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D3.2 – Optimum high-temperature heat exchanger design (solar heat input in power cycle)

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Executive Summary

The current deliverable (D3.2) is entitled “Optimum High-temperature Heat Exchanger Design (Solar Heat Input in Power Cycle)”. It is a public document produced within Task 3.2 “High-temperature heat exchanger design (heat input in power cycle) - recuperative vs. regenerative approach” (WP3) of the ASTERIX-CAESAR project.

The primary objective of WP3 is to identify the optimal heat exchanger (HEX) configuration for both low- and high-temperature applications within the ASTERIX-CAESAR concept. This includes the compression train intercoolers, and the heat transfer apparatuses required for heating and reheating the compressed air – from solar energy - prior to expansion. To achieve this, advanced computational models have been developed to accurately evaluate and design the most promising solutions while also estimating their associated costs.

An innovative concept for a regenerative heat exchanger (HEX) has been proposed and benchmarked against the conventional indirect recuperative HEX in specific scenarios. WP3 is closely linked to WP1, and the evolution of the project has led to an iterative methodology for optimizing heat exchanger configurations, considering the most effective and cost-efficient layouts and characteristics for the overall CAES-CSP plant.

This report focuses on the design of High-Temperature Heat Exchangers (HHEX), whose main objective is heating pressurized air to high temperatures (up to about 750 °C) before expansion in the turbines. Achieving this objective requires the implementation of advanced equipment to transfer heat using air as the heat transfer fluid while managing significant pressure differences between the two streams. In order to achieve a cost-effective solution, different approaches have been analysed: (i) an indirect shell-and-tube design, (ii) a regenerative approach with high specific heat transfer areas and very high heat exchange effectiveness, and (iii) an innovative heat exchanger similar to a conventional Heat Recovery Steam Generator (HRSG), featuring bundles of tubes at varying pressures arranged within a shell through which hot atmospheric air flows.

In the case of the regenerative approach, a length of 9 m has been assumed for both high pressure and low-pressure regenerative beds. For both beds, a flange diameter DN600 has been selected for charging operations. For discharging operations, instead, a flange diameter DN150 and DN300 has been selected for high pressure and low-pressure bed, respectively. An acceptable pressure drop is obtained, and the expected switching time is around 30 min. The resulting total area of heat transfer for one bed is about 1323 m². The total cost of the high-pressure and low-pressure regenerative systems is estimated at about 1 M€ each, i.e. some 1.160 M USD. The total cost of the regenerative system is thus estimated at about 2.32 M USD.

The Shell & Tube solution has a total heat transfer area of some 280 m² and an estimated cost of about 2.56 M USD. Therefore, from the economic point of view, both solutions, the regenerative design as well as the Shell & Tube design have very similar CAPEX, being the regenerative solution slightly cheaper. Nevertheless, considering the higher complexity of design and operation related to the regenerative approach, the classic Shell & Tube design seems to be the preferred choice. Additionally, the needed valves are size-limiting components and are very expensive. The availability of valves imposes a clear upscaling limit, since maximum valve sizes available for such demanding conditions are around DN600 to DN800. The regenerative solution is therefore only viable for the Small-Scale plant.

Heat transfer equipment for Medium and Large-Scale plants operating according to the original layout have been designed by adopting an approach based on Shell & Tube technology. In Medium-Scale plant, two heat exchangers are sufficient to heat and reheat the pressurized air stream, while the Large-Scale plant calls for multiple units operating in parallel arrangement. Estimated capital costs are in the order of 14.0 and 47.4 M USD, respectively.

Concerning the modified plant layout, in addition to the traditional Shell & Tube design an innovative approach based on the well proven HRSG technology has been addressed. The proposed approach is suitable only for Medium and Large-Scale plants. For the Small-Scale Plant,

the classic Shell & Tube solution is fully sufficient as the needed flow cross sections are small enough.

For the Medium-Scale Plant as well as for the Large-Scale Plant, the optimum design solution is the design according to the HRSG approach, that offers advantages in terms of cost and lesser plant complexity.

In conclusion, it must be pointed out that the relevant output of WP3 activities consists in the development of a set of general design indications and guidelines applicable to any plant layout established in compliance with the ASTERIX-CAESAR concept. Moreover, results here presented and discussed constitute the input for activities to be carried out in the last period of WP3.

Nomenclature

ACRONYM	DESCRIPTION
CAES	Compressed Air Energy Storage
TES	Thermal Energy Storage
CSP	Concentrated Solar Power
EU	European Union
HEX	Heat Exchanger
LHEX	Low-Temperature Heat Exchanger
HHEX	High-Temperature Heat Exchanger
U	Global Heat Transfer Coefficient
ΔP	Pressure Drop
RTE	Round-Trip Efficiency
HRSG	Heat Recovery Steam Generator
TOBV	Triple Offset Butterfly Valve
DOFs	Degrees of Freedom

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1 Introduction

1.1 Objectives

The objectives of the deliverables D3.1 and D3.2 include identifying the optimum heat exchanger configuration for both low- and high-temperature applications through a comprehensive design study. The benchmarking of the conventional indirect recuperative heat exchanger against regenerative heat exchangers is performed to assess the relative advantages of each approach. Additionally, an optimum material selection process is undertaken, considering both cost, and performance. Finally, potential process control challenges are identified, and suitable control methods are proposed to ensure robust and efficient operation. Finally, in D3.3 potential process control challenges are identified, and suitable control methods are proposed to ensure robust and efficient operation.

1.2 Activities

According to the proposal, two different approaches are taken into consideration for high temperature heat exchanger design: (i) the classical indirect recuperative approach, where the two air streams are separate from each other and heat transfer happens through duct walls; and the less conventional regenerative approach (ii) where air is heated and cooled in one single duct or vessel while passing through porous heat storage media. Both approaches have been analysed in detail and a fair benchmarking has been done based on the extensive heat exchanger design experience within the consortium (CENER, AAL, ROMA3, PSA).

Regarding the design approach (i), only tubular heat exchanger geometries are viable because of the very high pressure difference of up to 150 bar between the air streams. The classical shell-and-tube heat exchanger design is assumed as the base reference case, including passive heat transfer enhancement methods such as fins. Furthermore, alternative design options based on the Heat Recovery Steam Generators (HRSG) technology have been explored.

While the recuperative configuration (approach i) has been mainly addressed through detailed design analysis and definition of boundary conditions, the regenerative configuration (approach ii) required dynamic modelling to properly capture its transient thermal behaviour. To this end, dedicated distributed-parameter models have been developed in Modelica to simulate the performance of tubular regenerator-type heat exchanger configurations.

For both approaches extensive campaigns aimed at defining the optimal high-temperature heat exchanger configuration design have been carried out for the different plant sizes taken into consideration (small, medium and large-scale plants). Besides heat exchanger specifications, activities have been also addressed to the sizing of air piping and to the selection of the most appropriate valve types. All involved project partners contributed to critically review design activities and results. Such a process has covered material cost considerations, operating temperature limitations, off-design operation, high-temperature oxidation resistance, as well as manufacturing possibilities. Also, recyclability and sustainability are important and are to be evaluated coherently in collaboration with Task T1.6 (WP1).

1.3 Partners involved and role

According to the Grant Agreement, activities related to the optimum design of high-temperature heat transfer equipment have been carried out mainly by CENER and ROMA3. Their contributions focused on the sizing activities have been carried out by using in-house developed codes and commercial tools.

Industrial partners (AAL, NTT and SIW-S) as well as Research Institutions (CIEMAT-PSA, USE and IKTS), on the basis of their expertise, provided valuable suggestions concerning the most suitable design solutions, useful information about costs and checked the feasibility of the proposed designs, taking also into consideration the integration of the heat transfer equipment with the other plant sections.

1.4 Interaction with other WPs

This deliverable is part of WP3, “Advanced Heat Exchangers”, which is closely linked to WP1, “Concept Definition and Techno-Economic Optimization”. WP1 has defined the key boundary conditions, including operating temperatures, pressures, mass flows, and maximum allowable pressure drops, which serve as fundamental inputs for WP3.

Additionally, during the first two years of the project, a continuous iterative methodology has been implemented across WP1, WP3, and WP4. The interdependencies between these work packages necessitate an ongoing exchange of results, ensuring that design refinements in one WP align with the evolving requirements of the others.

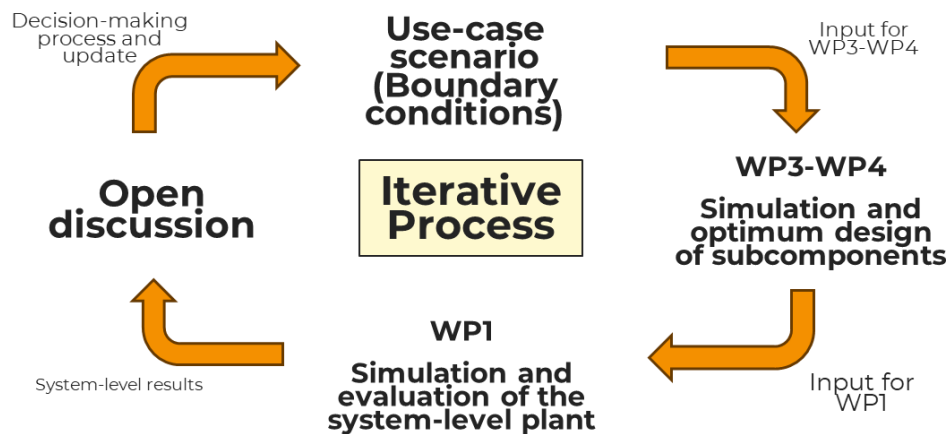


Figure 1. Iterative methodology carried out for interaction between WP1-WP3-WP4.

1.5 Deliverable outline and contents

The deliverable is structured as follows:

Section 2 introduces the ASTERIX-CAESar plant concept. Section 3 talks about general advantages and disadvantages of regenerative and recuperative heat exchangers. Section 4 deals with the regenerative heat exchanger design for the Small-Scale Plant only, since the available valves needed for flow control limit the application at higher power rating. Section 5 presents the recuperative heat exchanger design for all three power ratings. The original and the modified plant layouts are addressed. Designs based on Shell & Tube and HRSG concepts are presented and compared. Section 6 concludes.

2 ASTERIX-CAESar plant concept

The ASTERIX-CAESar concept aims to facilitate large-scale grid energy storage by integrating Compressed Air Energy Storage (CAES) with Concentrated Solar Power (CSP). The primary operation of the system involves purchasing electricity during periods of low electricity prices to drive a compression train. The compressed air is subsequently stored in an air reservoir, which can either be underground caverns or artificial aboveground vessels. During daylight hours, when solar radiation is abundant, the incident solar energy is converted into heat and transferred to the air through an open volumetric air receiver. This thermal energy is stored in a high-temperature thermocline Thermal Energy Storage (TES) system, utilizing a low-cost packed bed of rocks. During periods of peak electricity demand, when the need for power is elevated, the compressed air from the reservoir is released, and the thermal energy stored in the TES is transferred to the air via a heat exchanger. This process raises the air temperature, thereby enhancing the power

output of the system.

Regarding the low-temperature HEX (D3.1), different approaches have been analysed such as air-air HEX for reusing the compression heat in discharging process (see Figure 2) or for supplying heat to an industrial process (see Figure 4).

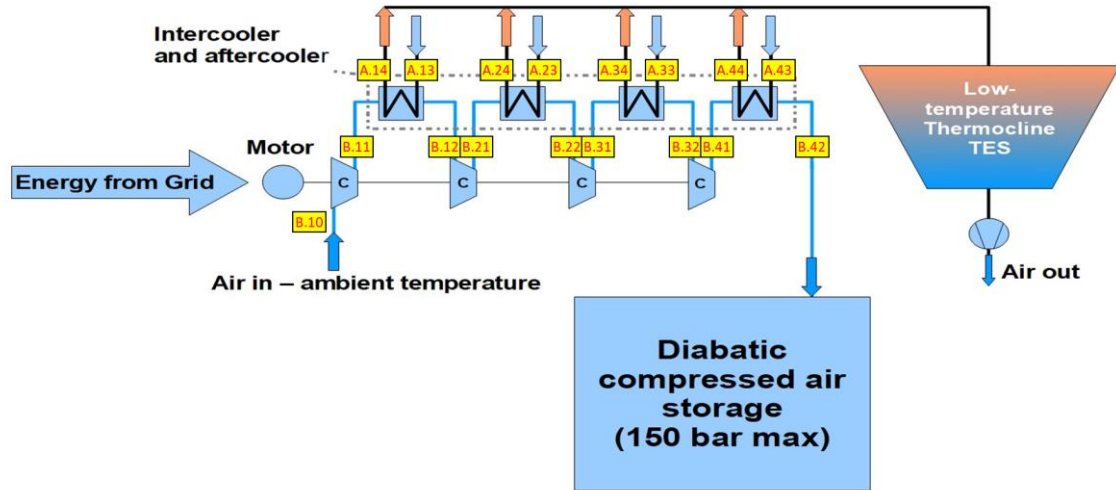


Figure 2. ASTERIX-CAESar original plant scheme with low-temperature TES. Charging process.

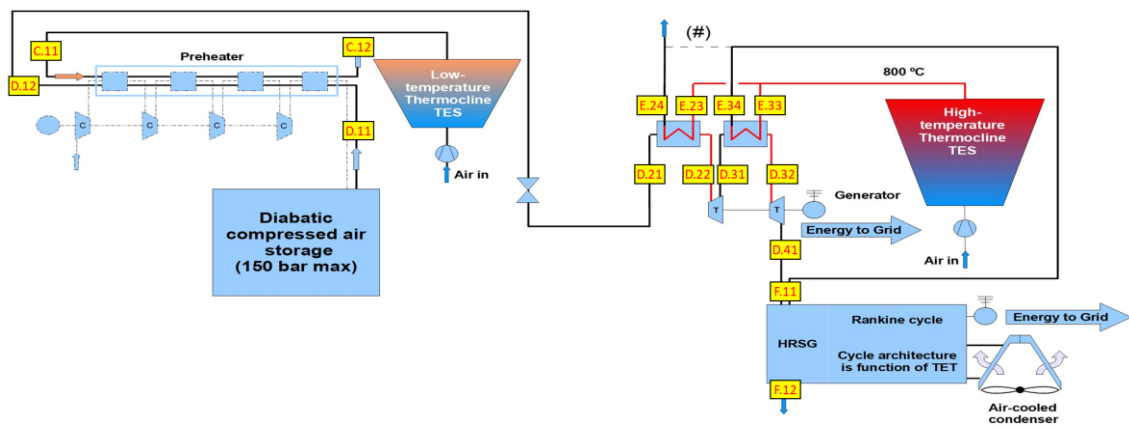


Figure 3. ASTERIX-CAESar original plant scheme with high-temperature shell-and-tubes HEX. Discharging process.

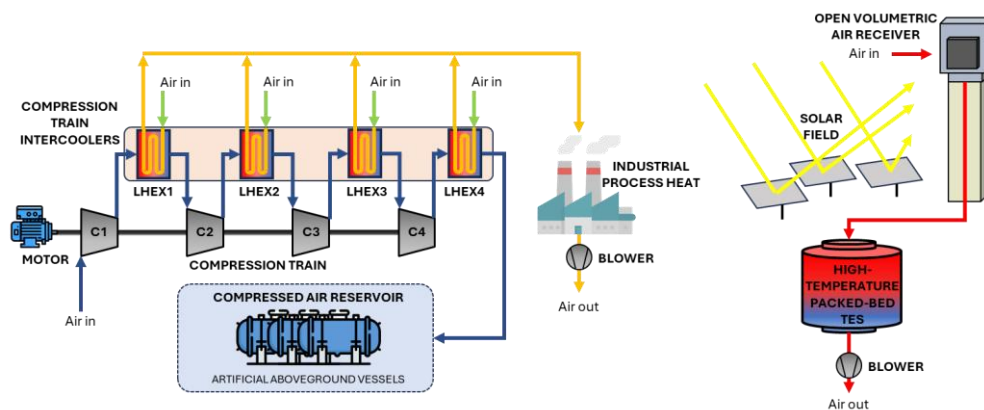


Figure 4. Modified plant concept with connected industrial process. Charging process.

