



Project 101122231 — ASTERIX-CAESar

D3.1 – Optimum low-temperature heat exchanger design (reuse heat of compression)

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Executive Summary

The current deliverable (D3.1) is entitled “Optimum low-temperature heat exchanger design (reuse heat of compression)”. It is a public document produced within Task 3.1 “Low-temperature heat exchanger design (reuse heat of compression) - 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 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 layouts and characteristics for the overall CAES-CSP plant.

This report focuses on the design of Low-Temperature Heat Exchangers (LHEX), which main objective is intercooling the pressurized air between the compressors of the compression train. In the proposal phase of the project, it was contemplated to reuse heat of compression for the discharging process through a Thermal Energy Storage (TES). Nevertheless, outcomes from WP1 and the design of an innovative High-Temperature Heat Exchanger (HHEX) similar to a Heat Recovery Steam Generator (HRSG) in D3.2, brings about a certain amount of waste heat from the solar field that can be directly used for the preheating function. Thus, the design of LHEX is no longer limited to its integration with a low-temperature TES, but the heat generated during the charging phase is proposed to be used for providing industrial process heat.

In the case of the Small-Size Plant, the regenerative design as well as a Shell & Tube design is considered. The total CAPEX of the regenerative system is estimated at 1.7 M USD. It must be noted that this cost already includes the cost of the low-temperature thermal energy storage unit. The Shell & Tube solution has an estimated cost of about 1.13 M USD. In the latter case, the additional cost of the low-temperature TES needs to be considered, when the heat should be stored for later use. However, it is expected that the cost of the TES will be substantially below the resulting difference of 570 kUSD, which makes the regenerative design economically not viable in any case. Hence, the Shell & Tube solution is the better choice, also in terms of simplicity of process control and valve operation.

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 as well as thermodynamic performance. For the Small-Scale Plant, the classical Shell & Tube solution is fully sufficient as the needed flow cross sections are small enough.

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 (W/m ² K)
ΔP	Pressure Drop (Pa)
RTE	Round-Trip Efficiency
HRSG	Heat Recovery Steam Generator

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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. A detailed parametric design study is conducted using 1-D Modelica models, which are further refined by integrating heat transfer and pressure drop parameters obtained from CFD simulations (if necessary) and experimental data. 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 cost, performance, environmental impact, and recyclability. 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 low 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.

Considering the design approach (ii), dedicated Modelica distributed parameter models have been developed to simulate the intrinsic dynamic behaviour of tubular regenerator-type heat exchanger configurations.

For both approaches extensive campaigns aimed at defining the optimal low-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, 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 low-temperature heat transfer equipment have been carried out mainly by CENER and ROMA3. Their contributions focused on the sizing activities 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.

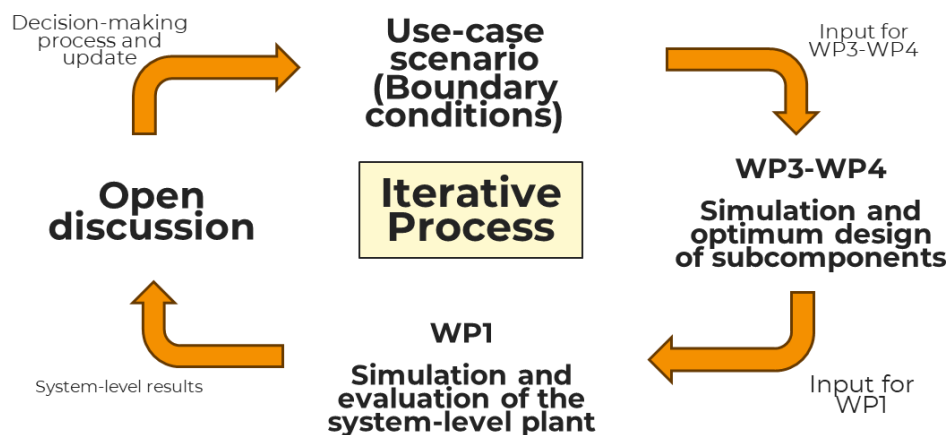


Figure 1. Iterative methodology carried out for interaction between WP1-WP3-WP4.

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 3 power ratings. 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.

Figure 2 and Figure 3 illustrate the initial plant scheme of the proposed concept, in which the heat generated during compression is stored in a low-temperature thermocline TES system. However, findings from WP1 have demonstrated that substituting the low-temperature TES with an industrial process heat supply may offer a more cost-effective solution with greater economic potential (see Figure 4 and Figure 5). Here, also alternative heat transfer fluids, such as water may be considered.

To explore the potentiality of the proposed concept 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.

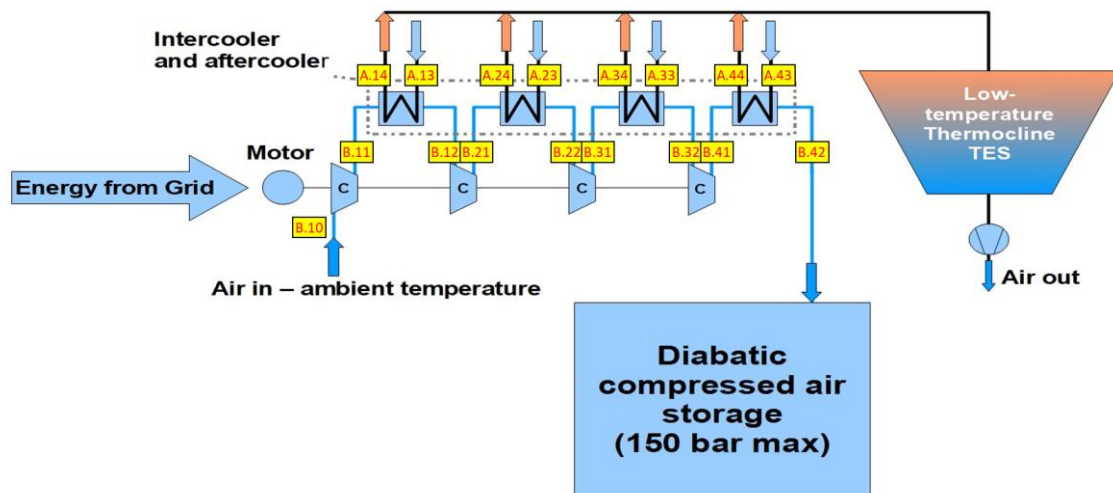


Figure 2. ASTERIX-CAESar original plant scheme with low-temperature TES. Charging process.

