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ABSTRACT

A modified recuperative supercritical CO₂ cycle for DEMO fusion reactor is analysed. A layout with just one recuperator is used, because of the relatively low temperature of the breeding blanket coolants. The proposed cycle is able to integrate several thermal sources with different temperature ranges and includes two kinds of thermal energy storage to overcome the pulse operation: a two-tank molten salt storage and a concrete storage. The design point is obtained using the Engineering Equation Solver (EES) environment, which allows to use pure substance models for CO₂, helium and water. Once the design point is calculated, the sizing of pipes, heat exchangers, turbomachines and energy storage systems has been performed in order to obtain a preliminary estimation of the layout and its complexity. So, the power conversion system has been split into four equal parts in order to use commercial pipe diameters. Even though Printed Circuit Heat Exchangers (PCHE) have been used, extremely large sizes have been obtained, especially for the HTR recuperator. These sizes contrast with the small values obtained for the turbomachines, whose preliminary design has been carried out, using the AXIAL™ meanline design software. In addition, the two largest heat exchangers are divided into serial and parallel configurations when assessing the allocation of the components of the power conversion system to reduce the footprint. An average electric efficiency (including pulse and dwell period) of 28.25% has been reached. The required mass of molten salt per each power conversion subsystem is 2,568 tonnes, while the mass of concrete is 833 tonnes. Regarding the footprint, 85 x 170 m² are required for each power conversion subsystem, including the storage systems.

KEYWORDS: Balance Of Plant, fusion power, supercritical CO₂ Brayton cycle, DCLL, DEMO

1. Introduction

Power conversion system is a key issue in the future DEMO fusion reactor power plant. Besides the scientific and technological challenges involved in DEMO, the existence of several thermal sources to feed the power cycle results in a concern which requires a tailored solution to the cycle characteristics. The medium temperature (300°C/500°C) found in the blanket allows the use of other power cycles different than the classical steam Rankine. So, Medrano et al. [1] compare steam Rankine power cycles with supercritical CO₂ (S-CO₂) for a helium-cooled based DEMO fusion reactor. The reactor analysed (Helium Cooled Lithium Lead, HCLL, model AB) is a clear example of the problem related to the different temperature ranges of the sources, being the blanket at 300°C/500°C and the divertor at 540°C/717°C. Authors work out the problem with separate S-CO₂ power cycles for each source.

The S-CO₂ power cycle is a Brayton power cycle using CO₂ as working fluid. The cycle operates between two pressures, both supercritical. The key idea of this cycle is that the compression process occurs in the vicinity of the critical point, which results in a consumption power lower than in a classical Brayton cycle, using ideal gas as working fluid. This reduction in the compression work enables the S-CO₂ power cycle to work with a medium temperature range in the thermal source, achieving good efficiencies in the range of 300°C/500°C. As the turbine exit temperature is still high (the enthalpy drop in the turbine is low) an efficient thermal recovery process is possible, although the strong variation of the CO₂ specific heat with pressure makes the process rather complicated. That is, the specific heat of CO₂ increases with pressure at medium-low temperatures (up to 200°C), while that increase is smaller at higher temperatures. In order to overcome such issue, different layouts have been proposed, being the recompression the most common approach. In such layout the recuperator is split in two parts, the high temperature recuperator (HTR), crossed by the same mass flow rate in each stream, and the low temperature recuperator (LTR), crossed by a lower mass flow rate in the high-pressure stream, with higher specific heat. This arrangement may achieve a nearly constant temperature difference between both streams inside the LTR, whereas a low difference occurs in the HTR recuperator, where the specific heat difference between both streams is lower due to the fact that it works in a higher temperature range. Admittedly, the recompression can achieve high efficiencies, but it requires a low temperature difference in the thermal source. If such difference is higher, the simple recuperative layout is a more efficient solution, as shown Linares et al. [2]. Figure 1 shows both layouts: simple recuperative in subfigure (a) and recompression in (b), where the recuperator has been split into two heat exchangers: LTR and HTR. In some layouts the main compressor is split into two units in serial, with an intermediate cooler to reduce even more the compressor consumption. Such arrangement is known as “intercooling layout”.

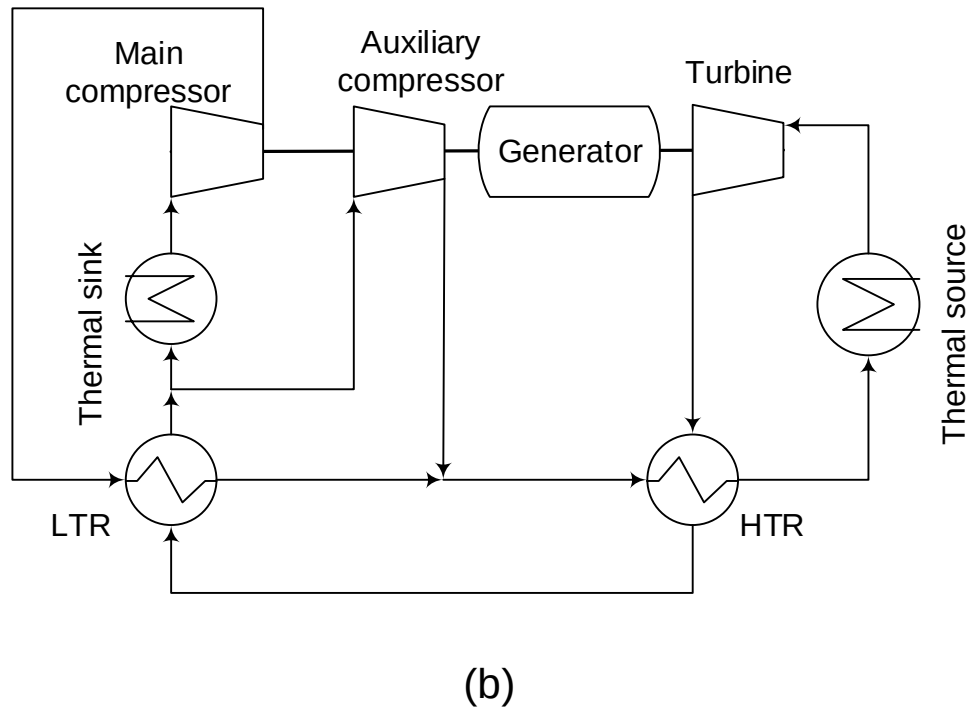
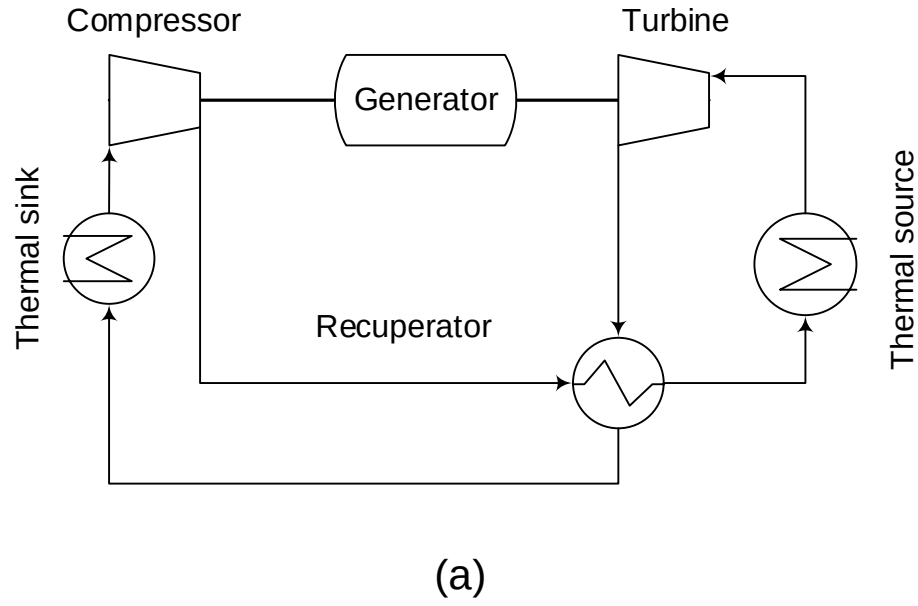


Figure 1. Classical layouts for Brayton S-CO₂: (a) simple recuperative; (b) recompression.

Other authors also propose the S-CO₂ for fusion reactors, as Herranz et al. [3] for a helium cooled lithium lead reactor (HCLL) and Serrano et al. [4] for a dual coolant lithium lead reactor (DCLL), where the classical recuperator arrangement is modified to achieve a better matching of the several thermal sources. Halimi et al. [5] take advantage of the high temperature divertor (717°C/502°C) of a HCLL reactor to propose a classical recompression arrangement, whose efficiency increases adding reheating. Isiyama et al. [6] analyse steam Rankine and Brayton power cycles with helium and supercritical CO₂ for a fusion reactor with a low temperature divertor (coolant outlet at 250°C). The study reveals again the proper integration of all the sources as a key factor to maximise the

efficiency. Estimations of sizes are included, achieving the S-CO₂ layout a reduction of 56% in the overall volume compared to a steam power plant. Recently, Vesely et al. [7] have compared steam Rankine power cycle with S-CO₂ for DEMO, concluding that S-CO₂ is more flexible to integrate thermal sources at different temperature levels.