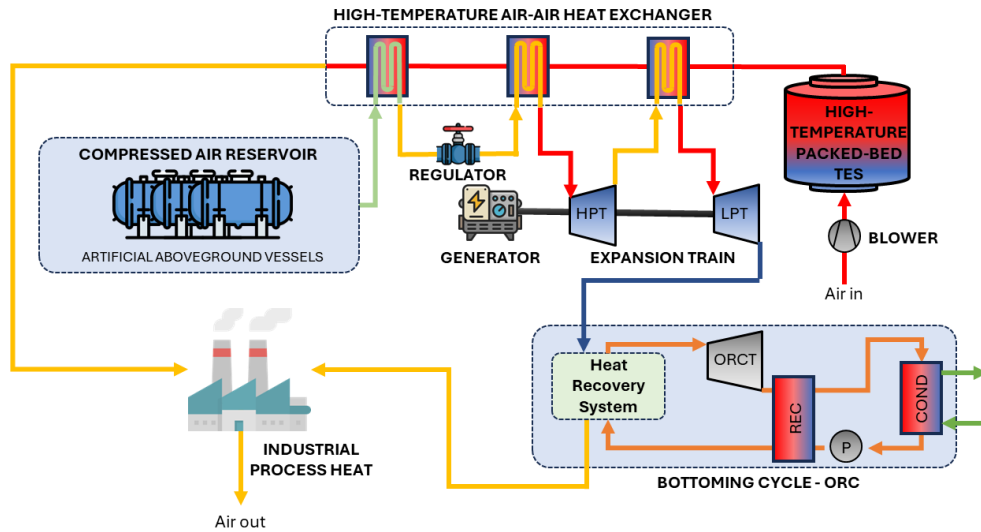


Figure 5. Modified plant concept with high-temperature air-air HEX similar to HRSG and with connected industrial process. Discharging process.

Regarding High-Temperature HEX (HHEX), different approaches have been evaluated to achieve the best option in both technical and economic terms. The original concept (see Figure 3) has two shell-and-tubes HHEX fed by two parallel atmospheric air streams that heat the pressurized air to both high- and low-pressure turbines, respectively. Atmospheric air streams exit the two HHEXs at a temperature of some 400-450 °C, suitable to feed the Heat Recovery System of the ORC bottoming cycle. This approach is based on conventional technology, although some modifications are required to match the specifications of pressure drop on the atmospheric side and higher thicknesses are required to handle with the high pressure inside the tubes.

Figure 5 presents the alternative plant layout, where a single stream of hot ambient pressure air is used to heat and re-heat the pressurised air entering the High Pressure (HPT) and Low Pressure Turbine (LPT). In this way, a lesser amount of atmospheric air is used. The air is cooled to some 150-160° C and then supplied to industrial users. According to WP1 findings, such a lay out is expected to bring to an improved overall techno-economic performance and, therefore, has been taken into consideration for further investigations.

To explore the potentiality of the proposed concepts in a wide range of possible applications, three different power classes for charging and discharging operation are addressed: Small Size Plant (1MW), Medium Size Plant (20 MW) and Large Size Plant (100+ MW). For each power class, main process quantities have been established by the consortium partners in WP1. Such quantities are used as inputs for heat transfer equipment sizing

3 Regenerative and recuperative approach

The regenerator is a storage-type heat exchanger. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. To have continuous operation, either the matrix must be moved periodically into and out of the fixed streams of gases, as in a rotary regenerator, or the gas flows must be diverted through valves to and from the fixed matrices as in a fixed-matrix regenerator [1]. In the ASTERIX-CAESAR project, fixed-matrix-type regenerators are addressed. Major advantages of the regenerators are the following [1]:

- A much more compact surface may be employed than in a recuperator, thus providing a reduced exchanger volume for given exchanger effectiveness and pressure drop and thereby making a regenerator economical compared to an equivalent recuperator.

- The cost of manufacturing such a compact regenerator surface per unit of heat transfer area is usually substantially lower than that for the equivalent recuperator.
- Similarly, material cost could be lower in a regenerator than in a recuperator. Hence, a compact regenerator usually has a smaller volume and is lower in weight than an equivalent recuperator.
- As it is a direct-contact heat exchanger, its effectiveness can be very high.

Regenerators are well known from the application in Stirling engines and blast furnaces for iron production, for example.

For example, in a Stirling engine, the regenerator occupies a volume around 120 cm³, the highest working temperature of this element is around 700 °C, and the heating/cooling cycle is really fast (50 Hz for engines that operate at 1500 rpm). Taking into account these specifications, mentioned developments have been always carried out in metallic materials, either by sintering or stacking wire mesh, or using metallic foams, that withstand required temperature and charge/discharge cycles, avoiding thermo-mechanical failures (fatigue, etc.).

On the other hand, in the hot blast process in blast furnaces, and open hearth furnaces used for glass production at industrial scale, the heat of hot exhaust gases from combustion is first transferred to the regenerative bed (firebrick chambers) and then it is used for preheating the inlet air for the combustion system. Working temperature in this application is much higher (1300 °C), charging/discharging cycle time is very slow (several hours), and regenerators are much bigger (2000 m³). Under these operating conditions, traditional furnace ceramic materials provide a good solution since they withstand very high temperatures, and in these cases the material is not subjected to high mechanical loads.

With respect to ASTERIX-CAESAR's regenerative system for heating of compressed air, around 550 to 750 °C of maximum nominal operating temperature is expected. The charging of the regenerator happens at ambient pressure, when hot air, coming from the TES system, heats the regenerative matrix. The regenerator is cooled under pressurized conditions, when cold pressurized air from the CAES enters the regenerator's matrix and is heated to high temperature. The pressurization and depressurization sequence requires several valves that must withstand high pressure (up to about 40 bar as maximum turbine inlet pressure) and high temperature (up to about 750°C) at the same time, which are very demanding conditions. The critical components are the valves, which impose a clear upscaling limit, since maximum valve sizes available for such demanding conditions are around DN600 to DN800. In order to provide a continuous heat transfer process, at least two regenerators need to be placed in parallel. Charge and discharge cycle times are expected to be most practical between 30 min and 1 hour. For the specific design, general knowledge of regenerator design can be applied [1]. Due to the high temperatures observed, matrix materials need to be selected well to guarantee long lifetime and no degradation during operation. Selected natural rocks may be used, as well as slag, ceramic or glass. Nevertheless, due to the high operating pressure (up to about 40 bar) special attention must be given to the design of the regenerator casing and valve system for efficient flow control.

On the other hand, the recuperative type heat exchanger will be based on tubular configurations due to the high operating pressures. Shell & Tube is the most widely used technology in heat exchanger manufacturing and represents in a wide range of applications the most effective solution in terms of techno-economic performance.

However, it has to be pointed out that these positive features subside when design and operating conditions become more challenging, particularly when the temperature approach between hot and cold streams reduces, heat transfer coefficients are not particularly high and volumetric flow rates are huge.

Generally speaking, the thermodynamic performance of a system improves by reducing the temperature difference between hot and cold stream in each heat transfer device. Therefore, especially for the modified plant layout, temperature approaches between hot and cold stream in the different heat transfer units are taken as low as possible, consistent with the need to contain

the extent of heat transfer surface and, consequently, the capital cost.

Values expected for the overall heat transfer coefficient are not particularly brilliant. The low value of the heat transfer coefficient on atmospheric air side constitutes a first barrier to improving the overall coefficient. Actions leading to a heat transfer enhancement on atmospheric air side (i.e. the use of extended surfaces and/or the increase of air velocity) has to be carefully checked because such actions lead to an increase of the pressure drop, on which depends the power absorbed by the fan to pass the air through the heat exchanger and the packed bed constituting the TES. Based on the results from WP1, the auxiliary consumptions of the plant can account for more than 10–15% of the gross electricity produced. Therefore, minimizing this pressure drop has been a crucial aspect in the design process of the high-temperature heat exchanger.

Moreover, with reference to Medium and Large Size plants, the relevant mass flow rates, together with the high thermal loads can lead to the need of huge heat transfer area resulting in a cumbersome design at limit of feasibility or in the necessity of installing several units in parallel arrangement.

Finally, a crucial design aspect is related to the choice of most suitable materials in order to conjugate an adequate level of reliability with affordable capital costs. Working temperature up to 750- 800° C are expected. Most material commonly used for lower temperature applications (steel, copper, aluminium) lose their strength at high temperature, typically when the operating temperature exceeds 500°C and particularly above 600°C. Depending on operating temperature and pressure, materials can range from ferritic steels to Ni-based alloys. The prices of material vary from 2-3 USD/kg for ferritic steels to some 70 USD/kg for Ni-based alloys. In addition, taking the characteristics of the addressed application into account consideration (i.e. cyclic operations), materials should be resistant to thermal fatigue and creep rupture for a combination of high temperature and high-pressure operating conditions.

In conclusion, the particular application here considered poses relevant design challenges, that may require innovative solutions for their overcoming.

3.1 Qualitative comparison between the two approaches

Major advantages of the regenerators in respect to recuperative heat transfer are listed below:

- A much more compact surface may be employed than in a recuperator, thus providing a reduced exchanger volume for given exchanger effectiveness and pressure drop and thereby making a regenerator economical compared to an equivalent recuperator.
- The cost of manufacturing such a compact regenerator surface per unit of heat transfer area is usually substantially lower than that for the equivalent recuperator.
- Similarly, material cost could be lower in a regenerator than in a recuperator. This aspect is particularly important in the present application, where, due to the high operating temperature levels, costly material have to be employed in manufacturing the regenerative heat transfer equipment.
- Hence, a compact regenerator usually has a smaller volume and is lower in weight than an equivalent recuperator.
- As it is a direct-contact heat exchanger, its effectiveness can be very high.

Conversely, the regenerative approach presents the following drawbacks:

- Though several noticeable applications of the regenerative concept can be found in the real world, such a technology is to be considered less consolidated in respect to the traditional recuperative approach.
- The maximum size of a single regenerative unit is limited by the maximum size of commercially available valves. Therefore, multiple units in parallel arrangement could be required also for small plant sizes.

- System based on the regenerative concept are expected to be more complex in respect to those based on recuperative heat transfer equipment.

The definitive design choices have been made on the basis of careful investigations carried out taking all the above aspects into consideration.

4 Regenerative approach

The idea of using a regenerative system to transfer heat between two air streams at different pressures originated from the predecessor project of ASTERIX-CAESAR, the CAPTURE (Competitive Solar Ar Power Towers) project [2]. The CAPTURE project investigated a regenerative system working in alternating modes (atmospheric heating, pressurized cooling – see Figure 6) in order to drive a solar-only powered combined cycle. This approach aimed at decoupling the high temperature and the high heat flux part (solar receiver) from the high-pressure part (compressed air stream of the Brayton cycle) via an air-air regenerative heat exchanger.

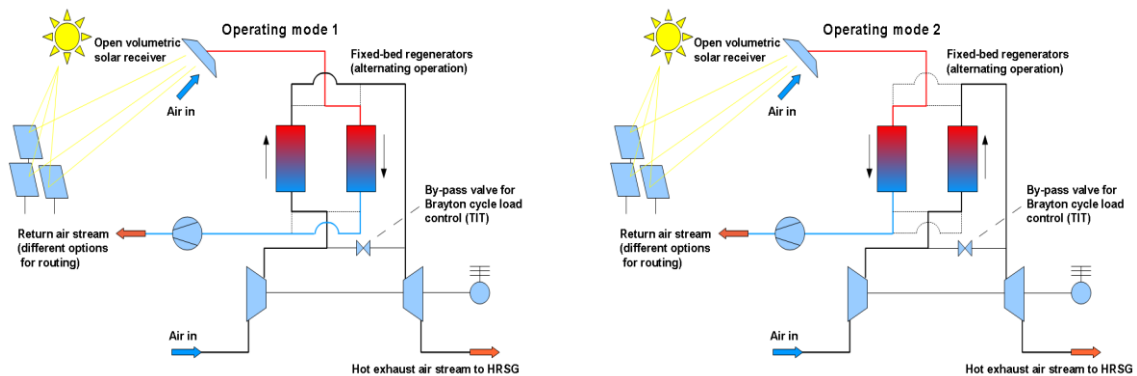


Figure 6: Innovative coupling of open volumetric air receiver and Brayton cycle (CAPTURE concept [3])

The main conclusion of the CAPTURE project was that such a regenerative system may be only suitable for small-scale application (due to valve sizes available and bed switching strategy). However, the alternating pressurization and depressurization sequence causes a non-negligible parasitic power consumption (decrease in efficiency) and significantly increases control complexity of the Brayton cycle, which needs to be offset by reduced costs, compared to conventional shell-and-tube heat exchangers. In the CAPTURE project, due to lack of further development time, no conclusion could be drawn regarding the economic suitability of such a regenerative system.

The ASTERIX-CAESAR project aims at analyzing this approach further, since the operational complexity is considerably simplified (compared to the CAPTURE concept) when connecting a CAES system to the plant. The main difference is that the pressure is already built up in the CAES vessel and the pressurization process and the regenerative beds' switching strategy is straightforward, implying only a certain loss of pressure energy and simple valve actuation (on/off). However, the upscaling limit, which is imposed by the available valve sizes, remains and such a regenerative system seems only be viable for the small-scale application (close to about 1 MWe). The cost analysis and comparison to conventional recuperative systems will provide sufficient details to draw conclusions.

Like the low-temperature regenerative system for intercooling, also here a tubular configuration in U-shape is proposed (see Figure 7). The U-shape design has the advantage of having the inlet and outlet flanges close together as turbine stages are usually not far from another and additionally, the U-shape allows thermal expansion of the regenerative system, without large displacements of turbine inlet flanges during warm up of the plant.

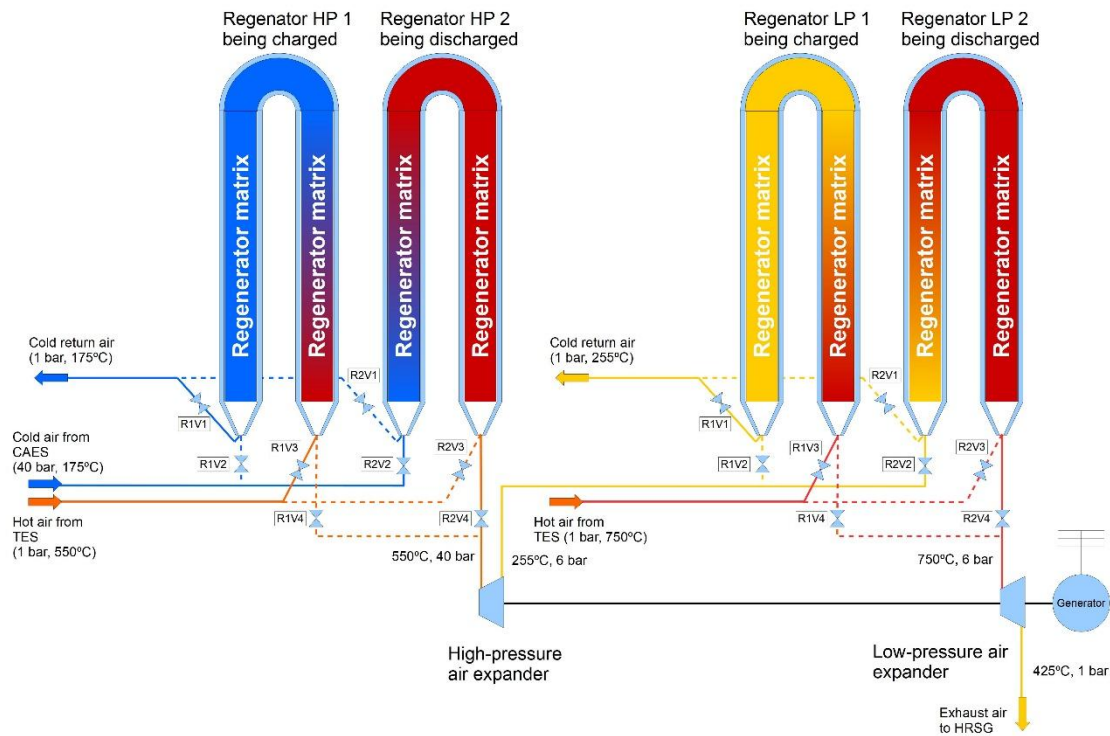


Figure 7: High-temperature regenerative system scheme for heating inlet air stream of high-pressure expander and low-pressure expander

Nominal diameters of flanges need to be chosen such that reasonable flow velocities are obtained in the piping system. The critical parameter is the flow velocity under atmospheric charging conditions, which needs to be kept low. Table 1 shows flow velocities in the piping system for the high-pressure regenerative bed and the low-pressure regenerative bed at the highest operating temperatures observed. At near atmospheric pressure charging conditions, DN600 piping is appropriate. While DN150 would be sufficient for discharging the high-pressure regenerative system, the piping size of the low-pressure discharging system would need to be increased to DN300.