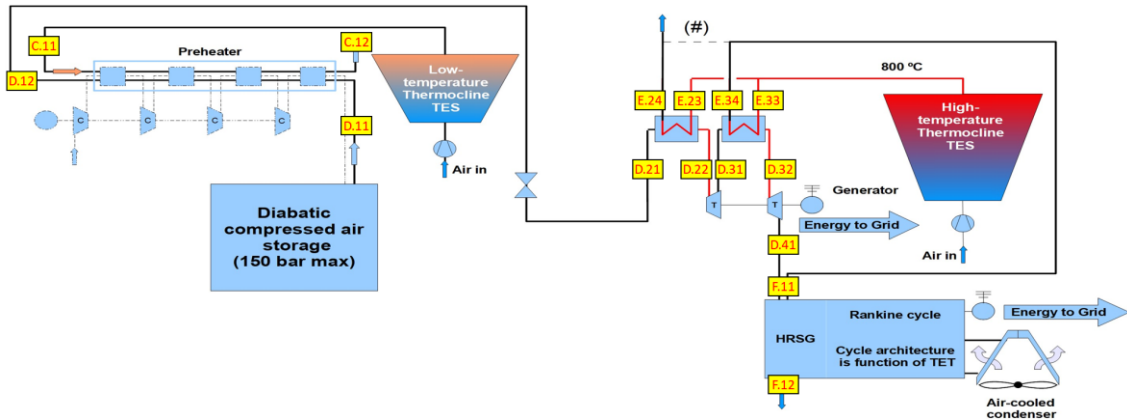


Figure 3. ASTERIX-CAESAR original plant scheme with high-temperature shell-and-tubes HEX. Discharging process.

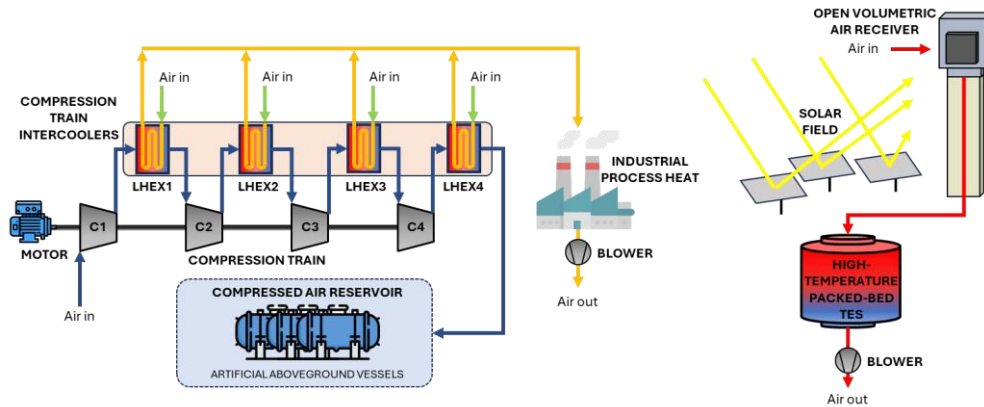


Figure 4. Modified plant concept with connected industrial process. Charging process.

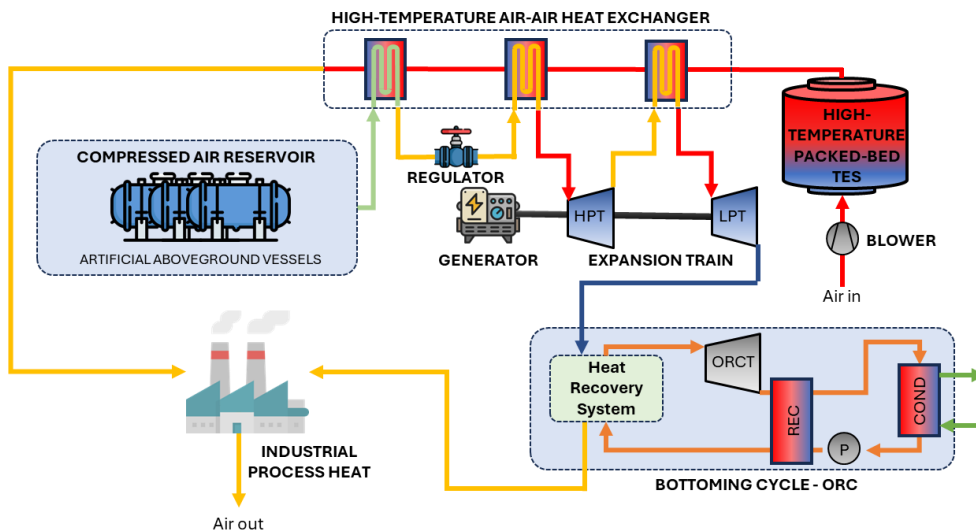


Figure 5. Modified plant concept with high-temperature air-air HEX similar to HRSG and with connected industrial process. Discharging process.

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 [2]. In the ASTERIX-CAESAR project fixed-matrix-type regenerators are addressed

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 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 intercooling of compressed air, around 200 °C of maximum nominal operating temperature is expected. Charge and discharge cycle times are between 3 and 6 hours. For the specific design, general knowledge of regenerator design can be applied [2]. Due to the rather low temperatures observed, low-cost matrix materials, such as rock, slag or glass can be used. Nevertheless, due to the high operating pressure (up to about 150 bar) special attention must be given to the design of the regenerator casing and valve system for 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, especially when the temperature approach between hot and cold streams is reduced, when overall heat transfer coefficients are not particularly high, or huge volumetric mass flow rates occur, which applies in the present case.

In fact, with reference to the plant lay out sketched in Figure 2, it is evident that the thermodynamic performance of the overall system can be improved: i) by reducing the compressed air temperature at the inlet of each compression stage, to reduce the compression work; ii) by increasing the temperature of the atmospheric air streams at the exit of each heat exchanger, in order to store thermal energy at the highest possible temperature. The attainment of such goals implies a reduced temperature difference between hot and cold stream in correspondence of each cross section of the heat exchanger.

