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A new conversion section for Parabolic Trough - Concentrated Solar Power (CSP-PT) plants

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Abstract

One of the most important challenges facing our future is the balance between energy needs and production, in the framework of the CO_2 commitments almost universally adopted. The total energy consumption is still a prerogative of fossil fuels, with a share close to 90 %, [1]; renewable energy, apart from the energy production from hydro, photovoltaic, biomass, waste and others, counts 3-4 % since many years ago. This discouraging result calls for new conversion technologies based on renewables, if the concept of sustainability is really adopted.
Concentrated Solar Power (CSP) plants technology could make the difference with respect to the other renewable technologies, thanks to "hybridity" in combining the concentrated solar energy source and the conventional power generation (actually steam turbine plants as energy conversion section). In the sector of energy production, parabolic trough (PT) type is more promising.
Recently, the Authors showed in [2,3] how convenient could be the utilization of gases as Heat Transfer Fluid (HTF) with advantages from a technological point of view in the heat collector section and, mainly, from the conversion section point of view, having the possibility to use gas turbines in which the HTF directly expands.
In this work, the Authors discuss some thermodynamic and engineering aspects concerning the use of gases as HTF, limiting the attention to air and CO_2 and they further discuss the performances of an innovative gas turbine power plant. It is based on a sequence of compressions and expansions, intercooled and reheated (inside linear solar receivers) respectively, in order to increase cycle specific work and efficiency. The paper focuses the attention on the optimum number of compressions and expansions: when it changes, pressure levels change too, requiring a series of reheating processes which operate in parallel, so increasing the overall solar receiver length and, definitively, investment costs. The optimization has been done adopting as design parameter the power per unit of collector length [kW/m], which is the most sensible parameter defining investment cost.

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Keywords - Solar Energy, Concentrated Solar Power, Parabolic-trough, Heat transfer fluid, Gas turbine.

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1. Introduction

A key challenge for the future is the balance between energy demand and production. It is known that almost 90 % of overall energy consumption is based on fossil fuels. This is in contrast with CO_2 commitments universally proposed in today's rule. Renewable energy sources apart from hydro-electric, biomass and waste, cover 3-4 %. In [4], IPCC invites to limit at 450 ppm the maximum CO_2 concentration in atmosphere, and the actual concentration is equal to 395 ppm [5]. A new energy era must be supported and it can be only based on renewables if the concept of sustainability would be conceptually adopted.

The CSP technology offers the lowest cost option for large, utility-scale solar energy today, with expected un-incentivized production costs for early commercial plants sited in locations with premium solar resources. As the cost of electricity from conventional generation technologies continues to rise, off takers are becoming increasingly interested in CSP as a viable alternative to other renewable technology options. Concerns over global warming, as mentioned before, have further increased this interest. As a matter of fact, large solar development plants and projects, such as the Desertec initiative [6] and the Mediterranean Solar Plan [7], which consider the installation of CSP plants in the Mediterranean area, are currently under discussion. Among these technologies, the parabolic trough type are the most commonly used in commercial plants and their performance continuing to attract energy sectors worldwide [8]. Unfortunately, actual CSP plants (mainly of parabolic trough type) suffer of many drawbacks: reliability, high costs, financing problems, use of pollutant or critical Heat Transfer Fluid (HTF), temperature limits related to the most used HTF, pressure drops, technological aspects concerning the management of actual HTF, the complexity of thermal storage, the downstream conversion section and etc. A strong simplification is expected, in order to increase reliability, safety, decreasing investment and maintenance costs, so attracting attention of financial funds.

This could be done adopting simpler HTF like gases and new downstream energy conversion sections more simplified with respect to the actual ones. The use of gases is not new: in [9] a pilot plant is presented having CO_2 as HTF. Moreover, in [10] a gas set is compared. One should acknowledge that these studies remained at a very early stage: main reason, probably, is that the gas as HTF choice must be accompanied by new energy conversion sections. Main drawback of gases as HTF, in fact, is the much lower thermal capacity i.e. the capacity to transport heat and to store it inside reservoirs. This is due to the lower density and specific heat: concerning density, their behaviour can be partially recovered by increasing operating pressure (up to 100 bar) but operating parameters make this choice questionable. It is in the Author's opinion that the use of gases is really suitable and able to simplify the plant and contribute to increase the potential applications (allowing a decrease in plant size) but only if the downstream conversion section is simplified, eventually renouncing to the thermal storage. So, a wider system approach should be adopted.

