Multi-stage configurations for central receiver hybrid gas-turbine thermosolar plants

R.P. Merchán[a], M.J. Santos[b], A. Medina[c] and A. Calvo Hernández[d]

[a] University of Salamanca, Dpt. of Applied Physics, Salamanca, Spain, rpmerchan@usal.es
[b] University of Salamanca, Dpt. of Applied Physics, Salamanca, Spain, smjesus@usal.es
[c] University of Salamanca, Dpt. of Applied Physics, Salamanca, Spain, amd385@usal.es
[d] University of Salamanca, Dpt. of Applied Physics, Salamanca, Spain, anca@usal.es

Abstract:

A thermodynamic model for hybrid Brayton thermosolar plants is proposed with the aim to analyze possible configurations with improved performance. In these plants an array of mirrors with a two-axis tracking system gathers solar power and redirects it to a central receiver. In turn the receiver acts as a heat exchanger that heats up a gaseous working fluid that runs a Brayton-like cycle. These plants also include a combustion chamber that ensures an approximately constant power output even during night or in periods with poor solar irradiance. Throughout the last years it has been demonstrated by means of experimental projects and prototypes that this concept is technically feasible but still R+D+i efforts are required in order to reach commercial feasibility. From the thermodynamic viewpoint it is necessary to increase overall plant efficiency. The model proposed in this paper is an extension of previous studies from our group that takes into consideration multi-stage configurations with an arbitrary number of compression steps with intercooling and expansion stages with reheating between turbines. The model is comprehensive and includes the main sources of losses in real plants: pressure decays in heat absorption and release, losses in compressors, turbines and heat exchangers, non-ideal recuperators and, of course, losses in the solar subsystem and combustion chamber.

A numerical application is done taking as reference the data from the project Solugas, developed by the Abengoa Solar at the south of Spain. Several plant configurations are analyzed and also different working fluids checked, including air, nitrogen, carbon dioxide, and helium at subcritical conditions. It is concluded that for air, nitrogen and carbon dioxide, plant configurations with 2-3 compression/expansion steps are capable of achieving improved overall plant thermal efficiency (about 25% above single step plants) and also fuel conversion efficiency, *i.e.*, lead to a considerable increase in power output without an appreciable increase in fuel consumption.

Keywords:

Thermosolar gas-turbines, Hybrid plants, Thermodynamic model, Multi-stage gas-turbines, Working fluids

1. Introduction

During the last years a great R+D effort is being done in order to develop energy production technologies that at the same time can meet two basic ingredients required by our society: clean and efficient generation on one side, and reliable, non-intermittent, and predictable on the other. Among other options thermosolar power plants with a central tower solar receiver are being investigated. In these plants a heliostat field collects and redirects solar power to a receiver located at the top of a central tower. This receiver, in turn, transfers this high temperature heat to a working fluid that runs a thermodynamic heat engine as a Rankine or Brayton one (or a combination of both) [1,2].

In these plants it is not difficult to include a combustion chamber in series with the solar receiver in such a way that during low irradiance periods due to meteorological conditions or during the nights, the combustion chamber provides the heat input required to keep approximately constant the turbine inlet temperature. And so, the plant power output. These plants are not completely carbon free, because of the combustion of a fossil fuel (that uses to be natural gas) but produce constant, reliable, and predictable electric power to the grid. These plants are so-called hybrid thermosolar plants [3,4].

Several projects during the last 10 years allowed for building prototype and pre-commercial scale plants [5,6]. All of them coincide in the feasibility of this kind of installations and the mentioned advantages. Particularly, those ones working (from the thermodynamic viewpoint) under Brayton-like cycles have additional benefits: very reduced water consumption, high efficiency rates, not too complex control systems, flexibility inherent to these engines, and reliability [7]. Nevertheless, another agreement among researchers and developing companies is that improvements are still necessary in order to produce energy at competitive prices, and so, to develop plants at commercial scales. There are several work lines to walk along: solar receivers design and materials, appropriate heat exchangers for very high temperatures, efficient Brayton cycle layouts, improved working fluids, and others [8]. This work is focused on the last two mentioned points.

Our research group has developed during the last times thermodynamic models in order to predict and analyze the performance of these plants [9,10,11]. One of our aims is to perform global models for the plant as a whole, keeping the simplicity as far as possible, but at the same time to be able to achieve predictions with accuracy when comparing with real plants. This approach makes possible to identify the main bottlenecks in plant output records, develop sensitivity analysis, check different plant configurations, and ultimately to suggest optimized plant designs.

In this work we present a thermodynamic model capable to forecast plant performance in the case of operating with a closed Brayton cycle for any number of compression/expansion stages. This is one plausible option to improve heat engine efficiency and so, that of the overall plant. Moreover, the model is also able to work with different gases as working fluids for the heat engine. Although the last years there has been a great amount of works about the possibilities of using supercritical CO_2 (apart from the most standard fluid, air) [7,12,13], we are interested in the analysis of subcritical CO_2 together with the concept of multi-stage compression and expansion. This strategy could avoid reaching too high pressures and the scarce technical experience in the design and performance of turbomachinery and heat exchangers working with transcritical or supercritical fluids.

