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Performance study of solar power plants with CO_2 as working fluid. A promising design window

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ABSTRACT

In this paper a systematic analysis is reported on the use of CO_2 as heat carrier fluid in solar thermal receivers and as thermodynamic working fluid. It includes the performance of close-to-critical regenerative Brayton cycles, which opens a broad field of cycle possibilities with low pressure ratios (very simple turbines) complemented with large but standard heat exchangers as regenerators. Radiation intensities needed to reach relevant efficiencies are in the range above 25 kW/m^2, but receiver efficiencies do not increase significantly beyond that value, featured as a threshold. Receivers are made of multi-tube bundles enclosed in glass-windowed collectors with compensated pressure and dilatation, which eliminates the problem of gas leakage through rotating joints and other non-hermetic fits. This leads to needing concentrators compatible with those collectors, which can be either finely optimized Linear Fresnel Reflectors or central minitowers. CO_2 was chosen for this study because its critical temperature (31 °C) is very close to environmental temperature, which conveys very positive features for the efficiency of the cycle. The overall result of the theoretical study is the identification of a set of different types of efficient, flexible and robust CSP plants with CO_2 as the only fluid which deserves further research at experimental level and in the design and construction of new plant components.

1. Introduction and background

One of the most critical decisions in the design of a solar-thermal power unit is the choice of the radiation concentration geometry [1–6], which is in turn connected with the thermal flux needed in the receiver to fulfill the working temperature of the powerplant. At this point, it must be taken into account the coherence in the design: if a high value is chosen for the concentrated radiation thermal flux, the global heat transfer coefficient between the receiver inner surface and the heat carrier fluid must have a similarly high value [7,8]. Otherwise, the temperature difference between the receiver and the fluid will be very large [9,10], enhancing the thermal losses from the receiver. Besides that, large temperature differences between different parts of the receiver will convey important differential expansion effects, which can be a major cause of concern in the durability of the receiver [9].

In this sense, linear receivers such as trough and Fresnel collectors, with lower concentration capabilities than towers or solar dishes [1–6,11,12] can be adequately connected with fluids with minor heat transfer coefficient values [7,8]. It is known that the usual fluid used in linear receivers, mainly in trough collectors as SEGS plants, is thermal oil [13]. This fluid has the operational limitation of its maximum working temperature, below 400 °C for the case of the most used one, Therminol VP-1 [14], and the chemical instabilities related with the use at high temperature for a long time. Another well-known possibility is the use of direct steam generation [15–20], but this option has the inconvenience when higher temperatures are wanted, that implies higher pressures and in turn implies higher pipe thickness [21], deteriorating the heat transfer. In relation with its own nature, both fluids have problems: thermal oil is an aggressive contaminant and water is typically expensive in the places where the thermal power plants can be placed, usually deserts.

Other fluids can be analyzed in relation with heat transfer requirements commented above: gases [22,23]. Moreover, properly chosen gases can fit quite well some kinds of Brayton cycles recently identified as very efficient ones [24] for the temperature levels achievable with simple and cheap solar fields.

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Nomenclature

a	constant involved in the theoretical relation between pressure drop and fluid velocity (–)
A	receiver surface (m^2)
B	pumping power (W)
c	constant involved in the equation of state for ideal gas (m^2/s^2-K)
CSP	concentrating solar plants
Cp	fluid specific heat (kJ/kg-K)
Δp	pressure drop (Pa)
ΔT	temperature difference between receiver pipe and fluid (K)
D_e	receiver pipes external diameter (m)
D_i	receiver pipes inner diameter (m)
e	pipe thickness (m)
E	total amount of energy impinging in the receiver (W)
ε	effectiveness (–)
η	Brayton cycle efficiency (–)
η_{elec}	mechanical to electricity conversion efficiency (alternator efficiency) (–)
η_{total}	solar receiver and Brayton cycle coupling efficiency including the efficiency to electricity conversion (–)
F	relation between fluid properties and film coefficient in the L/D relation (–)
h	specific enthalpy (J/kg)
h_{in}	tubes inner film coefficient (W/m_2-K)
I	heat flux intensity (W/m^2)
k	fluid thermal conductivity (W/m-K)
L	receiver pipes length (m)
$\dot{m}$	mass flow (kg/s)
n	receiver tubes number (–)
μ	fluid viscosity (kg/m-s)
P	fluid Pressure (Pa) in equations, (bar) in figures
P_{int}	fluid internal pressure (bar)
P_{ext}	fluid external pressure (bar)
Pr	Prandtl number (–)
$P1$	turbine inlet pressure (bar)
$P2$	turbine outlet pressure (bar)
P_{high}	higher pressure reached in the Brayton cycle (bar)
P_{low}	lower pressure reached in the Brayton cycle (bar)
q''	thermal flux impinging in the receiver (W/m^2)
ρ	fluid density (kg/m^3), pressure ratio (–)
SE	maximum admissible material stress at design temperature (ASME B31.1) (kpsi)
SM	thickness safety margin (ASME B31.1) (m)
T	temperature (K)
T_{cr}	fluid critic temperature (°C)
T_{fluid}	receiver fluid temperature (K)
T_{f0}	receiver fluid inlet temperature (K)
T_{t0}	receiver tube temperature at fluid inlet (K)
T_{fM}	receiver fluid outlet temperature (K)
T_{tM}	receiver tube temperature at fluid outlet (K)
T_{tube}	receiver tube inner temperature (K)
$T1$	turbine inlet temperature (K)
v	fluid velocity (m/s)
$V_{s,compressor}$	specific volume at compressor (isentropic compression) (m^3/kg)
$V_{s,turbine}$	specific volume at turbine (isentropic expansion) (m^3/kg)
w	turbo-machinery specific work (J/kg)
Y	coefficient related with the steel type and the temperature (ASME B31.1) (–)

1.1. Gas as a working fluid: pressure design window

Of course, gases have the inconvenience of low density, and are not usually used for cooling purposes. But if the nuclear energy field is studied, it can be considered that gases are usual coolers. Several nuclear reactors are cooled by gas at high temperature and pressure [26–28], and taking into account the safety requirements in nuclear devices, this means that the use of gases as a coolant is an interesting idea to apply in the solar industry.

From the point of view of linear receivers, it is worth citing the activity of the PSA (Plataforma Solar de Almería) in relation to this topic [22,23,29]: an experimental facility of 350 kW (thermal) has been developed with two Eurotrough-II collector to test the capabilities of the carbon dioxide to cooling the receivers at relatively high pressure and temperature (Fig. 1) [22,23]. High temperature is a design decision to increase the efficiency of the cycle, but it conveys the need to demonstrate the capabilities of concentrators for reaching radiation intensities yielding temperatures above 500 °C, and high pressure (100 bar) is required to reduce the energy to move the CO_2 through the circuit [22]. The main disadvantages associated to this idea are the needs of higher pipes thicknesses and the leaks associated with mobile parts and flanged connections.

