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Recuperative solar-driven multi-step gas turbine power plants

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ABSTRACT

An analysis on the influence of the recuperator effectiveness in a multi-step solar-driven Brayton engine is presented. The solar collector model includes heat losses from convection and radiation. The Brayton engine includes an arbitrary number of turbines and compressors, regeneration, and several realistic irreversibility sources. It is stated that the combination of both systems makes the evolution of the overall efficiency with the effectiveness of the regenerator not trivial. Such behavior is associated to the losses arising from the coupling of the working fluid with the collector and the surroundings. The overall efficiency admits a simultaneous optimization in regards to the pressure and temperature ratios. When the system is designed to work close to the optimum values of those parameters an increase in the effectiveness of the recuperator is always associated with an increase in the overall optimum efficiency. This holds for configurations from the simplest solarized Brayton up to arrangements with several turbines and compressors.

1. Introduction

The use of regeneration, in combination with intercooling and reheating, in gas turbine power plants is a key concept in order to improve efficiency and reduce fuel consumption by introducing a heat exchanger that transfers waste heat exhausted from the turbine to preheat air from the compressor. Additionally, when the sun energy is used to drive the gas turbine the main advantage of the recuperator is the preheating of the working fluid before the receiver. This leads to a reduction of the surface of the concentration system and then to a saving of investment and maintenance costs.

Solarized and recuperative stand-alone gas turbine power plants or in combination with a bottoming Rankine cycle, have been proposed as a real possibility in the development of efficient installations with significant cost reductions for solar electric power generation [1–6]. These plants use direct solar heating of pressurized air by means of central receiver technology to get working temperatures over 1100 °C [7,8]. The feasibility of gas turbines to operate in a solar-fossil hybrid way has been also proved by means of appropriate combustion chambers [1,3–5], thus allowing predictable performance even when little or no solar energy is available.

From a theoretical point of view different results have been reported for solar-driven Carnot-like models with external [9–12] and internal [13–16] irreversibilities, for Braysson [15,17,18], Ericsson [19], and Rankine [20,21] cycle models under different optimization criteria. However, for solar-driven gas turbine cycles the reported analytical results are quite scarce: they include works for regenerative [22–24] and non-regenerative solar cycle models with one turbine-one compressor standing alone [15,25] or in combination with steam cycles [26]. Together with regeneration, multi-step compression and expansion constitutes a direct way to improve the thermal efficiency of the plant, although with the consequent cost increase. An economical analysis becomes necessary in order to balance the increment on efficiency with respect to the increase of costs.

The main purpose of this paper is to present a detailed theoretical analysis on the influence of the regenerator effectiveness in a multi-step, solarized Brayton cycle model which incorporates the main irreversibility sources, both for the thermal engine itself as for the solar collector. For the concentrating collector we assume a model which accounts for radiative and convective losses [18,27,28]. For the thermal Brayton cycle we assume a regenerative multi-step Brayton heat engine model with an arbitrary number of turbines N_t and compressors N_c accounting for: the non-ideal behavior of turbines and compressors, pressure drops during the heat input and heat release stages, regenerator irreversibility, heat leakage through the plant to the surroundings, and non-ideal couplings of the working fluid with the external heat reservoirs [29,30].

Although our numerical model allows for the consideration of an arbitrary number of turbines and compressors the main focus will be on realistic, recuperative configurations ranging from

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Nomenclature

A_a	aperture area of the collector
A_r	absorber area of the collector
a_c	isentropic compressor pressure ratio
a_t	isentropic turbine pressure ratio
C	concentration ratio
C_w	heat capacity rate of the working fluid
C_i	internal conductance of the power plant
G	solar irradiance
$\dot{m}$	mass flow rate of the working substance
M_1	radiation heat loss coefficient for the solar collector
M_2	convection heat loss coefficient for the solar collector
N_c	number of compressors
N_t	number of turbines
$\lvert\dot{Q}_H\rvert$	heat-transfer rate between the working fluid and the heat reservoir at T_H
$\lvert\dot{Q}_L\rvert$	heat-transfer rate between the working fluid and the heat reservoir at T_L
$\lvert\dot{Q}_{HL}\rvert$	heat-leak between heat reservoirs
r_p	overall pressure ratio
T_H	working temperature of the solar collector
T_L	ambient temperature
U_L	convective losses of the solar collector
α	effective emissivity
η	overall efficiency
$\eta_{\max}$	maximum overall efficiency with respect to the temperature ratio
$\eta_{\max,\max}$	maximum overall efficiency with respect to pressure and temperature ratios
η_s	efficiency of the solar collector
η_h	thermal efficiency of the heat engine
η_0	effective transmittance-absorptance product
ϵ_c	isentropic efficiency of the compressors
ϵ_r	regenerator effectiveness
ϵ_t	isentropic efficiency of the turbines
ϵ_H	irreversibilities coming from the coupling of the working fluid with the heat reservoir at temperature T_H
ϵ_L	irreversibilities coming from the coupling of the working fluid with the heat reservoir at temperature T_L
γ	adiabatic coefficient
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
τ	heat reservoirs temperature ratio
ξ	heat leakage through the plant to the surroundings