We present in this paper results for several plant configurations with a variable number of compression/expansion stages and for 4 working fluids: dry air, nitrogen, helium, and carbon dioxide. To obtain numerical values particular parameters were taken from a pre-commercial scale plant recently developed by the company Abengoa Solar near Seville, Spain [6,14]. Optimum pressure ratios as a function of the number of stages will be presented for all the fluids and predicted numerical values of overall plant performance, power output, fuel conversion efficiency, and specific fuel consumption are analyzed in detail. It will be shown that considerable gains over the reference plant could be obtained with a Brayton cycle with two compression stages and one or two expansion stages. Moreover, subcritical carbon dioxide leads to quite interesting values of the fuel conversion rate because of the low associated fuel consumption.

2. Theoretical considerations for the simulation model

Our system is composed of a concentrated solar power (CSP) tower collecting solar energy from a heliostat field and acting as heat source for a hybrid gas turbine operating a closed Brayton cycle. The whole system is sketched in Fig. 1 and the corresponding T-S diagram in Fig. 2. The working fluid is compressed through an arbitrary number N_c of compressors (atmospheric pressure is assumed at the inlet of the first compressor). Between each pair of compressors an intercooler ensures that the temperature at the inlet of any compressor is always T_1. After the last compressor the fluid is subsequently heated up in three steps. First, by means of a recuperator that takes advantage of the high temperatures of the fluid at the exit of the last turbine. Second, the fluid is heated up in the solar receiver located at the top of the solar tower. And third, a main combustion chamber burning natural gas ensures that the turbines inlet temperature is always approximately constant. A control system is required to this aim. Particularly by night or if solar irradiance is scarce the combustion chamber provides all the energy to heat the fluid. Afterwards, the fluid is expanded by means of an arbitrary number, N_t, of turbines. Between them intermediate reheaters make that for each turbine the inlet temperature is fixed at a value T_3.