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, 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.

Finally, 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.

In conclusion, the particular application 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 [2]:

- 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.

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 motivation of the regenerative approach is to integrate the heat exchanger and the thermal energy storage unit in the same component, the regenerative bed. Furthermore, cheap materials such as natural rocks, glass or ceramic waste material can be used as filler material for the regenerative bed, which presents a cost advantage. However, the duct design is more complex, as several tubular ducts may need to be placed in parallel and the flow needs to be controlled with appropriate valves.

The general principle is the following, as shown in Figure 6 and Figure 7.

4.1 The regenerative L-HEX design approach

The basic idea is the integration of low-temperature thermal energy storage (L-TES) and heat exchanger (L-HEX) in the same component, by applying a direct contact regenerative heat exchanger concept, in order to reduce size and cost. Regenerators are well known for high specific heat transfer areas and very high heat exchange effectiveness [2]. Also, when ceramic material is

used as regenerator matrix, substantial cost savings can be achieved with respect to conventional shell-and-tube heat exchangers of similar power rating. For example, cordierite ceramic honeycomb structures (as widely applied in catalytic converters and high temperature gas filtration, etc.) are orders of magnitude cheaper than metal alloys that would be needed for the shell-and-tube heat exchanger design.

A key design requirement is the adequate insulation of the regenerator pressure vessels as the regenerator shell must support very high pressures and must therefore be kept at sufficiently low temperatures. For the case of the high temperature heat exchangers (covered in D3.2), internal insulation is the only way, since the operating temperature is as high as 800°C. However, for the case of the low-temperature system, where the maximum outlet temperatures of the compressors is around 200°C, an external insulation of the tubular system can be considered, which reduces the cost of insulation substantially.

The idea is to apply a network of parallel high-pressure pipelines that are externally insulated and filled with suitable material to obtain a packed-bed thermocline heat storage concept. Thus, between each compressor stage a packed-bed TES pipe network fulfils the task of air intercooling (aftercooling) as well as heat storage (see Figure 6, where only two compressor stages are shown for simplicity of the graphic). During charging state, the regenerative pipe-networks are connected in series, in between the compressor stages. During discharge operation, the regenerative networks are used in parallel, preheating the compressed air upstream the expander train (see Figure 7).

It is crucial to correctly design flow cross section, i.e. number of parallel pipes (only 3 are shown in Figure 6 and Figure 7), such that the pressure drop between compressor stages is acceptable. Since the pressure increases from stage to stage, and flow velocity should be kept constant, the number of parallel pipes is decreased for each consecutive stage. However, in order to keep the storage capacity similar, the length of each unit needs to be adjusted accordingly.

The cut-off conditions are chosen such that the charging of the regenerator is stopped once the outlet temperature rises to 50°C. The discharging is stopped once the outlet temperature falls to 150°C.

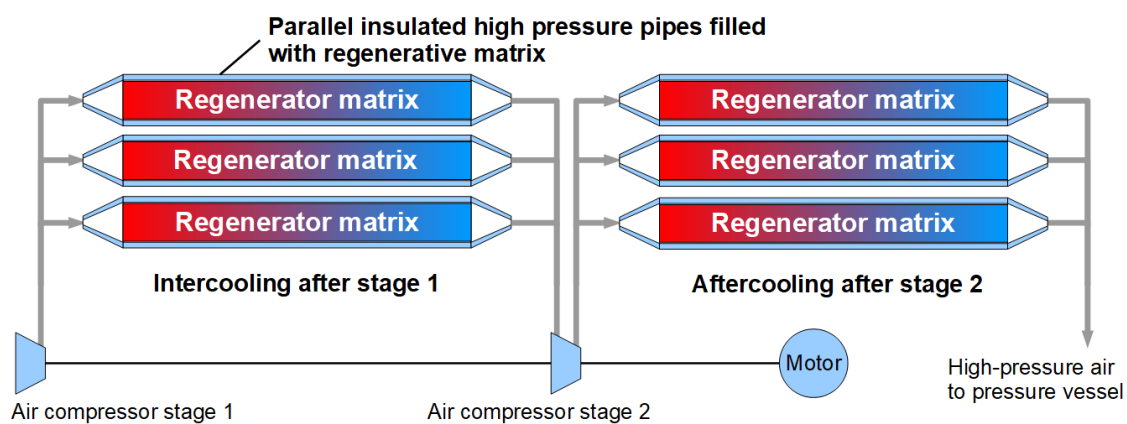


Figure 6: Low-temperature (adiabatic CAES) heat exchanger (L-HEX) and thermal energy storage (L-TES) as regenerative system (in charging state)

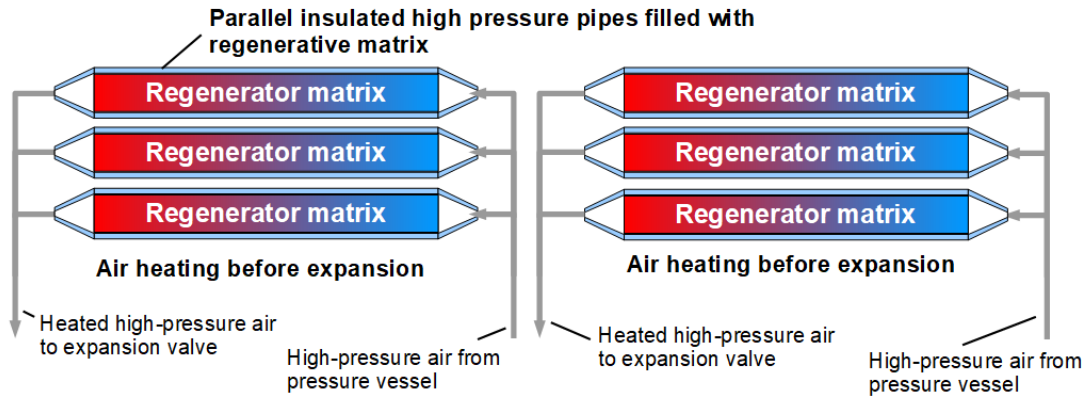


Figure 7: Low-temperature (adiabatic CAES) heat exchanger (L-HEX) and thermal energy storage (L-TES) as regenerative system (in discharging state)

4.2 The modelling methodology

Given the importance and already widespread usage of Modelica [3] 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 [3].