one-compressor and one-turbine to two-compressors and two-turbines. For all the configurations we survey in this work, theoretical results for the optimized efficiencies, optimized collector temperatures, and optimized pressure ratios will be presented. We shall see that our predictions for the efficiency, obtained in the basis of a purely thermodynamic analytical framework, are in accordance with those of experimental prototypes. Moreover, the optimum values for temperature and pressure ratios are attainable with currently available technology.

Thermodynamic models like the one we have developed can contribute to improve the design and optimization of this kind of solar power plants in order to make them interesting from a commercial viewpoint. This kind of studies constitute a pre-design analysis with the aim to avoid losses and define the useful intervals of the basic parameters of the plant. Particularly, in this work we show the necessity of a global knowledge of the plant parameters to get an advantage from the use of a regenerator. Our model does not require details of the recuperator, the only relevant parameter is its effectiveness, so it applies to different recuperators built up from different technologies.

2. Numerical model

The plant scheme we shall analyze is depicted in Fig. 1 and the corresponding thermodynamic sketch in Fig. 2. The Brayton heat engine absorbs a net heat rate $|\dot{Q}_H|$ from the solar collector at temperature T_H and releases a net heat rate $|\dot{Q}_L|$ to the ambient at temperature T_L. We also assume a linear heat leakage, $|\dot{Q}_{HL}|$ directly from the hot reservoir at T_H to the cold heat sink at T_L [31].

As usual for a concentrating collector we consider that heat losses at low and intermediate temperatures are essentially associated to conduction and convection while at high enough temperatures radiation losses are dominant. In this model the useful energy delivered to the heat engine, $|\dot{Q}_H|$, and the efficiency of the solar collector η_s can be written, respectively, as [18,27,28,32]

$$|\dot{Q}_H| = \eta_0 G A_a - \alpha\sigma A_r T_L^4(\tau^4 - 1) - U_L A_r T_L(\tau - 1), \tag{1}$$

$$\eta_s = \frac{|\dot{Q}_H|}{G A_a} = \eta_0[1 - M_1(\tau^4 - 1) - M_2(\tau - 1)] \tag{2}$$

In these equations $\tau = T_H/T_L$ denotes the heat reservoirs temperature ratio, G is the solar irradiance, A_a and A_r are, respectively, the aperture and absorber areas, η_0 is the effective transmittance-absorptance product (optical efficiency). $M_1 = \alpha\sigma T_L^4/(\eta_0 G C), M_2 = U_L T_L/(\eta_0 G C)$, where U_L is the convective heat loss coefficient, α is the effective emissivity of the collector, $C = A_a/A_r$ is the concentration ratio, and σ the Stefan–Boltzmann constant. Indeed, if $M_2 = 0$ the model with purely radiative losses is recovered [26].

The model for the closed multi-step Brayton thermal cycle was reported recently. Here, we briefly summarize the most important steps in the thermodynamic cycle (see [29] for details):

(1) The working fluid, a constant mass flow of an ideal gas with constant heat capacities and adiabatic coefficient γ, is compressed from the initial state 1 by means of N_c non-adiabatic compressors and $N_c - 1$ isobaric intercoolers. All the compressors are assumed to have the same isentropic efficiency ϵ_c and the same inlet temperature T_1. In a recent study for a solarized Braysson cycle, Wu et al. [18] have evaluated the influence of temperature dependent heat capacities in the overall efficiency. They reported differences below 2% (Table 1 in [18]) in comparison with the case where the heat capacity remains constant. Thus, in order to obtain analytical expressions for heat input and heat release we assume constant heat capacities for the working fluid.

(2) After state 2 the gas is pre-heated to state X in a regenerative counterflow heat exchanger with effectiveness $\epsilon_r = (T_X - T_2)/(T_4 - T_2)$. A non-regenerative cycle corresponds to $\epsilon_r = 0$ while ideal or limit regeneration corresponds to $\epsilon_r = 1$. This interval of values for ϵ_r ensures that the temperature of the working fluid after the last compressor, T_2 is lower than that at the end of the expansions in the turbines, T_4 [33].

After X the working fluid is heated up to the final maximum temperature T_3. The global irreversibilities in this hot-end heat exchanger are accounted by $\epsilon_H = (T_X - T_3)/(T_X - T_H)$. The overall heating process from state 2 to 3 is considered as non-isobaric, with a pressure drop quantified by ρ_H ($\rho_H = 1$ corresponds to a zero pressure decay) [33,34].