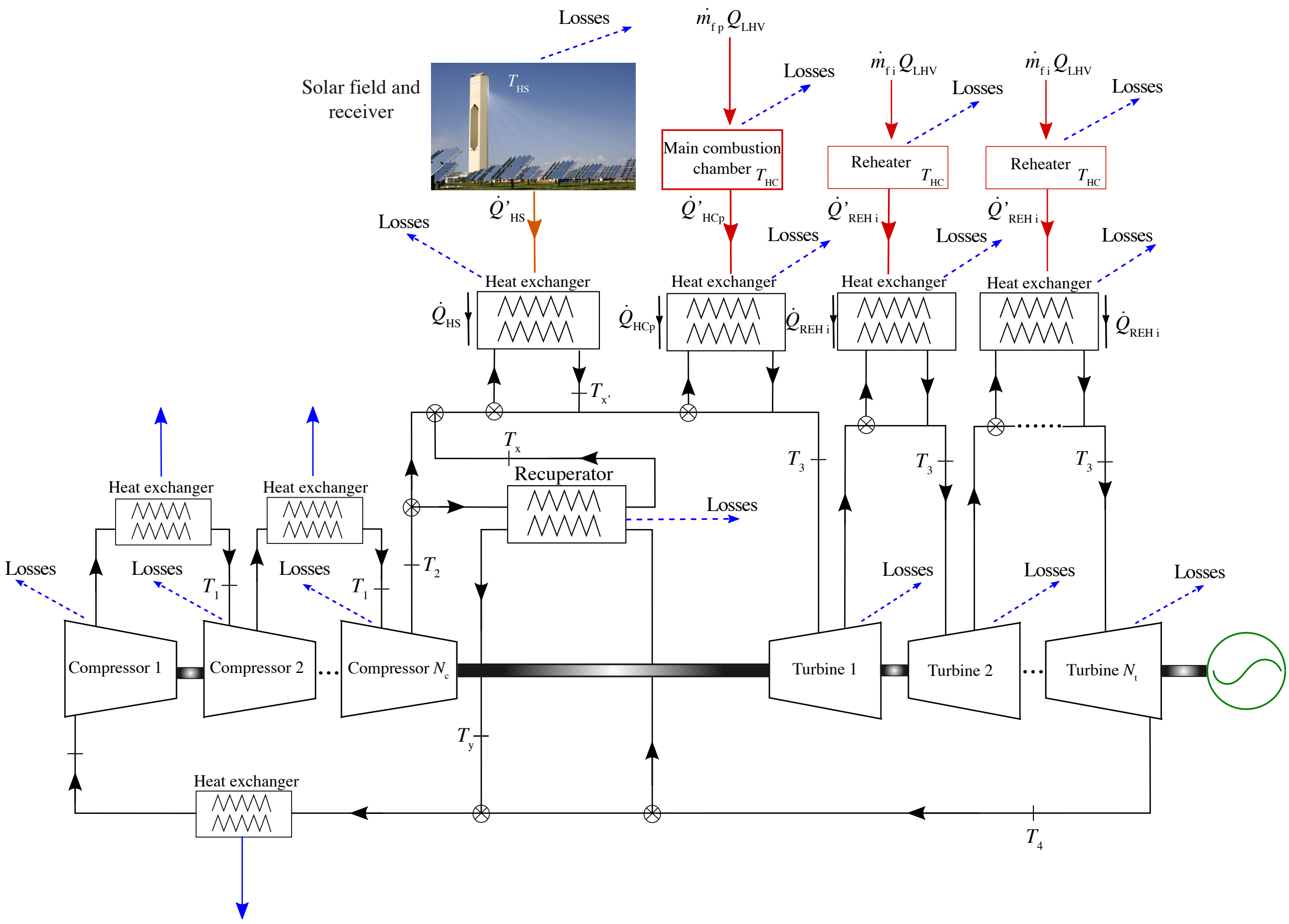

Fig. 1. Simplified picture of the hybrid plant. A CSP central tower solar collector provides heat for a hybrid multi-stage recuperated gas turbine.

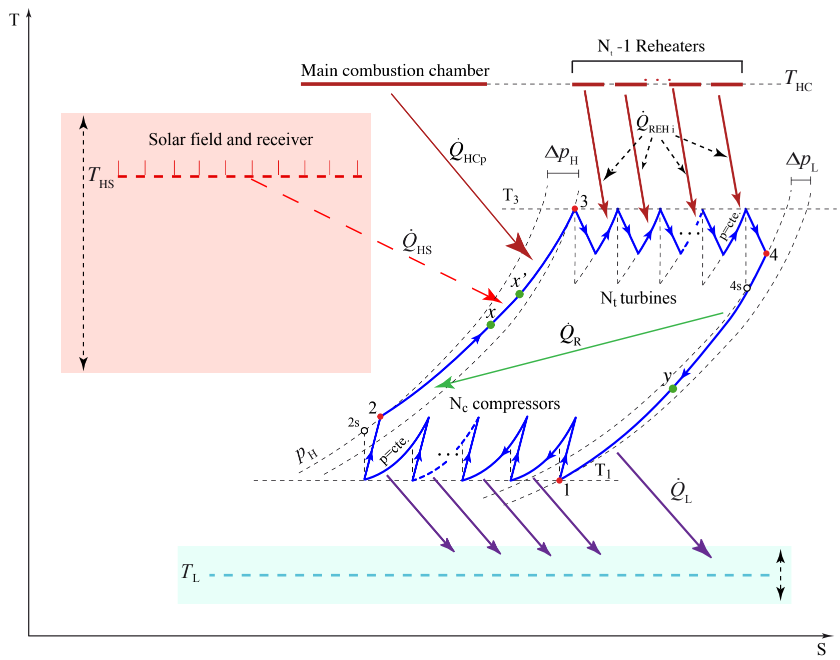

Fig. 2. T-S diagram of the system. Compressors, turbines, and recuperator are non-ideal. Pressure decays are assumed in heat input and release. The model could be applied to variable, off-design conditions although in this paper on-design parameters will be assumed.

Overall plant efficiency, η, is the ratio between the mechanical power output and the total heat input, from the solar collector, from the main combustion chamber, and the secondary reheaters in between turbines:

$$\eta=\frac{P}{GA_a+\dot{m}_f Q_{LHV}}$$

where G is the direct normal irradiance, A_a, the aperture area of the solar field, $\dot{m}_f$, the total fuel mass flow, and Q_{LHV} the lower heating value of the fuel. It is feasible to express this overall efficiency in terms of the efficiencies of all the subsystems involved and the heat exchangers connecting them. Details of the calculations can be found in [9,10].

$$\eta=\eta_h\eta_s\eta_c\left[\frac{\epsilon_{HS}\,\epsilon_{HC}}{\eta_c f\,\epsilon_{HC}+\eta_s(1-f)\,\epsilon_{HS}}\right]$$

In this equation η_s is the efficiency of the solar subsystem, the fraction of the solar power collected in the heliostat field that can be usefully released by the receiver. It depends on the optical efficiency and the heat losses in the receiver that include radiation terms [15]. It is important to underline that this efficiency depends on the temperature of the solar receiver, T_{HS}. An explicit expression can be found in [10]. ϵ_{HS} represents the effectiveness of the solar receiver as heat exchanger that transfers a fraction of the solar energy to the working fluid developing the thermodynamic cycle. The efficiency of the main combustion chamber and the intermediate reheaters between turbines is assumed identical and is represented by η_c. It depends on combustion inefficiencies and the fuel-air ratio of the fuel mixture. Usually is taken as a constant parameter. ϵ_{HC}, denotes the fraction of the heat released by the combustion chamber that actually yields to the working fluid. In other words, the effectiveness of the combustion chamber as a heat exchanger (note that we are considering a closed Brayton cycle and so a heat exchanger is necessary to release the energy to the fluid from the combustion chamber or intermediate reheaters). The factor f is the solar share, the ratio between the heat input to the thermodynamic cycle coming from the solar collector with respect to the total heat input (solar and combustion). In the case that the plant works in a pure combustion mode (by night or for not enough solar irradiance), f=0. The opposite case is that where all the heat input comes from the receiver if irradiance is good enough, f=1.