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) [4].

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 [5], 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.

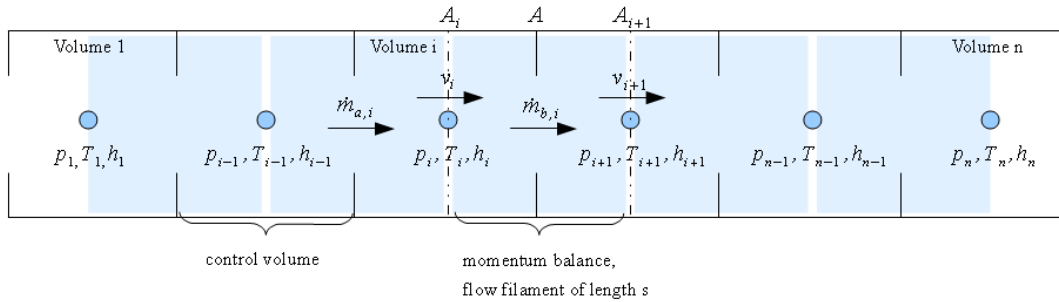


Figure 8: Finite volume discretization scheme according to the staggered grid approach [1]

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. [6]. 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 [7]:

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

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. [8]. 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. [7] 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. [9].

The pressure drop model

The pressure drop across a packed bed may be calculated according to the following correlation as published by Ergun [10]:

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

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. [7] 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. [11]. For the subsequent application of the model with air as HTF, air has been modelled as ideal gas, according to Refs [12, 13]. Note that the model has been also validated with air as HTF in the work of Ramacciotti [14].

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. [15] 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.

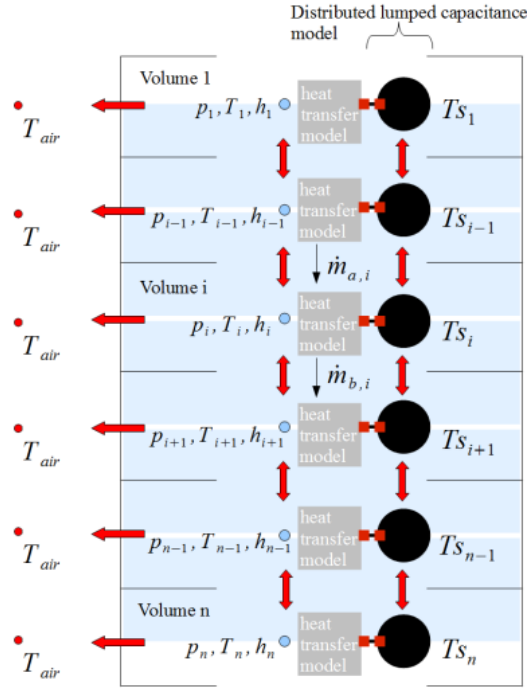


Figure 9: Thermocline TES model scheme

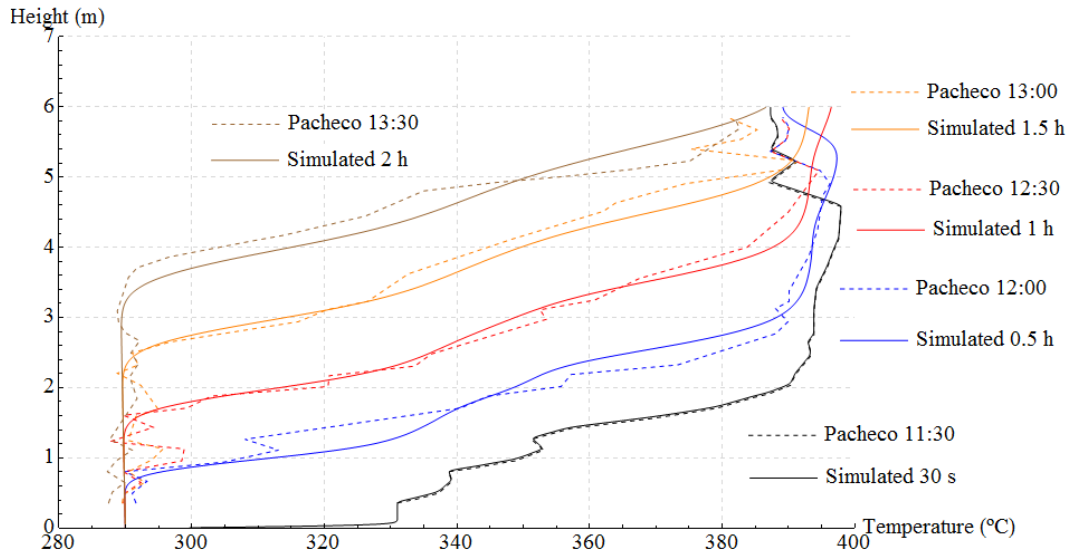


Figure 10: Thermocline TES model experimental validation against data from Ref. [9]

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 [16, 17]. 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 [16, 17].

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

10 [18]. Additionally, the model has been further validated in the work of Ramacciotti [14].

4.3 Discussion of results

The above-described model of a thermocline TES has been applied to simulate the behavior of regenerative beds that are assumed to be connected to the compressor stages. Since the dimensions of the regenerative beds will be quite long (long tubular ducts), the installation will be horizontally above ground level. Hence, the model scheme shown in Figure 9 should be inclined by 90 degrees. Nevertheless, the governing equations do not change.

As the limiting component is expected to be the control valve size, the regenerative concept is considered to be suitable only for the small-scale size of the plant (1MWe discharging). In this case, we would have a nominal air mass flow of approximately 1.4 kg/s during discharging (3 hours), and a nominal compressor mass flow of 0.7 kg/s (6 hours charging). By limiting the maximum flow velocity in the thermocline to reasonable values to below 3 m/s, and assuming a nominal diameter of the steel pipes of DN600, one duct would be needed for the first intercooling unit. The procedure is outlined in the following.