A contribution in this direction has been offered by the Authors recently, [11]. The result has been a comprehensive mathematical model which receives as input solar radiation and all the fluid and plant characteristics. The model takes benefits from previous studies and privileges a global approach, [12]: all the design parameters of a CSP solar field can be setup and the HTF temperature increase can be calculated, considering thermal oil, molten salt, gases. Receiver thermal efficiency prediction is a key point when the receiver temperature is increased in order to favour higher HTF temperature. More recently [3,12], the Authors focused their attention to the energy conversion section suggesting the use of a direct expansion of the compressed gas inside turbines. In order to increase cycle (and plant) efficiency, the proposal considered a series of intercooled compressions in order to reach maximum pressure and, similarly, a series of multiple expansions, reheated inside solar receivers. For an increase in terms of efficiency, with a positive correspondence on limiting receiver length, a regeneration processes have been conceived, with a heat transfer that involves the fluid at the end of the expansion and immediately after the end of the compression.

The optimization of the expansion levels is the subject of this paper considering that it affects sensibly overall receiver length and cost, as well as plant performances.

In this paper, a maximum temperature for HTFs (air and CO_2) is fixed at 1023 K from previous studies as well as maximum operating pressure (set at 50 bar), [11,13] which represents a good compromise between plant operational aspects and cycle (and plant) conversion performances. The paper discusses the effects of the values of the intermediate pressure stages of expansions on: (1) overall plant efficiency and specific work; (2) overall receiver length. In fact, the maximum expansion work which is realized, from basic thermodynamics, when the expansion stages are equal, would lead to a plant in which the overall receiver length will significantly increase plant cost: consequently, reliability will decrease and make the required investment less attractive.

In order to investigate this aspect, a parameter representing the Power-to-receiver Length ratio (P/L) has been introduced and discussed as a function of the number of expansions and of the intermediate pressure ratios.

The power plant which follows from these considerations, when compared with conventional CSP-PT using thermal oil as HTF, opens new ways to a greater flexibility and to a number of not-only energetic applications.

2. CSP-PT plant layout and characteristic parameters

In a CSP plant an important element is the Heat Collector Element (HCE). This component allows the absorption of the solar radiation and its transfer to the Heat Transfer Fluid (HTF) thanks to the high absorbance of the metallic tube for the solar radiation (short wavelengths), combined with low values of the emissivity of the glass envelope in the region of long wavelengths (so in the temperature range in which the surface emits).

Recently Authors showed the use of gases (in particular air and CO_2) [9,10] as HTF in place of conventional ones (diathermic oils and molten salts), [14]. This choice has been demonstrated with advantages from a technological point of view [12,13]: if the gas is directly expanded inside turbines, the downstream energy conversion section is strongly simplified with respect to the conventional thermoelectric conversion which represents the actual technology. So, a conversion cycle which makes use of gases has been proposed: considering that maximum temperature allowable at the exit of receivers (expansion inlet) has to be limited and can't reach actual inlet temperatures of gas turbine plants, the conversion gas cycle must approaches, as reference, to an Ericsson's cycle instead of the more conventional Joule's type.

This reference cycle, as known, presents isothermal compression and expansion phases. From an engineering point of view, this could be done by a sequence of intercooled compressions and reheated expansions. A cooling medium is required and external air is the most suitable: it is reheated by solar energy inside CSP receivers (solar fields). As these transformations are done in a discrete way, the corresponding cycle of the CSP plant has been named Discrete Ericsson Cycle (DEC). Thanks to this setup, the overall plant efficiency has values close, and even greater, to the actual technologies (thermoelectric plants), in spite of the great simplicity, which means cost reduction and reliability increase.

Key elements for this plant are the HCE and the capability to represent the processes inside. Considering that the maximum gas temperature must be few hundreds degrees greater than actual HTF and that convective heat transfer coefficients are lower than the actual values, the performances of the receiver in terms of thermal efficiency are modified and tend to decrease with respect to actual values. So, thermal efficiency must be recalculated according to a receiver model which represents also relevant quantities of high engineering interest (metal temperature, convective heat transfer coefficient, thermal elongation, pressure losses and etc). The Authors in previous works, [2,3] presented this comprehensive modelling.

Thanks to this model, Figure 1 has been done. It reports the behaviour of a conventional receiver which operates at 1.5 bar, 5 bar, 10 bar, 20 bar and 50 bar. These pressure values are inside the typical range of those resulting from intermediate expansions, starting from a maximum operating pressure of 50 bar. Air and CO_2 have been chosen as HTF. Metal temperature and thermal efficiency have been chosen as most important variables. A unitary mass flow rate has been fixed at input.

Metal temperature inside the collector has a great importance: in fact, when a gas is considered as HTF, and the operating pressure decreases, the convective heat transfer coefficient decreases too, and so a greater temperature difference between the metal and the fluid is necessary to heat the fluid; this means that, for a given HTF maximum temperature, the metal temperature can significantly differ from the one of the gas. If the metal become hotter, it emits a greater part of the radiation that it receives from the Sun, increasing energy losses and decreasing the efficiency of the collector: this happens because the emission wavelength of the metal goes beyond the opacity limit of the glass envelope. This important effect (which calls for new manufacturing technologies of the metal tube and of the glass outer tube) is evident in Figure 1: when the operating pressure decreases (as it happens on those receivers which operate at lower pressures), metal temperature increases if maximum HTF temperature must be restored (1023 K) in order to start a new expansion. Solar receiver thermal efficiency dramatically decreases, much more lower that of the conventional behaviour which a mean efficiency at 0.7 - 0.75. This will produce a very weak heating of the HTF inside the receiver till to a situation in which all the energy received by the fluid is lost. In any case, a thermal efficiency decrease means a length increase of the overall solar receiver in a plant. Receivers which operate with CO_2, thanks to a slightly greater heat transfer coefficient, demonstrate a greater thermal efficiency.

