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Roads to improve the performance of hybrid thermosolar gas turbine power plants: Working fluids and multi-stage configurations

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ABSTRACT

This paper presents a general thermodynamic model for hybrid Brayton central tower thermosolar plants. These plants have been proved to be technically feasible but research and development efforts need to be done in order to improve its commercial interest. From the thermodynamic viewpoint it is necessary to increase its performance to get larger power production with reduced fuel consumption, and so reduced emissions. A model for multi-step compression and expansion is developed with that aim. The model is flexible and allows to simulate recuperative or non-recuperative plants, with an arbitrary number of stages and working with different subcritical fluids. The results for multi-step configurations are compared with those obtained for a plant with one turbine and one compressor. Different working fluids are analyzed, including air, nitrogen, carbon dioxide, and helium. Several plant layouts and the corresponding optimal pressure ratios are analyzed. Configurations with two-stages compression with intercooling combined with one or two expansion stages can significantly improve overall plant efficiency and lower fuel consumption. Power block efficiencies can reach 0.50 and overall plant efficiency can attain values about 0.40 working with air or carbon dioxide. For instance, comparing with a single-stage plant running with air, a plant working with subcritical carbon dioxide and two compression stages with intercooling can reach an overall efficiency about 19% larger and a fuel conversion rate around 23% larger. For such configuration, the specific fuel consumption is predicted to be about 108 kg/(MW h) at design point conditions.

1. Introduction

Concentrating solar power (CSP) is one of the promising renewable energy technologies that can contribute to decrease the dependence on fossil fuels for the generation of electricity and so, the environmental impact of energy production. As mentioned by Nathan et al. [1], unlike other renewable resources this technology is suited to produce non-intermittent power with the implementation of thermal storage. Peterseim et al. [2] discuss which CSP technologies are best suited for hybridization. Powell et al. [3] have recently published an extensive work on hybridization possibilities, including geothermal and photovoltaic resources. In this work CSP plants in which solar heat input is complemented with the heat released by the combustion of natural gas in a combustion chamber are surveyed. This technology ensures an almost constant energy injection to the grid in the range of a few megawatts. These plants are not completely free of fossil fuel consumption and pollutant emissions but guarantee predictability. Olumayegun et al. [4] highlight that the plants which work following a closed Brayton-like thermal cycle require a reduced water consumption compared with those working on Rankine cycles and can reach similar efficiencies. This point is especially advantageous in arid regions with appropriate solar resources. To get those efficiencies quite high turbine inlet temperatures have to be reached in the solar receivers, about 800–1000 °C. Several experimental prototypes [5] have shown that this is feasible using ceramic materials in central tower volumetric receivers. Ho and Iverson [6] have summarized these advances. Pioneer demonstration size plants have arrived at the same conclusion: the technology is practicable but it is still necessary a R&D activity to look for ways to improve the overall plant efficiency in order to get commercially interesting levelized costs of electricity, as pointed out by Korzynietz et al. [7]. Particularly, as mentioned by Dunham and Iverson [8], thermo-economic studies show that there is still a wide margin for improvement in the power block.

Along this work line thermodynamic studies about possible refinements on the basic Brayton cycle and the effects of the working fluid are important to guide future plant designs, as stated by Osorio et al. [9]. McMahan et al. [10] modelled the plant in terms of a reduced number of parameters. Within a similar framework, Zare and Hasanzadeh [11]