Assuming a maximum storage pressure of 150 bar and by distributing the pressure ratio over all 4 stages, a nominal discharge pressure of about 3.5 bar is obtained for the first compressor stage. Hence, during charging, the first regenerator unit between stage 1 and 2 is operated at about 3.5 bar pressure.

A charging time of 6 hours and discharging time of 3 hours has been assumed. This means that the length of the regenerative bed has to be chosen accordingly, in order to provide sufficient thermal energy storage capacity at acceptable pressure drop across the packed bed.

By choosing a nominal diameter of DN600, a total pipe length of 40 meters would be required (one single pipe) to provide 6 hours of charging. The pressure drop is expected to be at around 0.09 bar, being the maximum flow velocity at around 2.6 m/s. By designing 2 ducts in parallel, with a duct length of 20 meters each, the mass flow per duct is reduced to 0.35 kg/s, being the expected pressure drop at around 0.012 bar (1.3 flow velocity) (see Table 1). Since one duct already provides a reasonable fluid velocity and acceptable pressure drop, 1 duct of 40 m length seems to be sufficient.

Table 1: Duct mass flow, velocity and pressure drop for regenerator after first compressor stage (3.5 bar)

Number of parallel regenerative ducts (-)	Length of DN600 duct (m)	Air mass flow per duct (kg/s)	Maximum velocity (m/s)	Total pressure drop (bar)
1	40	0.7	2.6	0.09
2	20	0.35	1.3	0.012

Assuming a maximum storage pressure of 150 bar and by distributing the pressure ratio over all 4 stages, a nominal discharge pressure of about 3.5 bar is obtained for the first compressor stage. The second compressor stage has then a discharge pressure of about 12.25 bar. Hence, during charging, the second regenerator unit between stage 2 and 3 is operated at about 12.25 bar pressure.

By choosing again a nominal diameter of DN600, a total pipe length would be required of 40 meters (one single pipe) to provide 6 hours of charging. The pressure drop is expected to be at around 0.026 bar, being the maximum flow velocity at around 0.75 m/s. By choosing 2 parallel ducts of 20 meter length, the pressure drop can be reduced to about 0.0035 bar, being the maximum flow velocity at around 0.375 m/s (see Table 2). However, one duct of 40 m length

seems to be enough, as well.

Table 2: Duct mass flow, velocity and pressure drop for regenerator after second compressor stage (12.25 bar)

Number of parallel regenerative ducts (-)	Length of DN600 duct (m)	Air mass flow per duct (kg/s)	Maximum velocity (m/s)	Total pressure drop (bar)
1	40	0.7	0.75	0.026
2	20	0.35	0.375	0.0035

The third compressor stage has a discharge pressure of about 42.9 bar. Hence, during charging, the third regenerator unit between stage 3 and 4 is operated at about 42.9 bar pressure.

By choosing again a nominal diameter of DN600, a total pipe length would be required of 40 meters (one single pipe) to provide 6 hours of charging. The pressure drop is expected to be at around 0.008 bar, being the maximum flow velocity at around 0.22 m/s. By choosing 2 parallel ducts of 20 meter length, the pressure drop can be reduced to about 0.001 bar, being the maximum flow velocity at around 0.11 m/s (see Table 3). Hence, also for the third stage of regenerative bed, just one duct seems to be sufficient.

Table 3: Duct mass flow, velocity and pressure drop for regenerator after second compressor stage (42.9 bar)

Number of parallel regenerative ducts (-)	Length of DN600 duct (m)	Air mass flow per duct (kg/s)	Maximum velocity (m/s)	Total pressure drop (bar)
1	40	0.7	0.22	0.008
2	20	0.35	0.11	0.001

The forth compressor stage has a discharge pressure of about 150 bar, the final maximum vessel pressure. Hence, during charging, the fourth regenerator unit after stage 4 and is operated at about 150 bar pressure.

By choosing again a nominal diameter of DN600, a total pipe length would be required of 40 meters (one single pipe) to provide 6 hours of charging. The pressure drop is expected to be at around 0.002 bar, being the maximum flow velocity at around 0.06 m/s. Hence, for the fourth stage of regenerative bed, just one duct seems to be sufficient, as well (see Table 4).

Table 4: Duct mass flow, velocity and pressure drop for regenerator after second compressor stage (150 bar)

Number of parallel regenerative ducts (-)	Length of DN600 duct (m)	Air mass flow per duct (kg/s)	Maximum velocity (m/s)	Total pressure drop (bar)
1	40	0.7	0.06	0.002

In summary, a total of 4 ducts of 40 m lengths are required for the charging phase. Each regenerative bed is assumed with the following parameters as given in Table 5.

Table 5: Packed-bed properties

Bed filler material	Size of crushed rocks (cm)	Bed porosity (-)	Bed heat transfer area (m ²)
quartzite rock	3	0.35	1470

The next step is the evaluation of the discharging operation, hence preheating of the pressurized air leaving the pressure vessel. Here, the nominal mass flow is increased to 1.4 kg/s and the operating pressure varies from 150 bar at the start of the discharging operation, down to approximately 40 bar, when discharging stops. According to the above, 4 fully charged ducts of 40 m lengths are available. To find the most suitable discharging strategy, some options need to be considered.

Since the discharging operation is only half of the time (3 hours), it is likely that not all four ducts are required for air preheating. In fact, according to the simulations, one of the four ducts seems to be fully sufficient for air preheating during discharging operation. That means that the remaining 3 can be used for process heat supply.

Table 6: Regenerative system discharging operation options

Number of parallel regenerative ducts (-)	Length of DN600 duct (m)	Air mass flow per duct (kg/s)	Duct pressure (bar)	Maximum velocity (m/s)	Total pressure drop (bar)	Discharging time (h)
1	40	1.4	40	0.46	0.035	3
1	40	1.4	150	0.12	0.009	3
2	40	0.7	40	0.23	0.008	6
2	40	0.7	150	0.06	0.002	6

In Table 6, velocities and pressure drops are shown for the highest storage pressure level and the lowest expected pressure level. The real velocity values will be in between these two boundaries as function of discharging state. Since one DN600 duct already provides sufficient thermal energy to cover the whole discharging phase (3 hours), the maximum velocity in the TES system will range between 0.46 m/s to 0.12 m/s (see Table 6).