Finally, η_h, is the efficiency of the heat engine itself, *i.e.*, of the recuperative and multi-stage Brayton cycle. Details on the mathematical expressions can be found in [9]. For the sake of brevity it is noteworthy that an analytical expression can be obtained for η_h including the main losses sources in this kind of installations: non-isentropic compressors and turbines, pressure losses in all heat input and heat release processes, non-ideal recuperator, and non-ideal heat exchangers. Moreover, the dependence with the temperature of the heat capacity of the working fluid is taken into account.

Apart from the power output P and the thermal efficiencies of the subsystems, η_s , η_c, η_h, and the overall thermal efficiency, η, for practical purposes it is interesting to consider a fuel conversion rate, r_e, that represents the quotient between the power output and the component of the input energy with an associated economical cost, *i.e.*, the input energy provided by the combustion chambers. It can be calculated as:

$$r_e=\frac{P}{\dot{m}_f Q_{LHV}}=\frac{\eta\eta_h\eta_s\,\epsilon_{HS}}{\eta_h\eta_s\,\epsilon_{HS}-\eta\,f}$$

3. Numerical considerations

A previous version of this model in the particular case of single stage compression and expansion, considering air as working fluid was previously presented by our group [9,10]. For validation purposes numerical data were taken from a prototype plant developed by the company Abengoa

Solar at the south of Spain, called *project Solugas* [6,14]. Design point conditions were considered for solar irradiance (G=860 W/m^2) and ambient temperature (T_L=288 K). In this work these conditions are also considered. It is important to notice that the model is also susceptible to be applied to variable conditions and perform off-design estimations. Detailed values for the main parameters can be found in [9,10]. We only summarize here the main ones. The commercial turbine used in the project Solugas (model Mercury 50 by Caterpillar) is single stage and works with a pressure ratio of 9.9 and air gas flow $\dot{m}_f$=17.9 kg/s. It is fuelled with natural gas. As in this work we analyze possible multi-stage configurations the pressure ratio will be considered as a parameter for optimization, but the working fluid mass flow will remain the same. The turbine inlet temperature is approximately 1423 K and its thermal efficiency after the generator is 0.385. Our model was validated by comparing its predictions with the commercial turbine in [9], where all the numerical parameters can be checked. When the turbine is modified to be incorporated to the solar tower in the Solugas project, the following records were predicted at design conditions [10,11]: solar share, 0.32; fuel conversion efficiency, 0.58; specific fuel mass consumption, 132 kg/(MW h); mechanical power output, 5.06 MW; and overall thermal efficiency, 0.32.

In this work we survey possible forms to increase the performance of the original prototype plant by considering the options to increase the complexity of the gas turbine layout by incorporating multi-stage compression and expansion, and also by considering possible alternative working fluids for the thermal cycle.

Different theoretical and experimental studies have reported the influence of different working fluids on closed Brayton cycles, for instance, nitrogen, helium, carbon dioxide, and other noble gases. Several papers summarize the advantages and disadvantages associated to these fluids [7,12,13]. We shall focus on the comparison of the predictions of the model for air and nitrogen, helium, and carbon dioxide at subcritical conditions, except for helium. In this case a transcritical Brayton cycle is performed by the fluid because of its low critical pressure (2.2761 bar) and temperature (5.1953 K). In the rest of fluids the conditions of the Brayton cycle exceed their critical temperatures but pressures remain below their critical pressures. For each fluid a temperature dependent fitting polynomial for the constant pressure heat capacity, $c_w(T)$, will be considered. The parameters of the fit are taken from [16] at a pressure of 5 bar. It is noteworthy that the average value of $c_p(T)$ in the temperature interval of the Brayton cycle is about 4.5 times larger for He [5.1965 J/(g K)] and its molar mass (4.00 g/mol) is quite below those for N_2 (28.01 g/mol), air (28.97 g/mol), and CO_2 (44.01 g/mol).

4. Optimal pressure ratios for multi-stage configurations

In this section we shall present the predictions of the developed simulation model with the aim to find plant layouts with improved performance. To get this objective three ingredients are surveyed: the number of compression/expansion steps (assuming N_c=N_t=N), the optimal overall pressure ratio in each case (r_p), and the nature of the working fluid developing the Brayton cycle. Figure 3 depicts the evolution with r_p of the overall efficiency, η, for the 4 working fluids considered and for several values of N. The single stage case corresponds to N=1. The limit case of infinite stages is depicted just as a hypothetical upper limit for the interval of values that η could eventually reach. Moreover, in the case of He, recuperative and non-recuperative plant schemes are plotted. Other cases are always recuperative. For He and a regenerative plant in the single-stage configuration the optimum pressure ratio is about r_p=4. This optimal pressure ratio increases to about 10 for a non-recuperative plant. For all fluids the overall efficiency increases with N. Only in the case N=1 the curves display a maximum at moderately low values of r_p. For multi-stage configurations η increases monotonically and very slowly for values of r_p above 8 – 10. In the plot corresponding to dry air the reference value of the thermal efficiency of the Solugas plant is marked for comparison η=0.32 for r_p=9.9. Table 1 contains for each fluid the numerical values of the maximum efficiency and the associated pressure ratio. The change from single-stage to two-stages provokes a large increase of overall efficiency, for all working fluids. Subsequent increases of *N* do not lead to so large efficiency rising.