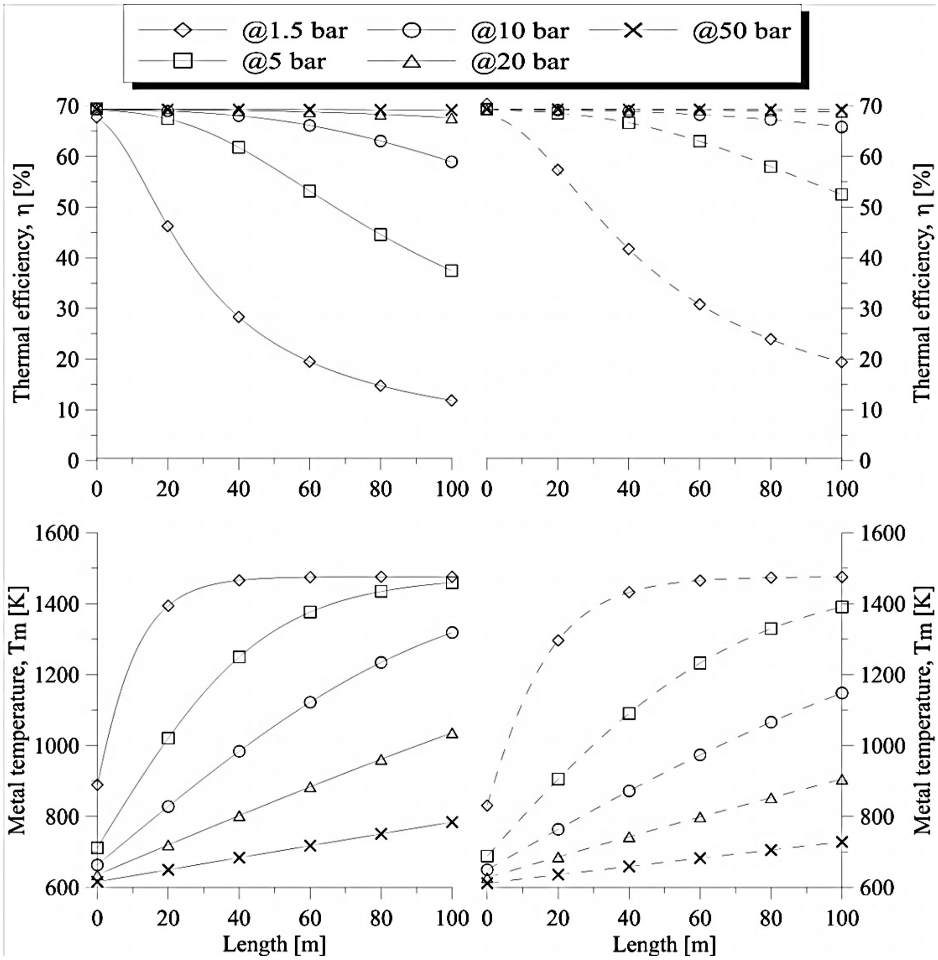

Fig. 1. Metal temperature and thermal efficiency of the PTC for different pressure for air (l) and CO_2 (r).

As the HTF goes through inside the receiver, temperature increases, pressure decreases and this leads to a lower density; a continuous speed increase is the direct consequence of all, in order to maintain the mass flow rate. An upper limit for speed must be fixed (25 m/s) to avoid pressure losses out of control, noise appearance, macroscopic vibrations and etc. When this limit is overpassed, the HTF has to be split in parallel branches, and the fluid continues to go through inside receivers restarting from an initial speed, obviously lower than the previous value. Considering that the different solar fields operate at very different initial pressure levels, being them placed downstream expansions, fluid medium speed increases towards low pressure values, having as a consequence a solar field in which several receiver branches operated in parallel.

Because of this, the greater dimensions of solar fields operates at lower pressure appear justified and evident, and so the real need of such longer receivers must be verified comparing the additional contribution produced by the subsequent low pressure expansions.

An example of a plant layout is represented in Figure 2, for 3 compressions and 3 expansions (3x3 plant): the HCEs operate at different pressure levels, so their performance are not the same and worsen as pressure decreases.