The first three regenerative beds can be used for process heat supply and need to be discharged close to atmospheric pressure. Since the blower consumption needs to be kept low, the discharging time should be as long as possible. Hence, having a charging time of 6 hours (compressor operation), about 18 hours would be available for process heat supply (discharging of the 3 process heat regenerators). According to simulations, the mass flow per bed should be kept at about 0.3 kg/s, which would cause a pressure drop per regenerative bed of about 5 to 6 kPa (flow velocities of about 4 m/s).

According to these findings, Figure 11 shows a possible design scheme, having the regenerative beds arranged in U-shape, so that the compressor stages can run on the same shaft. The nominal diameter of the regenerative bed ducts is DN600. 11 valves would be needed to allow the charging and discharging operation. V1 to V6 would be DN600 valves, V7 to V11 would be

DN200 valves. The diameters are chosen such that the air speeds are kept at a reasonable maximum value at about 6 to 7 m/s.

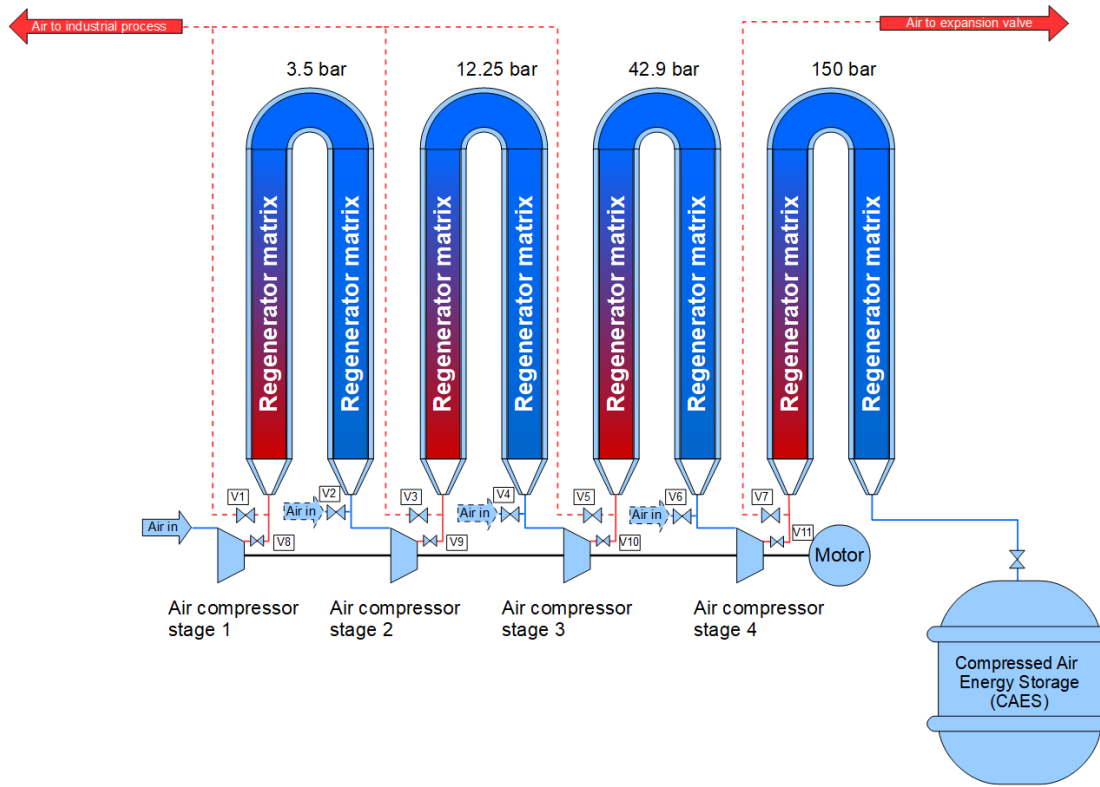


Figure 11: Regenerative system scheme with one regenerative bed after each compressor stage

Below, a list of all needed valves is shown (Table 7).

Note that the temperature-pressure rating of connecting flanges needs to be considered according to EN 1092-1. For example, when considering V7 in, PN250 needs to be taken, since an operating pressure of 150 bar at 200°C is not allowed with PN160 rating. When assuming AISI304 (1.4301) as flange material, the maximum allowed pressure at 200°C for PN160 would be 119.6 bar, whereas the maximum allowed pressure for PN250 is 186.9 bar.

Table 7: List of valves of regenerative heat exchange 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)
V1	DN600	PN16	2.5	1.4	1	3.5	200
V2	DN600	PN16	2.5	1.4	1	3.5	200
V3	DN600	PN25	11.25	1.4	1	12.25	200
V4	DN600	PN25	11.25	1.4	1	12.25	200
V5	DN600	PN63	41.9	1.4	1	42.9	200
V6	DN600	PN63	41.9	1.4	1	42.9	200
V7	DN150	PN250	149	1.4	40 - 150	150	200

V8	DN200	PN16	0	0.7	3.5	1	200
V9	DN200	PN25	0	0.7	12.25	1	200
V10	DN200	PN63	0	0.7	42.9	1	200
V11	DN150	PN250	149	0.7	150	40 - 150	200

Valves number V1, V2, V3, V4, V8 and V9 could be of type PFEIFFER BR 14b. Valves number V5, V6 and V10 could be of type SAMSON LTR 43. Valves number V7 and V11 could be of type SAMSON 3251-1. The latter design suggestion would be a globe valve, instead of butterfly valve, due to the high operating pressure.