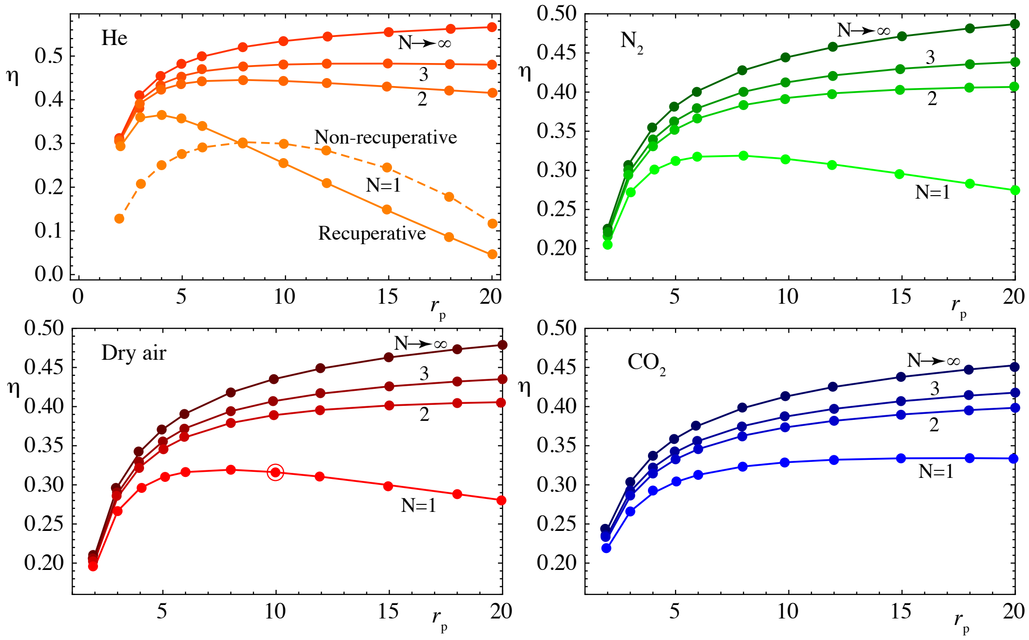

Fig. 3. Overall plant efficiency plotted against the overall pressure ratio, r_p, for different working fluids and several numbers of compression/expansion steps, considered identical, N. An open circle in the case of dry air marks the reference value for the plant Solugas [6]. In the case of He and N=1 two configurations are surveyed: recuperative (solid line) and non-recuperative (dashed).

Table 1. Maximum overall efficiency, η, for single (N=1) and multi-stage configurations and corresponding pressure ratios. In parenthesis the relative increments with respect to the reference case (Solugas plant, η=0.32, r_p=9.9) are shown as percentages.

Working fluid	*N*=1		*N*=2		*N*=3		*N* → ∞	
	η_{max}	r_p	η_{max}	r_p	η_{max}	r_p	η_{max}	r_p
He	0.37 (+15.6)	4	0.44 (+37.5)	8	0.48 (+50.0)	15	0.57 (+78.1)	20
N_2	0.34 (+6.2)	7	0.41 (+28.1)	20	0.44 (+37.5)	20	0.49 (+53.1)	20
Dry air	0.32 (0.0)	8	0.41 (+28.1)	20	0.44 (+37.5)	20	0.48 (+50.0)	20
CO_2	0.34 (+6.2)	18	0.40 (+25.0)	20	0.42 (+31.2)	20	0.45 (+40.6)	20

Power output for the cases considered is shown in Fig. 4. For multi-stage configurations the curves do not display a maximum, monotonically increase with the analyzed pressure ratios. For *N*=1, air, and N_2 show a maximum around r_p=10, that corresponds with the on-design power output of Solugas. In the case of CO_2, even for *N*=1, *P* is monotonic, larger values of r_p always lead to larger values of *P*. With respect to the numerical scales of *P*, the plot for He shows that it results in much higher values. This is a consequence of the setup we are considering for the comparison among fluids. The numerical magnitude of *P* is proportional to the product $\dot{m}\, c_w$, where $\dot{m}$ is the fluid mass flow and c_w its constant pressure specific heat. As the same mass flow for all fluids was assumed, power output for He is about 4.5 times larger than for the other fluids, the same than the ratio between its specific heat and the others. Another important feature, valid for all working fluids, is that there is a considerable increase of power output from *N*=1 to *N*=2. Higher number of stages also increases *P*, but not so markedly.

The fuel conversion rate, r_e (see Fig. 5), shows for all the fluids a narrow maximum at low values of pressure ratio for *N*=1. For He the height of this maximum is below the curves for multi-stage configurations, but in other fluids it is above. Moreover, for N_2, dry air and CO_2, there is a crossing point among curves and the comparison between different multi-stage configurations is not trivial. Physically this is associated to the simultaneous increase of power output and fuel consumption with the number of stages, associated to the heat required in the intermediate reheaters between turbines.