The possibilities of the technologies with moderate heat flux impinging in its receivers (trough and Fresnel collectors) can be found in the described analytical study of the following pages that has been carried out implementing the model in Engineering Equation Solver (EES) [31] already utilized in other related work [22]. Also, the direct coupling with a Brayton-cycle (Fig. 2) working in conditions near the critical point of the CO_2 will be described. This operation area in a Brayton cycle was partially studied in previous researches [24,25,32] showing interesting results for the cycle itself, and in this paper the results are specifically focused on its application in relation to the concentrated solar thermal energy, to linear thermal receiver as [22] troughs or Fresnel (lower temperatures than central receivers), an area the authors have not found developed from the state-of-the-art review carried out.

With these ideas, the presented work is developed through the following guideline: once the feasibility of CO_2 as heat transfer fluid is demonstrated, its application on CSP can be further studied (Section 1). The simpler electricity production with CO_2 as working fluid goes through Brayton cycles, whose behavior are studied near working conditions compatible with the already proven at CSP as Section 2 describes. Once the most adequate thermodynamic domain of CO_2 is selected, the direct coupling between CO_2 solar field with linear receiver and a Brayton cycle working at previously defined conditions can be studied (Section 3). Finally, main conclusions are resumed at Section 4.

2. Close-to-critical regenerative Brayton cycles

Available studies in relation with CO_2 Joule–Brayton cycle cover a wide range of pressures and temperatures, more recently with special attention to very high pressures, as the related ones with the nuclear field [28]. Other studies go to lower temperatures and pressures, but still with high temperatures and pressures and closer to the critical point [25,32–34], paying special attention to the thermal properties variation when through an isotherm line the pressure is varied. It can be seen in a pressure-enthalpy diagram like the one of Fig. 3, that the slope of the iso-temperature lines near the critic point is reduced until negligible values, looking almost horizontal.

Fig. 1. PSA test facility general view.

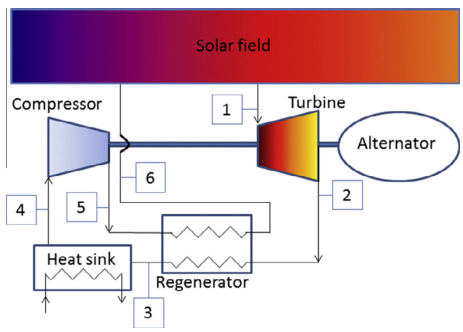

Fig. 2. Scheme of a Brayton cycle with regenerator.

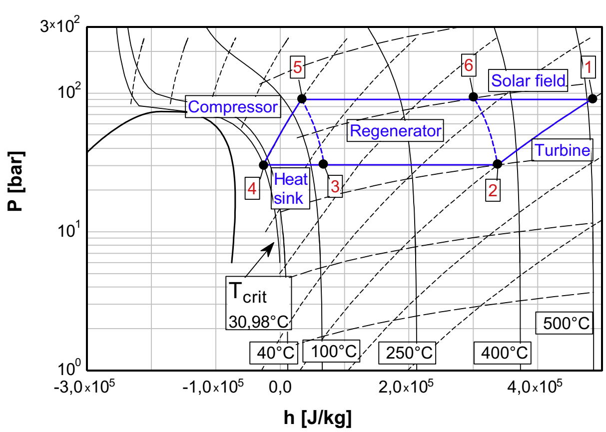

Fig. 3. CO_2 pressure-enthalpy diagram [35] with a close-to-critical Brayton cycle.

This tendency to 'almost horizontal' variation of iso-temperature lines vs. enthalpy drives the fluid specific heat value to relatively huge values as some works indicate [32–34] and can be seen in Fig. 4. It would be interesting to understand the behavior of thermodynamic cycles working in this area of the pressure-enthalpy diagram and compare its results with others possible cycles.

2.1. Sweeping the thermodynamic diagram of a real gas. The case of CO_2

From an extensive point of view, we can talk about five thermodynamic cycles (Fig. 5):

– Conventional Rankine cycle: the working fluid is pumped to a boiler where it evaporates and passes through a turbine and is finally condensed [36]. This one needs to condensate the working fluid, so in the case of CO_2, it is not adequate because of the cold sink temperature is the environment one.
– Rankine transcritical cycle: the working fluid goes through both subcritical and supercritical states [24,38,39]. As in the conventional Rankine cycles, the cold sink temperature makes it impossible with the proposed working conditions.
– Brayton cycle: the working fluid goes from high pressure and high temperature to low pressure and low temperature through a turbine, heat sink, compressor and heat source, without any phase change. Some alternatives could be found, the most representatives are:
  - Brayton supercritical: all the working fluid evolution through the cycle is over the critic pressure and temperature [34,37]
  - Brayton atmospheric: the low pressure value is the atmospheric pressure. Usually this type of cycle is open [40]. Taking into account the evolution of this cycle type, its use is more adequate where the hot source has higher temperature. The higher this temperature, the higher the slope difference between the iso-entropic lines related with compressor and turbine evolution for the same cold sink temperature, and for hence, it places the cycle in a thermodynamic diagram where the theoretical efficiency can be higher. As the solar linear receivers cannot increase enough the temperature, this configuration is not adequate with the couple CO_2 & solar linear receiver
  - Brayton near critical point: the working fluid low-temperature is near the critic temperature, and the low and high pressures of the cycle are near the critic pressure [24,25,28,34]. This cycle is especially interesting for solar applications, because the compressor thermodynamic evolution is close to an isentropic of badly compressible vapor with relatively low specific volumes; while the turbine expansion lay in a thermodynamic region where CO_2 behaves like an ideal gas and very high specific volumes. Note that the specific work of the cycle can be written (in a isentropic approach) as it is indicated in Eq. (1)

$$w\ (\mathrm{J/kg}) = \int_{P_{low}}^{P_{high}} (V_{s,tubine} - V_{s,compressor}) \cdot dP \tag{1}$$

In some cases, for these cycles the turbine outlet temperature is higher than the compressor/pump outlet temperature. Then, it is possible to take advantage of the turbine exit and reduce the heat source required through the addition to the system of a regenerator [40].