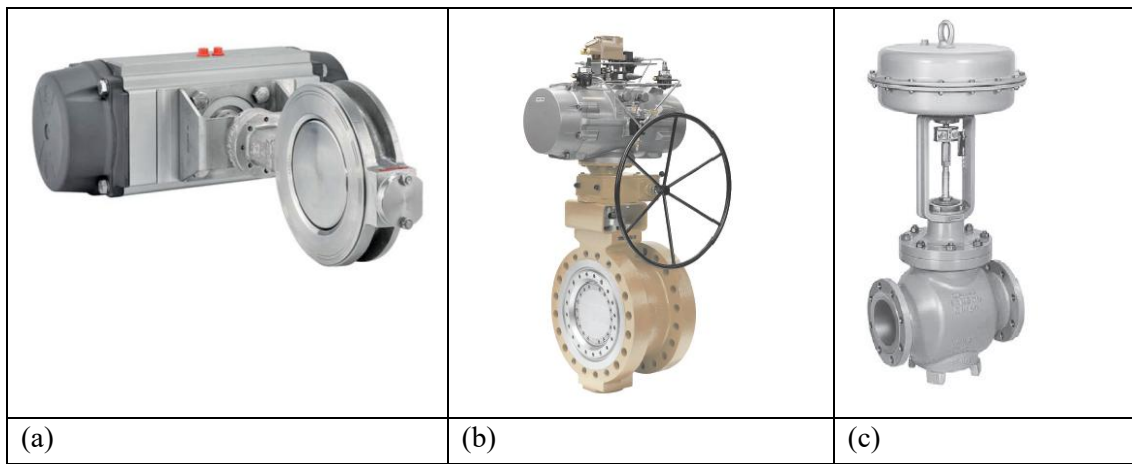


Figure 12: Valve suggestions – PFEIFFER BR 14b (a) - SAMSON LTR 43 (b) – SAMSON Type 3251 (c)

Taking into account available fittings with standard dimensions, the following 3-D figure has been created (Figure 13). The figure shows two DN200 flange connections, where the compressor outlet of stage 1 and the compressor inlet of stage 2 would be connected. Additionally, Figure 13 shows the large DN600 flange connections for the discharging operation close to ambient pressure for process heat supply. Since the pressure drop needs to be kept as low as possible to keep blower consumption at an acceptable level, the DN600 flow path is supposed to be designed without unnecessary flow restrictions. Additionally, the filling process of the regenerative duct needs to be considered. Several access flanges need to be foreseen to enable the filling of the regenerative matrix with small rocks for example. These access points need to be closed with blind flanges after the filling is completed (see Figure 13 and Figure 14). Figure 14 shows the general dimensions of the representative regenerative duct.

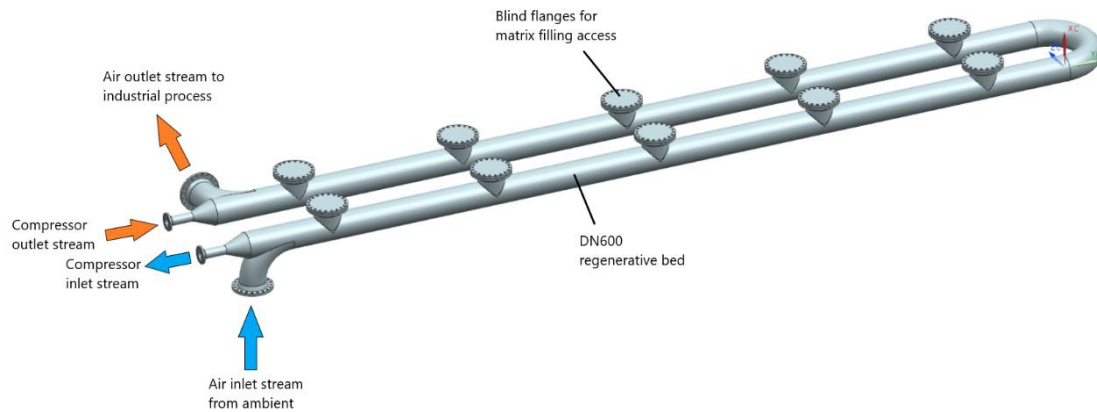


Figure 13: 3-D view of regenerative bed assembly of one U-turn (DN200 compressor connections – DN600 discharging connections) – see Figure 11.