The mathematical model considers compressors and expanders as defined by their adiabatic isentropic efficiencies. The Figure shows that, in order to fulfill maximum speed limit inside receivers, three parallel branches must be operated at high pressure (50 bar, after the regeneration), six branches at an intermediate pressure level (13.5 bar) and twelve branches at low pressure (3.7 bar). In order to predict the performances of the plant, the receiver model has been integrated with the thermodynamic models of transformations. Model receiver accounts for pressure losses which decrease expansion work. So, the thermodynamic conditions – in terms of pressure, being maximum temperature always restored – at the outlet of the turbines change too. The flow rate split, moreover, is a discrete process and it is managed in a way that, when inside the receiver the HTF reaches the maximum allowed value, calculation on that receiver is restarted from the inlet considering a new branch in parallel which divide by a factor of two the flow rate, reinitializing the inlet speed. An iterative procedure has been set up which governs all these aspects and, definitively, makes use of the model receiver as predicting (verification) tool. Figure 2 shows also the thermodynamic cycle which is done inside the plant.

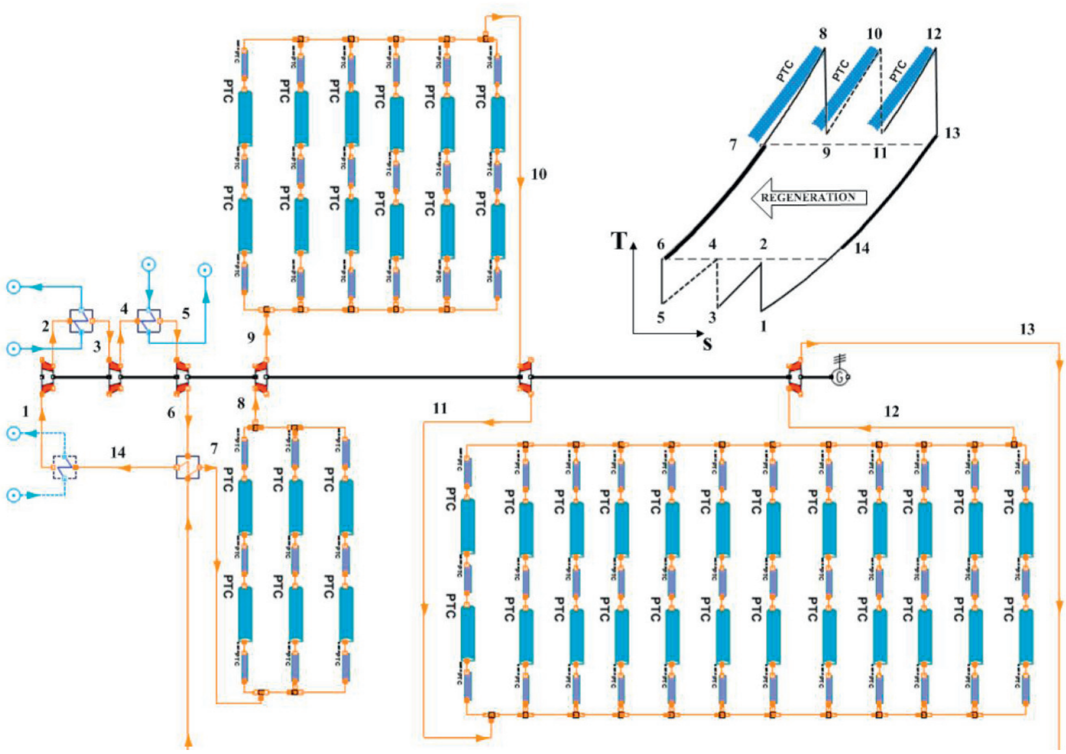

Fig. 2. 3x3 DEC cycle and plant layout.

Different plant arrangements have been analysed varying the number of compressions and expansions involved in the process. The aim was to evaluate the effect on specific work and plant efficiency as well as on overall receiver length which is proportional to plant cost.

It is known that increasing the number of compressions and expansions, the DEC approaches to a true Ericsson cycle which has, if completely regenerated, the highest efficiency allowable (Carnot) having fixed upper and lower temperatures. Varying number of compressions and expansions, the overall receiver length changes and an additional benefit in terms of performances must be verified in terms of additional receiver length increase. The analysis has been conducted starting from a 2x2 plant, up to a 5x5 plant, including combinations in which the number of compressions and expansions are not equal.

3. Expansions optimization

A first important evaluation can be done by analysing power and plant efficiency for a 3x3 plant. In Figure 3, for the cited arrangement, a maximum pressure of 50 bar and maximum temperature of 1023 K has been fixed, maintaining variable the lower pressure of the plant. Results are shown on the left axis for power and incoming heat, and on the right axis for cycle efficiency, while the two Figures are for air and CO_2.

In the Figure, the mass flow rate increase proportionally with base pressure (1 kg/s for 1 bar, 3 kg/s for 3 bar and 5 kg/s for 5 bar) and it shows that there is no direct proportionality between it and the power increase (dashed lines with symbols can be taken as a "proportional" reference, being roughly the product between mass flow rate and specific work). Pressure losses increase with mass flow rate and they cause that real power is always below the reference.

