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Alternative cycles based on carbon dioxide for central receiver solar power plants

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ABSTRACT

Research in concentrated thermal solar power plants of all types and, in particular, those based on central receiver, namely solar tower plants, has experienced great impetus in the last decade, reaching full commercial operation with the PS10 plant in Spain. In spite of previous demonstration plants testing different receivers and power cycle layouts, this first commercial power plant adopted a cavity receiver generating saturated steam and therefore penalising cycle efficiency in order to gain plant reliability. According to the experience gained, if a competitive Levelised Cost of Electricity is to be reached, capital and maintenance costs must be reduced and efficiencies must be increased. To achieve these goals, modifying the power cycle is deemed essential, whether using superheated steam or alternative fluids.

In this work, the use of supercritical and transcritical carbon dioxide cycles for this application is proposed. Three different cycles are considered, the first two of which are stand-alone closed cycle gas turbines using carbon dioxide. The third proposal is a combined cycle that comprises a topping carbon dioxide gas turbine and a bottoming Organic Rankine Cycle. Preliminary results show that these cycles are promising technologies for solar tower plants, having the potential to compete in terms of efficiency and costs with other conventional technologies.

1. Introduction

Concentrating solar thermal power (CSP) plants have been given great impetus in the last years, especially in countries like Spain where the installed CSP power capacity currently exceeds 430 MW and is expected to double by 2011, with plants in different construction stages [1]. The development of these power plants has been associated to the adaptation of proven steam power generation technologies combined with particular concentrating solar power components. Among others, the latter components include heliostats or solar receivers for central receiver solar power plants (CRS) [2–4] and linear collectors, oil pumps and oil to water/steam heat exchangers for parabolic trough power plants. For the first type of plants, the high solar flux hitting the receiver (averaging between 300 and 1000 kW/m^2) enables operating at rather high temperatures of up to 1000 °C [5], even if the maximum receiver temperature is limited to around 700 °C at the current stage of development [2]. Higher temperatures have nonetheless been obtained in experimental or demonstration plants like the Directly Irradiated Annular Pressurised (DIAP) facility [6] of the Weizmann Institute (Israel), with temperature exceeding 1300 °C when pressurised air at 10–30 bar and multistage receivers are used [7], with air temperature in the range of 800–1000 °C. In this sense, an analysis by Segal and Epstein [8] concluded that the optimum power plant performance would correspond to a receiver temperature close to 1600 K, what would allow using conventional gas turbine and combined cycle technologies.

Alternative power cycles, or cycles that make use of nonconventional fluids, are a different option to achieve higher efficiencies without reaching such high temperatures in the receiver. Among them, the supercritical and transcritical closed Brayton cycles working with carbon dioxide are deemed interesting. This cycle has been studied for the last 40 years, since firstly proposed by Feher and Angelino [8,9], for nuclear power production in gas reactors, though its applicability to solar power plants has also been explored [11–13]. Thermodynamically, the main advantage of the Brayton carbon dioxide cycle relies on its high useful to expansion work ratio (i.e. much lower compression work than expansion work) which is in the range 0.7–0.85 when compressor inlet is in supercritical conditions. At cycle level, different layouts were studied by Carstens et al. [14] and Dostal et al. [15] in order to increase cycle efficiency. From the point of view of major equipment, the necessary features of turbomachinery were analyzed by Vilim et al. [16] and Gong et al. [17] and heat transfer and heat exchanger layouts were analyzed by Utamura [18].

In the same category of alternative cycles, Organic Rankine Cycles (ORC) yield higher efficiencies than conventional steam cycles when heat delivery is at temperatures below 370 °C [19] and when a low power output does not allow exploiting the highest efficiency of more complex steam turbine designs (reheat or feedwater heating

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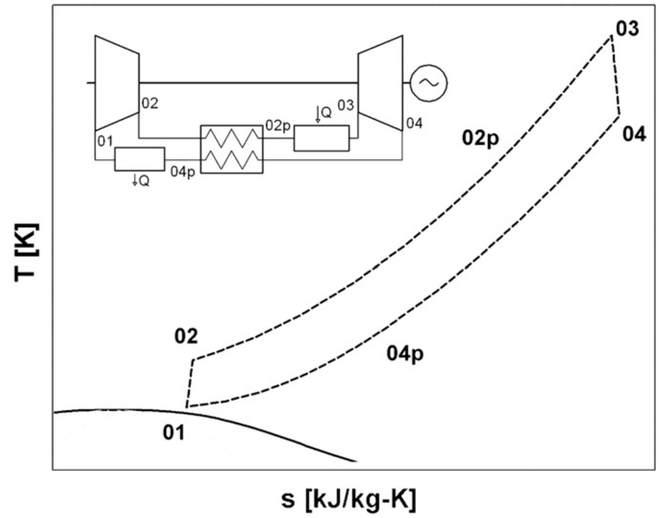

Fig. 1. T–S diagram of a simple recuperative carbon dioxide topping cycle (layout 1).

among others). Combined cycles with topping recuperative gas turbines and bottoming ORCs have been reported as an alternative to conventional combined cycles by Chacartegui et al. [20] and to low temperature solar thermal electric generation by Gang et al. [21].