Table 2 and Table 3 show a list of valves that are needed to control the air flow in the high-pressure and the low-pressure regenerative system of 2 parallel regenerative beds in each case.

Note that the temperature-pressure rating of connecting flanges needs to be considered according to EN 1092-1. For example, when considering R1V1 in Table 2, the maximum operating pressure would be 40 bar, and the maximum operating temperature would be 550°C (in the worst case, if thermocline degraded until the bottom end, or when overcharging). When assuming AISI304 (1.4301) as flange material, the maximum allowed pressure at 550°C for PN40 would be 17.5 bar. The maximum allowed pressure at 550°C for PN63 would be 27.6 bar. The maximum allowed pressure at 550°C for PN100 would be 43.8 bar. Therefore, a nominal pressure of PN100 is needed as a minimum when designing the regenerative system before the high-pressure turbine stage.

Table 1: Air speeds in regenerative system piping (charging/discharging highest temperature case)

Regenerator (operating mode)	Nominal diameter piping (mm)	Air mass flow (kg/s)	Operating temperature (°C)	Operating pressure (bar)	Flow velocity (m/s)
High-pressure (charging)	DN600	1.4	550	1	11.8
Low-pressure (charging)	DN600	1.4	750	1	14.7
High-pressure (discharging)	DN150	1.4	550	40	4.4
Low-pressure (discharging)	DN300	1.4	750	6	9

The following tables (Table 2 and Table 3) shows the valve specifications for the high-pressure regenerative system and the low-pressure regenerative system, respectively.

Table 2: List of regenerative system valves before high-pressure turbine (HP regenerative system)

Valve number	Nominal diameter flange (mm)	Nominal pressure flange (bar)	Max differential pressure when closed (bar)	Nominal mass flow (kg/s)	Operating pressure opened (bar)	Operating pressure closed (bar)	Max operating temperature (°C)
R1V1	DN600	PN100	39	1.4	1	40	550
R2V1	DN600	PN100	39	1.4	1	40	550
R1V2	DN150	PN100	39	1.4	40	40	550
R2V2	DN150	PN100	39	1.4	40	40	550
R1V3	DN600	PN100	39	1.4	1	40	550
R2V3	DN600	PN100	39	1.4	1	40	550
R1V4	DN150	PN100	39	1.4	40	40	550
R2V4	DN150	PN100	39	1.4	40	40	550

Table 3: List of regenerative system valves before low-pressure turbine (LP regenerative system)

Valve number	Nominal diameter flange (mm)	Nominal pressure flange (bar)	Max differential pressure when closed (bar)	Nominal mass flow (kg/s)	Operating pressure opened (bar)	Operating pressure closed (bar)	Max operating temperature (°C)
R1V1	DN600	PN100	5	1.4	1	6	750

R2V1	DN600	PN100	5	1.4	1	6	750
R1V2	DN300	PN100	5	1.4	6	6	750
R2V2	DN300	PN100	5	1.4	6	6	750
R1V3	DN600	PN100	5	1.4	1	6	750
R2V3	DN600	PN100	5	1.4	1	6	750
R1V4	DN300	PN100	5	1.4	6	6	750
R2V4	DN300	PN100	5	1.4	6	6	750

The diameter in the regenerative bed itself is even more critical than in the piping system. The whole system pressure drop should be around 5 to 8 kPa.

It is very likely that the diameter of the regenerative beds has to be chosen larger than in the case of the low-temperature regenerative system (D3.1). To find a suitable size that also agrees with component sizes available on the market, some parametric simulations have been performed varying bed length and diameter (Table 5).

The modelling methodology of the regenerative bed's thermocline performance is described in the next section.

4.1 The modelling methodology

Given the importance and already widespread usage of Modelica [4] as an interdisciplinary and extremely flexible modeling language, it has also been applied for this work's modeling procedure. Modelica is a multi-purpose physical system modeling language and has been developed in an international effort to unify already existing similar modeling approaches and to enable developed models and model libraries to be easily exchanged. The concept is based on non-causal models featuring true ordinary differential and algebraic equations, i.e. differential-algebraic equation (DAE) systems [4].

The modified Schumann equations used in this work

In the following, the governing partial differential equations of a one-dimensional thermocline packed-bed TES system will be stated. Subsequently, their discretization, implementation and numerical integration will be explained in detail.

$$M_f c_f \frac{\partial T_f}{\partial t} + M_f c_f v \frac{\partial T_f}{\partial x} = h_{fs} P_{fs} (T_s - T_f) + k_{fe} A_c \frac{\partial^2 T_f}{\partial x^2} - U P_w (T_f - T_a) \quad (1)$$

$$M_s c_s \frac{\partial T_s}{\partial t} = h_{fs} P_{fs} (T_f - T_s) + k_{se} A_c \frac{\partial^2 T_s}{\partial x^2} \quad (2)$$

where

$$M_f = \epsilon A_c \rho_f \quad (3)$$

$$M_s = (1 - \epsilon) A_c \rho_s \quad (4)$$

$$v = \frac{\dot{m}_f}{\rho_f \epsilon A_c} \quad (5)$$

The above set of coupled partial differential equations (PDEs) has to be discretized in order to obtain a set of ordinary differential equations (ODEs) suitable for the model formulation in Modelica. This has been done according to the finite volume method (FVM). In particular, the developed model is based on well-proven and freely available model structures and base classes, as provided by the Modelica Standard Library (MSL) [5].

The 1-D finite volume approach according to the MSL and the implementation of the

thermocline flow model

The MSL offers a rigorous implementation of a one-dimensional fluid flow model according to the finite volume method [6], which forms the basis of the thermocline flow model developed in this work. Basically, the total flow volume is discretized in n finite control volumes along the flow direction. In order to avoid a multiple definition of the basic mass and energy balances, the one-dimensional thermocline flow model extends from the MSL base class “partial distributed volume”. This generic base class defines the mass and energy balances in terms of net mass flow, net enthalpy flow, net heat flow $\dot{Q}_{net}$ and net work flow $\dot{W}_{net}$, for each control volume i (see Figure 8). This can be written as follows:

$$m_i = V_i \cdot \rho_i \quad (6)$$

$$\frac{dm_i}{dt} = \dot{m}_{a,i} - \dot{m}_{b,i} \quad (7)$$

$$U_i = m_i \cdot u_i \quad (8)$$

$$\frac{dU_i}{dt} = \dot{m}_{a,i} \cdot h_{a,i} - \dot{m}_{b,i} \cdot h_{b,i} + \dot{Q}_{net,i} + \dot{W}_{net,i} \quad (9)$$

Note: for a positive mass flow at the right control volume boundary ($\dot{m}_{b,i} > 0$, see Figure 8), $h_{b,i}$ is the specific enthalpy leaving the control volume, which corresponds to the specific enthalpy h_i of the control volume i ; for $\dot{m}_{b,i} < 0$, $h_{b,i}$ is the specific enthalpy h_{i+1} of the corresponding adjacent control volume upstream; for $\dot{m}_{a,i} > 0$, $h_{a,i}$ is the specific enthalpy h_{i-1} of the corresponding adjacent control volume upstream; for $\dot{m}_{a,i} < 0$, $h_{a,i}$ is the specific enthalpy h_i of the control volume i ; the net work flows $\dot{W}_{net,i}$ are set to zero in this work.

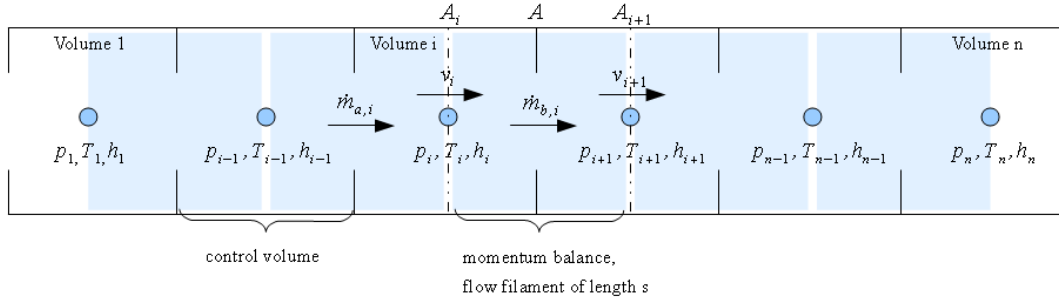


Figure 8: Finite volume discretization scheme according to the staggered grid approach [7]

To correctly implement Eq. (1), the net heat flow term $\dot{Q}_{net}$ of each control volume (in Eq. (9)) has to be defined correspondingly. According to Eq. (1), the net heat flow $\dot{Q}_{net}$ of each control volume depends on three factors, namely (i) on the convective heat transfer from the fluid to the solid phase (or vice versa), (ii) on the heat transfer via axial conduction in the liquid phase, and (iii) on the heat loss to the ambient:

$$\dot{Q}_{net} = \dot{Q}_{convective} + \dot{Q}_{net\ conductive} - \dot{Q}_{heat\ loss} \quad (10)$$

The forced convective heat transfer model

The forced convective heat transfer model has been implemented as a replaceable sub-model, similar to the corresponding wall heat transfer model in a simple tube model as explained in Ref. [8]. Basically, this model calculates the heat flow between the fluid and the solid via estimating the fluid-to-solid heat transfer coefficient h_{fs} using a Nusselt number correlation as published by Alazmi & Vafai [9]:

$$Nu = \frac{h_{fs} nc dp}{k_f} = 2 + 1.1 Re^{0.6} Pr^{1/3} \quad (11)$$

Additionally, since the particles are considered as lumped capacities at uniform temperature (see Section 0), the final heat transfer coefficient is obtained via a dimensionless correction factor that is a function of the Biot number as given in Eq. (7) of Ref. [10]. This defines the forced convective heat flow between the solid and the fluid as follows:

$$\dot{Q}_{convective} = \frac{h_{fs} nc}{1 + \frac{Bi}{5}} A_{fs} (T_s - T_f) = h_{fs} A_{fs} (T_s - T_f) \quad (12)$$

The model of the axial conduction within the fluid

Like the convective heat transfer, also the axial conduction has been implemented in a specific sub-model class that is then instantiated within the thermocline flow model. Basically, this model calculates the net heat flow for each control volume due to axial conduction according to a staggered grid approach, i.e. the axial conduction sub-model calculates the heat flows between the temperature nodes defined by the thermodynamic states (shaded dots in Fig. 1) within each control volume and determines then the final net heat flow for each volume via the difference between entering and leaving conductive flows. For instance, the heat flow $\dot{Q}_{conductive}$ between volume i and $i+1$ is calculated according to the following equation:

$$\dot{Q}_{conductive} = \frac{k_{fe,i} + k_{fe,i+1}}{2} \frac{A_c}{\Delta x} (T_{f,i} - T_{f,i+1}) \quad (13)$$

where k_{fe} is defined according to Ref. [9] as follows:

$$k_{fe} = \epsilon k_f \quad (14)$$

The model of the heat loss to the ambient

The model of the heat loss to the ambient has been implemented according to a quasi-steady approach considering an overall heat transfer coefficient U . This defines the heat flow to the ambient per control volume as follows:

$$\dot{Q}_{heat loss} = UA_w(T_f - T_a) \quad (15)$$

It has to be noted that only considering the heat loss between the fluid and the ambient is a simplification of the actual problem (since, strictly speaking, also the solid medium is in contact with the tank's wall), especially at offline mode of the storage system where the mass flow is set to zero and the heat transfer between the fluid and the solid is not dominated by convection. Nevertheless, this simplification has shown to be acceptable when validating the model against experimental data provided by Pacheco et al. [11].

The pressure drop model

The pressure drop across a packed bed may be calculated according to the following correlation as published by Ergun [12]:

$$\frac{\Delta p}{L} = 150 \frac{(1-\epsilon)^2}{\epsilon^3} \frac{\mu_f v_e}{d_p^2} + 1.75 \frac{1-\epsilon}{\epsilon^3} \frac{\rho_f v_e^2}{d_p} \quad (16)$$

The 1-D model of the packed-bed – A distributed lumped capacitance approach

Having formulated all the necessary equations for describing the fluid flow within the thermocline thermal energy storage system, the solid medium (the packed bed) inside the storage tank has to be defined to complete the model. This has been done according to a distributed lumped capacitance approach, i.e. the solid filling is discretized in sections along the flow assigning one

representative temperature variable to each of the discrete sections and lumping the solid mass per discrete section in this temperature node.

The thermal conduction within the solid material is considered in the same way as for the fluid, i.e. applying Eq. (13) correspondingly for the solid material, obtaining the heat flows between two neighboring nodes as follows:

$$\dot{Q}_{conductive, s} = \frac{k_{se,i} + k_{se,i+1}}{2} \frac{A_c}{\Delta x} (T_{s,i} - T_{s,i+1}) \quad (17)$$

where k_{se} is defined according to Ref. [9] as follows:

$$k_{se} = (1 - \epsilon) k_s \quad (18)$$

In order to connect the model of the solid medium to the fluid flow model, i.e. to the convective heat transfer sub-model, a Modelica heat connector (red squares in Fig. 2) is added to each node of the distributed lumped capacitance model. Thus, the governing equation for one of the n solid material nodes can be formulated as follows, using $\dot{Q}_{convective}$ from Eq. (12):

$$C_s \cdot \frac{dT_s}{dt} = \dot{Q}_{net\ conductive, s} - \dot{Q}_{convective} \quad (19)$$

Media properties

Within the MSL, all specific media property functions are decoupled from the library components by defining a replaceable “medium package” in each of them. Basically, all fluid property function names and interfaces are defined within the base class “partial medium”. In order to allow for a full replaceability, each specific medium model extends from this base class the “partial medium” and defines the specific medium related relationships by redeclaring each necessary medium property function. Thus, every single component of the library that covers the modeling of fluid is not limited to a single specific medium. In fact, it can easily be adapted for the use of different media, by simply replacing the default medium package when instantiating the final model.

In this work, two different media models were used. For the model validation, a model for Hitec XL has been implemented according to Siegel et al. [13]. For the subsequent application of the model with air as HTF, air has been modelled as ideal gas, according to Refs [14, 15]. Note that the model has been also validated with air as HTF in the work of Ramacciotti [16].

The solid media properties have been implemented similarly, also providing a replaceable solid medium model for the filler material. In this case, the solid medium properties have been implemented independent of temperature according to Xu et al. [17] for quartzite rock.

The final air-rock thermocline thermal energy storage model

The final comprehensive Modelica class of the thermocline thermal energy storage system consists of the thermocline flow model as described in Section 0 and the model of the solid medium as described in Section 0. When instantiating the model, the fluid medium model and the solid medium model are re-declared according to the specific modeling needs. Figure 9 displays the model scheme.