A comparative table can be made (Table 1) with characteristic values taking into account some limitations and conceptual simplifications of the most relevant cycle-types for this work:

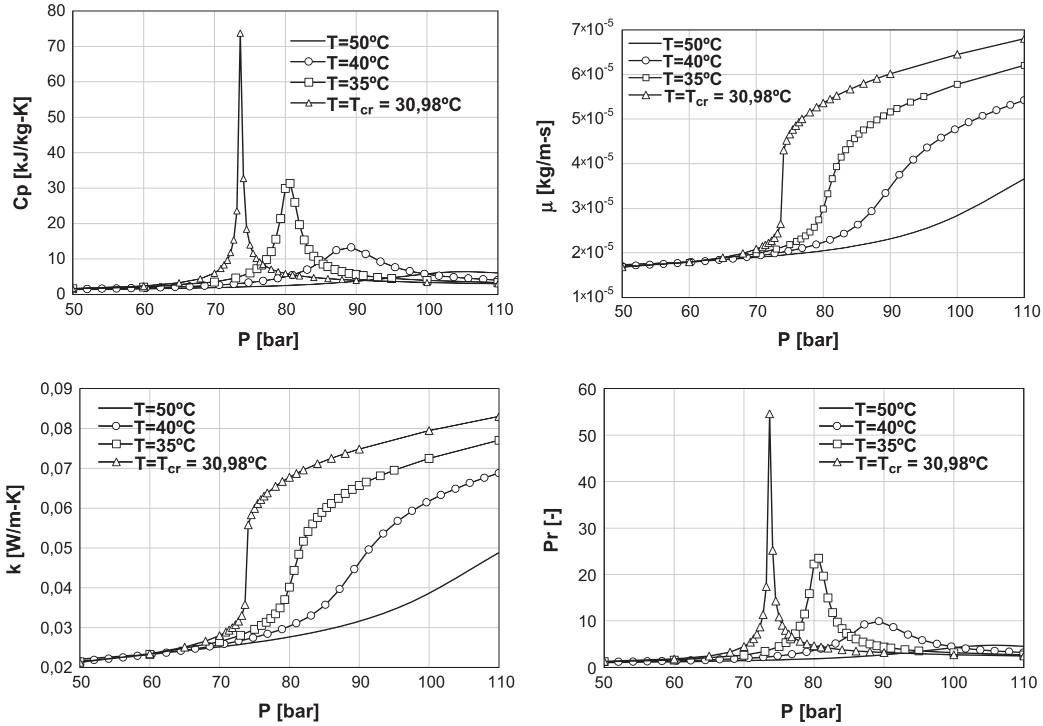

Fig. 4. Carbon dioxide main thermal properties variation near the critic point (T_{cr} = 30.98 °C, P_{cr} = 73.77 bar) [35]. The peak observed in Cp comes from a well-known property of the critical point, where enthalpy grows on the critical isobar without any changes in temperature.

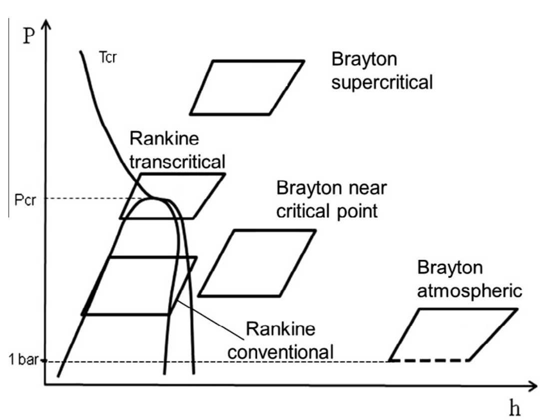

Fig. 5. Generic pressure-enthalpy fluid diagram to show the possible thermodynamic cycles to operate with.

Table 1
Indicative values for different identified types of Brayton thermodynamic cycles according to available bibliography.

Cycle type	Close-to-critical with regenerator	Supercritical with regenerator [28]
Turbine inlet T (°C)	500	500
Turbine inlet P (bar)	100	250
Tubine outlet P (bar)	50	75
Pressure ratio	2	3,32
Coldest cycle T (°C)	45	45
Cycle efficiency (%)	**33.3**	**33.4**
Indicative conditions	For solar thermal power plants	For nuclear power plants
Compressor politropic efficiency [24]	0.9	0.9
Turbine politropic efficiency [24]	0.9	0.9
Regenerator efficiency [24]	0.95	0.95
Alternator efficiency [20,24]	0.98	0.98
Mechanic efficiency [20,24]	0.987	0.987

– The studied cycles are totally different in the physics of its evolution from the heat source to the heat sink, this consideration in itself describes the limitation of comparing several different concepts.
– The features of each cycle drives to different turbo-machinery, then the inefficiencies related to each one will be considerably different, and also the size effect implies different behavior [41]: in order to obtain an adequate comparison, the author considers a special case found in available literature [28,34] in relation to the nuclear field, where the fluid thermal conditions are similar to the ones that can be found in the solar thermal energy field.
– Carbon dioxide has been chosen to obtain this table. Its election is related to the same criteria as pointed out in previous works [22,28].

In consequence to the results of Table 1, the reader can observe that the close-to-critical Brayton cycles drive the efficiency to values near supercritical Brayton cycles, but with considerably lower pressure values, that implies less mechanical requirements, mainly in the sense of pipe thickness. This pressure reduction also implies positive effects in the sense of heat transfer reducing the thermal resistance in the specific case of solar receivers, where the thermal resistance from the heat source to the fluid is a key point to improve the thermal efficiency [7,8].

2.2. The Brayton cycle: the need to improve efficiency

Nowadays, water/steam Rankine cycle is usually used in conventional thermal power plants and nuclear power plants and Brayton cycles are mainly used in gas combined cycle power plants. Also the Brayton cycle is used in another technological field such as aeronautics, due to its high specific power and thrust, quick response and, in general, to its great behavior in transient operation.

Gas combined cycle power plants and plane engines use open Brayton cycle, it is different from the closed Brayton cycle described on Fig. 2, so then, the transient behavior must be considerably different. This point is not the reason to choose a Brayton cycle to be coupled with the solar resource: the feasibility of Brayton when the application where it is used has transient heat source is the most relevant point to consider when coupling solar energy and a thermodynamic cycle.

These ideas must be taken into account when a designer tries to couple a thermodynamic cycle with concentrated solar plant for electricity production. There are two main aspects that characterize each one:

- Water/steam Rankine cycles gives adequate efficiency values even with low steam temperatures, that can be produced by the typical solar systems (250 °C for PS10 [42], near 380 °C with trough collector and HTF technology [42]).
- Brayton cycles can work with solar systems as several projects have demonstrated (for example Solgate project, [43]), but this cycle in its common design needs very high temperatures, and with solar systems, high temperatures produce very high thermal losses in the solar field (for trough collectors or Fresnel systems) or in the receiver (for central receivers).

If the options are analyzed in a deeper way:

- Water/steam Rankine cycles have the inconvenience of needing longer time to start-up than Brayton cycles [3], due to its inner working way.
- If we analyze Brayton cycle in other working conditions, for example, near the critical point (as Table 1 results show), it is possible to take advantage of the huge properties-values variations that appear in those conditions (Fig. 4) moving the cycle to be competitive with the typical Rankine yield values in solar power plants applications, and according to the inner possibility of Brayton cycles of unattended operation [44], it constitutes a really important cost reduction possibility for concentrated solar thermal power plants. The pressures and temperatures to be analyzed are the achievable ones taking into account the previous state-of-the-art already described, and can drive to values of 45 °C for the cold sink, and 400 °C or 500 °C for the solar field outlet temperature. On these values the variations between ideal gas (higher temperatures) and real gas (45 °C because of its proximity to the critical temperature) are relevant as can be seen on Fig. 6.

