There is assumed to be zero pressure loss across the combustor. As others have demonstrated [1], a simple Brayton cycle model will lead to a one-variable optimisation for optimal cycle pressure ratio.

As a baseline for the Brayton cycle, the above process limits, together with an 80% effective regenerator, yield an optimal pressure ratio of 4.5, and a resulting cycle efficiency of 40.88%. The cycle diagram is shown in Figure 3. This optimised cycle shows heat rejection at 320 °C.

It is interesting to note that h significantly lower than our upper limit cycle temperature for earlier Rankine cycle work. In fact, as the effectiveness of the regenerator increases, the optimal pressure ratio goes is reduced. With a 95% regenerator, the cycle efficiency is 50.9% and the pressure ratio is 2.4, and heat is rejected at ~200°C, but more effective regenerators are larger and larger, and more expensive.

5.4 Combined-cycle system

The addition of a gas turbine cycle as topping cycle for the above Rankine cycles is now investigated. At the outset, we would expect to see better combined-cycle performance with the toluene bottoming cycle owing to the straight-line heating curve for toluene at the operational conditions. For the combined-cycle system with water, an open feedwater heater was assumed. For the other systems, a closed heat exchanger is used.

T-s plots for combined-cycle systems with toluene, water, and ammonia are shown in Figure 4, with cycle efficiencies and the proportion of power output coming from the gas turbine shown in Table 3.

Table 3. Cycle efficiencies for the combined-cycle cycle models

Rankine fluid	Cycle efficiency	Gas turbine power fraction
water	45.61%	61.93%
toluene	46.17%	65.37%
ammonia	45.69%	66.40%

For the the water and ammonia cycle the gas turbine compression ration was optimised in the usual way. However, for the toluene cycle, the gas turbine compression ratio was a value defined by the fact the the gas turbine outlet temperature has to be compatible with the upper temperature limit for the toluene fluid.

It is quite surprising to see the way that the efficiencies of these different options has converged to similar values! While the exergy destruction in the water and ammonia systems is the main concern, the toluene system has a similar exergy destruction occurring in the condenser, due to the superheated turbine outlet state.

These systems all show a significant improvement in the cycle efficiency compared with the simpler regenerative Rankine cycles. Water has make only a modest improvement, due to the unavoidable exergy destruction in the heat exchanger. However, both ammonia and toluene bottoming cycles are working well in these cases. We should certainly expect to see increased efficiencies in all cases: with the addition of the gas