The first proposals of CO₂ Brayton power cycles come from the end of sixties of the past century, when Angelino [8] analysed several arrangements of transcritical cycles, that is, with condensation of the CO₂ in the heat rejection process, which requires a low temperature heat sink. At the same time, Feher [9] proposed the supercritical cycle, that is, all the cycle working at supercritical pressure. At that stage, Feher compared the S-CO₂ with the supercritical steam Rankine cycle (a transcritical cycle indeed), showing some advantages for the CO₂. The interest in the S-CO₂ appeared again in the beginning of the current century, boosted by the thesis of Dostal [10] where he analyses the recompression cycle for Generation IV nuclear reactors, performing parametric studies, comparisons with other cycles (helium-Brayton and steam Rankine) and cost estimation. Based on this work, a latter study [11] was carried out in gas cooled nuclear reactors, finding that S-CO₂ reduces the cost up to 30% compared with the indirect steam Rankine power cycle. According to the results, the S-CO₂ power cycle results specially recommended for molten salt or liquid metal cooled reactors, where the thermal source pumping consumption is reduced, compared to a gas cooled reactor. In fact, the application of the S-CO₂ to sodium fast reactor (SFR) is likely the most studied case in nuclear applications. So, Moiseyev et al. [12] investigate alternative layouts to the classical recompression S-CO₂ in a SFR application. They also carry out a study to maximise the efficiency through the optimisation of the compressor inlet conditions. No improvements are found adding reheating and intercooling, as confirmed by a latter analysis by Pérez-Pichel et al. [13]. Recently, Ahn et al. [14] analyse the use of S-CO₂ and other Brayton power cycles for a small modular reactor (SMR) based on a SFR to avoid the issues derived from the interaction between sodium and water when steam Rankine cycle is used. Several turbomachine arrangements were proposed, with radial and axial designs. The S-CO₂ cycles exhibit the highest efficiencies and the lowest overall heat exchanger size.

The S-CO₂ has also been proposed for nuclear applications other than SFR. So, Bae et al. [15] compared S-CO₂ and helium Brayton power cycles for future Small Modular Reactors (SMR) based on a small (20 MWth) high-temperature gas cooled reactor (HTGR). They analyse the effect of the size reduction in the performance of the main components of the power plant. Based on the reduction of turbomachines efficiency with size they found that the pressure ratio in S-CO₂ is hardly affected by this efficiency reduction, whereas in helium cycle the pressure ratio is reduced. This behaviour entails to an increase in the volume of heat exchangers in the helium Brayton cycle, while this volume is nearly constant in the case of S-CO₂. So, S-CO₂ reveals as a better option for future SMR than helium Brayton power cycle. Mohammadi et al. [16] also compare helium with S-CO₂ Brayton power cycles for a gas cooled fast reactor (GFR). On one hand, the use of S-CO₂ allows to reduce the investment costs by 18 – 24% compared to an indirect steam Rankine cycle and 7% compared to a direct helium cycle. On the other hand, fuel costs are higher in CO₂ cycles due to the required extra enrichment to compensate the moderating effect of CO₂. Recently, Santini et al. [17] propose a S-CO₂ cycle for a pressurised water reactor power plant. In this case, due to the low reactor temperature, it is necessary to adapt the cycle using intercooling and reheating. The condensation cycle (transcritical) is also considered. The results confirm that S-CO₂ is a feasible alternative to steam Rankine power cycle not only for medium to high future reactors (especially SFR) but to current light water reactors (LWR).

Although the vast majority of the analyses about the use of S-CO₂ as a power conversion system has been focused on nuclear field, other applications have also been considered. So, still in connection with current nuclear plants, Lee et al. [18] study the integration of a S-CO₂ cycle to a current-technology nuclear power plant which is coupled to a desalination plant. The proposal is to replace either the low-pressure turbine from which the steam is bled for the thermal desalination process or even the entire steam power plant by a S-CO₂. The use of S-CO₂ cycles for this cogeneration application implies operational advantages. In a more general point of view, Li et al. [19] carry out a comprehensive review of S-CO₂ applications, mainly in nuclear and solar energy sectors, but also covering geothermal and heat recovery from waste and fuel cell. The analysis includes theoretical and economic aspects, together with a survey about the experimental facilities around the world. Another review is found in Ahn et al. [20], which covers fossil, renewable and future nuclear power plant applications. The S-CO₂ is recommended for intermediate to high turbine inlet temperatures, achieving the recompression arrangement the highest efficiencies. However, the design of a bottoming cycle should be carefully performed due to the high temperature of CO₂ at the inlet of the recovery heat exchanger, as is also established by Linares et al. [2]. Ahn et al. [20] also compile a list of international laboratories with experimental facilities to test the actual performance of the cycles, although higher demonstration units (greater than 10 MWe) should be available to test the economy aspects. Sarkar et al. [21] explore the feasibility of the S-CO₂ cycles in low-grade heat sources. The analysis points out that transcritical cycle is more suitable for these type of resources, being the noticeable temperature variation in the heat addition process an advantage of S-CO₂ over steam or organic Rankine power cycles. In order to obtain the required low heat sink temperatures, some alternatives are proposed, as a liquefied natural gas production process.

Concentrated Solar Power plants (CSP) constitute another field where the S-CO₂ has revealed an interesting potential. So, Turchi et al. [22] study this topic, including the dry cooling issue. Recompression arrangement is assessed, with variations including reheating and intercooling. Intercooling is revealed as a good option for CSP due to the reduction of the CO₂ source inlet temperature, which enhances the matching to the sensible thermal storage system. Based on recompression with intercooling and reheating, Milani et al. [23] propose a hybrid fossil/solar power plant of 10 MWe. An analysis of a molten salt storage embedded in a cycle is performed by Wang et al. [24], who analyse the integration of a two-tank molten salt storage system with recompression, intercooling, reheating, partial cooling, and precompression arrangements. Intercooling achieves the highest efficiency, whereas the lowest is reached by recompression. Reheating is not worth if the high turbine inlet temperature is available and precompression might be the best option when compression inlet temperature is high, as in dry cooling.

Bottoming and combined cycles are other fields which are paying attention to S-CO₂. Bae et al. [25] propose many arrangements as bottoming cycle for a molten carbonate fuel cell (MCFC). The analysed layouts include recompression, simple recuperative and even a S-CO₂/steam Rankine combined cycle. The achieved temperature is very high, due to the temperature of the waste heat from the fuel cell (700°C/650°C). A comparison of the heat exchangers size is performed, showing that the recuperative layouts result in lower heat exchanger volume than in the recompression arrangement. The replacement of the steam Rankine power cycle by a S-CO₂ power cycle in a combined cycle is analysed by Cho et al. [26], concluding that classical layouts (simple recuperated and recompression) are not suitable for waste heat recovery. Authors propose new layouts that are able to produce a sufficiently low temperature of the exhaust gases leaving the heat recovery heat exchanger. S-CO₂ Brayton power cycles have been also proposed as topping cycle in a combined cycle arrangement using an organic Rankine cycle (ORC) as bottoming cycle, as performed by Akbari

et al. [27] who compare the combined cycle with the stand-alone S-CO₂, finding a higher efficiency in the former. The analysis covers several fluids as working fluid for the ORC. Wang et al. [28] apply the analysis of the combined cycles to future nuclear power plants, comparing stand-alone S-CO₂ with combined cycles. They find that in stand-alone arrangements the partial cooling exhibits better performance than recompression arrangements, being the partial cooling operated in transcritical condition, and the recompression in supercritical. Efficiency is always improved in combined cycles, in relation to the stand-alone cycles, using a recuperative arrangement in the S-CO₂ power cycle.

The optimisation process of a S-CO₂ cycle is a complex task and has been the aim of a number of studies focused in determining the key parameters of the cycle. So, Sarkar et al. [29] optimise the reheating process in a recompressed layout. The balanced condition (equal approach temperature at each end of the heat exchanger) in the low temperature recuperator is shown, and an empirical rule to determine the reheating optimum pressure is proposed, once verified that the geometric average rule does not work in recompression cycle. A comprehensive study about several layouts of S-CO₂ power cycles in similar conditions is carried out by Crespi et al. [30], highlighting the key parameters. Another parametric analysis carried out by Pham et al. [31] maps the performance of a recompression S-CO₂ cycle for nuclear applications ranging the temperatures from 250°C to 850°C and setting the pressures at 200 bara and 250 bara as parameters. The results are applied to the replacement of a current pressurised water reactor (PWR) for a SMR application and to the power conversion system of a future SFR. Other authors focus on the fluid. So, Cardemil et al. [32] investigate the use of CO₂ as working fluid compared with others, finding that CO₂ exhibits the lowest thermal conductance. Hu et al. [33] analyse the replacement of pure CO₂ by binary mixtures of CO₂ with another substance in a recompression S-CO₂ cycle for a future nuclear reactor. Both the fluid used in the mixture and its concentration affect the critical pressure and temperature, increasing or decreasing them. Heat transfer properties also are affected, being possible to reduce the amount of heat exchanged in the recuperators. Although such modifications would allow the designer to optimise the steady state operation, the lack of experimental results with such mixtures make them more suitable for long-term designs, being more reliable the use of the CO₂ as a pure substance in the current paper. Some dynamic analyses have been performed, as it is found in Floyd et al. [34], where off-design behaviour due to a sink temperature increase in a recompression layout used as a power conversion system in a future SFR reactor is assessed. No stability issues are detected, but an important loss in power and efficiency occurs. This concern can be overcome if the speed of the main compressor is controlled in the range from 50 to 56 Hz.

Heat exchangers, especially recuperators, are a key aspect in S-CO₂. Dostal [10] states that printed circuit heat exchangers (PCHE) exhibits a great advantage over shell and tubes, as, for instance, much higher compactness. Moreover, the bonding manufacturing process allows to endure high pressure differences [35]. Nikitin et al. [36] perform experimental studies about the behaviour of PCHE with supercritical CO₂ as working fluid, deriving some correlations. Results show an overall heat transfer coefficient from 300 to 650 W/m²-K and compactness of 1,050 m²/m³ with respect to the heat exchanger core. Ariu et al. [37] compare the design of a heat exchanger for a future molten salt fast reactor (MSFR) following the Tubular Exchange Manufacturer Association (TEMA) shell and tube criteria and PCHE. The analysis shows a large advantage in the use of PCHE for its high compactness. A guide to size PCHE in both parallel/counterflow and crossflow arrangement is given in Youn et al. [38], with application to SMR, which enlarges the method described by Dostal [10].