According to the thermo-physical properties of carbon dioxide in particular, and other gases in general, at environmental conditions (near of 20 °C and 1 bar), if this idea works directly with a thermodynamic cycle (without intermediate heat exchanger), it will be required to go to the close-to-supercritical region [24], the supercritical or at least between the two regions mentioned above and the atmospheric Brayton cycle, but always between Brayton or transcritical Rankine, depending on the fluid critical-temperature. For CO_2 the critical point corresponds to 31 °C and 71 bar (approximately). With these characteristic values, the temperature is very close to the typical environmental temperature. Since the heat rejection should take place at temperatures as low as possible, the critical temperature of CO_2 allows the design of Brayton cycles close to the critical point.

This consideration is really useful when the chosen heat source temperature is relatively lower than the usual values of the heat source of other applications of Brayton cycles, such as airplane reaction engines or combined cycle power plants. Then, in relation with this 'relatively low' heat source temperature, the heat sink temperature must be decreased as possible.

Both ideas drive the design to the region over the critical point, where lower achievable pressure is higher than 70 bar at these tem-

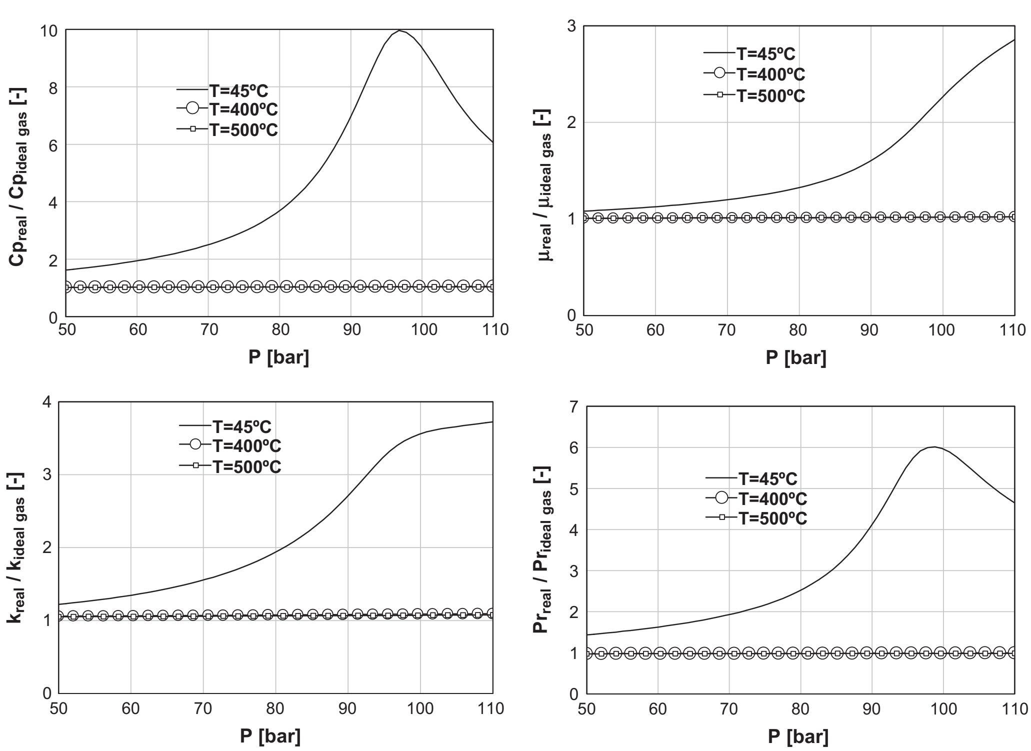

Fig. 6. Carbon dioxide main thermal properties ratio between real gas and ideal gas behavior at high temperatures (400 °C and 500 °C, similar behavior of real and ideal gas) and at temperature near the critical one (45 °C, when the critical temperature of CO_2 is 30.98 °C, approximately) a peak can be observed in Cp, that is maintained even 14 °C far from the critical temperature.

perature (that implies low pumping power [22]), and the higher pressure required in the cycle to obtain an adequate pumping power is near 100 bar. In this region the iso-temperature lines go near the horizontal slope that means very high specific heat values and then very high heat transfer with low temperature variation: higher heat transfer rate assuming fixed heat sink temperature.

According to this, it looks logical for the designer to go for the close-to-critical region to improve the efficiency of a Brayton cycle to be coupled with concentrated solar thermal receivers, technology that in linear receiver versions can reach temperatures near 500–550 °C [22], very low compared with the maximum temperatures that other technologies such as central receivers or solar dishes can reach.

2.3. A unique thermodynamic domain: close to the critical point

The previous paragraphs explain the necessity of going to this area of the thermodynamic diagram domain to couple concentrated solar thermal technology and the Brayton technology, characterized among other features with the possibility of unattended operation and a very quick response in start-ups and minor transients in open loops (aeronautics, conventional and combined cycle gas turbines).

Start-ups are an example Rankine cycles disadvantage coupled with solar energy, because every day the plant stops production at dusk until dawn. Even if the plant has thermal storage, this technology usually covers only some hours at night, except some new plants such as Gemasolar in Spain [45]. But these power plants have the necessity of oversize the thermal storage, selling electricity production at night, when electricity is cheaper and then the income is lower (very important point to take into account when the technology has not reached the conventional power plants electricity-prices).

With a model of the thermodynamic cycle composed of compressor, turbine, heat source and heat sink (even regenerator, if required), the thermal efficiency of the cycle could be evaluated as a close-to-critical cycle. It would be interesting to analyze the system behavior in a wide range of values to ensure the adequate working conditions that would be suitable for a solar thermal power plant.

Results for the thermodynamic cycle, assuming no pressure losses in any place of the loop are represented in Fig. 7 (cycle efficiency) and Fig. 8 (specific work). The required parameters to make the calculations, such as politropic efficiencies or regenerator and alternator conversion efficiencies are shown in Table 1.

In Fig. 7 a maximum value of efficiency can be seen for the three turbine inlet pressures ($P1$) considered. As can be expected for very low pressure ratio (ρ), defined as the ratio between turbine inlet pressure and turbine outlet pressure ($P2$), the efficiency decrease, and for relatively high values of pressure ratio the efficiency also decrease due to the effect that produces de politropic efficiency value on the turbo-machinery: the lost work during the compression and expansion processes grows with the pressure ratio [24]. This drives the efficiency to take a maximum cycle yield value below $\rho = 2$.