This paper is focused on the analysis of alternative cycles for solar power plants with intermediate temperature central receiver (TIT below 1100 K) in order to improve the performance of the power plant. In all cases, carbon dioxide cycles have been considered, ether in stand-alone or combined cycle layouts with ORC bottoming cycles. The results show the interest of these cycles, which are envisaged as promising technologies for solar tower facilities.

2. Carbon dioxide cycles

2.1. Cycle description and modelling

Two carbon dioxide cycles for intermediate temperature are analyzed in this paper. First, a stand-alone closed recuperative Brayton cycle is considered, namely layout 1 –L1-, Fig. 1. Then, a second system, namely layout 2 –L2-, incorporates a two stage compression. After an initial compressor (05-06), a fraction of the flow is bypassed, cooled down and compressed (06-01-02), while the remaining flow is compressed with the same pressure ratio without being cooled (06-07). This recompression cycle is aimed at enhancing heat transfer and attenuating internal pinch point problems in the low temperature heat exchangers [15,22](Fig. 2).

Table 1
Main assumptions of carbon dioxide and ORC cycles.

Maximum pressure	225 bar
Minimum temperature	303–308 K
Pinch Point (or minimum economiser temperature difference)	10 K
Heat recovery exchangers efficiency	85%
Heat exchangers pressure drop	2%
Mechanical losses	2%
CO_2 turbine efficiency	90%
CO_2 compressor efficiency	80%
ORC condenser temperature	303 K
ORC turbine efficiency	87%
ORC pump efficiency	80%

A lumped volume approach has been used to model cycle performance, applying mass and energy conservation to each individual component of the system. For the fluids, real dense gas behaviour has been considered when working in the vicinity of the critical point (low compressibility factor). All these features have been incorporated into a computer code using Engineering Equation Solver (EES®) in order to facilitate post-processing of results and genetic algorithms optimisation. Results from the model have been compared against available data in literature [10,15] showing satisfactory agreement. The main assumptions of the model are shown in Table 1:

The CO_2 cycles were studied under compressor transcritical and supercritical inlet conditions. The maximum value of the compressor intake pressure (CIP) was limited to 75 bar.

2.2. Stand-alone carbon dioxide gas turbine analysis

This section shows results of the genetic algorithm optimisation carried out to maximise cycle efficiency [23]. First, Fig. 3 illustrates the impact of compressor inlet pressure on specific work and hot delivery temperature from the recuperator for different turbine inlet temperatures (TIT). These results apply to transcritical, $P_{01} < 73.5$ bar, and supercritical, $P_{01} > 73.5$ bar, cycles following layout 1 and show that there is little effect on specific work whereas the recuperative potential is very sensitive to compressor inlet pressure. Actually, increasing pressure at state 01 brings about a dramatic drop in hot delivery temperature at the recuperator, T_{04p}.

Fig. 4 shows specific work vs efficiency plots for layouts 1 and 2, L1 and L2 respectively. For the same turbine inlet temperatures considered in Fig. 3, an optimisation process has been carried out

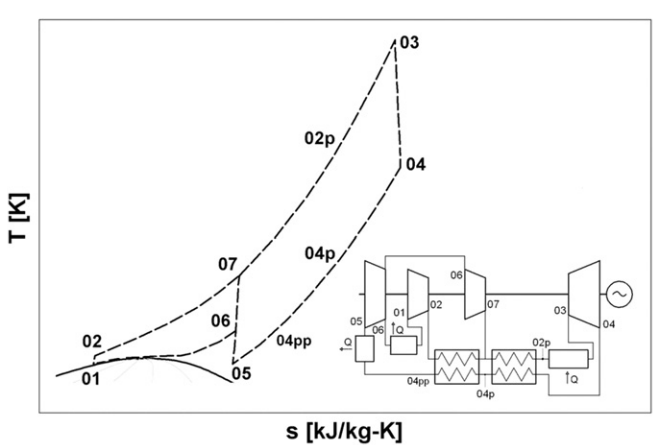

Fig. 2. T–S diagram of the supercritical carbon dioxide cycle (layout 2).

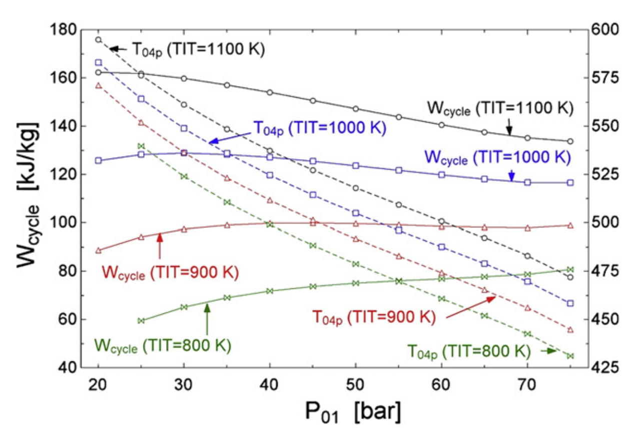

Fig. 3. CO_2 cycle (layout 1). Effect of compressor inlet pressure (P01) on specific work and recuperator hot delivery temperature (T_{04p}). Carbon dioxide cycle (layout 1).