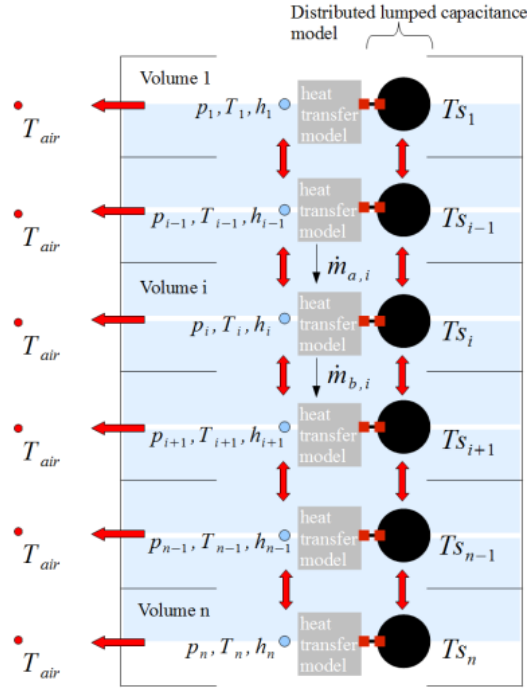


Figure 9: Thermocline TES model scheme

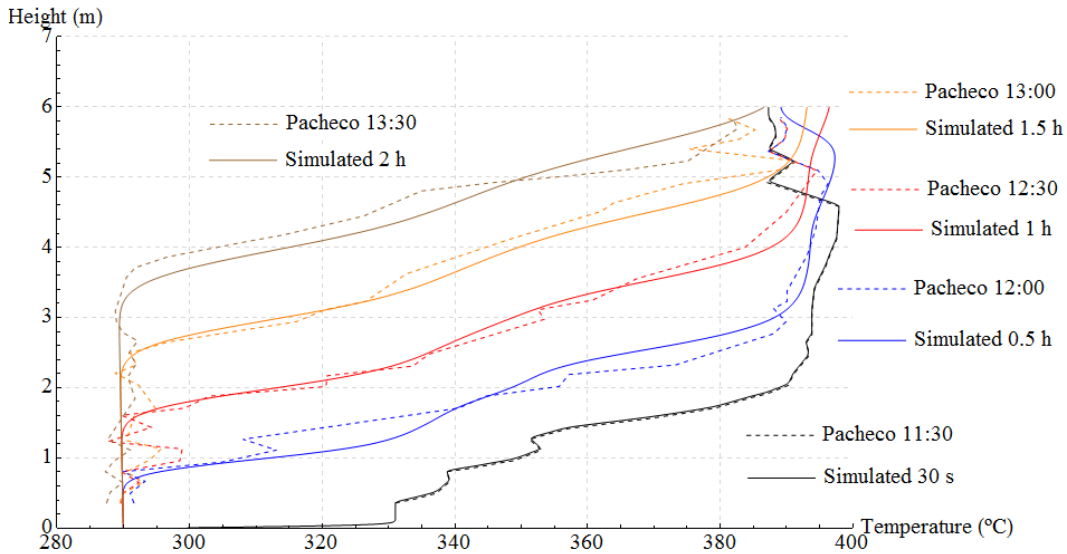


Figure 10: Thermocline TES model experimental validation against data from Ref. [11]

The model's translation and its numerical integration, i.e. simulation

The developed Modelica code has been translated into numerical simulation code using a state-of-the-art commercial Modelica tool, applying its differential-algebraic system solver DASSL [18, 19]. This algorithm applies an implicit method for the numerical integration of the governing ordinary differential equations. In particular, it approximates the derivatives using a k^{th} order backward differentiation formula, where k ranges from 1 to 5. At every step it chooses the order and the step size based on the behavior of the solution. Newton's method is used to solve the resulting equations for the solutions at each discrete point in time [18, 19].

The model has been validated against experimental data in the literature, which is shown in Figure

10 [20]. Additionally, the model has been further validated in the work of Ramacciotti [16].

4.2 Discussion of results

The results of the parametric simulations are shown in Table 5, showing a total of 6 cases. For case 1 for example we would have a bed diameter of 1.5 m and a bed length of 12 m. This case shows the lowest maximum flow velocity (8 m/s) and lowest pressure drop across the bed (1.8 kPa). However, the overall dimensions (diameter and length) seem to be critical as standard piping components will hardly be available. Furthermore, considering the pressurization process, the shell should be as compact as possible.

By reducing the duct diameter to 1 m, the maximum flow velocity and pressure drop becomes too high. Hence, the duct diameter shall be between DN1000 and DN1500. DN1200 seems to be appropriate. By choosing a bed length of 9 meters, an acceptable pressure drop is obtained and the expected switching time is around 30 min (slightly longer for the high-pressure system – case 6; and slightly shorter for the low-pressure system – case 6 LP). The resulting total area of heat transfer for one bed is about 1323 m² (see Table 4).

Table 4: Packed-bed properties

Bed filler material	Size of crushed rocks (cm)	Bed porosity (-)	Bed heat transfer area (m ²)
quartzite rock	3	0.35	1323

Table 5: Bed dimensions, flow velocity, pressure drop and estimated cycling time

Case number	Mass flow (kg/s)	Bed internal diameter (m)	Bed length (m)	Bed maximum flow velocity (m/s)	Bed pressure drop (kPa)	Bed cycling time (min)
1	1.4	1.5	12	8	1.8	111
2	1.4	1	12	20	9	39
3	1.4	1.5	6	8	0.9	34
4	1.4	1.2	12	14	4.3	60
5	1.4	1.2	6	14	2.1	18
6	1.4	1.2	9	14	3.2	38
6 LP	1.4	1.2	9	14	4	26

The following figures (Figure 11 to Figure 14) show 3D and 2D drawings of the high-pressure and low-pressure regenerative bed shells. Since the diameter of the regenerative bed is larger than the valve diameters, corresponding diameter reductions are required. Additionally, several blind flanges are needed to be able to fill the duct with crushed rock and to perform maintenance work if necessary.

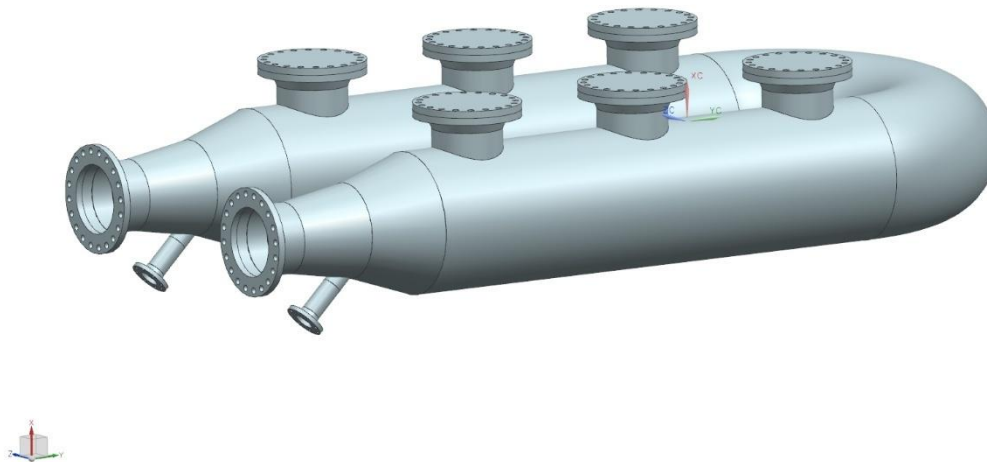


Figure 11: High-pressure regenerative bed 3D drawing (Flanges DN600 – charging- and DN150 - discharging)

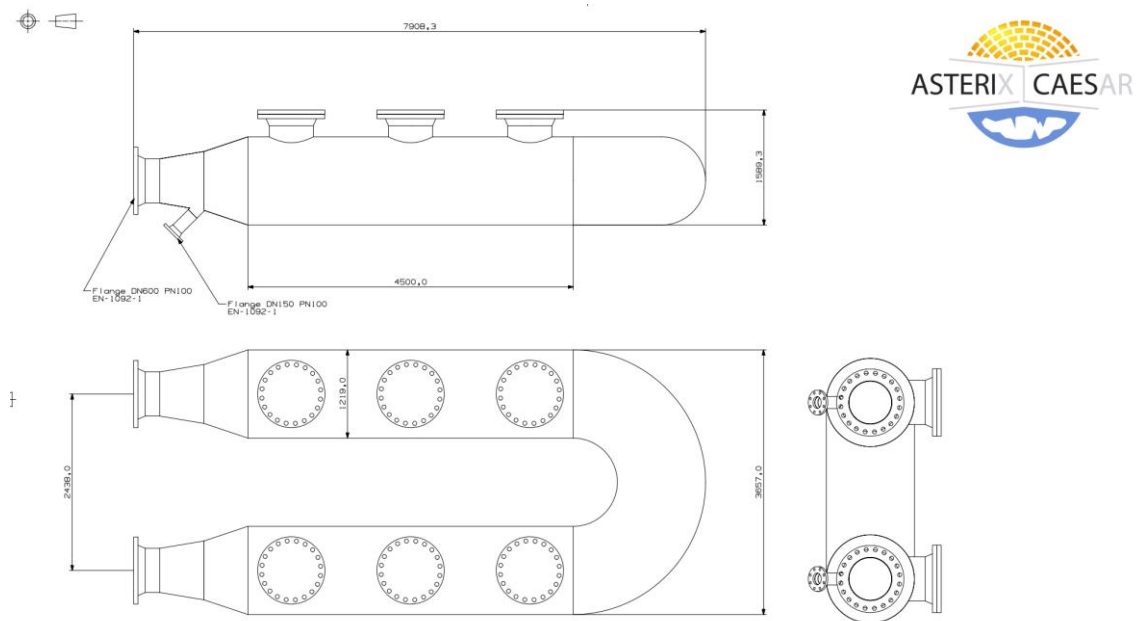


Figure 12: High-pressure regenerative bed 2D drawing (Flanges DN600 – charging- and DN150 - discharging) – dimensions in mm

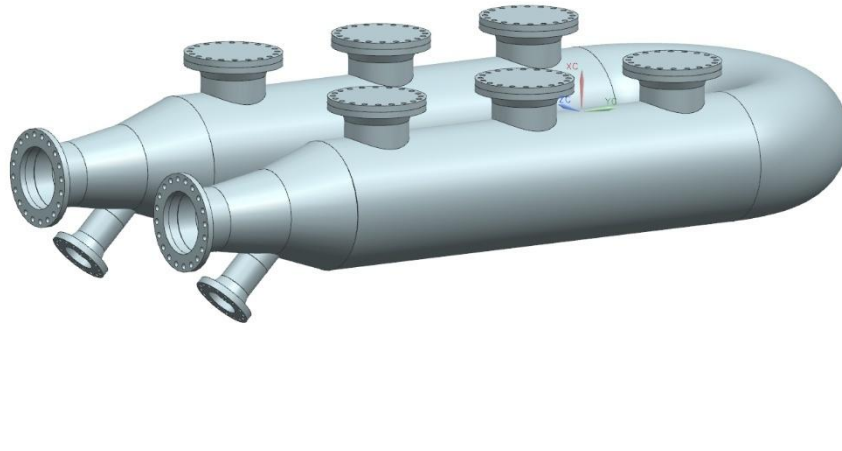


Figure 13: Low-pressure regenerative bed 3D drawing (Flanges DN600 – charging- and DN300 - discharging)

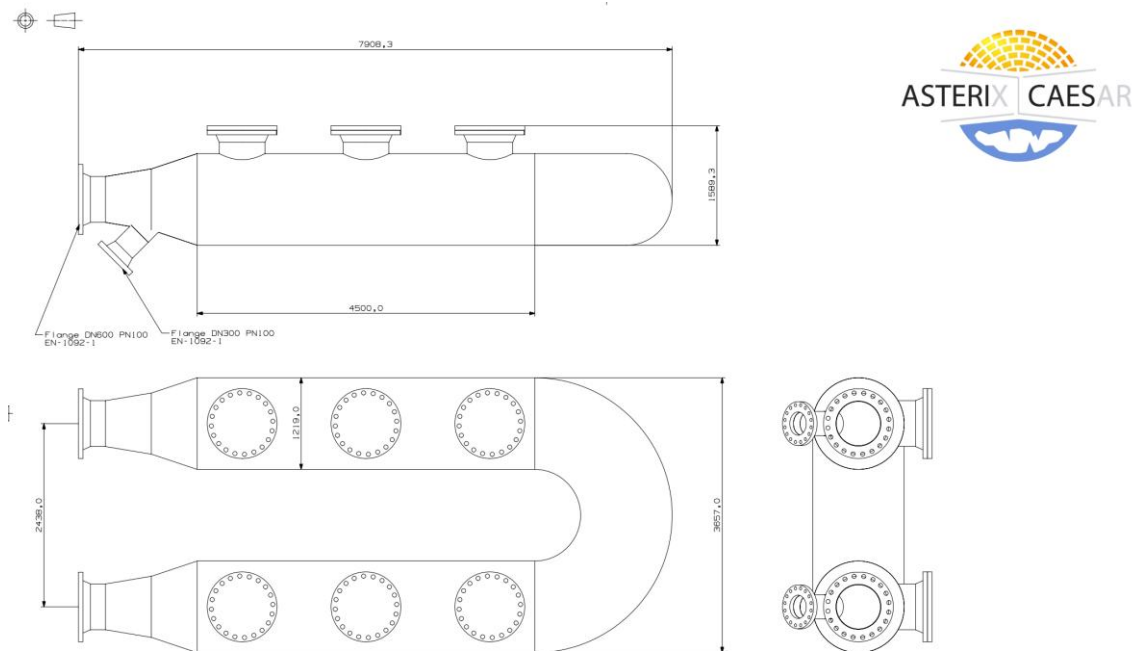


Figure 14: Low-pressure regenerative bed 2D drawing (Flanges DN600 – charging- and DN300 - discharging) – dimensions in mm

4.3 Valve design considerations

In principle, due to the high maximum system pressure of 40 bar (maximum turbine inlet pressure), ball valves may be the most suitable ones. However, the upper manufacturing limit for the valve balls leads to a maximum flow cross section size of about DN 200, which is very small considering the application with high-temperature air (very low fluid density). Apart from ball valves, another suitable valve type seems to be a high-performance butterfly valve, manufactured of expensive high-temperature alloys and metal-metal seat. The so-called triple offset butterfly valve would be the recommended design. A triple offset butterfly valve (TOBV), also known as

a triple eccentric butterfly valve, is a highly engineered type of quarter-turn valve designed for critical applications requiring bubble-tight shut-off and zero leakage, especially in high-pressure, high-temperature, or abrasive environments.

The "triple offset" refers to three distinct eccentricities in its design that work together to eliminate friction between the sealing surfaces during operation, leading to superior performance and extended lifespan:

- **First Offset:** The valve stem is positioned *behind* the center of the sealing surface of the disc. This ensures a continuous and uninterrupted sealing path.
- **Second Offset:** The stem is also offset *from the centerline of the valve bore/pipe*. This allows the disc to move away from the seat almost immediately upon opening, minimizing friction.
- **Third Offset (Conical Seating Geometry):** This is the most crucial offset. The sealing surfaces of the disc and seat are designed with a unique **conical or cammed profile**. This creates a right-angled cone shape, ensuring that the disc only makes contact with the seat at the very final moment of closure. As the valve opens, the disc "cams" away from the seat, eliminating rubbing.

In summary, a triple offset butterfly valve is a high-performance valve that provides superior sealing, durability, and a longer lifespan in challenging industrial environments due to its innovative three-offset design that eliminates friction between sealing surfaces.

The atmospheric pressure charging operation requires low air speeds (to keep the pressure drop low) and the largest possible valve diameter need to be chosen. Considering the demanding operating conditions of 40 bar differential pressure and up to 750°C fluid temperature, the maximum valve size seems to be limited between DN600 and DN800. Therefore, the regenerative system is clearly limited to the small-scale application (about 1 MWe discharging power).