It is very important to point out that the slope of the curves change near pressure ratios of 7. At this point the turbine outlet temperature becomes very similar to the compressor outlet temperature and regenerator cannot be used for higher compression ratios. Therefore, this is a frontier that separates regenerative cycles (low pressure ratios) from non-regenerative ones (high pressure ratios).The heat exchanger called regenerator is characterized in terms of 'efficiency' defined in [40].

In Fig. 8 the variation of the cycle specific work (w) can be seen, where a similar tendency to the cycle efficiency appears: for low pressure ratios the specific work increases, but for high pressure

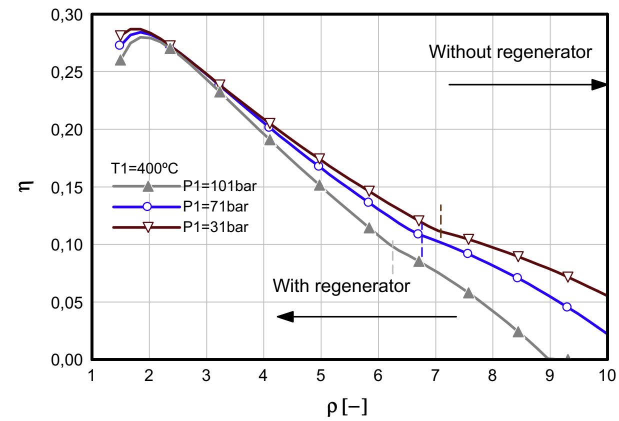

Fig. 7. Close-to-critical Brayton cycle behavior by means of thermo-electric conversion efficiency (η) vs. pressure ratio (ρ) for 400 °C of turbine inlet temperature and 45 °C of heat sink outlet temperature. Vertical-dashed line indicates the transition from configuration with regenerator to configuration without regenerator.

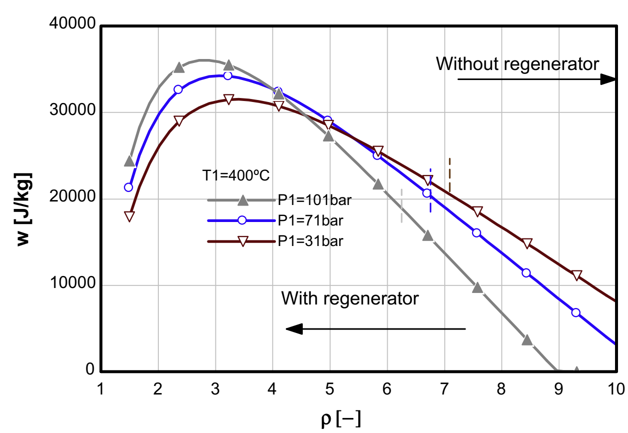

Fig. 8. Close-to-critical Brayton cycle behavior by means of specific work (w) vs. pressure ratio (ρ) for 400 °C of turbine inlet temperature and 45 °C of heat sink outlet temperature. Vertical-dashed line indicates the transition from configuration with regenerator to configuration without regenerator.

ratios decreases, appearing a maximum below $\rho = 3$. In this case, the maximum values of specific work are associated to the maximum turbine inlet pressure, but increasing the pressure ratio implies that lower turbine inlet pressure gives higher specific work: this is due to the relation between turbine and compressor behavior with its politropic efficiencies and the values of pressure considered for each studied value.

In order to make a decision about the adequate design conditions, it would be interesting to take into account that the efficiency is relevant, but also the cycle specific work because it is related to the size of the components. Then a figure of merit for a thermodynamic cycle can be constituted as the product of specific work and cycle efficiency vs. pressure ratio, as can be seen in Fig. 9. In this case the maximum values are displaced to the right of the cycle efficiency maximum and to the left from the maximum specific work (referring to the pressure ratio values). Depending on the final goal of the cycle, the maximum value found with Fig. 9 analysis could be the optimal.

From the results of Figs. 5–7 it can be deducted that the best efficiencies are in relation with lower pressures but the higher specific work is related with higher pressure values. This model does not consider transient effects, and for hence the transient (thermal inertia) effect of higher pipe thickness associated with higher working pressure.

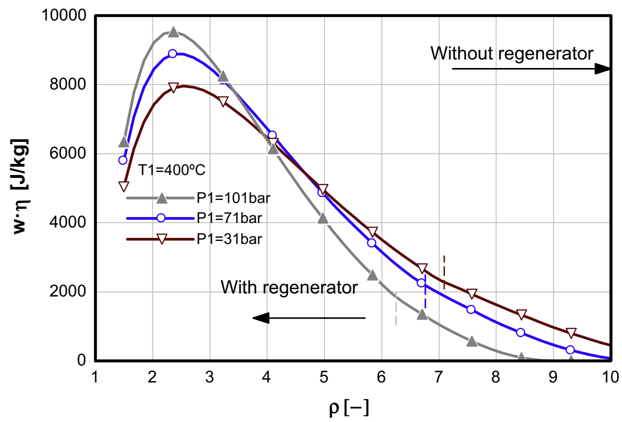

Fig. 9. Close-to-critical Brayton cycle behavior by means of the product of specific work (w) and efficiency (η) vs. pressure ratio (ρ) for 400 °C of turbine inlet temperature and 45 °C of heat sink outlet temperature. Vertical-dashed line indicates the transition from configuration with regenerator to configuration without regenerator.

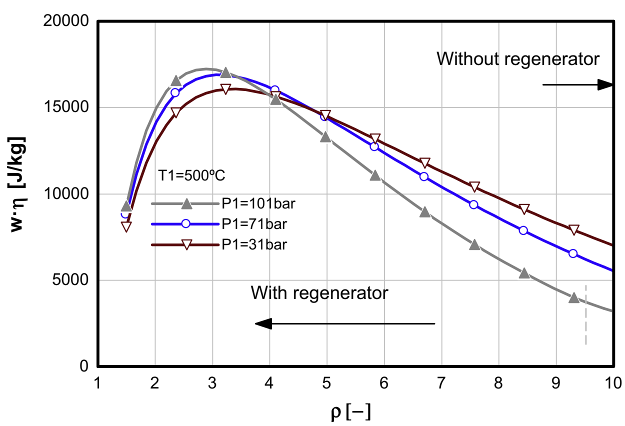

Fig. 10. Close-to-critical Brayton cycle behavior by means of the product of specific work (w) and efficiency (η) vs. pressure ratio (ρ) for 500 °C of turbine inlet temperature and 45 °C of heat sink outlet temperature. Vertical-dashed line indicates the transition from configuration with regenerator to configuration without regenerator.

