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Thermodynamic and design considerations of organic Rankine cycles in combined application with a solar thermal gas turbine

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Abstract. Concentrated Solar Power (CSP) technologies are considered to provide a significant contribution for the electric power production in the future. Different kinds of technologies are presently in operation or under development, e.g. parabolic troughs, central receivers, solar dish systems and Fresnel reflectors. This paper takes the focus on central receiver technologies, where the solar radiation is concentrated by a field of heliostats in a receiver on the top of a tall tower. To get this CSP technology ready for the future, the system costs have to reduce significantly. The main cost driver in such kind of CSP technologies are the huge amount of heliostats. To reduce the amount of heliostats, and so the investment costs, the efficiency of the energy conversion cycle becomes an important issue. An increase in the cycle efficiency results in a decrease of the solar heliostat field and thus, in a significant cost reduction. The paper presents the results of a thermodynamic model of an Organic Rankine Cycle (ORC) for combined cycle application together with a solar thermal gas turbine. The gas turbine cycle is modeled with an additional intercooler and recuperator and is based on a typical industrial gas turbine in the 2 MW class. The gas turbine has a two stage radial compressor and a three stage axial turbine. The compressed air is preheated within a solar receiver to 950°C before entering the combustor. A hybrid operation of the gas turbine is considered. In order to achieve a further increase of the overall efficiency, the combined operation of the gas turbine and an Organic Rankine Cycle is considered. Therefore an ORC has been set up, which is thermally connected to the gas turbine cycle at two positions. The ORC can be coupled to the solar-thermal gas turbine cycle at the intercooler and after the recuperator. Thus, waste heat from different cycle positions can be transferred to the ORC for additional production of electricity. Within this investigation different working fluids and ORC conditions have been analyzed in order to evaluate the best configuration. The investigations have been performed by application of improved thermodynamic and process analysis tools, which consider the real gas behavior of the analyzed fluids. The results show that by combined operation of the solar thermal gas turbine and the ORC, the combined cycle efficiency is approximately 4%-points higher than in the solar-thermal gas turbine cycle.

1. Introduction
Solar thermal power plants based on Concentrating Solar Power (CSP) technologies will become an important part within the future energy mixture in Europe and the MENA (Middle East and North Africa) region, as the DESERTEC [1] initiative shows. An important CSP technology is the central receiver technology. CSP plants based on central receiver usually consist of an area filled with heliostats (mirrors), which track the sun position and concentrate the solar radiation at a solar receiver located at the top of a tall tower (power tower system). The solar heat is then transferred from the receiver directly or indirectly (by a heat transfer fluid) to a thermal cycle, which is usually a water/steam Rankine cycle. An advantage of central receivers is the minimization of thermal losses which for example are caused by long ducts in parabolic trough applications. Power tower systems using a water/steam Rankine cycles as energy conversion cycle are on a commercial level since a few years, e.g. the GEMASOLAR power plant in Spain, which has a peak thermal efficiency of 40% [2]. nevertheless the implementation of a Rankine cycle as energy conversion cycle require a certain amount of fresh water and for independent power production large heat storage systems are necessary. This makes the solar power plant application much more depend from its region and of cause more expensive. An alternative is the implementation of gas turbine technologies in CSP power plants. The advantages of gas turbines are the higher flexibility regarding the place of location and in hybrid operation an additional heat storage system is not necessary. Besides these issues, gas turbine technologies, or Brayton cycles in general, offer the possibility of higher process temperatures and thus, of higher thermal cycle efficiencies. As nearly 50% of the investment costs for a central receiver power plant are caused by the expensive field of heliostats, an increase of the thermal efficiency of the energy conversion cycle is necessary to reduce the amount of heliostats. Muto et al. [3] have shown in their work on the application of a supercritical carbon dioxide gas turbine that their cycle can produce power at high thermal cycle efficiency of 43.4% even at comparable low temperature of 650°C. Turchi et al. [4] have shown that thermal efficiencies of even 51% and higher can be reached by closed Brayton cycles using direct supercritical CO$_2$ receiver, which generate gas outlet temperatures of 700°C. Disadvantages of such kind of CO$_2$ cycles are high operating pressures up to 25 MPa [4]. Furthermore the effect of dissociation of the gas, as well as the reaction of CO$_2$ with component materials at temperatures higher than 650°C can be seen as a challenge. As an alternative Kusterer et al [5, 6] have shown in their investigation that by using Helium or Argon as working fluid within a closed Brayton cycle, thermal efficiencies of higher than 46% can be achieved. Closed Brayton cycles need additional heat storage systems to achieve a stable and constant power production even when the sun is not shining. To get independent from weather conditions, the application of a solar hybrid gas turbine (open Brayton cycle) is a good alternative. To increase the overall cycle efficiency, a combined application with a Rankine cycle can be realized. Such combined cycle systems have been investigated intensively by Heide et al. [7, 8]. They have shown that combined cycle application including an ORC offer an additional potential for the improvement of the cycle efficiency. To achieve a high improvement in cycle efficiency, a maximum waste heat utilization from the gas turbine cycle is necessary. This paper will show the results of the application of an ORC to a solar thermal gas turbine within the 2MW class. Furthermore a first design concept of the ORC turbine will be presented.