An additional technical challenge rises in the power conversion system for DEMO. It is the pulsed operation, that is, the existence of a pulse period followed by a dwell period. Energy storage systems

might constitute a way to overcome this working scheme, allowing to maintain the same conditions in the power conversion system in both periods, as in CSP solar power plants with thermal storage. Gil et al. [39] conduct a review about a variety of thermal energy storage systems, covering sensible, latent and chemical storage materials, specifying properties and modelling methods. They include liquids (molten salts) and solids (concrete and castable ceramics) among the sensible systems. The analysis is completed in Medrano et al. [40] with some study cases. A more specific review, focused on CSP applications is found in Pelay et al. [41]. The study covers the most used systems, from sensible solid and liquids to future systems as latent, hybrids (sensible and latent) and thermochemical, analysing pros and cons. One important issue about the storage system is its ability to release the stored energy in the required time, that is, its dynamical behaviour. Zaversky et al. [42] perform a dynamic analysis comparing concrete and two-tank molten salt response, finding that the response in the concrete system tends to reduce the efficiency of the downstream power cycle. In order to solve this concern, Jian et al. [43] describe a control strategy which enhances up to 14% the material utilisation of a concrete storage system for a concentrated solar power plant. Regarding solid sensible systems, Bergan et al. [44] describe a new material named HEATCRETE, similar to concrete, developed by NEST, with good behaviour. The material is built in modules which can be connected to each other, and consists of inner steel pipes crossed by a heat transfer fluid (gas or liquid). The temperature range is 100°C to 500°C, achieving better structural integrity and much smaller footprint than usual concrete. Martins et al. [45] describe a solar power facility sited in Abu Dhabi to test this new material.

Authors have already explored the application of S-CO₂ power cycles to fusion reactors based on a DCLL blanket. So, in [4] they proposed some modifications in a recompression layout to match the thermal sources from a long-term DCLL design with high temperature divertors (566°C to 800°C). More recently, they have analysed the application for a mid-term DCLL design, the former version of the one considered in the current paper. So, in [46] they consider different arrangements in a recompression layout to integrate the different thermal sources. Finally, in [2] a comparison between simple recuperated and recompressed layout with only one recuperator is explored, considering the heat release of some low temperature sources. In this paper, a modified recuperated layout is proposed for DEMO fusion reactor based on a DCLL blanket. The layout integrates all the available thermal sources (helium and lithium-lead in the blanket, high temperature divertor (cassette), low temperature divertor (Plasma Facing Unit, PFU) and vacuum vessel) through two thermal loops according to the temperature ranges. Each thermal loop includes a sensible thermal energy storage system. Low temperature thermal loop is connected in parallel with the recuperator allowing it to overcome the known specific heat issues without employing a recompressed layout. Once the chain of efficiencies is defined, taking into account different self-consumptions, the steady state operation is obtained, so establishing the design point. For this point the main components (heat exchangers and turbomachines) are sized. The diameter of the pipes is also calculated according to the maximum velocity criterion. Drawings representing the space allocation and the heat exchangers arrangements are also obtained.

2. Methodology

2.1 System configuration

Figure 2 illustrates the proposed layout for the power conversion system of DEMO based on a DCLL blanket. The reactor operates in pulsed mode with a pulse period of 120 minutes followed by a dwell period of 30 minutes [47]. In order to balance the electricity output power, a thermal energy storage

system (TES) is required. In Figure 2, four different types of loops are shown: thermal source loops, storage system loops, power conversion system loop, and finally, a sink loop. There are five thermal source loops: two for the breeding blanket, two for the divertor, and one for the vacuum vessel (VV). The breeding blanket is cooled by two cooling media (helium (BB-He) and lead-lithium (BB-LL)) while the divertor is cooled by water. The divertor consists of two zones: the plasma facing unit (DIV-LT) and the cassette (DIV-HT), each operating at different temperature ranges.

Table 1 depicts the main characteristics of the thermal sources [47] at pulse period. It is noticeable the huge mass flow rate in the lead-lithium stream (23,500 kg/s). Such mass flow rate is divided into 16 thermal loops (represented as one equivalent in Figure 2), each with its own pump, so pumping 1,469 kg/s each one [48]. Utili et al. proposed mechanical centrifugal pumps to be used in the lead-lithium loops [49]. Later, Utili stated the specifications of the pump in [50], being Green Pump the company selected in 2017 as manufacturer.

Table 1. Specifications for thermal sources [47]

	Breeding Blanket		Divertor		Vacuum Vessel VV
	BB-He	BB-LL	DIV-HT	DIV-LT	
Coolant type	Helium	Lead-Lithium	Water	Water	Water
Pressure at reactor outlet [bara]	75	1	150	25	30.5
Temperature at reactor outlet [°C]	405	550	325	160.2	205
Temperature at reactor inlet [°C]	300	300	285	150	195
Pressure drop at reactor [bara]	5	12.9	25	25	1
Mass flow rate [kg/s]	1,146	23,500	416.6	2,886	2,007

Each source dumps its power, together with the source loop pump consumption, into the storage loop. According to the temperature range of each source, two storage loops have been defined: the high temperature storage loop (HTS) and the low temperature storage loop (LTS). The former uses a solar salt (60% NaNO₃ – 40% KNO₃) as working fluid [51] in a two-tank configuration, while the latter uses water and a concrete solid storage system. It has not been possible to use solar salt in the LTS loop due to its lowest temperature (150°C) being well below the freezing point of the salt (220°C). On the contrary, the lowest temperature in the HTS loop is 285°C, higher than the freezing point of the salt. During the pulse period (120 minutes), 80% of the salt mass flow rate passes through the HX-HTS and 80% of water through the HX-LTS while the rest of mass flow rate is charging the corresponding TES. In the dwell period (30 minutes), the stored energy is supplied to the HTS and LTS heat exchangers, resulting in the power cycle being fed with the same heat input as in the pulse period. In the dwell period, the heat released by the reactor (approximately 1% of the heat produced during the pulse period) is rejected to the sink (not represented in Figure 2).

The power conversion system is a modified recuperative supercritical CO₂ Brayton power cycle. The classical re-compressed layout has not been adopted as a result of the low temperature of both breeding blanket coolants (300°C) which results in a relatively low temperature in the high temperature storage loop heat exchanger (HX-HTS). The use of the high effective recuperation process given by a re-compressed layout entails to exceed the allowable temperature [2]. So, only one recuperator is used. The different value of CO₂ specific heat on each recuperator stream is overcome by the integration of the low temperature storage loop (HX-LTS), which enables a lower

mass flow rate in the high pressure (higher specific heat) stream, so achieving an equal temperature approach at each end of the recuperator (i.e., obtaining a balanced heat exchanger).

The performance of the layout has been simulated using the Engineering Equation Solver (EES) environment, which includes models of pure substances [52]. Therefore, CO₂, helium and water have been modelled as pure substances. The incompressible liquid model has been used for salt (properties included in the EES libraries) and for lead-lithium (specific heat as 189 J/kg-K and density as 9,726 kg/m³ [53]).

In order to determine the pump consumption, isentropic efficiency for pumps (water and lead-lithium) has been taken as 85 %; for circulator (helium) a value of 82 % has been considered. These values include the efficiency in the pumps and circulator engines. Pressure drop on the hot stream side of heat exchangers connecting the source loop with the storage loop has been assumed as 1 % [54], except for the lead-lithium where 10 kPa have been assumed due to the low pressure (1 bara) at the hot stream inlet. Regarding the sink loop, 5°C of temperature rise has been assumed in the cooling water passing through the pre-cooler and a pressure drop in the entire loop of 5 bara. With these hypotheses the pump consumption and heat power at each loop are calculated, obtaining the results shown in Table 2 for the pulse period. In this period (120 minutes) 20% of the thermal power dumped into the storage loop is stored: 391.23 MW in HTS and 44.44 MW in LTS. These are supplied during the dwell period (20 minutes) to the power conversion system, when the residual energy released by the reactor is released to the sink.

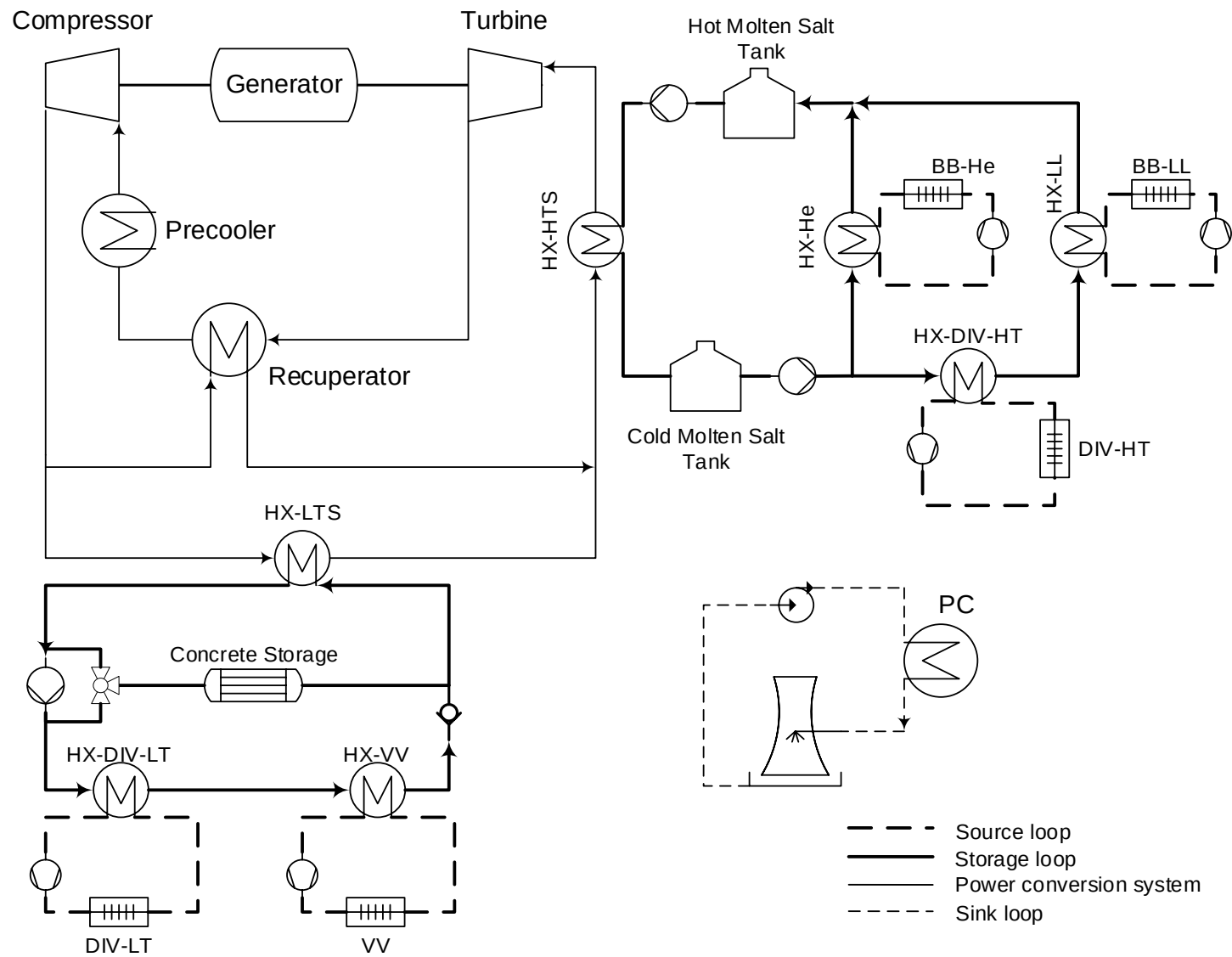


Figure 2. Layout proposed (pulse period).