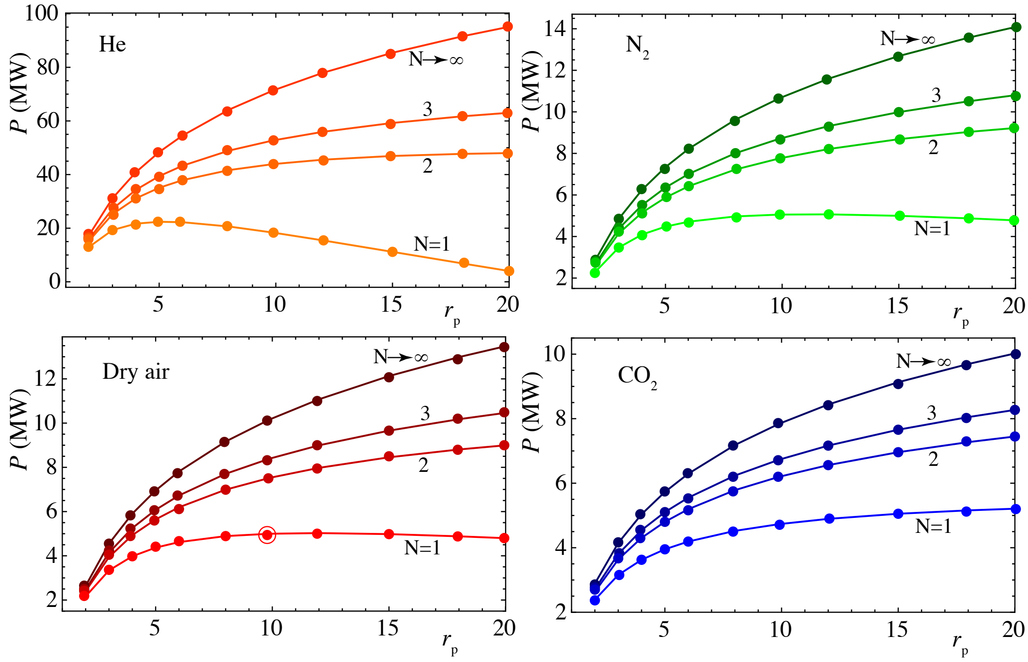

Fig. 4. Power output (P) plotted against the overall pressure ratio, r_p, for different working fluids and several numbers of compression/expansion steps, considered identical, N.

Except for He, for low pressure ratios, single-stage configurations are beneficial and for high pressure ratios multi-stage layouts are advantageous in terms of the fuel conversion rate. It is convenient to note that the curves for the fuel consumption (not shown) have a behavior opposite to those for r_e, the maxima turns to be minima and the monotonically increasing behavior to decreasing. In the next section these curves will be shown for particular cases.

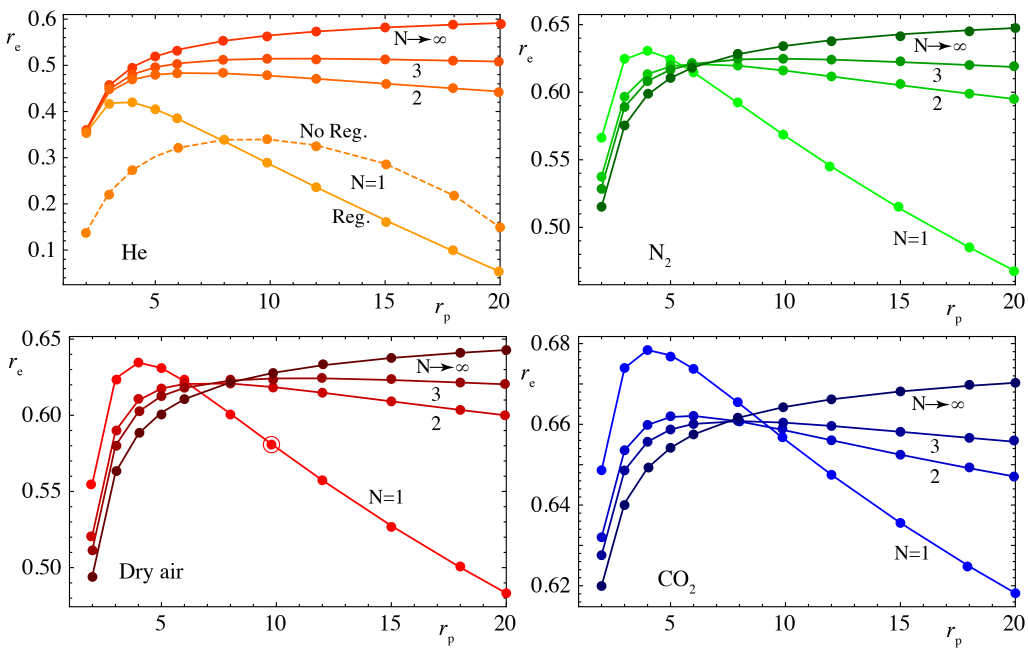

Fig. 5. Fuel conversion rate (r_e) plotted against the overall pressure ratio, r_p, for different working fluids and several numbers of compression/expansion steps, considered identical, N.