2. Thermodynamic process calculation of the solar energy conversion cycles

2.1 SICR-GT-model
The basis of the solar energy conversion cycle is given by the model of a 2 MW class solar intercooled and recuperated gas turbine (SICR-GT). The process of the SICR-GT has been modelled by consideration of typical component parameters of 2 MW class gas turbines. The SICR-GT cycle consist of a two stage radial compressor with a compressor interstage cooler, a recuperator, a heat exchanger which indicates the solar receiver, a combustor and a three-stage axial turbine. Within the gas turbine cycle it is assumed that the compressed air is heated up within the solar receiver to an air outlet temperature of 950°C. This fits to actual conditions of volumetric air receivers, which are intended for application in a solar hybrid gas turbine. Within the combustor the air is then heated up to
a turbine inlet temperature of 1100°C, so hybrid operation of the SICR-GT is assumed. The secondary air system (cooling, leakage, etc.) has also been considered within the thermodynamic process calculation of the gas turbine. The electrical efficiency of the SICR-GT, for an ambient temperature of 35°C, is 38.38% and the electrical power output is 2.26 MW [9].

2.2 SICR-GT-ORC-model

To achieve a further improvement of the overall energy conversion cycle efficiency, a combined cycle configuration has been analysed. Therefore an Organic Rankine Cycle (ORC) has been configured which utilized the waste heat from the SICR-GT process. The modelled ORC is connected with the SICR-GT cycle at two positions, at the compressor intercooler of the gas turbine and after the recuperator exhaust gas outlet. Thereby it is assumed that 95% of the discharged heat at the intercooler is transferred into the organic Rankine cycle. Thus, the organic medium acts as cooling fluid within the gas turbine intercooler (see Figure 1). At the second position (see Figure 1, position 9 of the GT cycle) the ORC is connected to the exhaust gas of the gas turbine cycle. It is assumed that the discharged heat from the exhaust gas is limited by an operation limit of the gas turbine. To avoid the production of hazardous substances within the downstream exhaust thread, a minimum exhaust gas temperature of 120°C has been considered. 95% of the discharged heat from the exhaust gas is assumed to be transferred into the ORC cycle. The configuration of the ORC is shown in Figure 2, where $Q_{\text{I-ORC}}$ indicates the transferred waste heat from the intercooler and $Q_{\text{EXH-ORC}}$ the transferred heat from the exhaust gas. Within this figure two heat exchangers are shown between the pump and the turbine (O1-O3), but for real application it is assumed that these heat exchangers can be designed as a two-stage heat exchanger component.

The thermodynamic process model of the ORC considers a pump with an isentropic efficiency of 75% and a turbine with an isentropic efficiency of 85%. The condensate temperature is assumed with 45°C and the condenser (O4 – O5) is modelled with a pressure loss of 3%. The pressure loss within the two-stage heat exchanger is 3% for each stage.

2.2.1 ORC parametric study and fluid sensitivity analysis

In order to obtain the best parameters to achieve the best waste heat utilization, a parametric study and fluid sensitivity analysis has been performed for the ORC. In table 1 an overview about the boundary conditions for the parametric study and sensitivity analysis are shown.
Table 1. Boundary conditions for ORC parametric study

|                | Q_{LORC} | Q_{EXH-ORC} | pinch point heat exch. | max. cycle pressure | condensate temperature |
|----------------|----------|-------------|------------------------|---------------------|-----------------------|
| ORC            | 1008 kW  | max. 1382 kW| 7 °C                   | 40 bar              | 45 °C                 |

As fluids n-butane, iso-butane, pentane and R245fa have been analysed. For each fluid the best operation parameters as pressure level, pressure ratio and mass flow rate have been determined in order to obtain a maximum power output and cycle efficiency. The mechanical losses of the ORC system have been considered with a power loss of 2% and the generator and gear losses in summary with 3%. The main results of the ORC parametric study and fluid analysis are shown in table 2.