Table 2. Thermal power and pump consumption at each loop in pulse period (pump consumption have been calculated according to the efficiencies given in [54]).

	Breeding Blanket		Divertors		Vacuum Vessel VV
	BB-He	BB-LL	HT-DIV	LT-DIV	
SOURCE LOOP					
Working fluid	Helium	Lead-Lithium	Water	Water	Water
Thermal power from the reactor [MW]	624.83	1,107.3	95.0	122.53	89.98
Thermal power to the storage loop [MW]	748.41	1,111	96.72	131.86	90.33
Pump consumption [MW]	123.58	3.7	1.72	9.33	0.35
STORAGE LOOP	HTS			LTS	
Working fluid	Salt			Water	
Thermal power to the cycle [MW]	1,565.5			178.3	
Thermal power to the thermal storage [MW]	391.23			44.44	
Pump consumption [MW]	0.75			0.70	
SINK LOOP					
Working fluid	Water				
Thermal power in the precooler [MW]	1,092.7				
Pump consumption [MW]	30.9				
Thermal power to the cooling tower [MW]	1,123.6				

2.2 Efficiency definitions

Figure 3 sketches the whole energy conversion process from the heat power released by the reactor to the busbar power. As it is seen, it is possible to identify three different mechanical/electrical powers:

- Cycle power (W_{cy}). Net mechanical power produced by the CO_2 power cycle, that is, power produced by the turbine minus power consumed by the compressor. This can be related to the heat supplied to the power cycle through the cycle efficiency (η_{cy}).
- Gross power (W_{gross}). Electrical power produced by the generator, assuming an efficiency of 97% [54]. This power can be related to the heat released by the reactor through the gross efficiency (η_{gross}).
- Electrical power (W_e). Electrical power obtained subtracting the power required to pump the working fluids in source, storage and sink loops from the gross power. This power can be related to the heat released by the reactor through the electrical efficiency (η_e).
- Net power (W_{net}). Electrical power measured at the plant busbars, obtained subtracting from the electric power the power required for auxiliary services (for instance, heating) and other parasitic loads. This power can be related to the heat released by the reactor through the net efficiency (η_{net}).

To take into account the effect of pulsed operation an average efficiency is defined based on the duration of each period. Pump consumptions are considered to be equal in both pulse and dwell period assuming that the pumps are also working on dwell period to reduce the thermal stress in the pipes (a by-pass around the pump allows to send a very low mass flow rate to the loop in order to maintain reasonable temperatures with very low heat released by the reactor). In such conditions, the electrical power is the same in both periods, and the average electrical efficiency can be determined using Eq. 1, where the thermal power taken from the reactor (the summation in the denominator) is 2,039.64 MWth, according with Table 2. This power includes both the fusion power and the nuclear energy multiplication in the blanket.

$$\eta_e^{avg} = \frac{150 \cdot W_e^{pulse}}{120 \cdot (Q_{BB-He}^{pulse} + Q_{BB-LL}^{pulse} + Q_{DIV-HT}^{pulse} + Q_{DIV-LT}^{pulse} + Q_{VV}^{pulse})} = 1.25 \cdot \eta_e^{pulse} \quad (1)$$

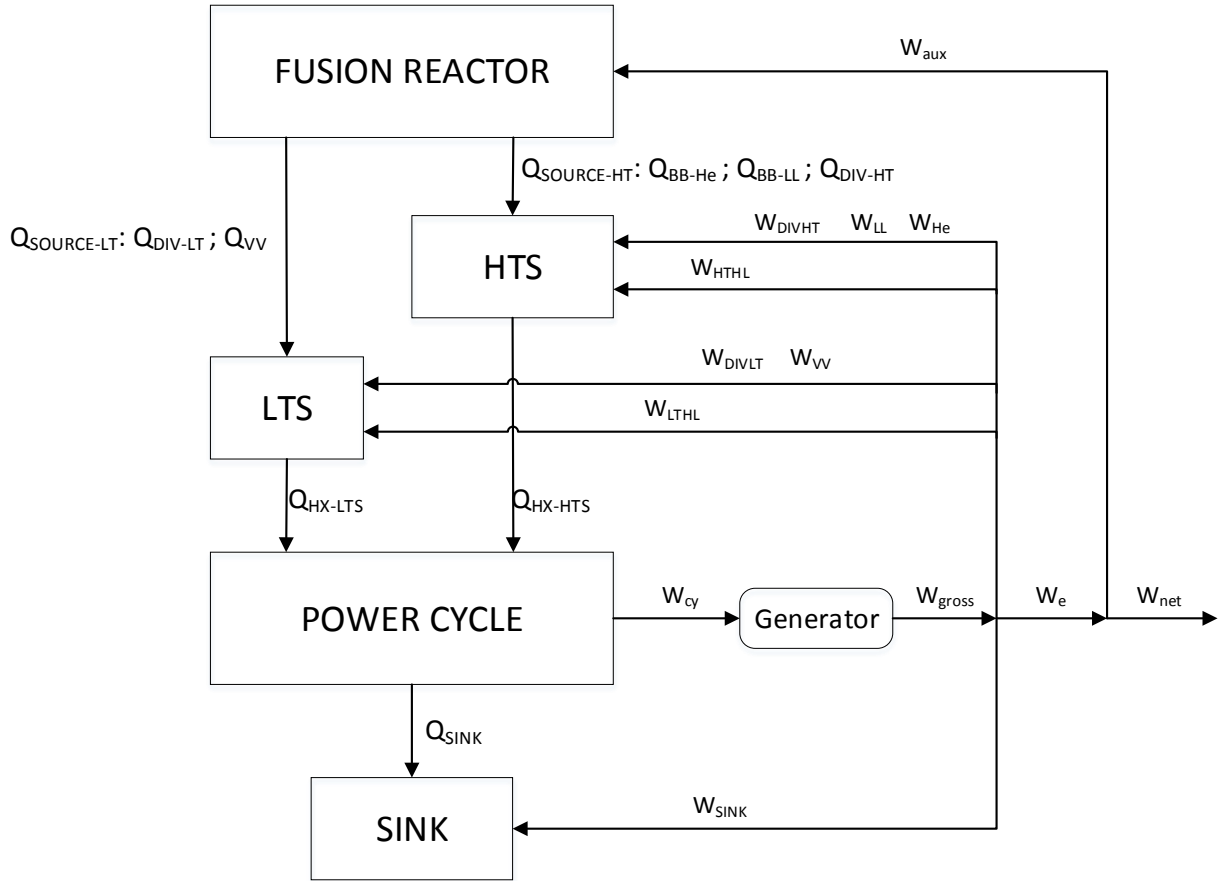


Figure 3. Energy conversion diagram in the fusion power plant.

2.3 Sizing of heat exchangers

Printed circuit heat exchangers (PCHE) have been considered because of their excellent strength to high pressures and their high compactness, with the ability to achieve pinch points as small as 2°C

(effectiveness around 98%) [5]. Pinch point has been fixed at 5°C in some heat exchangers (HX-VV, HX-DIV-LT, HX-He, HX-DIV-HT, HX-HTS, and Recuperator). In the others, the temperatures are determined by energy balance and the pinch point achieved is always higher than 5°C. Pressure drop in CO₂ stream is 40 kPa; in cold water 60 kPa and in salt 60 kPa.

The heat exchangers have been discretized in sub-heat exchangers using suitable correlations [55]. Pressure drop and energy conservation equations are solved iteratively. At each time step, heat transfer coefficients are determined and then, using the Log Mean Temperature Difference (LMTD) method, the heat exchanger length is estimated. If the pressure drop condition is not satisfied, the number of channels is modified and the whole iterative process is initiated again. At the end of the calculation, the heat transfer surface area and the dimensions of the heat exchanger are obtained.

The heat exchangers are manufactured in modules of 600 mm wide and 600 mm high, with a maximum length of 1.5 m [35]. Up to 14 modules can be linked in parallel, so obtaining a maximum height of the heat exchanger of 8.5 m. For supporting reasons, the heat exchangers lie down, with the streams flowing in vertical directions. So, the flow path length corresponds with the physical height of the PCHE and the flow path height with the physical length of the PCHE. The maximum nozzle diameter is 900 mm.

2.4 Turbomachines

When performing a turbine design some initial considerations have to be taken into account, like the values of the flow coefficient, the stage-loading coefficient, and the number of stages. Some empirical expressions and charts might help with the decision of their preliminary values, like the Smith chart, where an optimum loading coefficient can be estimated for a certain flow coefficient. In addition, seal type and tip clearance are other factors that affect the performance of the turbine. An unshrouded structure is chosen as a first approximation, because of its simplicity and lower cost, even though the efficiency decreases compared to the values obtained with shrouded ones. The inlet pressure has been set to 300 bara. The isentropic efficiency has been set to 93% for power conversion system calculations [56], being this value the objective in the design calculations.