4.4 Cost estimation

For the high-pressure and low-pressure regenerative system 8 valves are needed in each case (see Table 2 and Table 3). For the high-pressure system, 4 DN600 valves, and 4 DN150 valves are needed. For the low-pressure system, 4 DN600 valves, and 4 DN300 valves are needed.

The cost estimation for one DN600 butterfly valve is around 100 k€, based on requested quotes. The cost estimation for one DN150 butterfly valve is around 50 k€, based on requested quotes. The cost estimation for one DN300 butterfly valve is around 70 k€, based on requested quotes.

This would give a total cost of valves of 600 k€ for the high-pressure system, and 680 k€ for the low-pressure system.

The cost of one regenerative system shell (metallic shell + thermal insulation) is estimated at 200 k€.

Hence, the total cost of the high-pressure and low-pressure regenerative system is estimated at about 1 M€ each. This figure will be compared to the cost of the small-scale recuperative system design proposal according to section 5.

5 Recuperative approach

5.1 High temperature heat exchangers design issues

The plant layouts sketched in Figure 3 and Figure 5 have been taken as reference for the design of high temperature heat transfer equipment. In such plant arrangements, air streams at nearly atmospheric pressure coming from the High Temperature Thermal Energy Storage (HTES) is used to heat and reheat a pressurised air stream taken from a compressed air reservoir before its expansion in High Pressure Turbine (HPT) and Low-Pressure Turbine (LPT).

Overall plant thermodynamic performance improvements can be achieved by increasing the inlet temperature of both HPT and LPT. According to WP1, taking the current technology level into consideration, the above temperatures - at nominal conditions - have been set at 550 and 750° C, respectively.

To reduce the required heat transfer surface (which is strongly related to the cost of HEXs), high values of the overall heat transfer coefficient are beneficial. A first obstacle to the achievement of satisfactory values of such a coefficient is represented by the intrinsically low value of the heat transfer coefficient characterizing the heat transfer from the tube surface to the atmospheric air stream. Heat transfer rate can be improved by using extended surfaces (i.e. fins) and/or by assuming relatively high flow velocities. Both actions, however, contribute to increase the pressure loss, which affects the power absorbed by the fan to circulate the atmospheric air mass flow rate from HTES through the heat exchanger. Due to the relevant atmospheric air volumetric flow rate, the amount of power required by the fans is critical. Another aspect to keep in mind concerns the pressure loss on pressurised air side, which affects the power produced by the expansion train. Therefore, constraints on allowable pressure drops on both sides have been introduced in the design procedure.

With reference to the original plant layout (see Figure 3), temperature differences between hot atmospheric air and pressurised air are high enough in both heat exchangers. Such differences are significantly reduced when the modified plant layout is addressed, leading to reduced values of the Logarithmic Mean Temperature Difference (LMTD) in the different units constituting the heat exchanger network. As a result, low values of LMTD together with the strict requirement on the allowable pressure drop can pose a real design challenge.

Finally, for medium and large-scale applications, the expected large sizes can lead to unfeasibility issues and/or the need of several heat transfer unit in parallel or serial arrangement to fulfil the relevant heat duty.

The above concerns can be avoided or mitigated by adopting an alternative and innovative approach that seems more suitable for applications characterized by large mass flow rates and large thermal duties. Such an approach is based on a design like Heat Recovery Steam Generator (HRSG) technology. The reasons leading to this choice and the adopted design methodology are widely discussed in 5.3.

A proper material selection is a crucial design aspect, affecting the reliability and the cost of the heat transfer equipment. Metallic materials suitable for high temperature applications can be essentially grouped in iron-based and nickel-based alloys.

Among the iron-based alloys, austenitic stainless steels show favorable characteristics such as ductility, formability, high tensile strength and good capabilities to operate in hot and corrosive environments. Ferritic stainless steels are suitable for use at lower temperatures than austenitic steels, as they present lower strengths. Moreover, they show less corrosion resistance due to their lower chromium content. Iron-based alloys such as SS316 and SS347 constitute a common choice for heat transfer equipment operating at temperatures up to 650°C [21].

Nickel-based alloys show higher corrosion resistance and superior mechanical properties in respect to iron-based ones. If temperature does not exceed 750°C, alloy 803 and HR 120 are appropriate choices. For applications up to 900° C, most suited materials are alloy 230 and alloy 617. For temperatures higher than 1000°C, oxide dispersion-strengthened alloys are

recommended. It has to be pointed out that nickel-based alloys are 3-10 times more expensive than iron-based ones [22]. Therefore, their use should be limited, with a preference for adopting iron-based alloys whenever possible [22]. Other materials suited for high temperature applications are Haynes alloys, mainly based on nickel and cobalt, particularly indicated for their resistance to high temperatures and corrosion. Haines 120 and Haines 230 alloys are the most commonly used ones. Super alloys with a larger amount of nickel are well suited for temperatures characterizing the present application. Modified alloy 803, Inconel 625 and Haines 120 can be used when temperature are around 750 °C.

The quality of fluids involved in the heat transfer process (air taken from the atmosphere) does not pose any particular design issue, while attention should be paid to the high operating pressures and temperatures. In the present application, a maximum operating pressure of 150 bar and temperatures up to some 800° C have to be addressed. Therefore, according to what is reported in technical literature, carbon steel has been selected for heat transfer units operating at temperatures lower than 350°C, type 347 stainless steel for temperatures up to 650°C and Inconel 625 for temperatures up to 800°C.

Discharge operations at constant mass flow rate and constant HPT inlet pressure and temperature have been assumed as a reference. Anyway, at the end of the discharge phase, a temperature reduction of the atmospheric air coming from the LTES is noticed, and, consequently the HPT and LPT inlet temperature tend to decrease. Therefore, the HHEXs behavior has been assessed on the entire expected operating range. The assessment is performed on the basis of operating data provided by WP1.

5.2 Description of the adopted/developed models for shell & tube heat exchangers

For sake of brevity, only few notes are here provided to recall main characteristics of the methodology adopted to design Shell & Tube heat exchangers. A detailed description is given in Chapter 5.2 of Deliverable D3.1 Optimum low-temperature heat exchanger design (reuse of heat of compression).

The optimized design of low temperature S&T heat exchanger has been accomplished by using an in-house developed code written in MATLAB language [23] and by a commercial code (CC-Therm) included in the CHEMCAD package. Basically, the MATLAB code has been used to perform a first tentative thermal design of the heat exchanger aimed at defining the value of some relevant Independent Design Variables (IDV). Such values are assumed as input quantities to run CC-Therm in order to achieve a more accurate and reliable design. In the following sections, a short description of the adopted codes and of the strategy adopted to achieve the optimized design is given and commented.

Preliminary analyses aimed at choosing the most appropriate design approach have been carried out. It has been noticed that results achieved by adopting the Kern approach are in general more conservative (in terms of required heat transfer area) in respect to those achieved by applying the Bell-Delaware and the Stream Analysis methods. In conclusion, the Bell-Delaware method has been adopted for the final design, also because it is considered as the most reliable among those available in open literature [1, 24, 25].

Besides design calculations, CC-THERM can perform the heat exchanger rating. In such a calculation mode, the performance of a heat exchanger whose geometry is completely specified can be assessed by varying the operating conditions, i.e. the type of fluid, fluid mass flow rates, inlet temperatures and so on. In the present application, the rating calculation mode is used to evaluate and check the behaviour of each heat exchanger in the entire expected operating range.

5.3 Innovative HEX design based on heat recovery steam generator (HRSG) concept

As anticipated in Chapter 5.1, the Modified Plant layout is expected to be more demanding in terms of required heat transfer area in respect to the Original Plant concept. In fact, temperature differences between hot and cold streams in the heat transfer units constituting the heat transfer network are significantly reduced compared to Original Plant. Therefore, for Medium and Large-Scale plants, the adoption of the conventional Shell & Tube based approach can lead to design solutions which pose feasibility issues. In order to overcome these issues, a solution based on the Heat Recovery Steam Generator (HRSG) technology is proposed.

Basically, a HRSG is constituted by a rectangular duct containing the required number of tube bundles. Liquid water or steam flow inside the tubes, while the Gas Turbine exhaust gas flows externally. The present application shows relevant similarities with the heat recovery process occurring in a HRSG:

- streams flowing on duct side (Gas Turbine exhaust gas and air) are gas mixtures at atmospheric pressure with a similar chemical composition and, consequently, similar thermodynamic and thermal behavior;
- high-pressure streams are placed on tube side. In the present application, water/steam on tube side is replaced by pressurized air. Of course, in this case no phase change occurs. Anyway, the resulting arrangement is similar to that of tube bundles used to heat and re-heat the superheated steam in conventional HRSG.

Interesting aspects are related to the reduced duct side pressure losses, the possibility of using externally finned tubes, the capability of managing relevant heat duties and large atmospheric air volumetric flow rates. Moreover, the particular architecture of HRSG has the main potential of avoiding problems connected with the large size of nozzles that can occur when a Shell & Tube design is addressed.

A computational code for the design of LHEXs and HHEXs based on the HRSG concept has been developed in Visual Basic for Application (VBA). A detailed description of the developed calculation tool is given in the following.

The Logarithmic Mean Temperature Difference (LMTD) approach was adopted for the calculation of global heat transfer surface. Unlike combustion gases in HRSGs, in this application there are no triatomic gases that can make a noticeable contribution to heat exchange by radiation. Therefore, the only heat exchange mechanism considered is by convection.

The code allows the definition of the arrangement of each tube bundle in terms of required tube rows and number of parallel paths, i.e. number of rows fed in parallel by the tube side fluid. A typical arrangement and the most relevant geometric quantities characterizing a tube bundle are depicted in Figure 15. In order to achieve an adequate velocity of the pressurized air, the tube bundle is generally characterized by a certain number of parallel passes: tube side stream can be distributed on more than one row to reduce the velocity, as shown in the left part of Figure 15, where two tube rows are fed in parallel.

Input quantities are atmospheric air mass flow rate, pressure and temperature. Moreover, for each tube bundle, pressurized air mass flow rate, inlet pressure, inlet and outlet temperatures must be assigned. Other input quantities are Degrees Of Freedom (DOFs), i.e. quantities chosen by the user which completely define the design. The atmospheric air velocity at the duct inlet (referred to the free cross section area, i. e. not considering the presence of tubes) and the height-to-width ratio of the duct have to be given to determine the duct height and width. Further DOFs are required to specify the design of each tube bundle: tube outer diameter, tube thickness and bundle arrangement (in line or staggered, transversal and longitudinal tube pitches). Bare as well as finned tube can be selected. In the latter case, the type of fin (solid or serrated fin) and all elements required to specify the fin geometry have to be provided: fin pitch, diameter and thickness. Finally, the material used for tubes and fins has to be selected.

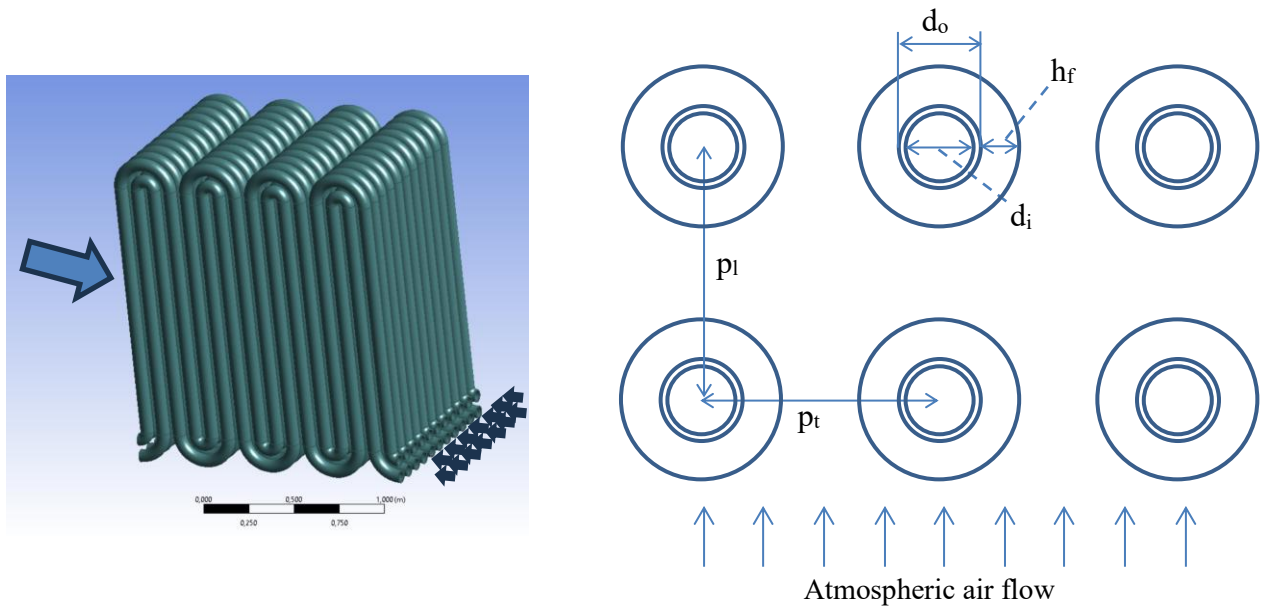


Figure 15: Typical tube bundle arrangement and main quantities characterizing the bundle geometry.

Correlations available in literature [26, 27] are adopted to estimate the heat transfer coefficient and the pressure drop on tube and duct side.

Constraints related to the maximum tube-side velocity and pressure drop on duct side are introduced based on the literature.

The developed tool works according to the flowsheet shown in Figure 16. An iterative procedure is implemented to define the geometry of all tube bundles. Input data are provided through excel sheets. The tool relates to NIST REFPROP v. 10.0 libraries for the accurate evaluation of the working fluids thermodynamic properties (e.g. density, thermal conductivity, viscosity, enthalpy, and Prandtl number). The duct size and, consequently, the number of tubes that can be placed in a transversal tube row are initially evaluated. Then, the number of parallel rows is evaluated according to the constraint on maximum allowable tube-side velocity. The process is iterative, and an additional parallel path is added whenever the calculated velocity exceeds the allowable velocity.

The overall heat transfer coefficient and the total exchange surface area are then calculated. For finned tubes, a specific procedure given in [26] is implemented to calculate the heat transfer coefficient on the external side of tubes [27].

To achieve an integer number of parallel passes, the number of fins per meter is iteratively varied until an integer number of passes is obtained with a tolerance of 3%.

In parallel with the procedure for calculating the overall heat transfer coefficient, the tube-side and duct-side pressure drops are evaluated according to [27].

The code has been successfully validated against a real existing tube bundle whose geometry and design operating condition are given in [28].

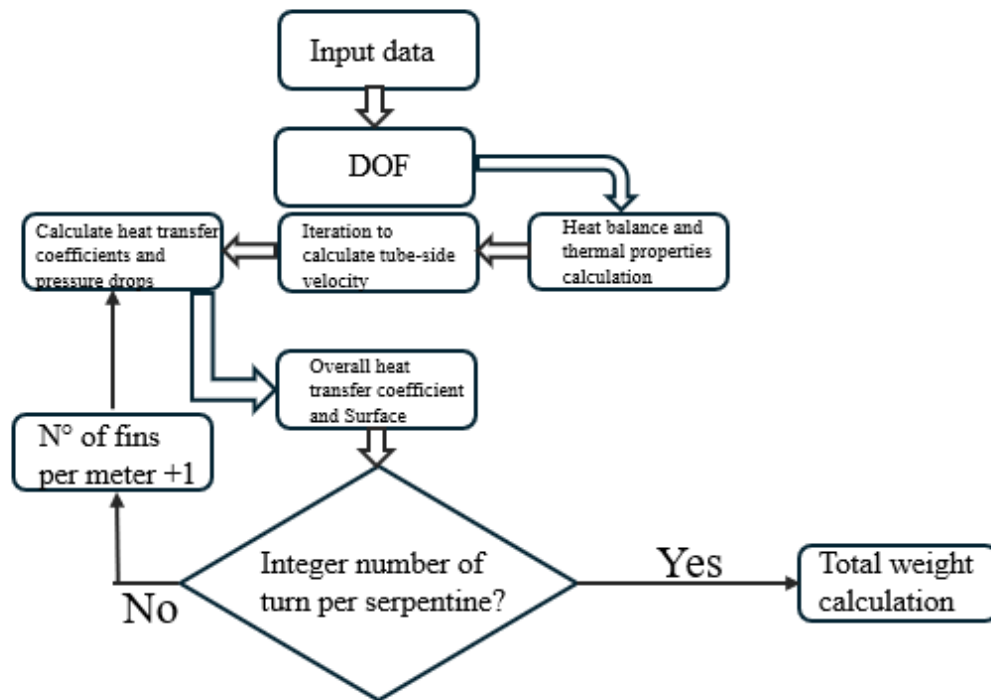


Figure 16: Flow chart of the developed computer code

Table 6 shows the values of process and geometric quantities of the existing superheater that have been used as input data. Free flow exhaust gas velocity and height-to-width ratio have been chosen in order to obtain the same height and width of the duct of the existing HRSG.