In addition to the comparison example of Table 1, mainly according to the heat source temperature, and to the maximum temperature obtained in previous experiences with gas and concentrated solar thermal energy [22,23], similar variations can be found when the turbine inlet temperature is increased to 500 °C and the heat sink outlet temperature is reduced to 45 °C, obtaining higher efficiencies in relation with the higher temperature difference between heat source and heat sink. Fig. 10 (with $T1$ = 500 °C) shows the analogous to Fig. 9 (with $T1$ = 400 °C): the 'figure of merit' maintains the same tendency referring to the maximum values and the pressure which is obtained. The specific values of specific work and efficiency can be read for the maximum values on Table 2.

As can be read from described results: the lower the temperature the lower the need of a regenerator. This is related to the following idea: the higher the turbine inlet temperature the higher the turbine outlet temperature and then, if the heat sink is the same, it will be easy to obtain a turbine outlet temperature higher than the compressor outlet temperature (see Fig. 3).

3. Solar receiver: heating CO_2 up to Brayton cycle requirements

According to the previous promising results, look for the heating system of hot sink that can be coupled with this close-to-critical Brayton cycles is a relevant issue. As the introduction proposes, and point 2 is repeated several times, the main advantage of Brayton cycles vs. Rankine cycles is the good behavior in transients, simplicity and unattended operation. Nowadays the main thermodynamic cycle used in solar thermal power plants is Rankine, with the problems related with the daily start-ups and clouds transient effect on turbines during operation when the plant does not have thermal storage or an energy buffer.

In such scenarios the steam turbines suffer considerably, decreasing its useful life. In comparison with the usual behavior of gas turbines for open Brayton cycles, the most typical example is plane engines transients, and the engines response is quite good with a long operational life. According to this idea, the first application of this new thermal domain for Brayton cycles can fall into the field of solar thermal power plants, of course, with high preference according to the latest experiences using carbon dioxide as a coolant in trough collectors. To obtain general conclusions, the basis of the following analysis is the study of a linear receiver in the same way as previous works by some of the authors [7,8].

For this goal, it is required to analyze the thermal and hydraulic behavior of the receiver in different conditions. The selected frame considers fixed thermal energy along the linear receiver (E). This drives the variation coupling thermal flux (q'') and receiver surface (A) by $E = q'' \cdot A$.

As the considered receiver surface is a tube, it will be defined by two geometrical parameters: outer diameter (D_e) and length (L) and the area that receives the solar energy is $A = L \cdot D_e \cdot \pi$.

The material through which the thermal flux (q'') will pass should be defined to evaluate the tube thermal effect on the system: thermal conductivity and pipe thickness. The thermal conductivity (a typical one has been chosen: 20 W/m K) and the mechanical features, in the sense to define the pipe thickness (e) with the information available. A simplified method is provided by ANSI B31.1 [46] that can be used as Eq. (2).

$$e = \frac{(P_{int} - P_{ext}) \cdot D_e}{2 \cdot (SE + (P_{int} - P_{ext}) \cdot Y)} + SM = \frac{D_e - D_i}{2} \quad (2)$$

Table 2
Main values of the simulations for Brayton cycle with 500 °C of turbine inlet temperature and 45 °C of heat sink outlet temperature.

	P1 Bar	ρ –	P2 Bar	Regenerator? –	η –	w J/kg	$\eta \cdot w$ kJ/kg
Maximum η	101	2.19	46.04	Yes	0.3341	47,680	15.93
Maximum $\eta{\cdot}w$	101	2.89	34.98	Yes	0.3212	53,660	17.23
Maximum w	101	3.76	26.90	Yes	0.2932	55,320	16.22
Maximum η	71	2.02	35.14	Yes	0.3403	41,670	14.18
Maximum $\eta \cdot w$	71	3.06	23.19	Yes	0.3227	52,360	16.90
Maximum w	71	4.28	16.61	Yes	0.2870	54,750	15.72
Maximum η	31	2.02	15.34	Yes	0.3473	37,520	13.03
Maximum $\eta \cdot w$	31	3.41	9.10	Yes	0.3175	50,600	16.07
Maximum w	31	4.80	6.46	Yes	0.2795	52,880	14.78

where P_{int} is the pressure inside the tubes, P_{out} is the pressure outside the tubes, D_e is the outer diameter, D_i is the inner diameter, SE is the maximum admissible material stress at design temperature, Y is a coefficient related with the steel type and the temperature, SM thickness safety margin (erosion, corrosion, etc.).

For the considered stainless steel the chosen values appear in Table 3 and comes from the *Asme Code for Pressuring Piping* [46].

For the correct evaluation of the system 'linear receiver + Brayton cycle' the thermo-hydraulic model used has been applied by previous authors' works [7,8]. In these studies the receiver inlet temperature has a fixed value, but in a complete system, this temperature depends on the thermodynamic cycle features. This means that the two models must be coupled to reach the adequate evaluation of the whole system.

As has been described in previous results [7,8], the thermal flux impinging in the receiver value has a huge relevance in the performance values, the same as the mass flow that must be controlled by the regulation system in order to maintain the turbine inlet temperature. Another temperature to maintain is the outlet temperature of the working fluid.

The selected values for these simulations are 500 °C (turbine inlet temperature) and 45 °C (heat sink outlet temperature) in order to maintain the argumentation line of previous points and the basis of referenced works.

An important detail that in this case appears is that the compressor outlet pressure is different from the turbine inlet temperature due to the pressure drop that the fluid develops through the linear receiver. Its calculation can be evaluated by the usual expressions with friction factors [30] and which have been used previously by the authors [7,8,22].

Some opposite tendencies can be studied, the most important the turbine inlet temperature:

– Linear receiver: the higher the turbine inlet temperature (and for hence, the receiver mean temperature) the higher the receiver thermal losses, and then the lower the thermal efficiency.
– Brayton cycle: the higher the turbine inlet temperature the higher the efficiency and the specific work [24].

All these effects can be studied separately, to obtain concrete conclusions for each one. But then can also be studied with the effect that produces the variation of the geometrical and thermal load parameters of the receiver: length, diameter and receiver impinging thermal flux:

– Longer length can produce higher pressure drop, as smaller diameter. But smaller diameter increases the convective heat transfer coefficient (or film coefficient), and for hence, the thermal efficiency of the linear receiver.
– Longer length and bigger diameter contributes to lower film coefficients and lower pressure drops; then, another opposite tendency appears here, that can be studied in a simplified way by the ratio length/diameter.

To achieve ideas of an adequate design, one restriction can be made: fix the value of the surface where the solar radiation impinges per tube. This means: fixed the surface area per tube, and fixed the total solar energy available, by varying the concentrated solar thermal flux, means obtaining a variation with the receiver pipes number, obtaining a wide range of applications:

– Lower number of tubes means high thermal flux → central receiver.
– Higher number of tubes means low thermal flux → trough collector or even Fresnel receiver.