Regarding the compressor design, there are also many restrictions to consider, such as the loading limit, the rotor inlet tip Mach number, etc. The relationship between stage loading and flow coefficient should be selected within the restrictions of surge and choke. In [57], a maximum value of 0.5 for the flow coefficient is suggested. With these restrictions, the compressor should also have a high degree of reaction [58], mainly to achieve a higher efficiency with a reduced cost and complexity. Also, the tip clearance has an effect on both compressor stability and performance. Inlet conditions for compressor have been taken as 85 bara and 35°C. This pressure exhibits a reasonable balance between the closeness to critical pressure and prevention of potential stability problems, if any [59]. An isentropic efficiency of 88 % has been set for compressor in the power conversion system calculations [56], being this value the objective of the design process.

AXIAL™, from Concepts NREC, a meanline design and analysis tool for multistage axial turbomachinery has been used to carry out the preliminary design of the turbine and the compressor. AXIAL™ includes two major procedures: a design mode and an analysis mode. The design mode generates geometry, based on flow and performance data and allows obtain preliminary designs which could be optimized using the analysis mode. In addition, NIST property database is included in the software, required when using CO₂. Other authors have used this

software to analyse and design this type of turbomachines, as can be found in [60], [61], [62], [63], [11].

As mentioned above, AXIAL™ have been used to obtain preliminary designs for the turbine and the compressor that will allow an estimation of the turbomachinery volume. Optimized designs do not differ significantly in terms of size from preliminary calculations. Therefore, at this stage of the power plant sizing, these values are enough to define the layout.

2.5 Sizing of pipes

Several criteria may be selected in order to size a set of pipes, mainly pressure drop or maximum velocity. Both criteria have been used: first the maximum velocity criterium is applied to obtain the pipe diameter and after that, the pressure drop is calculated to verify that it is under the recommended values.. In CO₂ pipes the NORSOK P-001 code has been followed [64], which is accepted to perform designs of CO₂ infrastructure [65]. The maximum allowable velocity in a gas pipe to avoid noise and vibration problems which is called erosional velocity is given by Equation 2, where ρ is the gas density, in kg/m³, and V_{max} is obtained in m/s.

$$V_{max} = \min \left\{ 175 \cdot \left(\frac{1}{\rho} \right)^{0.43} ; 60 \text{ m/s} \right\} \quad (2)$$

This equation can be used for any type of gas, including natural gas ([66], [67]).

In the same standard NORSOK P-001 [64], the recommended maximum velocity of a liquid in a carbon-steel pipe (material selected for the water pipes) is 6 m/s, which has been taken as the criterion for water pipes. Finally, in the molten salt pipes, velocity is more critical due to erosion or abrasion, and the maximum allowable velocity is assumed as 3 m/s, as found in [68]. Once the maximum allowable velocity is set, the minimum pipe diameter (inner diameter) is determined using the law of conservation of mass.

The following step is the material selection. Carbon dioxide pipes operate at high pressure (around 300 bara), so nickel-based alloy Inconel 740H has been chosen. This alloy, approved by ASME Code Case 2702 [69], has been developed for applications in advanced power plants, mainly ultra-supercritical steam cycles and supercritical CO₂ cycles [70], [71]. Its allowable stress properties can be found in [69], as a function of the temperature. As in water pipes there are no special strength requirement, carbon steel A-106B has been selected. This is a material commonly used in process and power piping. Its stress properties can be found in [72]. Corrosion is the main issue involved in the selection of the pipe material when working with solar salt. Following the recommendations given in [73], carbon steel (A-106B) has been selected for the cold pipes (up to 300°C) and moderate to high chromium stainless steel (SS-347H) for the hot pipes (above 300°C). The stress properties of both materials can be found in [72]. Once the material is selected, the maximum allowable stress as a function of temperature for each one is read from the standards [74].

For every pipe of the cycle, an iterative process has been carried out starting by choosing a nominal diameter (DN) greater than the value of the minimum inner diameter already obtained [74]. For that nominal diameter, t_{min} can be calculated [72], and a nominal thickness (characterized by its schedule) greater than that value is chosen [74], if it exists. If there is no nominal thickness equal or greater than the calculated, which was the case for some pipes in the CO₂ cycle, the mass flow rate was divided into several parallel pipes. The number of parallel pipes is the minimum necessary to

find an existing nominal thickness. A final calculation is performed to check that the maximum velocity criterion is met with the DN and the schedule selected.

Then, pressure drop (in kPa/m) is calculated for every pipe, to check that it is below the recommended values. These recommended values have been obtained from Norsok P-001 and are different for gases and liquids. For gases it depends on the pressure of the gas and for liquids it is a constant value. To perform pressure-drop calculations, Darcy equation was used along with Colebrook equation to obtain the friction factor. If the pressure drop is higher than the recommended value, then a higher diameter is selected and checked again. If there is no diameter that fits, which sometimes happens if the required thickness is high, a parallel pipe is added. In the salt pipes, pressure drop was not an issue because of the low velocity (3 m/s) fixed to prevent abrasion.

3. Results

As a consequence of the pipes sizing, the power plant has been divided in four equal parts in order to obtain pipe commercial diameters. This new arrangement also attains smaller mass flow rates, more suitable for turbomachinery design. This split criterion is the opposite to the common practice in steam power plant, where for the installed power of the current case a single turbine would have been installed for scale economy reasons. However, S-CO₂ power plants exhibit a different cost breakdown, being the investment in turbomachines only one fourth of the investment in the heat exchangers (PCHes) [75]. So, the absolute numbers given in this section are referred to a quarter of the overall plant, assuming that the source heat exchangers have also been split in this way.

3.1 Layout

The information is presented in three blocks:

- High temperature heating loop (HTHL), which consists on the high temperature sources and the salt storage loop.
- Low temperature heating loop (LTHL), which consists on the low temperature sources and the water storage loop.
- Power conversion system (PCS), which comprises the thermodynamic cycle.

The sink loop has not been studied.

Figure 4 shows the size of the pipes in the HTHL. The size is specified using three numbers: number of pipes x nominal diameter (DN, in mm) x schedule. The thickness ranges from 4.2 to 5.5 mm and the material is A-106B for pipes s1, s2, s4, s5, s10, and s11; for the rest of the pipes the material is SS347H. Table 3 depicts the properties of the salt at each state point of the loop. Table 4 compares the actual pressure drops and velocities with the allowed velocity and recommended pressure drop.

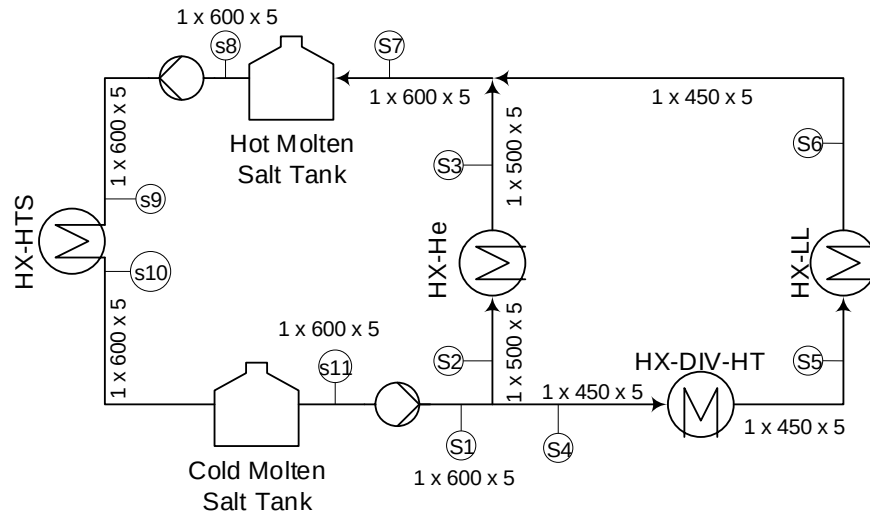


Figure 4. Size of HTHL pipes.

Table 3. HTHL state points (salt).

	T [°C]	p [bara]	h [kJ/kg]
s1	274.5	2.2	21.79
s2	274.5	2.2	21.79
s3	400	1	210.8
s4	274.5	2.2	21.79
s5	294.9	1.6	52.3
s6	525.7	1	402.8
s7	456.1	1	296.1
s8	456.1	1	296.1
s9	456.1	1.6	296.2
s10	274.5	1	21.72
s11	274.5	1	21.72

Table 4. HTHL Pressure and velocity criteria for pipes sizing (salt).

	Mass flow rate [kg/s]	Actual velocity [m/s]	Allowed velocity [m/s]	Actual pressure drop [kPa/m]	Recommended pressure drop [kPa/m]
s1	1,426	2.64	3.00	0.15	0.90
s2	990	2.65	3.00	0.19	0.90
s3	990	2.77	3.00	0.18	0.90
s4	793	2.62	3.00	0.21	0.90
s5	793	2.64	3.00	0.21	0.90
s6	793	2.86	3.00	0.21	0.90
s7	1,426	2.82	3.00	0.15	0.90
s8	1,426	2.82	3.00	0.15	0.90
s9	1,426	2.82	3.00	0.15	0.90
s10	1,426	2.64	3.00	0.15	0.90
s11	1,426	2.64	3.00	0.15	0.90

Density of molten salt at the hot tank is 1,798 kg/m³ and at the cold one 1,915 kg/m³. The 20% of the mass flow rate entering the HTHL heating loop (state s7 in Figure 4) is stored into the hot tank during the pulse period (120 minutes), resulting in 2,568 tones. This quantity will be moved from the hot tank to the cold one in 30 minutes during the dwell period. Such mass occupies a volume of 1,428 m³ in the hot tank and 1,341 m³ in the cold one. Considering that both tanks have the same volume, and increasing it by 20%, a volume of 1,714 m³ is necessary in each tank. Due to restrictions in the dimensions of the pumps (maximum axial length of 15 m [76]), a height of 13 m is stated, which entails to a diameter of 13 m too.

Figure 5 illustrates the size of the pipes in the LTHL. The thickness and the material for all the pipes are 18.26 mm and A-106B respectively. Table 5 depicts the properties of water at each state point of the loop. Table 6 compares the actual pressure drops and velocities with the allowed velocity and recommended pressure drop.