5. Two-stage compression layouts

One important conclusion from the preceding section is that the thermodynamic model predicts a considerable increase in the plant output records (overall efficiency, power output, and fuel conversion rate) when a single-stage Brayton cycle is substituted by a two-stages one. In this section we try to deep in this point by analyzing in detail two particular configurations, that with N_t=N_c=2 (N=2) and another with two compression stages with intercooling (N_c=2) and single expansion (N_t=1) (avoiding the fuel consumption associated to reheating). Figure 6 displays the evolution with r_p of several plant efficiencies for both layouts. Overall plant efficiency, η, for all fluids is smaller for single expansion. For N_t=N_c=2, overall efficiency for N_2 is slightly above that for dry air, but differences are scarce. In this case, for air, N_2 and CO_2 it is necessary to take high values of r_p to get the largest efficiencies. Nevertheless, curves for He (for both layouts) show maxima at smaller values of the pressure ratio (about r_p=8 for N_t=N_c=2 and about 5 for N_c=2, N_t=1).

The curves for the efficiency of the solar subsystem, η_s, never display maxima, always increase with r_p. In this case curves for N_t=N_c=2 are always below the corresponding ones for N_c=2, N_t=1, but looking at the scale of the vertical axis, numerical differences are small. The curves for the heat engine thermal efficiency, η_h, resemble those for the overall efficiency, η. The main difference comparing both sets of curves is the case of He. This working fluid leads to clearly larger values of η than the others, but numerical values for η_h are similar than for the other fluids. This fact arises partially from the solar efficiency, η_s, that is higher for He because the receiver temperatures are smaller for this fluid.

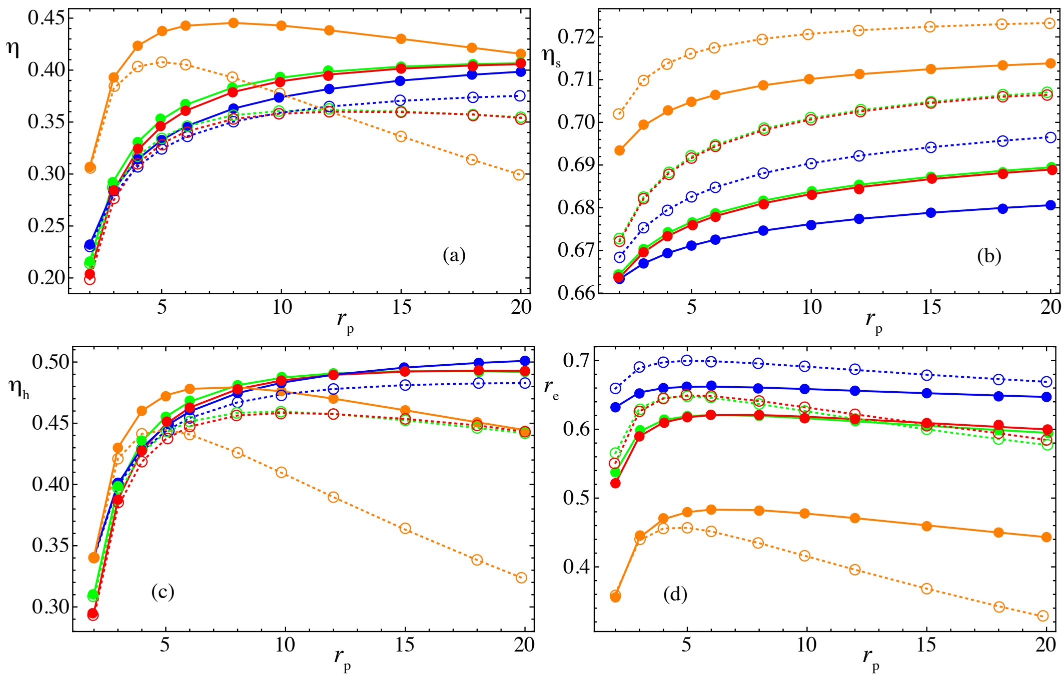

Fig. 6. Some plant efficiencies (η, overall plant efficiency; η_s, solar subsystem efficiency; and η_h, Brayton cycle efficiency) and fuel conversion rate (r_e) as functions of the overall pressure ratio for two particular configurations: N_t=N_c=2 (solid lines) and (N_c=2, N_t=1) (dashed). Different working fluids are displayed with different colours: He, orange; N_2, green; dry air, red; and CO_2, blue.

At the bottom right of Fig. 6 the evolution of the fuel conversion rate is depicted. This plot is interesting because shows that CO_2 leads to better values than the other fluids. Actually, the configuration N_c=2, N_t=1 reaches a maximum at r_p=5 that gives r_e=0.70, which is a noticeable result. Table 2 contains some particular values of maximum overall efficiencies and fuel conversion rates. Curves for r_e also show that for all fluids there exist a maximum at relatively low compression ratios, $r_p \approx 5-6$. For instance, N_2 and air show maxima for N_c=2, N_t=1 at r_p=5 with values about 0.65.