In relation to the total efficiency of the conversion 'solar concentrated thermal energy → electrical energy', the effect of the thermal flux impinging in the linear receiver can be seen in Fig. 11 for a set of given pressures. Then, the total efficiency mathematical definition corresponds with $\eta_{\text{total}} = \varepsilon \cdot \eta \cdot \eta_{\text{elec}}$. Where the coefficient related with the conversion from mechanical to electricity, alternator efficiency or η_{elec}, takes the value 0.98 (Table 1).

For lower ratios L/D_i (i.e. small linear receivers) it is recommended to use relatively 'high' values of thermal flux. For higher ratios L/D_i (i.e. large linear receivers) it is recommendable go to relatively 'low' radiation thermal flux. This is a coherent result with previous work [7]. The parameter selection L/D_i is related to the importance of this parameter with the pressure drop and the heat transfer in a pipe, and above all, related to the thermal characteristics of the fluid heating process, which follows the pattern given by Fig. 12, approximately.

It is considered almost constant ($\Delta T = T_{\text{tube}} - T_{\text{fluid}}$) which is valid for solar applications with constant radiation intensity, and we define the effectiveness (ε) of the collector in the way it is defined in heat exchanger theory (Eq. (3)).

$$\varepsilon = \frac{T_{fM} - Tt_{f0}}{T_{tM} - T_{f0}} \tag{3}$$

Therefore $\Delta T = T_{tM} - T_{fM} = (1 - \varepsilon)\cdot(T_{tM} - T_{f0})$.

Now the enthalpy balance in the fluid is equaled to the total heat transfer from the cycle (Eq. (4)):

$$\dot{m} \cdot Cp \cdot (T_{fM} - T_{f0}) = h_{in} \cdot A \cdot \Delta T \tag{4}$$

where $A = n \cdot \pi \cdot D \cdot L$, and also take into account Eq. (5):

$$\dot{m} = \rho \cdot v \cdot n \cdot D^2/4 \tag{5}$$

where n is the number of tubes of diameter D, and it holds the Eq. (6).

$$\rho \cdot v \cdot (D/4) \cdot Cp \cdot \varepsilon = h_{in} \cdot L \cdot (1 - \varepsilon) \tag{6}$$

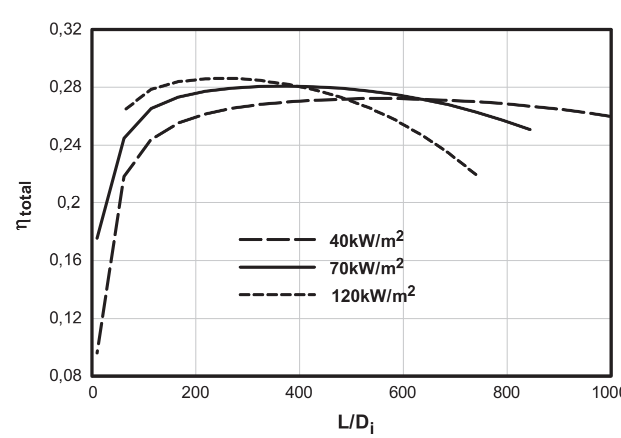

Fig. 11. Close-to-critical Brayton cycle coupled with linear receiver result for global efficiency vs. ratio length/diameter (L/D_i) for different thermal flux impinging in the receiver. It is considered for this result that the turbine inlet temperature is 500 °C and the heat sink outlet temperature is 45 °C. Turbine inlet pressure is 70 bar and turbine outlet pressure is 30 bar.

Table 3
Selected values for the pipe thickness evaluation.

SE	14.9 kpsi
Y	0.4
SM	0.001 m

Eq. (6) can be expressed with more physical sense as Eq. (7) shows.

$$\frac{L}{D} = \frac{\rho \cdot v \cdot Cp \cdot \varepsilon}{h_{in} \cdot (1 - \varepsilon)} \tag{7}$$

The aspect ratio (L/D) condenses all the information of the fluid heating process and it can also be related to some variables, if the rest remains constant. In particular, if all physical variables are kept fixed, the former equation becomes simpler (Eq. (8)).

$$\frac{L}{D} = F \cdot \frac{\varepsilon}{(1 - \varepsilon)} \tag{8}$$

where factor F is defined as can be seen in Eq. (9).

$$F = \frac{\rho \cdot v \cdot Cp}{h_{in}} \tag{9}$$

Of course, the effectiveness can be written as Eq. (10)

$$\varepsilon = \frac{L/D}{F + L/D} \tag{10}$$

As can be seen in Fig. 11, each thermal flux corresponds to maximum efficiency value in a more adequate set of values of the parameter L/D_i. It could be interesting to pay attention to the variation of the associated values of impinging thermal flux in the receiver and the L/D_i with the maximum global efficiency. In Fig. 13 this result is represented, as the reader can expect, the global efficiency increase with the impinging thermal flux with two tendencies: dramatic increase for low thermal flux values and low slope for high thermal flux values, following the tendency previously detected in other works [7,8]. Also, the optimal L/D_i ratio reduces its value with higher thermal fluxes.

It can be seen in Fig. 13 that going from 25 kW/m^2 to 100 kW/m^2 only implies an increase of efficiency of 10% (from 0.26 to 0.285). Of course this is an important fact, because the size of the solar field is inversely proportional to the cycle efficiency, for a given type of solar field. However, the efficiency of the receiver decreases dramatically when increasing the tube temperature, and thus is the main problem for very high radiation intensities. Note that approximately, the intensity I (W/m^2) is $I = h_{in} \cdot \Delta T$.

For gases, h_{in} will always be below 1000 W/m^2 K and 500 W/m^2 K would be a more acceptable limit. This means that for 100,000 W/m^2, the temperature jump between the gas and the tube would be around 200 °C. If the gas is at 500 °C, the tube would be at 700 °C, that will not only be a problem because of the increase of thermal losses, but a severe risk for the integrity of the tube.

It is interesting to take into account that one of the multiple degrees of freedom that appear in this system is one of the main parameters of the Brayton cycle: the pressure ratio (ρ). Some consequences of choosing different values for this variable are:

- The higher the pressure ratio, the more complex turbo-machinery. For hence, more expensive the cycle.
- The lower the pressure ratio, the cheaper the cycle, but lower values for the cycle efficiency and cycle specific work (see, i.e., Figs. 8 and 9).

The adequate selection of this parameter will play an important role in the final features of the entire system. The effect of the pressure ratio variation for a determined case can be seen in Fig. 14, appearing a maximum value of the global conversion efficiency more defined.

To do the calculations with a turbine inlet pressure of 70 bar it would be logical according to the results referred in [22], but turbine inlet pressure value is fundamental taking into account the consequences in the linear receiver design previously commented. Then, the analysis of this variable is already done and presented in Fig. 15.