Table 6: Existing superheater data used as input for the developed tool

Input data	Unit	
Process quantities		
Flue gas mass flow rate	Kg/s	409.56
Flue gas inlet temperature	°C	631.15
Steam mass flow rate	Kg/s	67.5
Steam outlet temperature	°C	523
Steam outlet pressure	bar	97.5
Steam inlet temperature	°C	439
Geometric quantities		
Duct width	m	7.536
Duct height	m	15.736
Tube arrangement	--	Staggered
Tube outer diameter	mm	38,1
Transversal pitch	mm	96
Longitudinal pitch	mm	90
Fin type	--	Solid
Fin height	mm	10
Fin thickness	mm	1,2

Table 7: Comparison between real and calculated data

	Unit	Existing superheater	Calculated with VBA code	Δ %
Number of rows	--	3	3	--
Tubes per row	--	78	78	--
Fins per meter	--	120	120	--
U	W/m ² K	61,8	61,0	-1,3
S	m ²	1805,4	1843,5	2,1
Flue gas pressure drop	kPa	0,079	0,072	-8.9

In Table 7, main results obtained by the design tool are compared with real data. It can be noticed that the same tube bundle geometry has been found, i. e. the same number of tubes and the same fin pitch. The calculated overall heat transfer coefficient U is therefore in good agreement with the real one, as well as the heat transfer surface S. A quite satisfactory agreement is achieved also for flue gas pressure drop. The percentage error is higher in respect to those affecting the overall heat transfer coefficient and the heat transfer surface. Anyway, the absolute error is small (some 0.007 kPa), leading to a really reduced increase in evaluating the back pressure at the exit of the GT expander.

5.4 Original plant layout: design results

In agreement with WP1 outcomes, the same plant layout has been assumed for Small, Medium and Large size plants.

According to Chapter 1.4, ultimate design specifications agreed among the Consortium partners have been established also on the basis of preliminary analyses carried out also in the frame of WP3 activities.

The Original Plant layout (see Figure 3) has two shell-and-tubes HHEXs fed by two parallel atmospheric air streams. In the first one (HHEX1), high pressure air is heated up to a temperature of 550°C. In the second one (HHEX2), the air stream exiting the High-Pressure Turbine (HPT) at 6 bar is heated up to 750°C before entering the Low-Pressure Turbine (LPT) to complete the expansion process. Atmospheric air streams at the exit of HHEX1 and HHEX2 are mixed and then fed to the Heat Recovery Steam Generator (HRSG) of the bottomer Organic Ranking Cycle (ORC). On the basis of abovementioned preliminary analyses carried out in WP1, a value of 450°C was chosen for the temperature of air entering the HRSG. Such a value is high enough to ensure a satisfactory efficiency level of the bottomer cycle and, consequently, the efficiency of the overall system.

Since the atmospheric air is fed to HHEXs at 800°C, LMTDs are of some 260°C for HHEX1 and of 110°C for HHEX2. Such values are significantly higher than those found for Low Temperature Heat Exchangers (see Deliverable D3.1) and of those which will be found when the Modified Plant layout will be addressed. Therefore, an easier design characterised by lower heat transfer surfaces is expected.

The power absorbed by fans to circulate atmospheric air constitutes a parasitic loss which adversely affect the plant overall performance. Therefore, in agreement to WP1 suggestions, an allowable shell side pressure drop limit of 4 kPa across each HHEX has been assumed.

Compressed air pressure losses (tube side) affect the expansion work, but represent an issue of minor importance in respect to shell side losses. Anyway, a maximum allowable pressure loss equal to 4 % of the inlet pressure has been assumed.

Taking the above aspects in mind, with a particular focus on the reduced need of heat transfer area due to relatively high LMTDs, it is believed that the approach based on Shell & Tube heat exchanger can lead to a satisfactory design and, therefore, alternative approaches are not considered.

Discharge operations at constant mass flow rate and constant HPT inlet pressure and temperature have been assumed for the design of the high temperature heat transfer equipment. In the last part of the discharge phase, a temperature reduction of the atmospheric air coming from the HTES occurs. The HHEXs performance has been assessed on the entire expected operating range. The check is performed on the basis of data provided by WP1.

5.4.1 Small-Scale plant

Flowsheet specifications required to design high temperature heat transfer equipment for Small-Scale plant are reported in Table 8.

Table 8: Original Plant layout. Design specifications for Small-Scale plant.

Compressed air mass flow rate (kg/s)	1.5
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First High Temperature Heat Exchanger (HHEX1)	
Compressed air inlet temperature (°C)	125
Compressed air inlet pressure (bar)	40
Compressed air outlet temperature (°C)	550
Second High Temperature Heat Exchanger (HHEX2)	
Compressed air inlet temperatures (°C)	250
Compressed air inlet pressure (bar)	6
Compressed air outlet temperature at the end of the charging phase (°C)	750

HHEX1 and HHEX2 optimized design has been performed by adopting the procedure outlined in Chapter 5.2. Results are summarized in Table 9 and Table 10. In both cases, pressurized air is placed on tube side and 16 mm OD bare tube and rod baffles have been adopted to lower the shell side pressure drop.

Table 9: Original Plant layout. Small-Scale plant. Main process quantities.

HEX	m _c (kg/s)	m _H (kg/s)	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	Q (kW)
HHEX1	1.5	1.8	125.0	550.0	800.0	452.0	698
HHEX2	1.5	2.1	250.0	750.0	800.0	444.0	832

m_c: compressed air mass flow rate; m_H: atmospheric air mass flow rate; T_{CI}: compressed air inlet temperature; T_{CO}: compressed air outlet temperature; T_{HI}: atmospheric air inlet temperature; T_{HO}: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 10: Original Plant layout. Small-Scale plant. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s x n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HHEX1	3.0	0.4	16	1×1	169	25.6	2.3	95.1
HHEX2	7.7	0.8	16	1×1	652	251.1	3.7	31.7

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.

The design of HHEX1 does not pose any particular issue. The relatively low compressed air exit temperature (550 °C) leads to a LMTD value of some 275° C.

The design of HHEX2 is more challenging. The not particularly high LMTD (about 110 °C) result in a large heat transfer surface in the order of 250 m². The design led to a quite cumbersome heat transfer unit. Anyway, both thermal duty and pressure drop constraint are fulfilled, demonstrating the feasibility of the proposed design.

HHEX1 design shows a satisfactory overall heat transfer coefficient of some 90 W/m²K. A significantly lower value of about 30 W/m²K is achieved for HHEX2. This is essentially due to the lower shell side heat transfer coefficient achieved in HHEX2 (some 45 W/m²K compared to the value 180 W/m²K estimated for HHEX1). Such a small value derives from the lower value of the shell side velocity (some 10 m/s against 30 m/s evaluated for HHEX1) which had to be adopted to reduce the shell side pressure loss.

According to the expected operating conditions, the maximum temperature that can be reached by the metal parts is of some 650°C in HHEX1 and 780° C in HHEX2. Therefore, in accordance with Chapter 5.1, type 347 stainless steel and Inconel 625 are used for the manufacture of HHEX1 and HHEX2, respectively. In order to avoid repetition, it is underlined here that the same choice is made for Medium and Large-Scale plants.

Bonnet type (integral cover) is adopted for the front-end

head, while a P type floating head (outside packed type) is selected for the rear end. Both designs are based on E shell type (one pass).

Table 11: Small-Scale plant off design behaviour

HEX	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
HHEX1	125.0	517.3	750.0	428.0	94.3	2.2	641.8
HHEX2	231.4	702.3	750.0	412.4	31.2	3.5	783.8

T_{CI}: compressed air inlet temperature; T_{CO}: compressed air outlet temperature T_{HI}: atmospheric air inlet temperature; T_{HO}: atmospheric air outlet temperature; U: overall heat transfer coefficient; ΔP_s: shell side pressure drop; Q: exchanged thermal power.

The behaviour of HHEXs at the end of the discharge phase - when the temperature of the atmospheric air coming from the HTES drops to 750°C – has been checked. It can be noticed that the HPT inlet temperature decreases of some 35°C in respect to the nominal value of 550°C. An LPT inlet temperature drop of the same entity of the atmospheric air temperature decreases is observed.

5.4.2 Medium-Scale plant

The design of Medium-Scale plant high temperature heat transfer equipment has been carried out according to data reported in Table 12.

Main process quantities assumed for the design are given in Table 13, while main design results are reported in Table 14. In this case also, the design of HHEX1 does not pose any particular issue. The design of HHEX2 is more challenging. The relevant heat duty (about 16 MW) and the not particularly high LMTD (about 110 °C) result in a large heat transfer surface in the order of 2700 m². Anyway, the heat duty can be met by a single unit, avoiding the need to install multiple units in parallel. Manufacturers have evidenced the same points of attention mentioned when dealing with Low Temperature Heat Exchangers (see Deliverable D3.1): i) high capital cost linked to the large heat transfer area and to the need of employing expensive materials and, ii) the large size of shell side nozzles, required to accommodate with acceptable pressure losses the huge atmospheric air volumetric flow rate, as shown in Figure 17.

The off-design behavior of Medium-Scale plant heat exchangers is similar to that found for the Small-Scale plant reported in Table 11.

Table 12: Original Plant layout. Design specifications for Medium-Scale plant.

Compressed air mass flow rate (kg/s)	30
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First High Temperature Heat Exchanger (HHEX1)	
Compressed air inlet temperature (°C)	125
Compressed air inlet pressure (bar)	40
Compressed air outlet temperature (°C)	550
Second High Temperature Heat Exchanger (HHEX2)	
Compressed air inlet temperatures (°C)	250
Compressed air inlet pressure (bar)	6
Compressed air outlet temperature at the end of the charging phase (°C)	750

Table 13: Original Plant layout. Medium-Scale plant. Main process quantities.

HEX	m _c (kg/s)	m _H (kg/s)	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	Q (kW)
HHEX1	30.0	33.0	125.0	550.0	800.0	430.0	13610
HHEX2	30.0	42.0	250.0	750.0	800.0	453.0	16250

m_c: compressed air mass flow rate; m_H: atmospheric air mass flow rate; T_{CI}: compressed air inlet temperature; T_{CO}: compressed air outlet temperature; T_{HI}: atmospheric air inlet temperature; T_{HO}: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 14: Original Plant layout. Medium-Scale plant. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s × n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HHEX1	7.0	1.7	16	1 × 1	2184	760	2.9	64.9
HHEX2	9.4	2.4	16	1 × 1	6202	2868	2.9	52.4

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.



Figure 17: HHEX2 sketch with main geometric dimensions.

5.4.3 Large-Scale plant

Large-Scale plant reference design quantities are given in Table 15. In the present case, a lower pressure of 80 bar (instead of 150 bar) has been set. The reason is that the size of the plant is not compatible with the adoption of an artificial reservoir because of the relevant capital cost to be incurred for the air storage tank. Therefore, the traditional approach based on the availability of a natural reservoir is adopted. The storage pressure has been chosen in agreement with those of Huntorf and McIntosh existing plants [29]. On the basis of analyses carried out in WP1, a higher value of the LPT inlet pressure (8 bar instead of 6 bar) have been selected in respect to Small and Medium-Scale plant layouts.

Table 15: Original Plant layout. Design specifications for Large-Scale plant.

Compressed air mass flow rate (kg/s)	142
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First High Temperature Heat Exchanger (HHEX1)	
Compressed air inlet temperature (°C)	125
Compressed air inlet pressure (bar)	40
Compressed air outlet temperature (°C)	550
Second High Temperature Heat Exchanger (HHEX2)	
Compressed air inlet temperatures (°C)	290
Compressed air inlet pressure – (bar)	8
Compressed air outlet temperature at the end of the charging phase (°C)	750

Table 16: Original Plant layout. Large-Scale plant. Main process quantities.

HEX	m _c (kg/s)	m _H (kg/s)	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	Q (kW)
HHEX1	23.7	26.0	125.0	550.0	800.0	430.0	10725
HHEX2	35.5	48.8	290.0	750.0	800.0	473.0	17818

mc: compressed air mass flow rate; mH: atmospheric air mass flow rate; TCI: compressed air inlet temperature; TCO: compressed air outlet temperature; THI: atmospheric air inlet temperature; THO: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 17: Original Plant layout. Large-Scale plant. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s × n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HHEX1	5.5	1.7	16	1 × 1	1423	391	2.4	99.5
HHEX2	7.3	2.7	16	1 × 1	6060	2217	2.1	78.9

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.

As expected, multiple parallel heat transfer units are required to fulfil the Large-Scale plant thermal duty. In particular, six units are required to heat the high-pressure air up to 550°C (HHEX1), while the re-heating of low-pressure air before entering the LPT is accomplished by installing four parallel units. Main design outcomes for the single unit are given in Table 16 and Table 17.

The design is characterized by strong similarities with the design of the heat transfer equipment of the Medium-Scale plant: one shell side pass, one tube side pass, rod type baffles. As a

consequence, the design is affected by same issues discussed in dealing with the Medium-Scale plant: high costs and relevant dimension of inlet and outlet shell side nozzles. Anyway, in order to contain the overall heat transfer area, design solutions characterized by high values of the heat transfer coefficient have been preferred, as it can be seen on Table 17. In principle, the heat duty could be fulfilled by installing five HHEX1 and HHEX2 units in parallel sized for the Medium-Scale plant. The present solution is preferred because it leads to a noticeable reduction of the overall heat transfer area. According to Table 14, the heat transfer area resulting from the installation of five Medium-Scale HHEX1 units is in the order of some 4000 m², versus 2350 m² of the present design. The heat transfer area saving achieved for HHEX2 is even more significant: 8870 versus 14340 m², corresponding to a reduction of about 40%. This result is of great importance if the relevant cost of the material required to manufacture the HHEX2 heat transfer units is considered.

In this case also, compressed air temperature drops at the end of discharge phase are similar to those found for the Small-Scale plant discussed in 5.4.1.

5.5 Modified plant layout: design results

The alternative modified plant layout is shown in Figure 5. High Pressure Turbine (HPT) and Low Pressure Turbine (LPT) inlet temperatures, pressures and mass flow rates are the same of the original plant layout. In this case, a single stream of hot ambient pressure air is used to heat and re-heat the pressurized air entering the HPT and the LPT.

According to WP1 findings, such a layout is expected to bring to an improved overall techno-economic performance and, therefore, has been taken into consideration for further investigations.

The resulting heat exchanger network for Small and Medium-Scale plants, obtained by applying the principles of the pinch analysis, is shown in Figure 18 (as discussed in 5.4.3, slightly different assumptions have been made for Large-Scale plant). The hot atmospheric air coming from high temperature packed bed TES enters the unit HT1, where the low-pressure air stream is heated up to the LPT inlet temperature. Then, the atmospheric air stream is equally split to feed the parallel units HT2 and HT3, where high- and low-pressure air streams are heated up to 550°C, which represents the HPT inlet temperature. Finally, in HPT4, high pressure air taken from the reservoir is preheated up to 250°C before entering HP3. The atmospheric air leaves HT3 at around 160°C and then is supplied to industrial users.