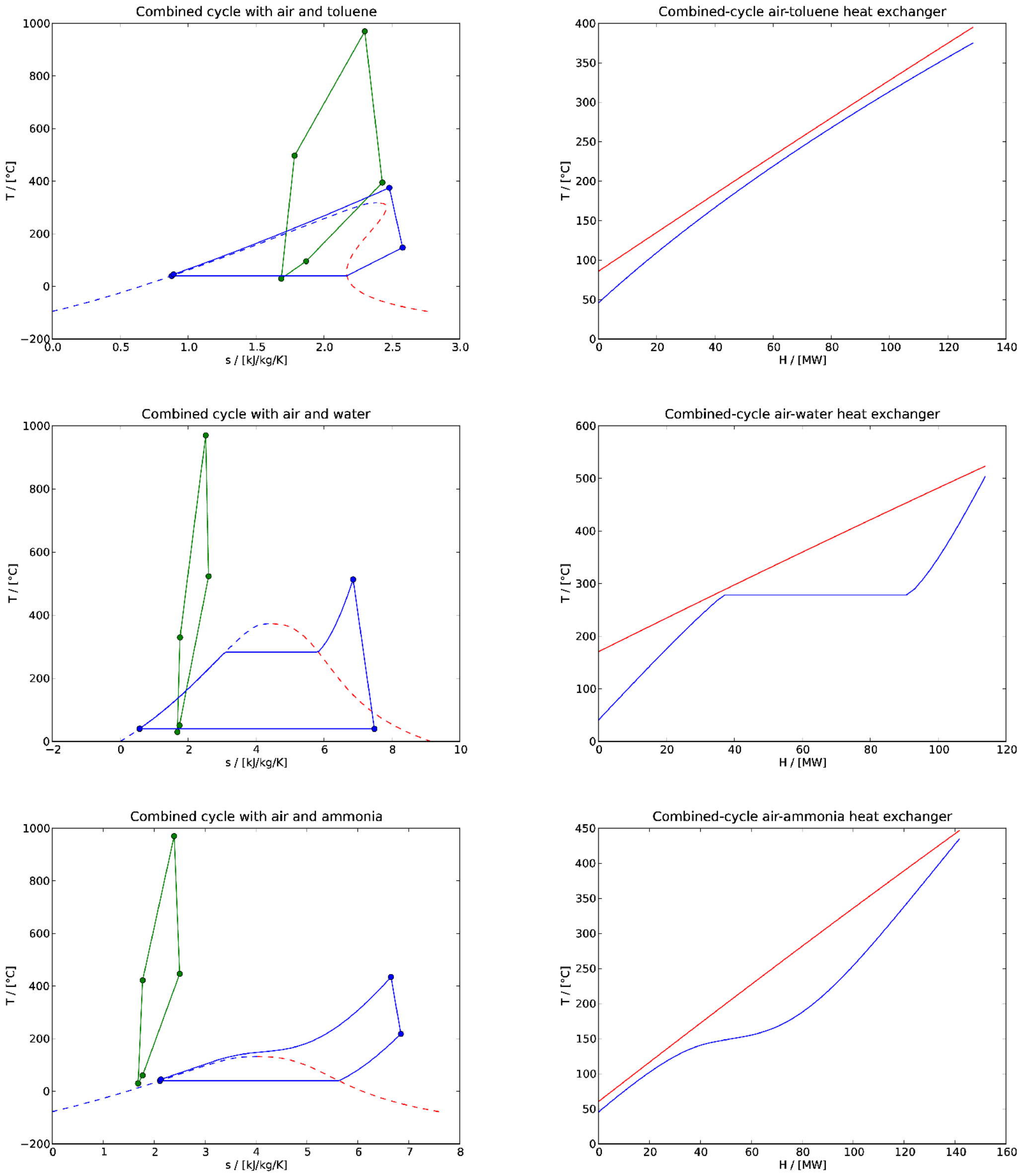

Figure 4. Combined-cycle configurations with toluene and water.

turbine, the maximum cycle temperature has increased from 580 °C to 970 °C, giving a much higher-exergy heat input.

5.5 Solar receiver

With a significant improvement in overall cycle efficiency shown for the combined cycle system, the problem now becomes one of designing a solar receiver able to capture heat at 970 °C or higher. This work has not been performed as part of the present study. A great deal of work has been done by Buck and his colleagues [19] on designs for volumetric air receivers using quartz windows. Tubular receivers are the alternative option, but the low convection coefficient for air means that large surface areas are required, which with the necessary pressures means large heavy structures. Work is continuing in this area.

Hotter receivers will have greater radiative losses and convective losses, but with suitable optimisation it is hoped that these greater thermal losses can be offset by higher power cycle efficiencies.

5.6 Further optimisation

It seems likely that the simplistic assumptions of constant isentropic efficiency for turbines, pumps and compressors used in this study would need to be investigated in greater detail. Factors such as the density of the different fluids are likely to have an effect on the peak achievable efficiency for turbines when different fluids are used, and this might be a factor that would cause greater spread between the options. A much more detailed understanding of turbine design and selection is required.

Optimising on cycle efficiency should always be secondary to optimising on cost. The different cycles presented here have very different operating parameters; this will affect the cost of the pumps and turbines, heat exchangers, pipework and condensers. This requires an extensive database of component costs, something which is unfortunately very difficult to access from academia. An optimisation based on levelised energy cost is a much better basis for design decisions, but was not undertaken as part of this work.

When considering combined-cycle gas turbines for dish concentrator application, the intent would be to mount a compressor and gas turbine at the focus of each dish, then a heat exchanger to transfer heat to the bottoming cycle working fluid, which would pass back to a single, shared bottoming cycle power block. An accurate model requires modelling of the pipework thermal losses, which would be significant over a large dish field.

This study has not yet considered the addition of a thermal storage system. It is proposed that the above combined-cycle system with toluene could be used with a molten-salt storage system. During sunlight hours, the gas turbine would be operated with some fraction (about 70%) of its output being committed to storage, and the remainder driving the tolene cycle directly. Then, during non-sunlight hours, the toluene cycle would be run from storage. The toluene cycle would therefor run continuously, and the (cheaper) gas turbine cycle would provide peaking power output. The toluene cycle will aleviate the pinch-point issues associated with steam (as discussed previously by Pye and Coventry [20]) although the thermal limits of molten salt will not be acheived here due to the thermal limits of toluene, with the result that a larger storage unit would be required.

6 Conclusion

A range of Brayton and Rankine cycles were investigated subject to a set of defined process limits. The fluids carbon dioxide, toluene, ammonia, and water were considered as alternatives.

The benefit of combined-cycle solarised gas turbines was outlined, by comparison of alternative cycles. Within assumed process limits, the best option was an open air gas turbine with a toluene bottoming Rankine cycle. Cycle efficiencies of about 46% were achieved. The main advantage of the toluene cycle is the flat heat addition and rejection curves, allowing heat exchange with minimal exergy destruction. A concept for thermal storage was introduced.

Further work is required to compare these alternatives on the basis of cost, to investigate thermal storage configuration, and to investigate other possible cycle improvements.

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