As can be seen, the global efficiency variation is approximately flat when the turbine inlet pressure value is varied due to the opposite effects commented above: increase the pressure improves the cycle features, but penalize the thermal behavior of the linear

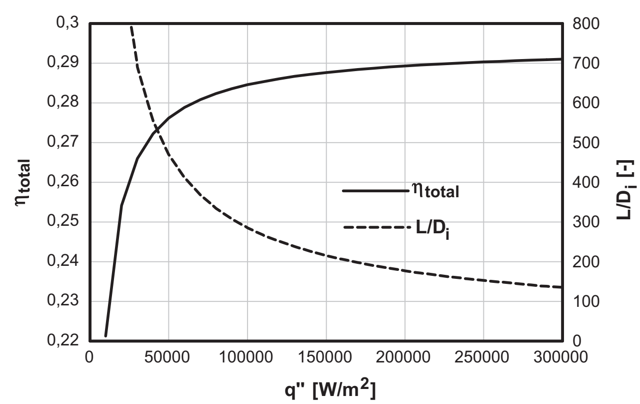

Fig. 13. Close-to-critical Brayton cycle coupled with linear receiver result for optimal global efficiency vs. thermal flux impinging in the receiver, and in the secondary 'y' axis, L/D ratio vs. thermal flux impinging in the receiver. It is considered for this result that the turbine inlet temperature is 500 °C and the heat sink outlet temperature is 45 °C. Turbine inlet pressure is 70 bar and turbine outlet pressure is 30 bar.

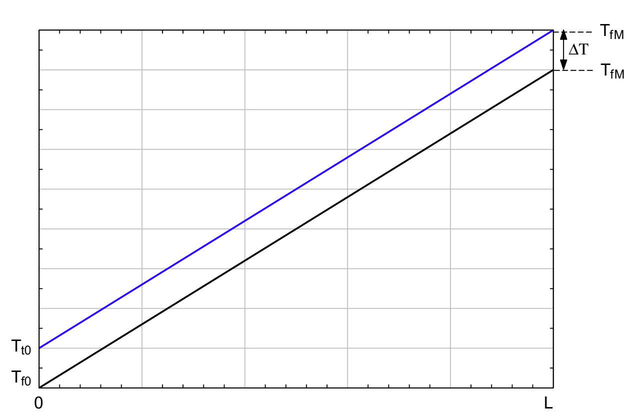

Fig. 12. Internal surface and fluid temperature vs. length through a pipe (approximated representation).

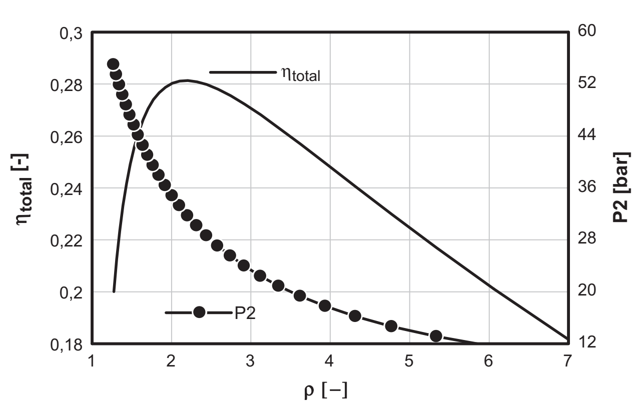

Fig. 14. Close-to-critical Brayton cycle coupled with linear receiver result for optimal global efficiency and turbine outlet pressure vs. pressure ratio. It is considered for this result that the turbine inlet temperature is 500 °C and the heat sink outlet temperature is 45 °C. Turbine inlet pressure is 70 bar.

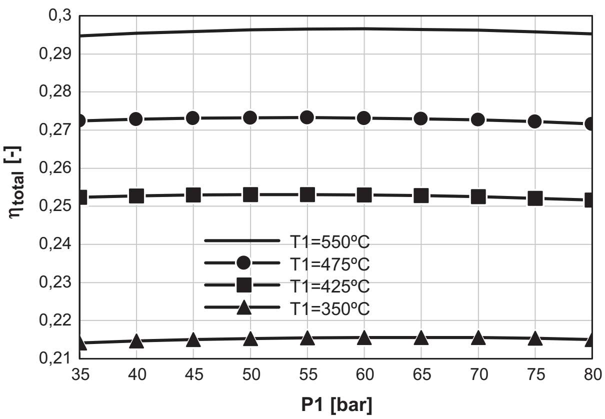

Fig. 15. Close-to-critical Brayton cycle coupled with linear receiver result for optimal global efficiency vs. turbine inlet pressure. It is considered for this result that outlet temperature is 45 °C.

receiver increasing the thermal resistance through pipe thickness. This means that for the whole system the pressure is not a key value for the system performance: the decision of the working pressure can be adapted to the cycle and linear receiver manufacturer lines.

4. Conclusions: overall results for CSP plants using CO_2 – pushing down the learning curve

The present work has studied the possibilities of the use of gas cooling linear receivers with high temperature and high pressure coupled with a close-to-critical Brayton cycle. The main features of this coupling has been explored and analyzed in order to obtain the maximum efficiency achievable. Results provide interesting possibilities for the coupling 'linear receiver – Brayton cycle' in the sense of efficiency, and taking into account the advantages in transient operation start-ups and unattended operation of Brayton cycles comparing to Rankine cycles, the studied configuration in this paper appear as a very interesting option:

- Rankine cycle is characterized by an important thermal inertia due to the phase change that complicates the start-ups in power plants [3], mainly when the start-up is produced every morning, as in the case of solar thermal technology. Also, the high pressure ratio that implies mechanical stress complexity.
- Brayton cycle turbo-machinery is characterized for its long life in transient processes scenarios, as can be seen in plane engines; and also, the obtained pressure ratios for this coupling with concentrated solar energy is considerably lower ($\rho \sim 2$–3), that means lower cost. The main properties of this considered cycle is that the compressor works with vapor near the critic point area and the turbine in the pressure-enthalpy area of the diagram where the fluid behavior is very close to the ideal gas, but it needs a regenerator.
- The use of gas in solar field has a main advantage with thermal-oil: there is no temperature limitation [3], and then it is possible to increase the temperature and then the cycle efficiency according to the possibilities of the considered solar receiver, for this case: a linear one [35].
- The total performance of a plant has a very important dependency to the thermal flux intensity in the receiver (Fig. 13). It must be taken into account that fluids with limitations in the film coefficient (h) must be in relation with the tubes limitation in the temperature differences, and for hence, in the values of the impinging thermal heat flux. Here appears the parameter L/D as a key point in the design.

Then, this technology adapted to the solar field can produce a very important effect in comparison with Rankine cycles: less risk to the turbo-machinery integrity in relation to the changes in the working conditions due to the solar radiation behavior. Of course, some economic aspects appear in relation to this effect: the maintenance budget can be reduced and the durability of the turbo-machinery is improved.

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