According to WP1 outcomes, a pinch point temperature difference of 60°C has been assumed, as shown in Figure 19, where the hot and cold stream composite curves are given for the Small-Scale plant case. In the present case, LMTD values are less than those found for the original plant layout. In fact, LMTDs of HT1, HT2 (which is equal to that of HT3) and HT4 are 87, 96 and 90°C, respectively. As a result, a larger overall heat transfer area is expected in respect to the original plant layout. This is counter-balanced by the higher plant overall efficiency resulting from the lesser amount of hot atmospheric air needed to heat and reheat the same compressed air mass flow rate. Moreover, the lesser amount of atmospheric air required entails further advantages in terms of size (and cost) reductions of other plant sub-systems, such as the solar field and the high temperature packed bed TES. Finally, the fact that atmospheric air leaves at 160°C represents an additional advantage, as it allows for more effective integration with conventional industrial process heat.

In sizing the heat transfer equipment, the same allowable pressure drop adopted for the original plant layout is imposed (maximum of 4 kPa on shell side, 4% of inlet pressure on tube side). Additionally, in order to ensure a similar flow split between HT2 and HT3, both heat exchangers have been designed with comparable pressure drops. The overall pressure drop affects each stream, namely:

- the overall pressure drop of the atmospheric air stream from HT1 inlet to HT4 outlet;
- the pressure drop of the high-pressure stream in units HT4 and HT3;
- the pressure drop of the low-pressure stream in units HT2 and HT1.

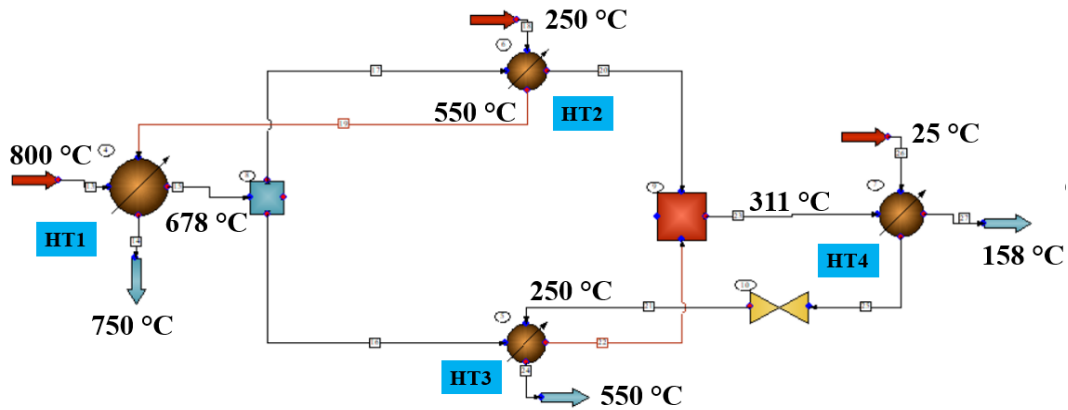


Figure 18: Small and Medium-Scale plant heat exchanger network.

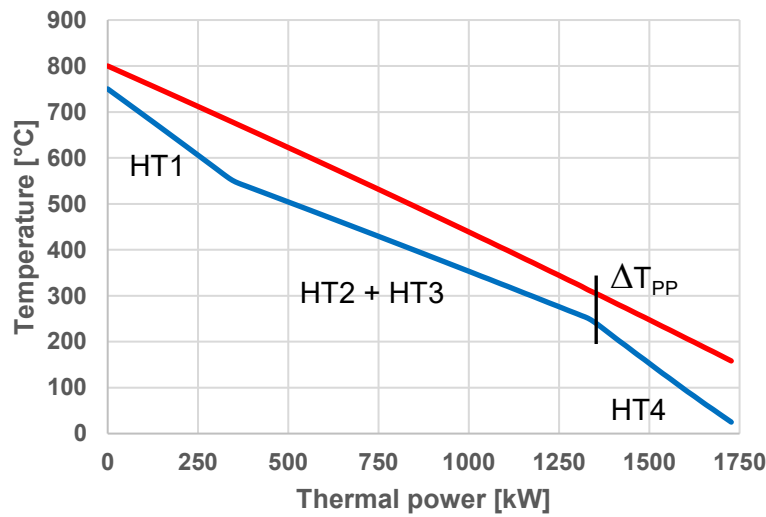


Figure 19: Small-Scale plant. Composite curves.

In the following, results achieved for Small, Medium and Large-Scale plant are presented and discussed. Since the HRGS based design is not suitable for Small-Scale plant, only the Shell & Tube based design is addressed. For Medium and Large-Scale plants, both approaches based on the Shell & Tube and HRSG technologies will be addressed and compared.

In the following, the same material will be used to manufacture the different thermal units, regardless of the adopted design approach: Inconel 625 for HT1, type 347 stainless steel for HT2 and HT3 and carbon steel for HT4.

Finally, in this case also, discharge operations at constant mass flow rate and constant HPT inlet pressure and temperature are assumed as a reference. Anyway, at the end of the discharge phase, a temperature reduction of the atmospheric air coming from the HTES occurs. Therefore, the thermal performance of each heat transfer unit has been checked on the entire expected operating range.

5.5.1 Small-Scale plant: Shell & Tube design

As previously stated, reference values of pressurized air quantities are the same of the original plant layout (see Table 8). According to the assumption made for the pinch point temperature difference, the required atmospheric air mass flow rate is of 2.50 kg/s, as reported in Table 18.

Table 18: Modified Layout. Small-Scale plant. Shell & Tube design. Main process quantities.

HEX	m _c (kg/s)	m _H (kg/s)	T _{ci} (°C)	T _{co} (°C)	T _{Hi} (°C)	T _{Ho} (°C)	Q (kW)
HT1	1.50	2.50	549.9	750.1	800.0	677.9	346
HT2	1.50	1.25	250.0	549.9	677.9	313.6	492
HT3	1.50	1.25	248.6	549.7	677.9	309.0	498
HT4	1.50	2.50	25.0	249.9	311.3	158.1	391

m_c: compressed air mass flow rate; m_H: atmospheric air mass flow rate; T_{ci}: compressed air inlet temperature; T_{co}: compressed air outlet temperature; T_{Hi}: atmospheric air inlet temperature; T_{Ho}: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 19: Modified Layout. Small-Scale plant. Shell & Tube design. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s × n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HT1	5.0	0.6	16	1×1	231	57.6	0.9	72.4
HT2	7.3	0.5	16	1×1	383	140.1	1.3	38.2
HT3	7.5	0.5	16	1×1	383	143.7	1.3	38.5
HT4	7.3	0.5	13	1×1	192	57.1	1.0	74.0

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.

Main design outcomes are summarized in Table 19. On the basis of results achieved for the original plant layout, pressurized air is placed on tube side. Bare tubes and rod baffles have been adopted to reduce the shell side pressure drop, which in the modified plant layout constitutes a relevant issue, representing the value of 4 kPa the overall allowable atmospheric air pressure drop across all heat transfer units.

Bonnet type (integral cover) is adopted for the front-end head, while a P type floating head (outside packed type) is selected for the rear end. Both designs are based on E shell type (one pass). To avoid repetition, it is anticipated here that the aforementioned choices are made for all the Shell & Tube heat exchangers addressed in the following.

No particular problems arisen in designing HT1 and HT4. Relatively high values for the overall heat transfer coefficient are achieved, as evidenced in Table 19. The design of parallel units HT2 and HT3 has proved more problematic. The fulfilment of the constraints on tube side pressure drop called for a significant reduction of the tube side velocity, which led to a decrease of the tube side heat transfer coefficient. This is evidenced by the number of tubes, which is considerably higher than that of HT1 and HT4. An acceptable overall atmospheric air pressure drop of some 3.2 kPa has been found.

The amount of hot air required to heat and reheat the high and low pressure air up to the target temperature is 40% lesser than that of the original plant layout (2.50 kg/s instead of 3.90 kg/s, as can be seen from Table 9 data). On the other hand, the required overall heat transfer surface shows and increase of the same entity, going from a value of 280 m² estimated for the original layout (see Table 10) to a value of some 400 m².

5.5.2 Medium-Scale plant: Shell & Tube and HRSG design

High- and low-pressure air process quantities for Medium-Scale plant are shown in Table 12. The required atmospheric air mass flow rate is of 48.6 kg/s, as reported in Table 20.

At first, results achieved by adopting the Shell & Tube design approach are presented and commented (see Table 21). The design of units HT2 and HT3 is characterized by the same issues discussed in the previous chapter, which led to a not so brilliant overall heat transfer coefficient.

Table 20: Modified Layout. Medium-Scale plant. Shell & Tube design. Main process quantities.

HEX	m _c (kg/s)	m _H (kg/s)	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	Q (kW)
HT1	30.0	48.6	550.5	750.2	800.0	678.6	6716
HT2	30.0	24.3	250.0	550.5	678.6	314.7	9609
HT3	30.0	24.3	250.0	550.5	678.6	310.8	9708
HT4	30.0	48.6	25.0	249.9	312.8	160.1	7625

m_c: compressed air mass flow rate; m_H: atmospheric air mass flow rate; T_{CI}: compressed air inlet temperature; T_{CO}: compressed air outlet temperature; T_{HI}: atmospheric air inlet temperature; T_{HO}: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 21: Modified Layout. Medium-Scale plant. Shell & Tube design. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s × n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HT1	5.8	2.9	16	1×1	3550	1028	0.8	78.8
HT2	9.8	2.4	16	1×1	5650	2759	2.2	37.5
HT3	7.0	2.3	16	1×1	4896	1713	2.2	61.9
HT4	7.0	2.1	10	1×1	3848	842	0.8	96.0

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.

Moreover, additional issues related to the larger size arise. Units HT2 and HT3 show relevant similarities with the Medium-Scale HHEX2 designed for the original plant layout: i) similar size, in terms of shell diameter and tube length, ii) similar heat transfer area, in the order of 2000-2500 m², iii) a large size of shell side nozzles, required to accommodate with acceptable pressure losses the huge atmospheric air volumetric flow rate. The above points are expected to lead to considerable manufacturing cost.

Finally an overall atmospheric air pressure loss of 3.8 kPa is evaluated. It has to be pointed out that such a value is really close to the maximum allowable one (4.0 kPa).

For Medium-Scale plant, also the approach based on the HRSG concept is addressed. According to this approach, each HT unit of Figure 18 constitutes a tube bundle placed inside a rectangular duct. The design is carried out according to procedure outlined in 5.3.

Values of Degrees of Freedom (DOF) assumed for the design are given in Table 22. Free flow velocity and duct height/width ratio have been assumed in order to have a 5×5 m cross flow section. The tube thickness is assumed in order to withstand the inside pressure. An inline arrangement has been preferred for all the bundles. Bare tubes are selected for the bundle operating at the highest temperature (HT1), finned tubes for other bundles.

Main results are shown in Table 23. A relevant cross flow area is required to allocate the huge volumetric flow rate of the low-pressure air. To ensure a counter-current heat transfer, HT1 has been designed by adopting a large tube diameter to reduce the number of parallel path and, consequently, increase the number of turns per serpentine. The same number of turns per serpentine is achieved in HT2 by adopting a smaller tube diameter and a larger number of parallel paths. High pressure bundles HT3 and HT4 can be successfully arranged by adopting smaller tube

diameters and a lesser number of parallel paths.

For all tube bundles, the calculated heat transfer coefficient is in line with typical values found for actual HRSGs given in [28, 30, 31]. Moreover, in analogy with actual HRSGs, a substantially reduced value of duct side pressure drop (in the order of 1.2 kPa) is found.

Table 22: Modified layout. Medium-Scale plant. HRSG design. Assumed DOFs.

	HT1	HT2	HT3	HT4
Free flow velocity [m/s]	6.0	5.29	5.29	3.27
Duct height/width ratio	1	2	2	1
Tube diameter [mm]	73.0	60.3	42.2	48.3
Tube thickness [mm]	5.15	3.18	4.85	7.14
Tube arrangement	Inline	Inline	Inline	Inline
Tube type [Bare, Finned]	Bare	Finned	Finned	Finned
Transversal pitch	1.25	1.33	1.45	1.40
Longitudinal pitch	1.25	1.33	1.85	1.40
Fin type [Solid, Serrated]	Solid	Solid	Solid	Solid
Fins height [mm]	8.0	8.0	8.0	8.0

Table 23: Modified layout. Medium-Scale plant. HRSG design. Main results.

	HT1	HT2	HT3	HT4
Duct width (m)	5.0	-	-	-
Duct height (m)	5.0	-	-	-
Bundle length (m)	2.74	4.81	2.19	0.68
Tubes per row x Parallel rows	51 x 5	28 x 10	38 x 4	71 x 1
Number of turns x Serpentine	6	6	7	10
Number of tubes	1530	1680	1064	710
Total surface	1756	2773	2141	1524
Fin per meter	-	40	102	95
Overall heat transfer coefficient ($W/m^2\text{°C}$)	46.3	37.3	50.0	52.8
Shell side pressure drop (kPa)	1.13	0.90	0.90	0.16
Tube side pressure drop (kPa)	10.40	14.05	92.51	81.1

Table 24: Modified layout. Medium-Scale plant: comparison between S&T and HRSG approaches.

	Shell & Tube	HRSG
Overall heat transfer surface (m ²)	~ 6350	~ 8200
Atmospheric air pressure drop (kPa)	3.8	1.2
High pressure air tube side pressure drop (kPa)	~ 222	~ 105
Low pressure air tube side pressure drop (kPa)	~ 20	~ 24.5

Main results achieved by adopting the two different design concepts are compared in Table 24. It can be noticed that the HRSG based approach requires a larger overall heat transfer surface in respect to that based on Shell & Tube heat exchangers. Anyway, the diverse characteristics of the installed heat transfer surface have to be pointed out. For unit HT1, bare tubes have been chosen for both design concepts. For other units, bare tubes are preferred for the Shell & Tube approach to contain the shell side pressure drop, as discussed earlier. Since the HRSG based approach is characterized by reduced pressure losses, finned tubes can be successfully adopted. This is expected to bring to advantages in terms of capital expenditure, because finned surface is significantly less expensive in respect to that of bare tubes.

Pressure losses play a relevant role in terms of net plant electric output and efficiency, since they affect both the net turbine output (tube side pressure losses) and the parasitic consumption related to the power absorbed by the fan to circulate the atmospheric air stream (shell or duct side pressure losses). It can be noticed as the HRSG approach gives better results, especially with regard to atmospheric air pressure losses.

Finally, the off-design performance at the end of discharge phase of both Shell & Tube and HRSG designs is given in Table 25. It can be noticed as results show really little difference, denoting in both case a satisfactory behaviour, comparable to that characterizing the original plant layout (see Table 11).

Table 25: Modified layout. Medium-Scale plant off design behaviour.

HEX	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
HT1 – S&T	522.9	704.3	750.0	639.7	77.8	0.70	6059
HT2 – S&T	250.0	522.9	639.7	308.6	37.4	2.20	8701
HT3 – S&T	240.8	519.8	639.7	298.4	61.5	0.80	8959
HT4 – S&T	25.0	242.3	303.5	155.6	95.8	0.80	7379
HT1-HRSG	522.5	705.1	750.0	638.9	46.3	1.13	6100
HT2-HRSG	250.0	522.5	638.9	308.3	37.3	0.90	8687
HT3-HRSG	239.6	519.7	638.9	296.2	50.0	0.65	8996
HT4-HRSG	25.0	241.2	302.3	155.0	52.8	0.16	7342
T _{CI} : compressed air inlet temperature; T _{CO} : compressed air outlet temperature T _{HI} : atmospheric air inlet temperature; T _{HO} : atmospheric air outlet temperature; U: overall heat transfer coefficient; ΔP _s : shell side pressure drop; Q: exchanged thermal power.							