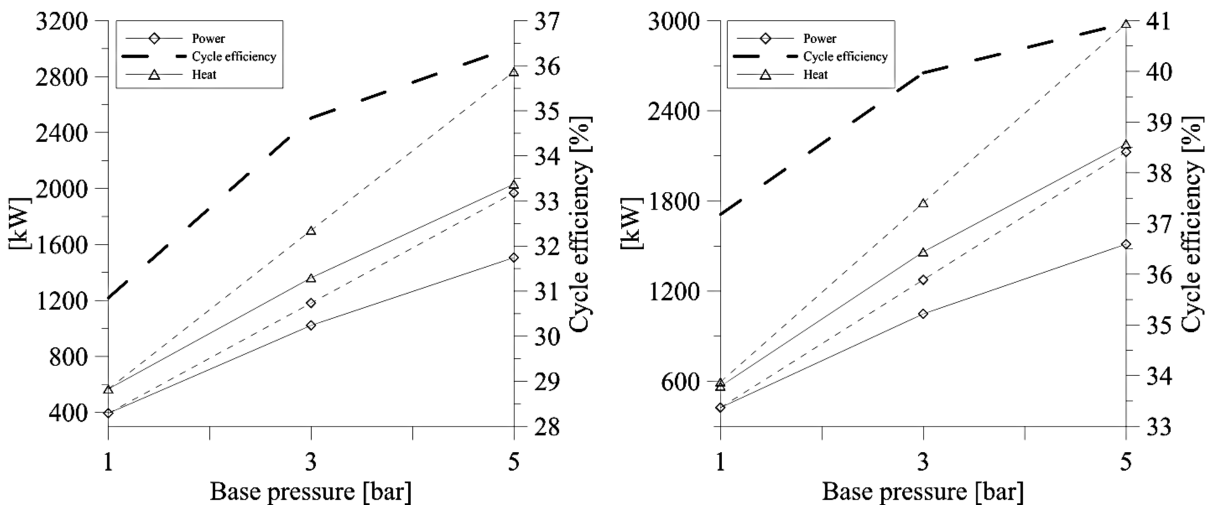

Fig. 3. Power, heat and cycle efficiency for air (l) and CO_2 (r) for 3x3 case.

Figure 4 reports a more general analysis in which number of compressions and expansions are varied in a range from 1 to 5, with all the possible combinations. The data refer always to a unitary HTF flow rate.

It can be noticed that, as predictable, the specific work decreases with the base pressure increase, becoming asymptotic after a certain value of the number of transformations involved in the cycle (Ericsson limit which approaches). However, plant efficiency increases as base pressure increases. In fact, if base pressure is increased, the pressure ratios (assumed equal each other) are lower (maximum pressure remains fixed): this leads to higher temperature at the collector inlet (after the regeneration). Higher inlet temperatures require lower heat fluxes received by the Sun. The work done on the expansions also decreases but the effect of the heat flux exchanged is greater, so plant efficiency increases. Additionally, the receivers operate at higher pressures and this increases thermal efficiency, decreasing overall receiver length. A higher base pressure allows to increase mass flow rate and, therefore, power, even if not in a proportional way.

Considering, for example, the 3x3 case, for air a specific work of 394 kJ/kg is obtained with a base pressure of 1 bar, a value which is reduced respectively of about 13 % and 23 % increasing base pressure at 3 and 5 bar (342 kJ/kg and 303 kJ/kg). Similarly, this happens for CO_2: the decrease is of 17 % and 28 %, starting from a value of 425 kJ/kg at 1 bar and reaching 351 kJ/kg at 3 bar and 303 kJ/kg at 5 bar.

An opposite behaviour is observed in cycle efficiency (solid lines): it increases varying base pressure towards higher values. This is due to what previously said: collectors that operates with higher temperatures at the inlet (consequences of lower values of the pressure ratios) need less heat to bring the fluid to 1023 K; if specific work decreases less than heat does, cycle efficiency increases.