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Nomenclature

A_a	aperture area of the solar field (m^2)
A_r	solar receiver area (m^2)
a_c	isentropic compressor pressure ratio
a_t	isentropic turbine pressure ratio
C	solar collector concentration ratio
c_w	specific heat of the working fluid [J/(mol K)]
f	solar share
G	direct solar irradiance (W/m^2)
h_1	radiation heat loss coefficient for the solar collector (K^{-4})
h_2	effective convection and conduction loss coefficient for the solar collector (K^{-1})
$\dot{m}$	mass flow rate of the working substance (kg/s)
$\dot{m}_f$	fuel mass flow rate in the main combustion chamber (kg/s)
$\dot{m}_{fi}$	fuel mass flow rate in reheaters (kg/s)
P	power output (W)
$\lvert\dot{Q}_C\rvert$	heat losses at the combustion chamber (W)
$\lvert\dot{Q}_H\rvert$	total heat-transfer rate absorbed from the working fluid (W)
$\lvert\dot{Q}_{iHC}\rvert$	heat losses at the heat exchanger associated to the combustion chamber (W)
$\lvert\dot{Q}_{HC}\rvert$	heat rate input from the combustion chamber (W)
$\lvert\dot{Q}'_{HC}\rvert$	heat rate transferred from the combustion chamber to the associated heat exchanger (W)
$\lvert\dot{Q}_{HS}\rvert$	heat rate input from the solar collector (W)
$\lvert\dot{Q}_{iHS}\rvert$	heat losses at the solar receiver (W)
$\lvert\dot{Q}'_{HS}\rvert$	heat rate transferred from the solar collector to the associated heat exchanger (W)
$\lvert\dot{Q}_l\rvert$	losses associated to heat transfers in the solar field (W)
$\lvert\dot{Q}_L\rvert$	heat-transfer rate between the working fluid and the ambient (W)
Q_{LHV}	lower heating value of the fuel (J/kg)
$\lvert\dot{Q}_{reh}\rvert$	heat rate input from the reheaters (W)
r_e	fuel conversion rate
r_p	overall pressure ratio
T_{HC}	working temperature of the combustion chamber (K)
T_{HS}	working temperature of the solar collector (K)
T_L	ambient temperature (K)
T_x	working fluid temperature after the heat input from the recuperator (K)
$T_{x'}$	working fluid temperature after heat input from the solar collector (K)
T_y	working fluid exhaust temperature (K)
T_1	compressors inlet temperature (K)
T_2	temperature after last compressor (K)
T_3	turbines inlet temperature (K)
T_4	temperature after last turbine (K)
$\overline{U}_L$	effective conduction–convection heat transfer coefficient [W/(m^2 K)]
α	effective emissivity
ε_{HC}	combustion chamber heat exchanger effectiveness
ε_{HS}	solar collector heat exchanger effectiveness
ε_L	cold side heat exchanger effectiveness
ε_c	isentropic efficiency of the compressors
ε_r	recuperator effectiveness
ε_t	isentropic efficiency of the turbines
γ	adiabatic coefficient of the working fluid
η	overall energy efficiency
η_c	combustion efficiency
η_h	thermal efficiency of the Brayton heat engine
η_s	solar collector efficiency
η_0	optical efficiency
ρ_H	irreversibilities due to pressure drops in the heat input
ρ_L	irreversibilities due to pressure drops in the heat release
σ	Stefan–Boltzmann constant (W m^{-2} K^{-4})

predicted realistic values for efficiencies. Thus, sensitivity studies and optimization analyses can be done in more general terms than those done, for instance, with simulation software, as performed for instance by Barigozzi et al. [12,13]. Both techniques are complementary. Probably, general thermodynamic models are to be developed first in order to select adequate plant concepts and then detailed component-to-component simulations, are required to solve technical issues as done by Milani et al. [14] and to get to very detailed predictions of plant performance as shown in the work by Kalathakis et al. [15].

One of the main drawbacks of considering Brayton cycles in CSP applications is that for the compression stage much power is required, so the net power output becomes reduced. This point is detailed by Iverson et al. [16]. One possibility to avoid this handicap is to operate at supercritical conditions as suggested by Al-Sulaiman and Atif [17]. Extensive work has been devoted to this issue, specially considering carbon dioxide as working fluid, as done by Luu et al. [18]. Near the critical region fluids show numerical values for compressibility similar to liquids. Compression work can be reduced but as critical pressure for CO_2 is about 74 bar, high pressures have to be used. Vasquez et al. [19] point out that this leads to several technical problems. Moreover, wide fluctuations of thermodynamic properties near the critical point make difficult to develop thermodynamic models relying on ideal gas approximations. With respect to the turbomachinery much scarce experience has been acquired in components working with critical or transcritical fluids. A thorough review on this point is due to Ahn et al. [20]. An alternative way to reduce compression work is by joining these concepts: recuperation and multi-stage compression with intercooling. Recent works on these issues have been developed by Reyes-Belmonte et al. [21]. Additionally, if expansion is performed in several turbines with intermediate reheaters, temperature at the exit of the last turbine is high and so the potential for recuperation, as shown in the paper by Sánchez-Orgaz et al. [22].

Even though there is a great amount of works on the possibilities of using supercritical CO_2 in CSP systems, to our knowledge there are much scarce thermodynamical investigations on subcritical fluids as CO_2 together with multi-stage compression with intercooling and multi-stage expansion with reheating. Our work deals with this point. Plant configurations for central tower hybrid CSP plants working on closed atmospheric Brayton cycles for several working fluids shall be investigated, including subcritical CO_2, helium, nitrogen, and air. Plant performance will be compared by taking similar conditions for all fluids. Although the peculiarities of heat exchangers and turbomachinery of course rely on the type of fluid, components with similar effectivenesses or isentropic efficiencies will be assumed, *i.e.*, details on the design and performance of plant components are not analyzed, but it is assumed that with the appropriate design particularities components can have similar effectivenesses or isentropic efficiencies. To get that aim it is developed a thermodynamical model that incorporates the main irreversibilities existing in all the subsystems in these plants: solar, combustion chamber, and thermal engine. A simplified model was developed and validated in previous works by our group for the case of air and single-stage compression and expansion [23,24]. In this work it is extended for an arbitrary number of compression/expansion steps, recuperation, and for subcritical fluids by explicitly considering the temperature dependence of specific heats. Although the model allows for on-design and off-design analyses as shown in the study by Santos et al. [25], in this work design point parameters summarized by Quero et al. [26] from an experimental facility will be considered as reference case to compare with. The compression ratio is a key parameter in the design of any plant involving Brayton-like cycles. In our