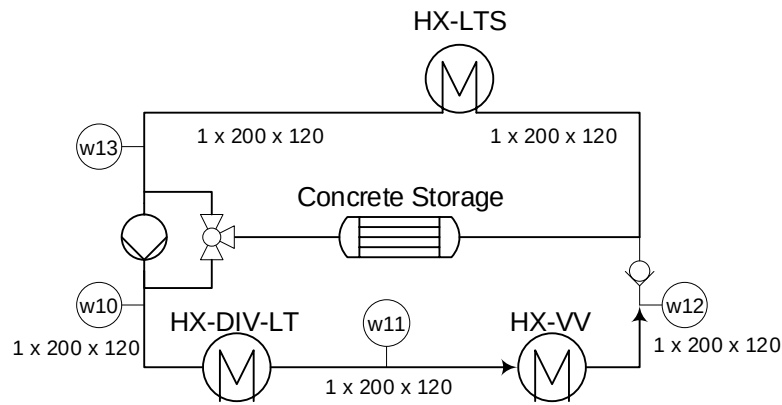


Figure 5. Size of LTHL pipes.

Table 5. LTHL state points (water).

	T [°C]	p [bara]	h [kJ/kg]
w10	87.03	30	366.8
w11	155	29.4	655.2
w12	200	28.8	852.8
w13	87.01	28.2	366.6

Table 6. LTHL Pressure and velocity criteria for pipes sizing (water).

	Mass flow rate [kg/s]	Actual velocity [m/s]	Allowed velocity [m/s]	Actual pressure drop [kPa/m]	Recommended pressure drop [kPa/m]
w10	114	4.51	6.00	0.81	0.90
w11	114	4.78	6.00	0.85	0.90
w12	114	5.04	6.00	0.89	0.90
w13	91	3.61	6.00	0.52	0.90

The concrete storage has been sized according to the modules described in [42]. Each module is formed by a matrix of concrete containing a tube bundle in staggered configuration. Water flows inside the tubes, exchanging heat power with the surrounding concrete. The maximum length of each module is 18 m, with 4 m high and 2.6 m wide, having 528 tubes of 18 mm of outer diameter and 15 mm of inner diameter. The density of concrete is 2,200 kg/m³ and the specific heat is 0.85 kJ/kg-K. The 20% of the mass flow rate leaving the HX-VV (state w12 in Figure 5) crosses the tubes of the concrete module releasing 22.22 MWh during the pulse period (120 minutes). A concrete mass of 832.8 tonnes is required to store such amount of energy, resulting in 2 modules of 18 m long with a third one of 875 mm long in serial configuration. The maximum pressure drop along this arrangement (reached in the dwell period) is 23.2 kPa.

Figure 6 illustrates the size of the pipes in the PCS. The thickness ranges from 9.5 mm to 38.89 mm and the velocity from 7.4 m/s to 25.7 m/s. The material is Inconel 740H for all the pipes. Table 7 depicts the properties of CO₂ at each state point of the loop. Table 8 compares the actual pressure drops and velocities with the allowed velocity and recommended pressure drop. Figure 7 shows the T-s diagram. It is shown how the state 10 comes from the mixing of the stream leaving the recuperator (7) and the one leaving the HX-LTS (9). The entropy balance determines a rate of entropy generation of 28.29 kW/K, showing the irreversibility of the process. This entropy generation represents an exergy destruction of 8.57 MW (taking 30°C as dead state temperature).

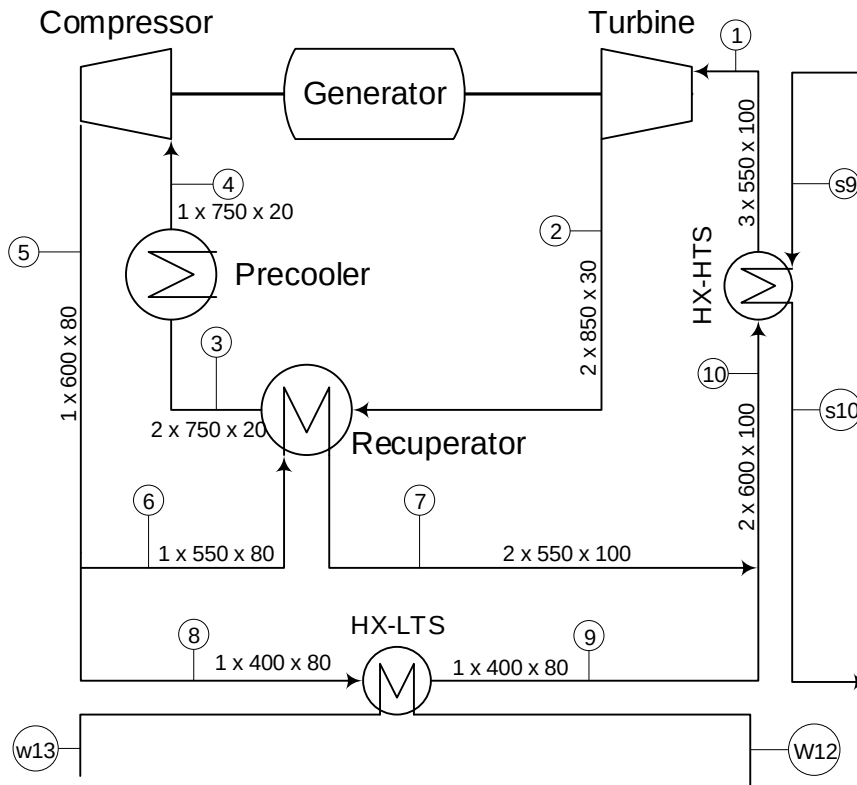


Figure 6. Size of PCS pipes.

Table 7. PCS state points (CO₂).

	T [°C]	p [bara]	h [kJ/kg]	s [kJ/kg-K]
1	451.1	300	397.1	-0.2474
2	309.1	85.8	255.1	-0.2288
3	81.62	85.4	-17.91	-0.8286
4	35	85	-197.9	-1.388
5	76.62	300.8	-163.2	-1.377
6	76.62	300.8	-163.2	-1.377
7	304.1	300.4	208.8	-0.5384
8	76.62	300.8	-163.2	-1.377
9	131.3	300.4	-53.12	-1.084
10	252.4	300.4	139.1	-0.665

Table 8. PCS Pressure and velocity criteria for pipes sizing (CO₂).

	Mass flow rate per pipe[kg/s]	Actual velocity [m/s]	Allowed velocity [m/s]	Actual pressure drop [kPa/m]	Recommended pressure drop [kPa/m]
1	505.7	12.78	17.53	0.43	0.60
2	758.5	17.50	26.62	0.16	0.19
3	758.5	10.26	19.06	0.14	0.19
4	1,517	5.82	11.08	0.16	0.19
5	1,517	8.47	10.10	0.59	0.60
6	1,113	7.40	10.10	0.50	0.60
7	556.6	10.51	15.47	0.38	0.60
8	403.8	5.11	10.10	0.35	0.60
9	403.8	7.11	11.63	0.49	0.60
10	758.5	10.51	14.55	0.40	0.60

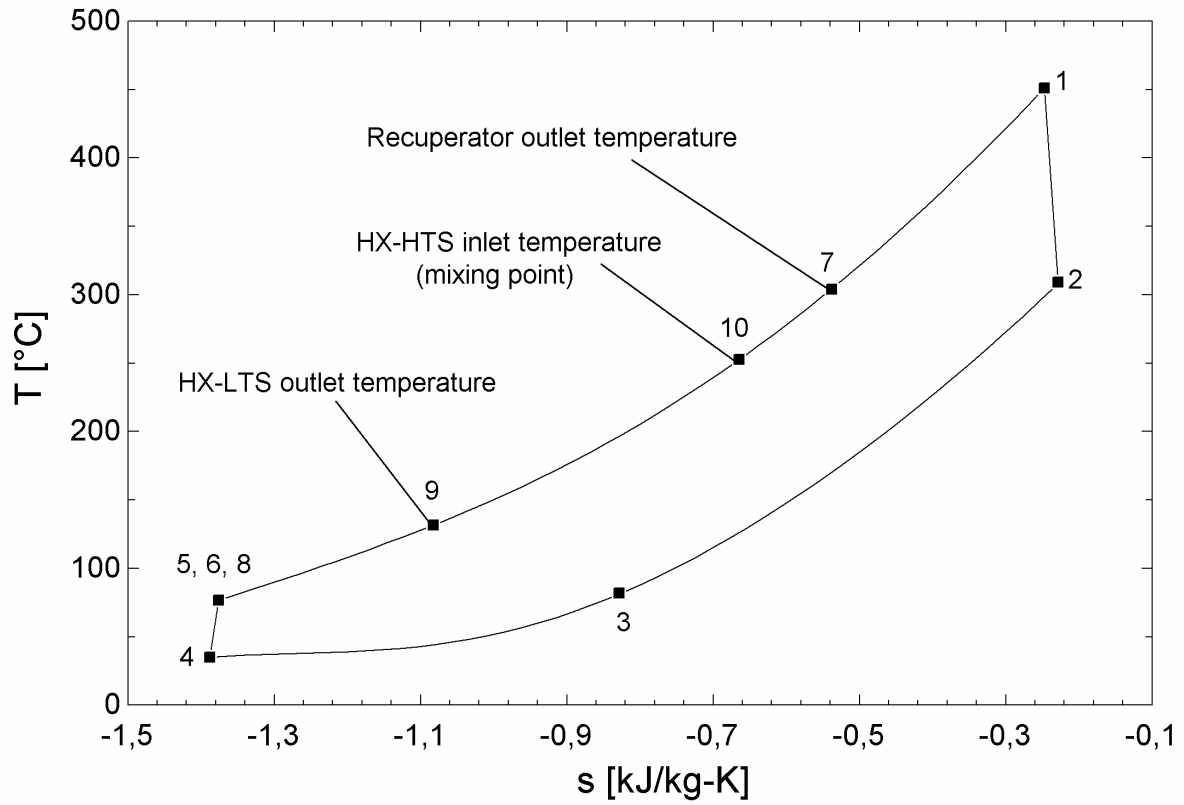


Figure 7. T-s diagram of the power cycle.