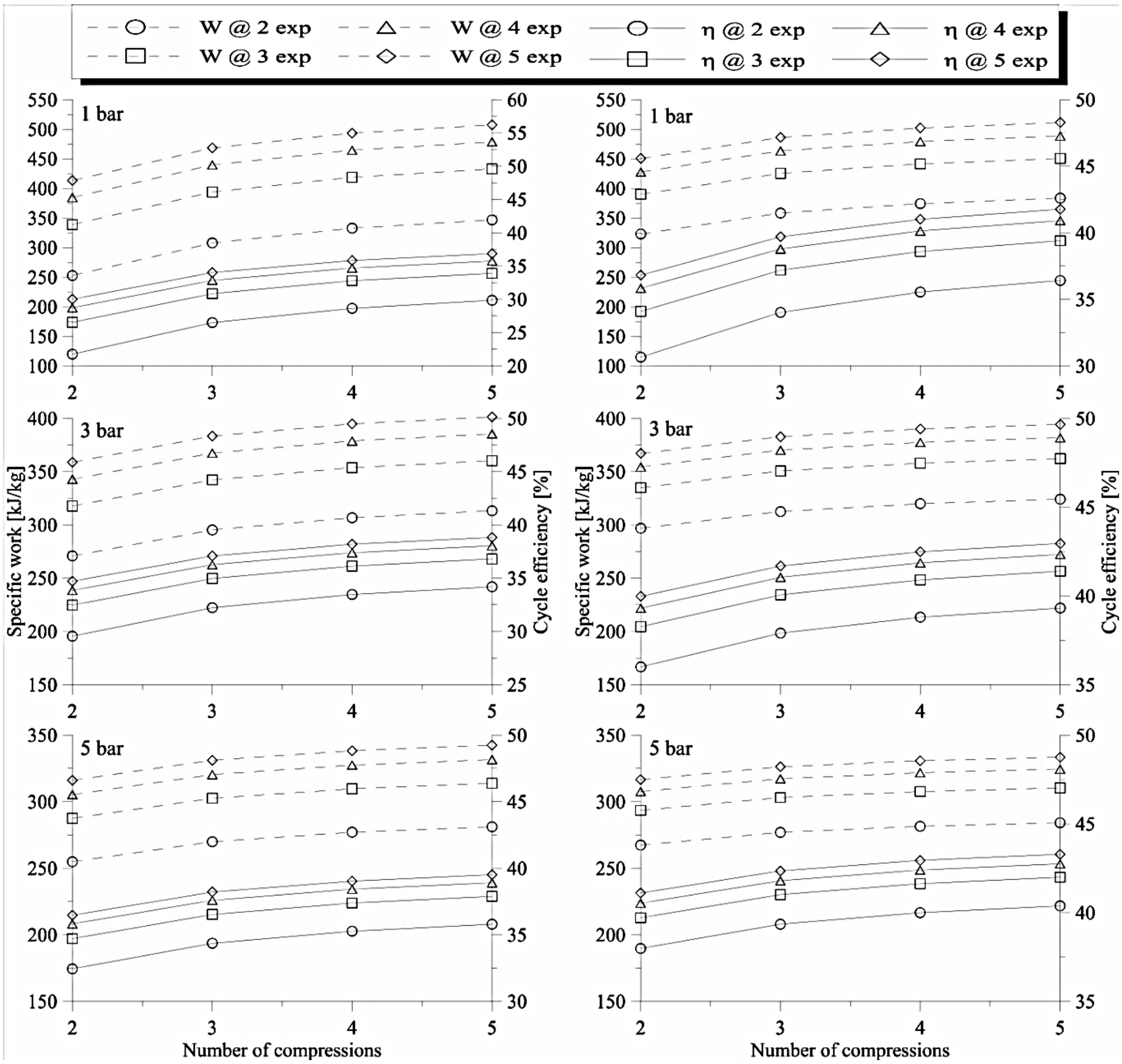

Fig. 4. Specific work (W) and cycle efficiency (η) for different combinations of expansions and compressions for air (l) and CO_2 (r).

As said before, when number of compression and expansion increase, specific work and efficiency tend to the values of the Ericsson's cycle. Considering expansions and compressions singularly, Figure 4 puts in evidence that expansions produce a beneficial effect greater than compressions. For instance, in the case 3x3 at 1bar, specific work is 394 kJ/kg for air and 425 kJ/kg for CO_2. Adding one compression to this case produces a specific work increase to 419 and 441 kJ/kg respectively (+ 6 % and + 3 %). If one expansion is added, specific work grows up to 440 and 463 kJ/kg, with an increase of 12 % and 9 % respectively. Changes in base pressure don't modify this behavior.

An increase in the number of expansions, which results in more specific work (and, therefore, power), produces an increase in the number of collectors and overall collector length.

For a base pressure equal to 1 bar, Figure 5 reports the change in collector length for each solar field when expansions number increases.

The lengths are grouped for each solar field: considering the 3x3 case (white bars, up to SF3), the first solar field as a total length of about 160 m, the second 170 m and the third more than 230, resulting in a total length of about 567 m. For CO_2 a total length of 596 m is needed.