5.5.3 Large-Scale plant: Shell & Tube and HRSG design

High- and low-pressure air reference quantities for Large-Scale plant are given in Table 15. The resulting heat exchanger network is shown in Figure 20. As discussed in 5.4.3, slightly different assumptions have been made for Large-Scale plant. The higher LPT inlet pressure (8 bar instead of 6 bar assumed for Small and Medium-Scale plants) leads to a higher temperature at the HPT exit of 289°C instead of 250°C. The HPT exit temperature defines the pinch section of the heat exchanger network. The assumption of a pinch point temperature difference of 60° C (the same assumed for Small and Medium-Scale plants) leads to a higher atmospheric air temperature at HT4 inlet, which implies an exhaust temperature of some 178 °C, as shown in Figure 20. The required atmospheric air mass flow rate is of 225 kg/s, as reported in Table 26.

As for the original plant layout, several units in parallel arrangement have to be installed to exchange the relevant thermal power. The final part of the low-pressure air heating (HT1) can be accomplished by installing six parallel units, while the fulfilment of thermal loads in HT2, HT3 and HT4 require for each one four heat exchangers in parallel arrangement. The heat exchanger design found for Large-Scale plant is similar to that found for the Medium-Scale one, as can be seen by comparing Table 27 with Table 21 data. Therefore, considerations set out in 5.5.2 about shell side pressure losses, shell side nozzle dimensions and expected costs still stand.

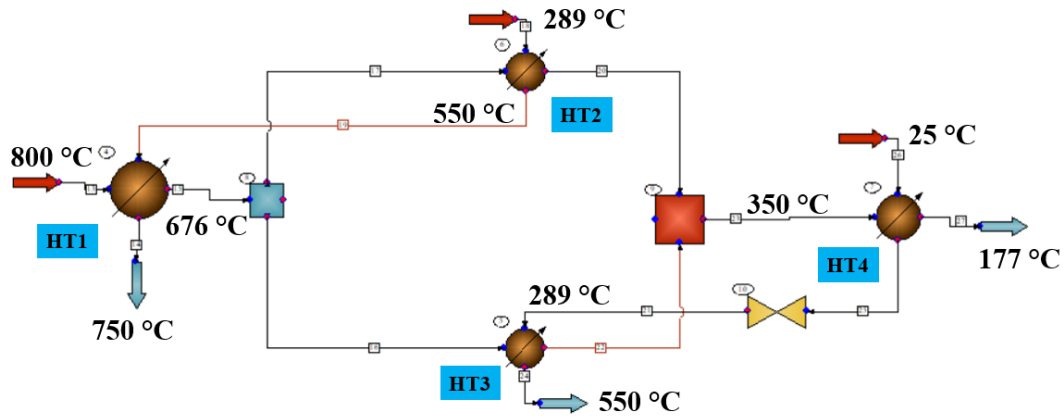


Figure 20: Large-Scale plant heat exchanger network.

Table 26: Modified Layout. Large-Scale plant. Shell & Tube design. Main process quantities.

HEX	mc (kg/s)	mH (kg/s)	T _{CI} (°C)	T _{CO} (°C)	T _{HI} (°C)	T _{HO} (°C)	Q (kW)
HT1	142.0	225.0	550.3	750.2	800.0	675.7	31830
HT2	142.0	112.5	288.7	550.3	675.7	351.6	39770
HT3	142.0	112.5	288.6	550.2	675.7	349.3	40040
HT4	142.0	225.0	25.0	288.7	350.4	176.6	40410

mc: compressed air mass flow rate; mH: atmospheric air mass flow rate; T_{CI}: compressed air inlet temperature; T_{CO}: compressed air outlet temperature; T_{HI}: atmospheric air inlet temperature; T_{HO}: atmospheric air outlet temperature; Q: exchanged thermal power.

Table 27: Modified Layout. Large-Scale plant. Shell & Tube design. Main design results.

HEX	L (m)	D _s (m)	do (mm)	n _s x n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
HT1 (x6)	6.0	2.4	16	1×1	2578	772	0.8	83.9
HT2 (x4)	7.3	2.4	16	1×1	6660	2436	1.8	45.1
HT3 (x4)	8.5	2.4	16	1×1	4100	1744	1.8	64.3
HT4 (x4)	8.5	2.4	16	1×1	2850	1217	0.9	83.0

L: tube length; D_s: shell diameter; N_t: number of tubes; do: tube outlet diameter; n_s: shell side number of passes; n_t: tube side number of passes S: heat transfer area; ΔP_s: shell side pressure drop; U: overall heat transfer coefficient.

The design based on the HRSG concept has been carried out by assuming DOFs given in Table 28. Free flow velocity and duct height/width ratio have been assumed in order to have a 10×10 m cross flow section. Basically, choices already made for the Medium-Scale plant have been confirmed. Bare tubes are selected for the bundle operating at the highest temperature (HT1), finned tubes for other bundles operating at lower temperature.

Main results are provided in Table 29. A relevant number of parallel paths is required to allocate the huge volumetric flow rate in low pressure bundles HT1 and HT2. In order to achieve an adequate number of turns per serpentine, largest available tube diameters have been selected for such tube bundles. Anyway, according to [26], a sufficient number of turns per serpentine is achieved in order to reach the low-pressure air target temperatures at HT1 and HT2 exit. High pressure bundles are less problematic. Due to the higher air density, volumetric air flow rates are considerably lower, and therefore bundles HT3 and HT4 can be arranged with smaller diameter tubes and with a lesser number of parallel paths. In this case, HT3 and HT4 tube bundles are really similar to superheating tube bundles commonly installed in real HRSG.

Table 28: Modified layout. Large-Scale plant. HRSG design. Assumed DOFs.

	HT1	HT2	HT3	HT4
Free flow velocity [m/s]	6.90	6.10	5.29	3.27
Duct height/width ratio	1	2	2	1
Tube diameter [mm]	73.0	60.3	42.2	48.3
Tube thickness [mm]	5.15	3.18	4.85	7.14
Tube arrangement	Inline	Inline	Inline	Inline
Tube type [Bare, Finned]	Bare	Finned	Finned	Finned
Transversal pitch*	1.44	1.33	1.45	1.40
Longitudinal pitch**	1.44	1.33	1.85	1.40
Fin type [Solid, Serrated]	Solid	Solid	Solid	Solid
Fins height [mm]	8.0	8.0	8.0	8.0
* transversal pitch to tube diameter ratio; * longitudinal pitch to tube diameter ratio				

Table 29: Modified layout. Large-Scale plant. HRSG design. Main results.

	HT1	HT2	HT3	HT4
Duct width (m)	10.0	-	-	-
Duct height (m)	10.0	-	-	-
Bundle length (m)	4.20	4.10	2.57	0.88
Tubes per row x Parallel rows	92 x 10	59 x 17	56 x 4	115 x 1
Number of turns x Serpentine	4	3	7	11
Number of tubes	3680	3009	1568	1265
Total surface	8495	9257	8765	7567
Fin per meter	-	33	104	115
Overall heat transfer coefficient (W/m ² °C)	46.1	47.4	51.3	53.5
Shell side pressure drop (kPa)	0.95	1.18	1.18	0.23
Tube side pressure drop (kPa)	15.79	16.51	143.00	282.6

Table 30: Modified layout. Large-Scale plant: comparison between S&T and HRSG approaches

	Shell & Tube	HRSG
Overall heat transfer surface (m ²)	~ 26000	~ 34000
Atmospheric air pressure drop (kPa)	3.5	2.4
High pressure air tube side pressure drop (kPa)	~ 103	~ 425
Low pressure air tube side pressure drop (kPa)	~ 27.5	~ 32.2

5.6 Capital cost estimation

Cost has been estimated by using two different correlations for Shell & Tube heat exchanger and for heat transfer equipment based on the HRSG concept. Additionally, the cost estimation has been validated through consultation with an industrial partner with high expertise in the field.

Shell & Tube heat exchangers costs have been calculated by adopting a formulation proposed by [32] achieved by reworking and improving a correlation proposed by NETL [33]. First of all, a base component cost is estimated as a function of heat transfer area. Base cost refers to a component operating at ambient pressure and made of carbon steel. Suitable factors are then applied to account for actual operating pressures, materials and labour. The correlations applicable in a real wide range (25-5000 m²), and therefore, can cover Small, Medium and Large-Scale plant designs. The correlation is applicable also in case of externally finned tubes.

The investment cost estimation for heat transfer equipment based on HRSG concept has been carried out by adopting the methodology developed by Foster-Pegg in 1986 for HRSG [34] subsequently updated by other Authors [35, 36]. The base cost of such component is given by the sum of three terms, related to the heat transfer surface area, the equipment necessary for the appropriate management of the compressed air (manifolds, valves, couplings and so on) and of the atmospheric air stream (enclosure, thermal insulation and chimney). A cumulative factor is introduced to take account of direct and indirect costs.

Both for Shell & Tube and HRSG approach, actual cost is evaluated by introducing a material cost factor assumed in agreement with [33]. The base cost evaluated based on the heat transfer area is corrected by introducing material factors given in Table 31.

Table 31: Material cost factors.

Material	Material factor
Carbon Steel C45	1
Stainless Steel 347	2
Inconel 625	8

Costs have been updated by applying the Chemical Engineering Plant Cost Index (CEPCI). All cost figures are given in 2024 USD.

Capital costs evaluated for the original plant layout are provided in Table 32. In this case, only the Shell & Tube based approach has been addressed. It can be noticed that the cost of HHEX2 constitutes almost the entire capital cost. This is because of two reasons: i) the larger heat transfer surface of HHEX2 in respect to HHEX1, ii) the more costly material adopted for HHEX2 to withstand the higher working temperature.

Table 32: Original plant layout. Shell & Tube heat exchangers capital cost.

HEX	Cost ₂₀₂₄ [\$]
Small size	
HHEX1	280.000
HHEX2	2.280.000
Total	2.560.000
Medium size	
HHEX1	1.250.000
HHEX2	13.750.000
Total	14.000.000
Large size	
HHEX1	3.100.000
HHEX2	44.300.000
Total	47.400.000

The cost per unit of surface decreases from 9200 to some 4000 \$/m² moving from Small to Medium-Scale plant. For Large-Scale plant, the cost per unit of installed surface is close to that found for Medium-Scale plant. In fact, as discussed in 5.4.3, the Large-Scale plant heat transfer equipment is constituted by parallel units in parallel arrangement similar to those designed for the Medium-Scale one.

Table 33: Modified plant layout. Shell & Tube heat exchangers capital cost.

HEX	Cost ₂₀₂₄ [\$]
Small size	
HT1	1.200.000
HT2	430.000
HT3	470.000
HT4	200.000
Total	2.300.000
Medium size	
HT1	5.700.000
HT2	3.300.000
HT3	2.400.000
HT4	800.000
Total	12.200.000
Large size	
HT1	22.100.000
HT2	12.300.000
HT3	9.200.000
HT4	3.400.000
Total	47.000.000

Table 33 shows capital costs evaluated for the modified plant layout in the case of Shell & Tube based design. For different plant sizes taken into consideration (Small, Medium and Large-Scale), costs are of the same order of those found for the original plant layout.

Table 34: Modified plant layout. HRSG based design capital cost.

HEX	Cost₂₀₂₄ [\$]
Medium-Scale	
HT1 – material	2.300.000
HT2 – material	700.000
HT3 – material	700.000
HT4 – material	300.000
Total	10.000.000
Large-Scale	
HT1 - material	8.000.000
HT2 - material	2.200.000
HT3 - material	2.300.000
HT4 - material	1.000.000
Total	37.000.000

Cost per unit of installed surface, instead, show noticeable differences. In fact, a value of some 6000 \$/m² is found for the Small-Scale plant, which is significantly less than that of 9200 \$/m² estimated for the original plant layout. Similarly, Medium and Large-Scale plants show a cost of about 2000 \$/m², that is half of the cost found for the original layout. This is mainly due to the following reasons: i) the possibility of using a less expensive material (C45 carbon steel) to manufacture the thermal unit HT4 operating in the lowest temperature interval, ii) the lesser extent of costly heat transfer surface required for high temperature operation in HT1.

Finally, capital costs evaluated for HRSG based design are given in Table 34. It has to be pointed out that cost figures provided for HTs refer to costs related to the installed heat transfer area, while the total cost includes the cost of equipment required for the management compressed and atmospheric air streams.

Compared to the Shell & Tube based design, capital costs show a significant decrease. For both Medium and Large-Scale plant, the HRSG based design shows a cost that is roughly 80% of that found applying the Shell & Tube design approach. Such a result, taking also other aspects previously discussed into consideration, makes the design approach based on HRSG technology preferable.

6 Conclusions

This document considers the design of high-temperature heat transfer equipment needed to heat and reheat the compressed air before the expansion in High-Pressure and Low-Pressure turbines. The design has been carried out for 3 different power scales: Small Size Plant (1MWe discharging power), Medium Size Plant (20 MWe discharging power) and Large Size Plant (100+ MWe discharging power).

Two different plant layouts are considered. In the original plant layout, two parallel atmospheric air streams are used to heat and reheat the pressurized air before expansion in High- and Low-Pressure turbines. In the modified plant layout, the above task is accomplished by using a single stream of hot atmospheric air.

In the case of the Small-Size Plant, original plant layout, the regenerative design as well as a Shell & Tube design is considered.

In the case of the regenerative approach, a length of 9 m has been assumed for both high pressure and low-pressure regenerative beds. For both beds, a flange diameter DN600 has been selected for charging operations. For discharging operations, instead, a flange diameter DN150 and

DN300 has been selected for high pressure and low-pressure bed, respectively. An acceptable pressure drop is obtained, and the expected switching time is around 30 min (slightly longer for the high-pressure system – case 6; and slightly shorter for the low-pressure system – case 6 LP). The resulting total area of heat transfer for one bed is about 1323 m². The total cost of the high-pressure and low-pressure regenerative systems is estimated at about 1 M€ each, i.e. some 1.160 M USD. The total cost of the regenerative system is thus estimated at about 2.32 M USD.

The Shell & Tube solution has a total heat transfer area of some 280 m² and an estimated cost of about 2.56 M USD. Therefore, from the economic point of view, both solutions, the regenerative design as well as the Shell & Tube design have very similar CAPEX, being the regenerative solution slightly cheaper. Nevertheless, considering the higher complexity of design and operation related to the regenerative approach, the classic Shell & Tube design seems to be the preferred choice. Additionally, the needed valves are size-limiting components and are very expensive. The availability of valves imposes a clear upscaling limit, since maximum valve sizes available for such demanding conditions are around DN600 to DN800. The regenerative solution is therefore only viable for the Small-Scale plant.

Heat transfer equipment for Medium and Large-Scale plants operating according to the original layout have been designed by adopting an approach based on Shell & Tube technology. In Medium-Scale plant, two heat exchangers are sufficient to heat and reheat the pressurized air stream, while the Large-Scale plant calls for multiple units operating in parallel arrangement. Estimated capital costs are in the order of 14.0 and 47.4 M USD, respectively.

Concerning the modified plant layout, in addition to the traditional Shell & Tube design an innovative approach based on the well proven HRSG technology has been addressed. The proposed approach is suitable only for Medium and Large-Scale plants. For the Small-Scale Plant, the classic Shell & Tube solution is fully sufficient as the needed flow cross sections are small enough.

For the Medium-Scale Plant as well as for the Large-Scale Plant, the optimum design solution is the design according to the HRSG approach, that offers advantages in terms of cost and lesser plant complexity.

In conclusion, it has to be pointed out that the relevant output of WP3 activities consists in the development of a set of general design indications and guidelines applicable to any plant layout established in compliance to the ASTERIX-CAESAR concept. Moreover, results here presented and discussed, constitute the input for activities to be carried out in the last period of WP3.

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