Table 2. Results of ORC parameter study and fluid analysis

|                              | mass flow [kg/s] | min. pressure [bar] | pump pressure ratio | el. eff. ORC | el. eff. combined | combined power output [kW] |
|------------------------------|------------------|---------------------|--------------------|--------------|------------------|---------------------------|
| n-butane                     | 5                | 4.5                 | 7                  | 12.20%       | 43.22%           | 2549                      |
| iso-butane                   | 5.3              | 6.5                 | 5.5                | 10.88%       | 42.76%           | 2521                      |
| pentane                      | 4.5              | 1.5                 | 9                  | 12.54%       | 43.17%           | 2545                      |
| r245fa                       | 10               | 3                   | 8                  | 11.81%       | 43.13%           | 2543                      |

The thermodynamic analysis has shown that with n-butane as working fluid, the best combined cycle efficiency can be achieved. Indeed the thermal efficiency of the ORC operated with pentane is higher than for the n-butane cycle, but within the pentane cycle not the total available transferable heat from the exhaust gas waste heat can be utilized due to pinch point boundaries. Thus, more heat can be transferred into the n-butane cycle and so the combined efficiency is higher.

3. Design consideration of the ORC turbine

3.1 1D ORC turbine design

The thermodynamic process calculations have shown that with n-butane, as working fluid for the organic Rankine cycle, combined cycle efficiencies of more than 43% can be expected. Within this chapter a first 1D design approach of the ORC turbine will be presented. The procedure of the design of organic turbines is similarly to the design of steam turbines. Nevertheless, because of the strong real gas behaviour of most organic fluids (so for n-butane) the design principles and methods of steam turbine design can only be used as a first approach. Such first approach has been performed for the design of the n-butane operated turbine.

As first step the rotational speed and the numbers of stages have been determined and as result a 4 stage axial turbine with a rotational speed of 24900 rpm has been evaluated. For the detailed stage design a constant flow coefficient $\varphi$ of 0.34 and an enthalpy reaction rate $\rho_h$ of 0.5 have been prescribed, which means that the stages are reaction stages. Equation 1 gives the definition of the flow coefficient and equation 2 of the enthalpy reaction rate:

\[
\varphi = \frac{c_m}{u} \quad (1)
\]

\[
\rho_h = \frac{\Delta h'''}{\Delta h} \quad (2)
\]

where $\Delta h'''$ is the enthalpy conversion within the rotor and $\Delta h$ the enthalpy conversion of the entire stage (stator and rotor). Due to the moderate load of reaction stages, higher efficiencies can be achieved in comparison to impulse stages, where the enthalpy conversion within the stator is much...
higher than within the rotor, which occurs to higher flow velocities and as a consequence to higher flow losses. The incidence flow of each stage is designed with zero degree versus machine axis, thus, the stages are designed as repeating stages. This is of advantage, as only one profile has to be designed, which can be used for all stages. Table 3 shows the boundary conditions, taken from the thermodynamic consideration, and table 4 the fragmentation of the reversed specific work and the enthalpy differences for each stage. It can be seen, that within the stages the enthalpy differences decline and so the load of each stage. Based on these conditions, the design of each stage has been performed and the characteristic parameters of the turbine cascade have been evaluated. The meridian diameter $d_m$ of the turbine has been held constant within the turbine design. Thus, the lower and upper shroud contours are variable and addicted to the expansion behaviour of the organic medium for the given enthalpy difference. The diameters of the upper and lower shrouds of each stage inlet are also presented in Table 4.

**Table 3.** Boundary conditions for ORC turbine design

| $\dot{m}$ | $\pi$ | $\Delta h_{\text{turbine}}$ | $\eta_{\text{is}}$ | $\varphi$ | $\rho_h$ | $d_m$ | $n$ |
|-----------|-------|-----------------|-----------------|--------|--------|------|----|
| 5 kg/s    | 6.2   | -66.8 kJ/kg     | 85%             | 0.34   | 0.5    | 0.099m | 24900 min$^{-1}$ |

**Table 4.** ORC turbine stage load fragmentation

| $a$       | $\Delta h_{\text{stage}}$ | $\pi_{\text{stage}}$ | $d_{\text{upper shroud}}$ [mm] | $d_{\text{lower shroud}}$ [mm] |
|-----------|-----------------|----------------|------------------|------------------|
| [kJ/kg]   | [kJ/kg]         |                 |                  |                  |
| -17.026   | -20.117         | 1.66            | 103.7            | 94.7             |
| -16.946   | -20.003         | 1.58            | 107.83           | 90.57            |
| -16.896   | -19.936         | 1.55            | 113.7            | 84.7             |
| -16.057   | -19.524         | 1.53            | 122.23           | 76.18            |

As already mentioned, the stages are designed as repeating stages, so the flow angles and velocities are similar in each stage which leads to similar airfoil profiles of the vanes and blades. In Figure 3 the velocity triangle for each stage of the ORC turbine is shown as well as the flow angles. The inflow and outflow ($\alpha_0$ and $\alpha_2$) of each stage, as well as the inflow of the rotor in relative frame of reference ($w_1$) are zero degree versus machine axis (or 90° to the normal of the machine axis) and the deflection of each vane and blade is 71.3°. Figure 4 gives the definition of the velocity triangle within a turbine cascade.