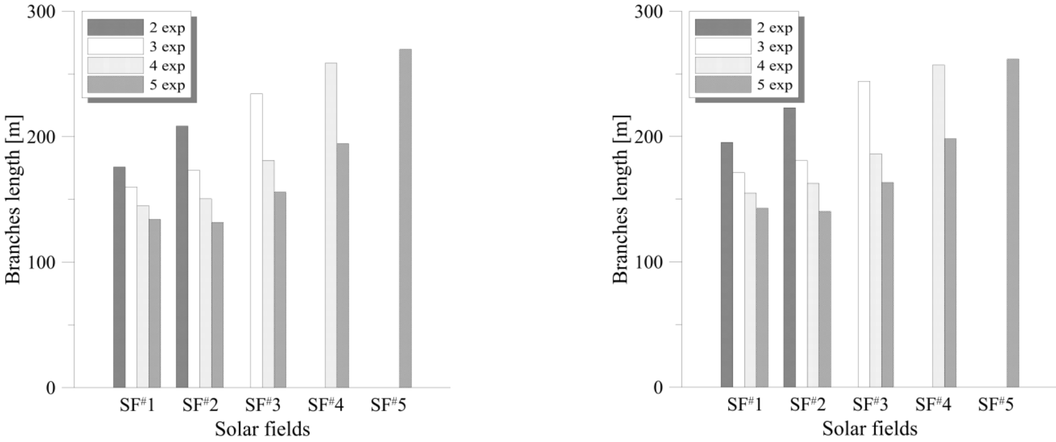

Fig. 5. Branches length [m] relative to the 1 bar case, for air (l) and CO_2 (r).

Table 1 shows overall receiver length as base pressure is changed for a unitary mass flow rate, keeping the same compression number (overall receiver length is practically independent from compressions number).

Table 1.Total collector length [m] per number of expansions and base pressure for air and CO_2 .

	2 exp		3 exp		4 exp		5 exp	
	Air	CO_2	Air	CO_2	Air	CO_2	Air	CO_2
1 bar	384	418	567	596	735	761	886	906
3 bar	341	370	463	496	570	596	664	678
5 bar	313	337	412	442	503	525	573	591

The total length of the collectors decreases when base pressure increases, and obviously depends when HTF is changed. In fact, for air with 3 expansions, the total length starts from 567 m at 1 bar base pressure reaching a value of 463 m at 3 bar, and 412 at 5 bar. Same behaviour, with different values (596, 496 and 442 m) , can be observed for CO_2.

When the number of expansions increases, the absolute receiver length changes and the lengths of the different solar field rearranges in percentage, in order to fulfil the reheating process among expansions. In those solar fields which operate at lower pressures, the condition that a maximum speed is reached inside receiver occurs more frequently and this requires a split of the HTF flow rate, as said before. Collector total length decreases as base pressure increases, when a unitary HTF flow rate is considered. When flow rate increases, due to the base pressure, total collector length increases but remains lower than the flow rate factor. This means that there is no direct proportionality between mass flow rate increase and collector total length increase, as seen also with power.

From an economical point of view, a decrease of the power reflects prominently on the economic upturn of the plant, while a decrease of the collector length reflects positively (and sensibly) on the money investment.

With these premises, power per unit receiver length ratio, (P/L), assumes an economic key role for these CSP plants, and so a correct individuation of the best set of plant parameters is fundamental for a further grow in importance of this proposal. The values of this parameter have been calculated in kW/m and reported in the Table 2.

Table 2. Power-to-Length ratio [kW/m] relative to different pressure cases, for air (l) and CO_2 (r).

air 1bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.66	0.60	0.52	0.47
	3	0.80	0.69	0.60	0.53
	4	0.87	0.74	0.63	0.56
	5	0.90	0.76	0.65	0.57

CO_2 1bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.77	0.65	0.56	0.50
	3	0.85	0.71	0.61	0.54
	4	0.89	0.74	0.63	0.55
	5	0.91	0.75	0.64	0.56

air 3bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.79	0.69	0.60	0.54
	3	0.86	0.74	0.65	0.58
	4	0.90	0.76	0.67	0.59
	5	0.92	0.78	0.68	0.60

CO_2 3bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.80	0.68	0.60	0.54
	3	0.85	0.71	0.62	0.56
	4	0.87	0.72	0.63	0.58
	5	0.88	0.73	0.64	0.58

air 5bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.81	0.70	0.61	0.55
	3	0.86	0.73	0.64	0.58
	4	0.89	0.75	0.65	0.59
	5	0.90	0.76	0.66	0.60

CO_2 5bar@750 °C		# exp			
		2	3	4	5
# comp	2	0.79	0.66	0.59	0.54
	3	0.82	0.69	0.60	0.55
	4	0.84	0.70	0.61	0.56
	5	0.84	0.70	0.62	0.56

Generally, values for CO_2 are higher than the corresponding ones for air, but it has to be observed that this ratio is referred to a unitary mass flow rate. Authors noticed in [13] that the condition of having the same speed at the collector inlet (between air and CO_2) means that carbon dioxide mass flow rate is about 43 % greater than air.

Table 2 shows interesting results. For a given base pressure, having fixed the number of expansions, P/L increases with the number of compressions, due to the overall more efficient compression, which increases the net power produced by the plant; collector length, moreover, depends mainly from the number of expansions. The overall P/L ratio has a benefit from this improvement and this applies to all the base pressures considered.