3.2 Performances

Table 9 shows the energy balance at PCS. Taking into account the pump consumption, the performances of the PCS are obtained, summarised in Table 10.

Table 9. Energy balance at PCS.

	CO ₂ mass flow rate [kg/s]	W [kW]	Q [kW]
Compressor	1,517	52,692	
Turbine	1,517	215,434	
HX-HTS	1,517		391,376
HX-LTS	403.8		44,457
Recuperator	1,113		414,094
Precooler	1,517		273,091

Table 10. Performance breakdown (definition according to Section 2.2).

	Pulse			Average
	Cycle	Gross	Electric	Electric
Power, each unit [MW]	162.7	157.9	115.3	115.3
Power, overall plant [MW]	650.8	631.6	461.2	461.2
Efficiency [%]	37.34	30.96	22.6	28.25

3.3 Heat exchanger sizing

The heat exchangers of the thermal sources (HX-LL, HX-He, HX-DIV-HT, HX-DIV-LT, and HX-VV) are implemented by the designers of the breeding blanket, divertor, and vacuum vessel, looking for a space integration with the Tokamak. So, only the heat exchangers of the power conversion system, including the connection to the high and low temperature sources (HX-HTS and HX-LTS) have been sized.

Table 11 gives the main dimensions of the heat exchangers. The largest one is the Recuperator, because it is a balanced heat exchanger (same temperature approach on both ends), being this temperature approach of 5°C. Figure 8 illustrates its temperature profile, showing a high approach along its entire length. The second largest heat exchanger is the HX-HTS, due to the closeness of both temperature profiles. So, although the heat exchanger is not balanced, the pinch point is 5°C and the approach in the other end is 22.1°C. It is noticeable that the duty of both heat exchangers is similar (391 MW for the HX-HTS and 414 MW for Recuperator), but the slight divergence in the temperature profiles results in the size of HX-HTS being 2/3 of the Recuperator. Regarding the other heat exchangers (Precooler and HX-LTS), the duty is lower and the divergence in the temperature profiles higher, which entails to lower sizes.

Table 11. Size of heat exchangers in the PCS (1/4 of power plant).

	HX-HTS	HX-LTS	Recuperator	Precooler
Heat transfer area [m ²]	54,647	846.7	80,395	5,131
H [m]	4.318	0.9378	4.311	0.7666
W [m]	0.6	0.6	0.6	0.6
L [m]	30.8	2.2	45.3	16.3
Number of modules	52	4	76	28
Volume [m ³]	79.71	1.235	117.3	7.484
Hot stream actual pressure drops [kPa]	39.38	2.64	40.04	17.49
Cold stream actual pressure drops [kPa]	40.00	39.97	6.23	500

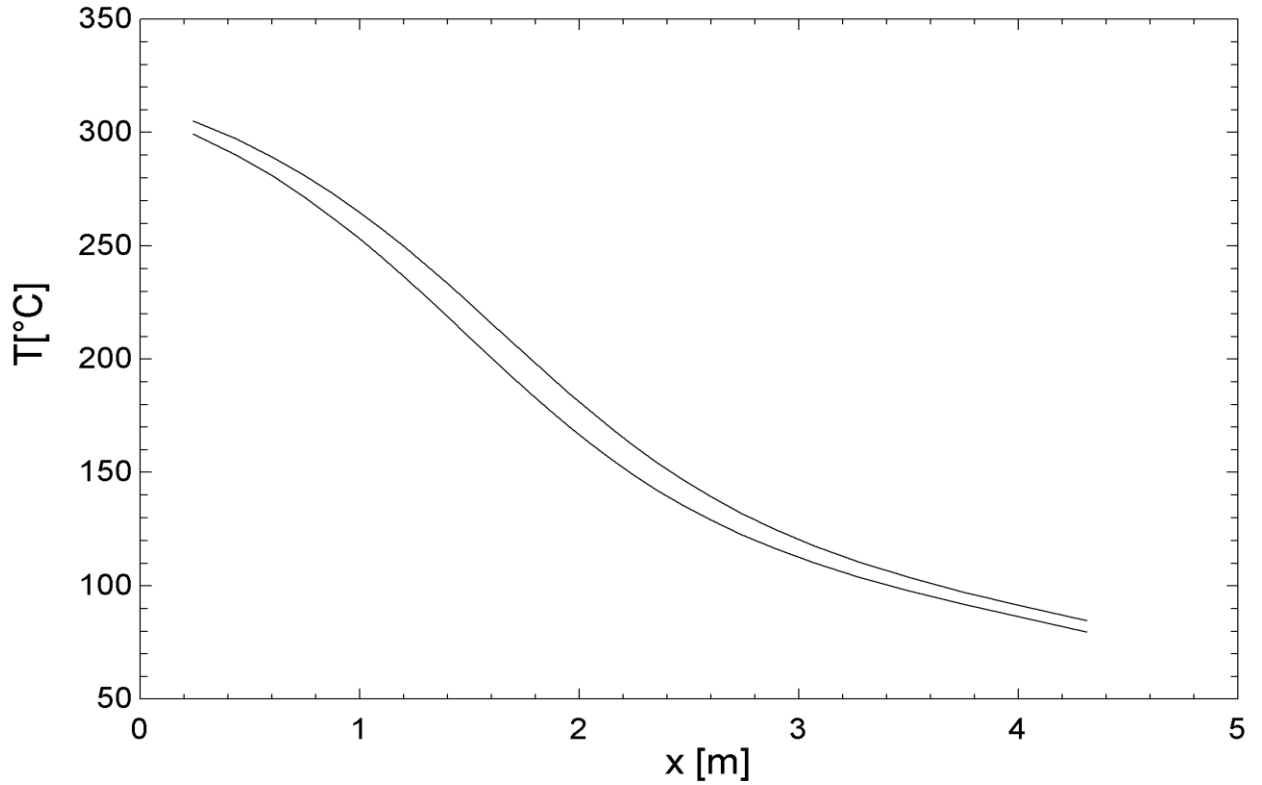


Figure 8. Temperature profiles in the Recuperator versus the flow path.

3.4 Turbomachines

A preliminary design of the turbine has been performed, according to the values found in literature [63], [77]. Nevertheless, in order to increase the efficiency, a rough sensitivity study was carried out, where many parameters were varied: rotational speed, degree of reaction, incidence angle, etc. The tip radius of the turbine was considered to be 600 mm and 3 were the number of stages chosen for this predesign. The main dimensions are 500 mm of length and 1,200 mm of diameter. The design is not optimised, achieving an isentropic efficiency of 65%.

In the compressor, a thorough preliminary study has been carried out, varying many parameters in order to achieve convergence. A high number of stages has been found in literature [78], [79] as a solution to convergence problems. Therefore, the design was performed with the number of 10 stages. The compressor length is 630 mm and the diameter is 800 mm. An efficiency of 70% is achieved, below the objective.

It has to be mentioned that these preliminary designs have been performed only to contribute to the space allocation evaluation. Further optimization is required but the optimized designs will not differ much in terms of size and in this way, these values are suitable for obtaining a rough estimation of their volume. In addition, to prove that the current preliminary designs are adequate, a comparison with similar turbomachines found in the literature can be seen in Table 12 for turbines and Table 13 for compressors. The preliminary designs found in the tables have been selected

according to their proximity to the values of the current design, although in some cases, some data are missing.

Table 12. Turbine characteristics comparison.

	[63]	[80]	[81]	[82]	[10]	[83]	Current design
Mass flow rate [kg/s]	3,749.5	1,950	3,843	-	3,749	3,600	1,517
Rotational speed [rpm]	3,600	3,600	3,600	3,600	3,600	3,000	3,000
Power [W]	370	150	413.4	450	246	479	215
Tip Diameter [mm]	1,200	1,300	960	1,100	1,200	1,310	1,200
Length [mm]	550	-	2,000 ¹	-	550	2,720 ¹	500

¹ Total length

Table 13. Compressor characteristics comparison (AC stands for “Auxiliary compressor” and MC “Main Compressor”).

	[63]		[80]		[81]		[10]		Current design
	AC	MC	AC	MC	AC	MC	AC	MC	Compressor
Mass flow rate [kg/s]	1,175	2,574	780	1,170	2,416	2,416	1,145	2,604	1,517
Rotational speed [rpm]	3,600	3,600	-	-	3,600	3,600	3,600	3,600	3,000
Power [W]	-	-	-	-	55.7	55.7			53
Tip Diameter [mm]	800	800	900	500	520	520	800	800	800
Length [mm]	530	300	-	-	860	860	1,000	3,700	630

3.5 Space allocation

The main restriction in space allocation for the PCS is the footprint [84]. From Table 11 it can be concluded that the largest dimension in the heat exchangers is the length (L), which represents the pile of the different plates of the heat exchanger, that is, a parallel configuration. Besides, the maximum length which can be manufactured is 8.5 m, so larger lengths should be split into the necessary number of parts. In order to reduce the footprint, these parts will be placed close to the others, forming a columns structure. Figure 9 shows the scheme of a PCHE with the main dimensions and the indication of the flow paths. Figure 10 takes the HX-HTS as an example. According to Table 11, the height (H) of the conceptual design is 4.3 m, the width (W) is 0.6 m and the length (L) is 30.8 m. As the height is larger than 1.5 m it is also necessary to split the height in three parts, which will be connected in serial configuration. As explained before, the length is split in four parts, which are connected in a parallel configuration. So, the HX-HTS is arranged in a 3 serial/4 parallel configuration, designated as “3s 4p”. Figure 10 shows the parallel modules in a two columns configuration and Figure 11 shows the plan of the PCS.