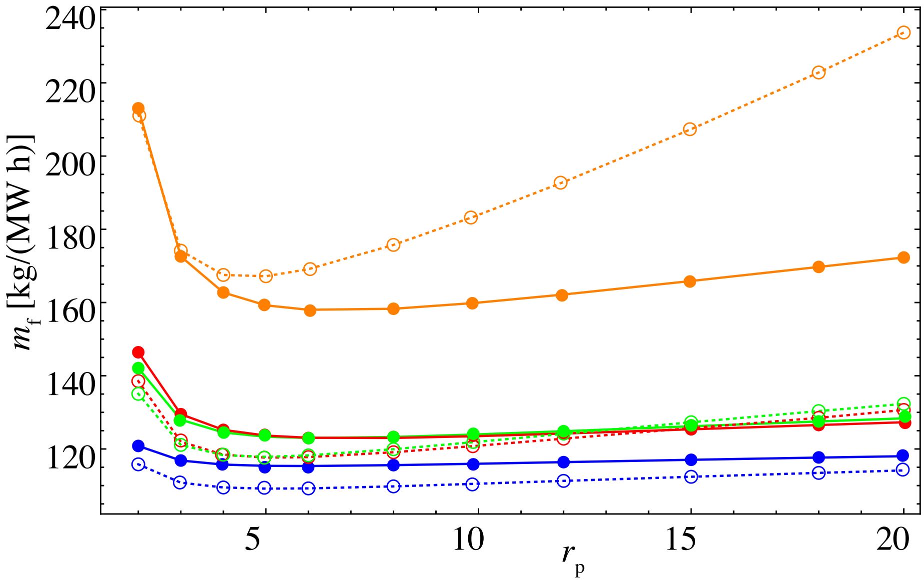

Fig. 7. Specific fuel consumption for the considered fluids and two-compressors configurations. Colours and dashing are the same that in Fig. 6.

The specific fuel consumption, m_f, is depicted in Fig. 7. The behaviour of the curves is opposite to the curves for r_e. Maxima move to be minima and the fluids sequence is reversed. CO_2 leads to the smallest values of m_f [about 120 kg/(MW h)] and He to the largest ones [between 160 – 170 kg/(MW h)].

Table 2. Maximum values of overall efficiency (η_{max}), fuel conversion rate ($r_{e,max}$), power output (P_{max}), and minimum values of specific fuel consumption ($m_{f,min}$). The reference value of the Solugas plant is also included for the sake of comparison (Ref.) [6,14].

	η		r_e		P		m_f	
	η_{max}	r_p	$r_{e,max}$	r_p	P_{max} (MW)	r_p	$m_{f,min}$ [kg/(MW h)]	r_p
Ref.	0.32	9.9	0.58	9.9	5.1	9.9	132	9.9
							N_t=N_c=2	
He	0.45	8	0.48	6	48.0	20	158	6
N_2	0.41	20	0.62	6	9.2	20	121	6
Dry air	0.41	20	0.62	6	9.0	20	121	6
CO_2	0.40	20	0.66	6	7.4	20	118	6
					N_c=2; N_t=1			
He	0.41	5	0.45	5	29.3	8	170	5
N_2	0.37	12	0.65	5	6.6	20	118	5
Dry air	0.36	12	0.65	5	6.4	20	118	5
CO_2	0.38	20	0.70	5	6.1	20	108	5

6. Summary and conclusions

A thermodynamic model has been applied in order to obtain numerical predictions about a hybrid thermosolar central tower plant working on a closed Brayton cycle. The model is a step forward over previously versions developed by our group. Particularly, it is feasible to incorporate an arbitrary number of compression/expansion stages and recuperation. It includes the main irreversibilities in the subsystems assembled in the plant: optical and thermal losses in the solar collector and receiver, losses in the combustion chambers, and irreversibilities in the Brayton heat engine. The latter includes non-isentropic turbines and compressors, real heat exchangers, and non-ideal recuperator, as well as, pressure decays in heat input and heat release processes. Moreover, temperature dependent polynomials for the working fluid are considered. The objective of the paper is to investigate possible plant layouts in combination with appropriate working fluids to enhance the performance of this kind of installations looking for better economic and commercial outcomes.

The numerical application was made on the basis of an experimental prototype plant developed by Abengoa Solar near Seville, Spain (project Solugas). On-design conditions were considered and reference values were taken from this project. Model predictions for several plant outputs were presented (overall plant efficiency, power, fuel conversion rate, and specific fuel consumption) as functions of two parameters, the overall pressure ratio and the number of compression/expansion stages. Several working fluids at subcritical conditions were surveyed: dry air, N_2, CO_2 and He (transcritical). For all of them a noticeable increase on overall efficiency, power output and fuel conversion rate is observed when a two-stages layout is considered with respect to the simplest single-stage reference configuration. Helium leads to the highest overall efficiencies (about 0.45) and power output, but requires high fuel consumption. Dry air and N_2 lead to similar values for all records and CO_2 results in very interesting values of the fuel conversion efficiency (about 0.7).

So, as suggestion for future developments, two-stages compression configurations (with one or two expansion stages) working with transcritical He are predicted to lead to high overall thermal efficiencies and power output at a relatively high specific fuel consumption. Subcritical CO_2 cycles, on the contrary, lead to appealing low fuel consumption rates. In all cases pressure ratios are not high, so, as atmospheric pressure was assumed at compression inlet, not so high pressures are reached during the cycles.

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