A last consideration refers to a further aspect. All the cycle arrangements presented before, have been set up considering the same compression ratio for compressors and the same expansion ratio for expanders. This means, in a rough way, that all the pressure ratios agree with the Equation 1, where n represents the number of machines considered for the transformation, either compression or expansion.

$$\beta_{\text{stage}} = \sqrt[n]{\beta_{\text{tot}}} = \sqrt[n]{\frac{p_{\max}}{p_{\min}}} \qquad \text{(Eq. 1)}$$

Having pressure levels of the cycle a different influence on power and length, expansion pressure ratios have been modified, while the compression pressure ratios have been maintained the same value and equal each other.

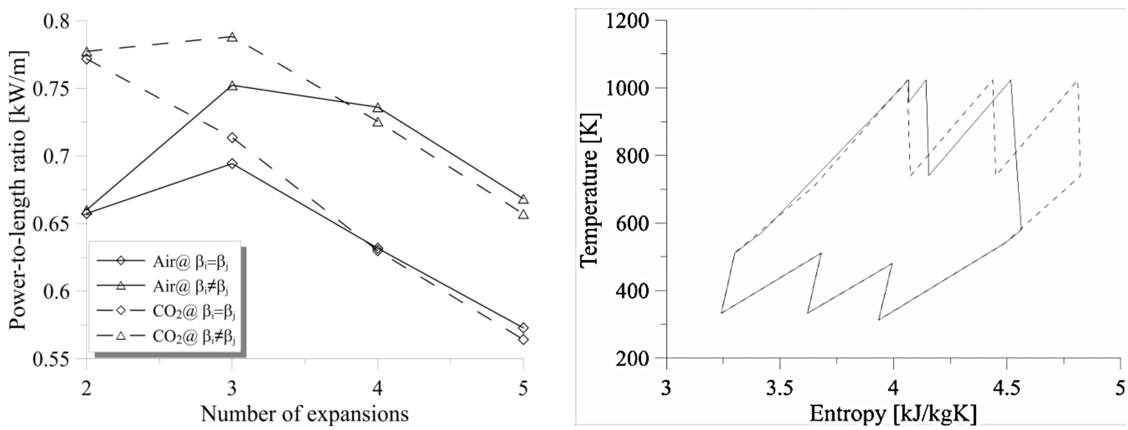

Fig. 6. Power-to-Length ratio with same and different β for air and CO_2 (l) and DEC cycle with different β for air (r).

This condition doesn't guarantee maximum useful work but it has an impact on the total collector length. A trade-off between performance improvement and overall collector length has a useful engineering importance. This analysis has been preliminary developed examining all the possibilities (in a discrete way), and the results of such optimization are represented in Figure 6 (l) in terms of P/L ratio. Number of compressions was always kept equal to the number of expansions. The transformation with equal pressure ratios is represented in dashed lines, while the solid lines represent the diagram as resulting from the optimization of the parameter P/L. The reduction of the specific work, which passes from 394 kJ/kg to 325 kJ/kg (-17.5 %) is less important than the reduction of the collector length that results from the optimization, which passes from 567 m to 431 m (-24 %), so P/L ratio increases of about 16 %. The three optimum expansion ratios are of about 1.3, 3.7 and 10.4.

4. Conclusions

CSP-PT plants have a great potential in the mean and long term and, among the other renewable based energy conversion technologies, they appear able to produce a step change ahead. Actual CSP-PT technology are conceived thinking to diathermic oil (and more recently molten salt) as heat carrying fluid; they have a thermal storage and a downstream energy conversion section based on a conventional thermoelectric power plants. These features represent themselves some intrinsic advantages of CSP-PT (matching the solar energy capture with conventional conversion technologies). On the other hand, the same features surprisingly represent important weak points from an industrial point of view: the heat carrying fluids (even though characterized by very high thermal capacity) are, when using oil, pollutants and unsafe; when using molten salts, they are not suitable with standard materials, they solidify at a given high temperature, they introduce leakage problems in between the connecting elements, they require a management of the plant complex and, first of all, not at all fail safe. The use of gas as heat carrying fluid would simplify the plants and calls for new energy conversion sections always making profit from actual fossil fuels technologies. The paper discusses the use of air and CO_2 and goes deep, with respect to previous works, into a new energy conversion technology in which the fluids expand directly into turbines, after being compressed and heated inside solar receivers. Being maximum temperature limited at 1023 K, the plant has a series of intercooled compressions and reheated expansions thanks to the passage inside solar receiver. From an industrial point of view, power produced to receiver length ratio has been assumed as most relevant parameter to reduce cost and increase reliability. Authors further investigate this parameter in order to find an optimum situation and varied relevant design plant choices. The analysis have been conducted varying wisely the parameters that influence both specific work and collectors length: number of compressions and expansions could not exceed three stages and, changing the expansions ratios (instead of having a constant ratio which would imply maximum work), an increase of P/L ratio about 16% has been found.

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