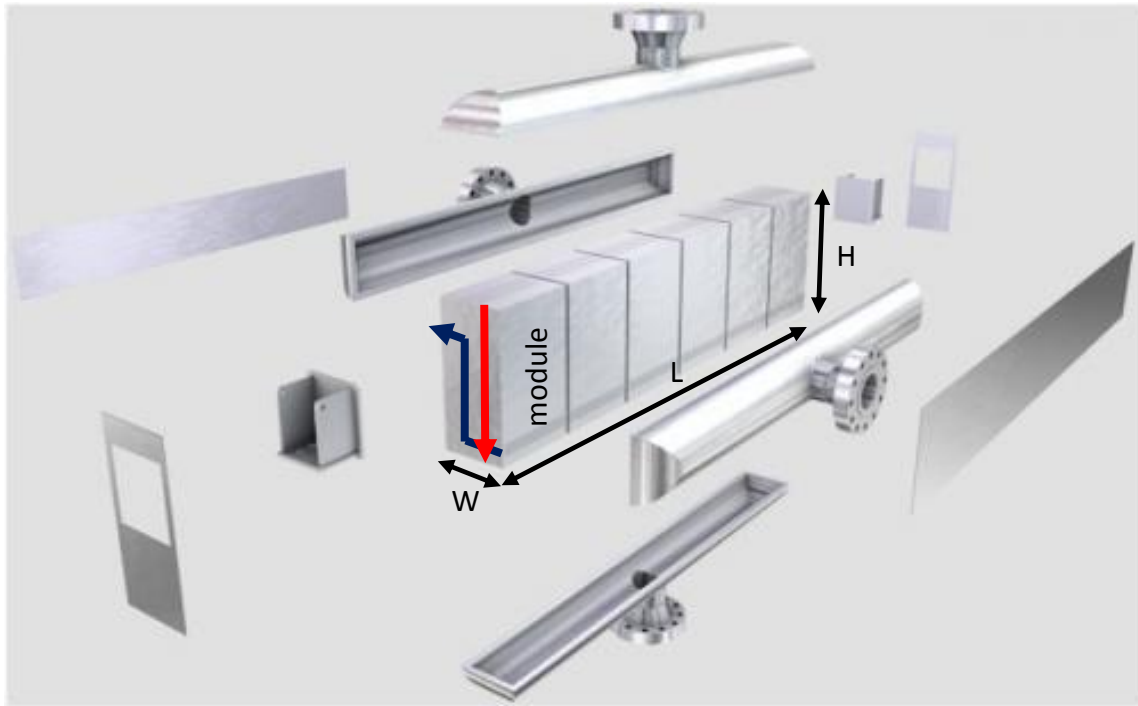


Fig 9. Scheme of a PCHE with indications of the physical dimensions: length (L), width (W) and height (H). The flow paths (blue and red arrows) are vertical (except for the change in direction to connect with the lateral manifolds), being the physical length (L) the height of the heat exchanger from the streams point of view and the physical height (H) the length gone over by the streams (adapted from [35]).

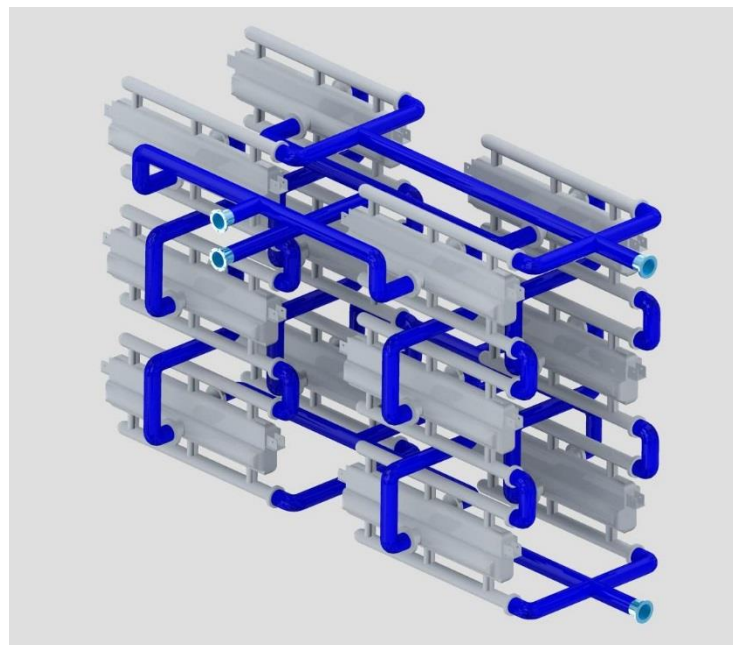


Figure 10. 3D rendering of the HX-HTS heat exchanger showing the “3s 4p” arrangement. Grey pipes represent the manifolds, considered as a part of the heat exchanger; blue pipes represent the connections between each sub-heat exchanger.

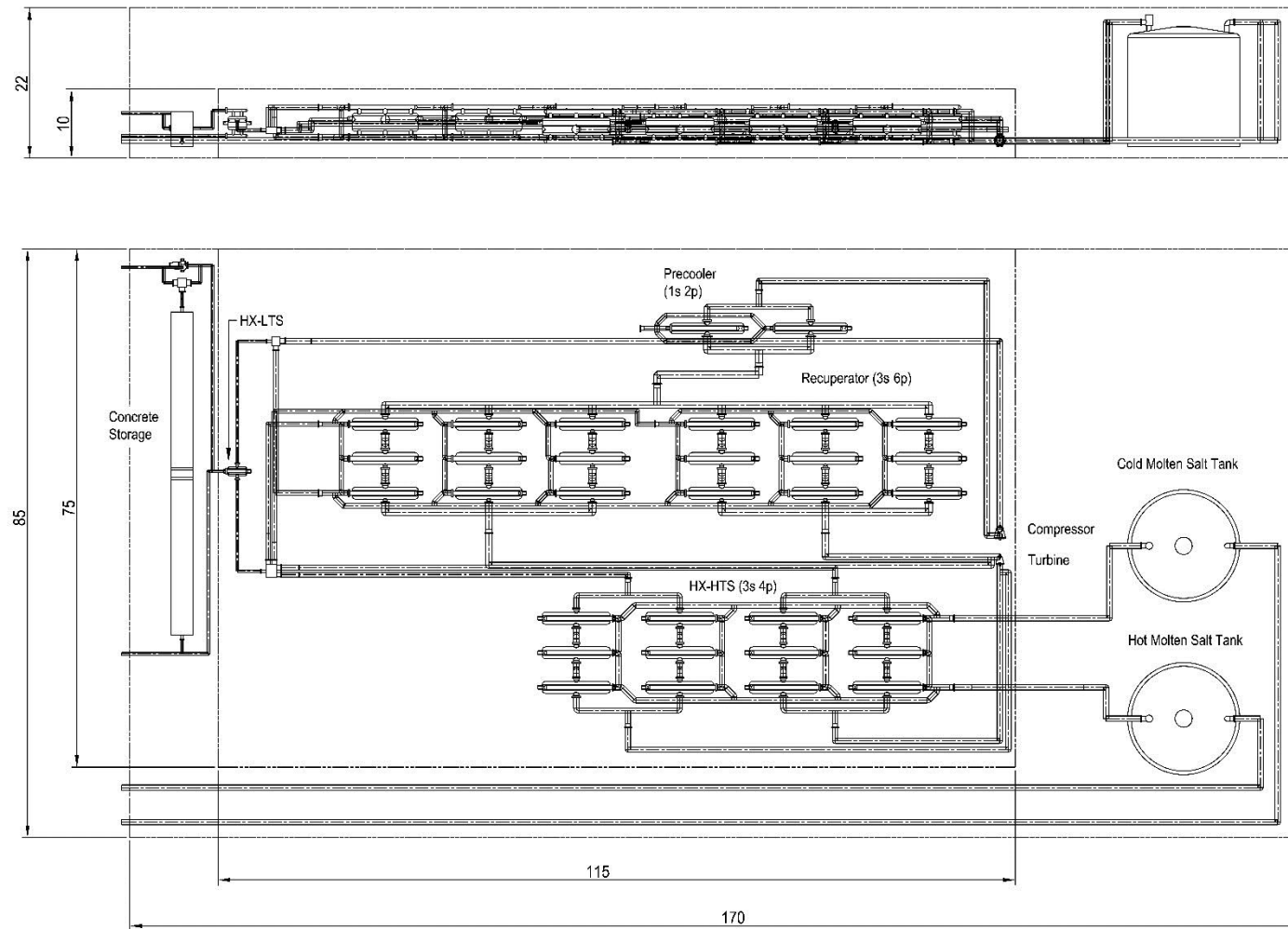


Figure 11. Plan view (lower figure) and elevation view (upper figure) of the PCS (1/4 of power plant).

4. Conclusions

A layout adapted to the specifications of DEMO based on a DCLL breeding blanket has been proposed, including thermal energy storage to obtain continuous electricity production. The average electrical efficiency obtained is 28.25%. The sizing of pipes, heat exchangers, and turbomachines has been conducted. In order to keep the pipe diameters in commercial range, the power conversion system has been split into four equal parts. The allocation of the components of the power conversion system has been arranged to reduce the footprint. In any case, the size of heat exchangers, especially the HTR recuperator, is extremely large, which strongly contrasts with the extremely small size of the turbomachines.

Piping of three different parts of the layout has been carried out: CO₂, water, and molten salt pipes. The maximum allowable velocity criterion has been used, establishing 6 m/s for water, 3 m/s for molten salt, and a velocity that depends on the density for CO₂, with values from 10 to 27 m/s. As pipe material, carbon steel A-106B has been chosen for the water pipes and for the molten salt pipes in which the temperature is under 300°C. Where molten salt has a higher temperature, a stainless steel (SS-347H) has been selected, due to the higher risk of corrosion. For the CO₂ pipes, a high strength nickel alloy (Inconel 740H) has been chosen, used in applications such as ultra-supercritical steam cycles, where high pressure and high temperature are present, as in this case. Nominal diameter and thickness have been determined for every pipe following an iterative procedure, checking that all the requirements (allowable velocity, allowable stress) are met.

A turbine preliminary design has been obtained with an acceptable size and efficiency. Nevertheless, an optimization process is needed in order to approach a better efficiency and will be performed in future works. With respect to the compressor design, the elevated number of restrictions coupled to the CO₂ behavior, makes its design more intricate. An approach with a high number of stages is proposed due to convergence problems. Again, an optimization is necessary to achieve a better value of efficiency. Radial compressors will also be evaluated for this type of cycles. However, these preliminary designs provide a good estimation of the turbomachinery size, which is the main objective of the current work.

5. Acknowledgment

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