**Figure 3.** Velocity triangle of ORC turbine stages

**Figure 4.** Definition of velocity triangle in turbine cascade [10]

3.2 Design of 2D and 3D turbine vanes and blades profiles

Based on the flow angles given in Figure 3, the airfoil of the turbine can be designed and is shown in Figure 5. Because of the small size of the turbine, a specified 3D profile in dependency of the radial position is not necessary. Thus, the 3D profiles of the vanes and blades are extruded profiles based on
the 2D profile without any torsion and limited by the upper and lower shroud contour. In Figure 5 it can be seen that the profiles for the vanes and blades are identical. The profiles have been designed for the constant meridian section. In this figure the 2D profiles of the 1st stage are shown, the profiles for the other stages are equal. As mentioned, the 3D profiles are extruded from the 2D meridian section profiles and bordered by the contour of the upper and lower shroud. The contour of the turbine can be seen in Figure 6.

![Figure 5](image)

**Figure 5.** 1st stage profiles and velocity triangles

![Figure 6](image)

**Figure 6.** 4-stage ORC turbine contour

4. Summary and conclusion
A thermodynamic and design consideration of an organic Rankine cycle application for improved cycle efficiencies for a combined solar thermal power plant has been presented. Based on a concept of a solar thermal intercooled and recuperated gas turbine, the application of an organic Rankine cycle for waste heat utilization of the gas turbine cycle has been analysed. It has been shown, that cycle efficiencies of more than 43.2% can be achieved for the combined cycle operation, even at higher ambient temperatures of 35°C, as typical for regions of high solar radiation. Within a first axial turbine design of the ORC turbine an overview about the turbine dimensions, the numbers of stages, the flow conditions and the profiles have been presented.

References
[1] “Executive Summary-The Case for desert Power”, 2050 Desert power report, Dii GmbH, Report 2012
[2] Garcia E and Calvo R 2012 One year operation experience of GEMASOLAR plant Solar Paces 2012(Morocco, 11-14 September 2012)
[3] Muto Y, Ishiyama S, Kato Y, Ishizuka T and Aritomi M 2010 J. Energy & Power Eng. 4 7-15.
[4] Turchi C S, Ma Z, Dyreby J 2012 Supercritical carbon dioxide power cycle configurations for use in concentrating solar power systems Proc. of the ASME Turbo EXPO 2012(Copenhagen, Denmark, 11-15 June 2012) GT2012-68932
[5] Kusterer K, Braun R, Moritz N, Lin G and Bohn D 2012 Helium Brayton cycles with solar central receivers: Thermodynamic and design considerations Proc. of the ASME Turbo EXPO 2012(Copenhagen, Denmark, 11-15 June 2012) GT2012-68407
[6] Kusterer K, Braun R, Moritz N, Bohn D, Sugimoto T and Tanimura K 2013 Comparative study of solar thermal Brayton cycles operated with helium or argon Proc. of the ASME Turbo EXPO 2013(San Antonio, Texas, USA, 3-7 June 2013) GT2013-94713,
[7] Heide S, Gampe U, Orth U, Beukenberg M, Gericke B, Freimark M, Langnickel U, Pitz-Paal R, Buck R and Giuliano S 2010 Design and operational aspects of gas and steam turbines for the novel solar hybrid combined cycle SHCC Proc. of the ASME Turbo EXPO 2010(Glasgow, UK 14-18 June 2010) GT2010-22124
[8] Heide S, Felsmann C, Gampe U, Freimark M, Langnickel U, Buck R and Giuliano S 2012 Parameterization of high solar share gas turbine systems Proc. of the ASME Turbo EXPO 2012(Copenhagen, Denmark, 11-15 June 2012) GT2012-68608
[9] Kusterer K, Braun R, Koellen L, Bohn D, Sugimoto T and Tanimura K 2013 Combined solar thermal gas turbine and organic Rankine cycle application for improved cycle efficiencies
Proc. of the ASME Turbo EXPO 2013 (San Antonio, Texas, USA, 3-7 June 2013) GT2013-94713

[10] Bohn D and Gallus H D 2007 Energiewandlungsmachinen (University lecture, Aachen)