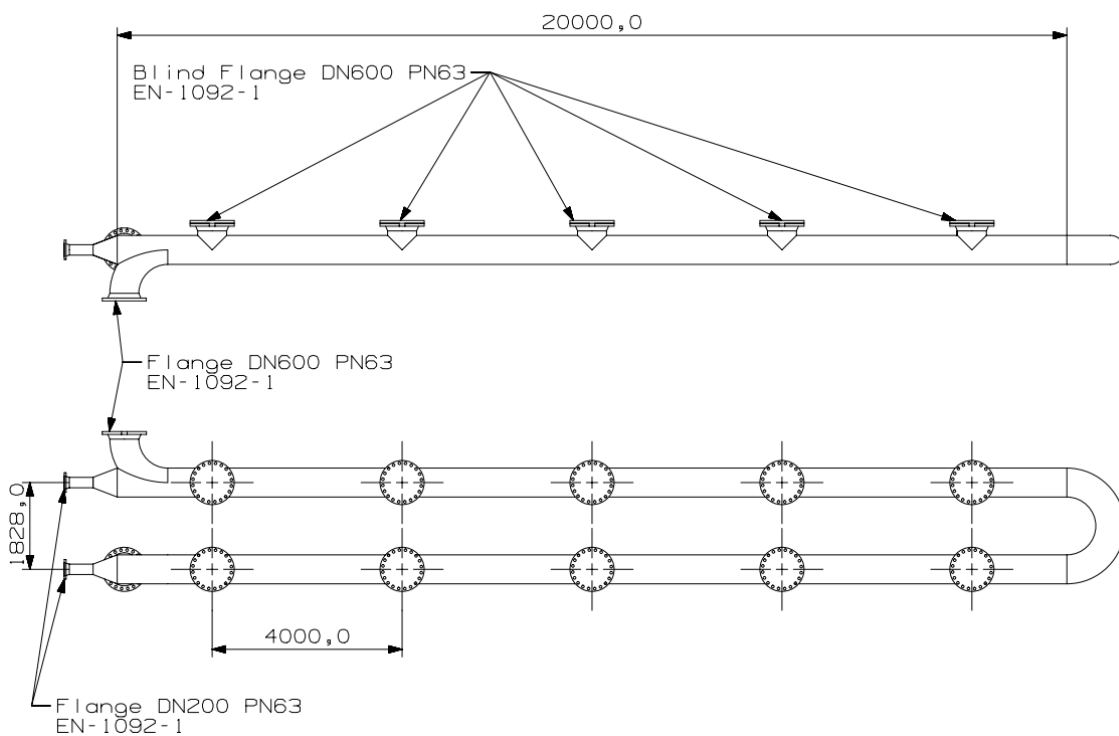


Figure 14: General dimensions (mm) of one regenerative bed unit according to

Regarding material selection of the regenerative bed unit, the application of high-pressure pipeline grade seamless pipes (API 5L X80 grade or similar with maximum allowable stress above 200 MPa [19]) is proposed. The choice of adequate steel type is important since the wall thickness should be kept reasonable (< 25 mm). High-strength, low-alloy steel (e.g. API 5L X80) used primarily in the oil and gas industry for pipeline transportation systems seems to be a good choice.

4.4 Cost estimation

The cost of all 11 valves can be estimated at around 450 k EUR based on obtained manufacturer quotes.

The cost of one regenerative duct (metal shell + filler material + thermal insulation) is estimated at 250 k EUR.

These figures give a total cost of about 1.5 M EUR (1.7 M USD) for the whole regenerative system.

5 Recuperative approach

5.1 Low temperature heat exchangers design issues

The plant arrangement sketched in Figure 2 has been taken as reference for the design of low temperature heat exchangers. In such a plant arrangement, the heat removed by cooling the compressed air after each compression stage is stored in a Low Temperature Thermal Energy Storage (LTES) to be re-used during the discharge phase. At the exit of each compression stage, heat is re-moved from compressed air by means of a stream of air at atmospheric pressure passed through the heat transfer device by using a fan. Subsequently, the hot atmospheric air is sent to a thermocline LTES, where the thermal energy of the air is transferred to a rock packed bed.

Overall plant thermodynamic performance improvements can be achieved by increasing the temperature of the stored heat and, therefore, by increasing the temperature of the atmospheric air exiting each intercooler. Another way to improve the overall plant performance is the reduction of the specific compression work, achievable, inter alia, by lowering the temperature of the compressed air fed to each compression stage.

In exploring the recuperative approach potentialities, the classical Shell & Tube (S&T) arrangement has been addressed first. As an example, a schematic representation of a recuperative intercooler is given in Figure 15, together with a typical temperature profile of both streams involved in the heat transfer process. Data refer to a typical medium size plant application.

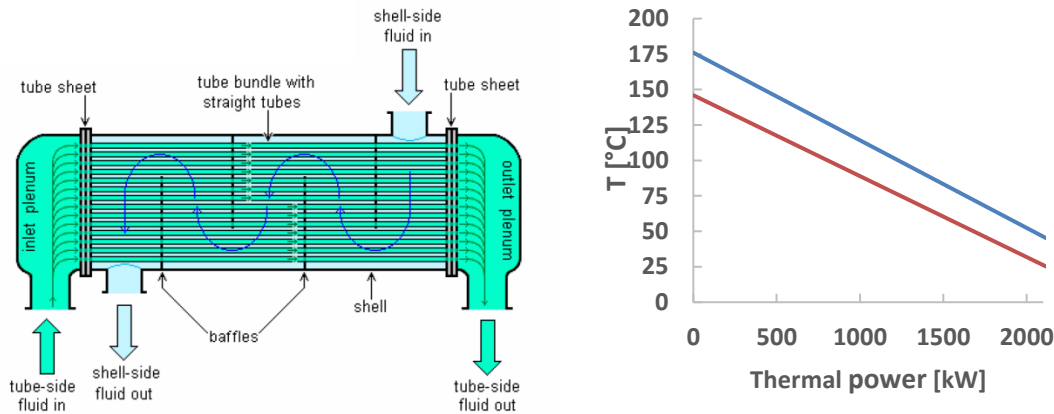


Figure 15: Shell & Tube heat exchanger schematic representation (left). Typical intercooler temperature profile. Red line: atmospheric air, blue line: compressed air (right).

The need to store heat in the LTES at a sufficiently high temperature level and the achievement of the target value for compressed air exiting the intercoolers lead to really close and almost parallel temperature profiles.

Moreover, really reduced values of the Logarithmic Mean Temperature Difference (LMTD) - which approximatively equals the temperature difference between the two fluids in correspondence to any axial abscissa of the HE - are expected if the atmospheric air exit temperature must reach a high value.

With this premise, to reduce the required heat transfer surface (which is strongly related to the cost of HEs), 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 through the heat exchanger and the TES. Due to the relevant atmospheric air

volumetric flow rate, the amount of power required by the fans is not negligible.

As a result, the really reduced LMTD and the strict requirement on the allowable pressure drop constitute major design challenges. Design solutions able to contain the heat transfer surface and parasitic power losses must be addressed.

Another aspect to be taken into consideration concerns the formation of condensed water in the colder section of the heat exchanger, which can represent an additional design complication.

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 heat duty.

The quality of fluids involved in the heat transfer process (air taken from the atmosphere) and operating temperatures (ranging from 20 to some 150-170°C) do not pose any particular design issue, while attention should be paid to the high pressure reached in the aftercooler placed downstream the last compression stage.

Since the charging phase is characterized by non-stationary operation, a satisfactory behaviour of each heat exchanger must be assured along the entire expected operating range. At the beginning of charging, operating pressure and temperatures assume the lowest values. Such values increase as the charging phase progresses, until to reach the target values at the end of the process. Therefore, the design of each heat exchanger is accepted if a satisfactory level of performance is assessed throughout the entire charging phase. The check is carried out on the basis of operating data provided by WP1.

5.2 Description of the adopted/developed models for Shell & Tube heat exchangers

The optimized design of low temperature S&T heat exchanger has been accomplished by using an in-house developed code written in MATLAB language [20] and by a commercial code (CC-Therm) included in the CHEMCAD package. Basically, as better explained in the following, 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.

5.2.1 The in-house developed MATLAB code

The code works according to the flowchart reported in Figure 16. The optimal solution is found by the genetic algorithm (GA) provided by MATLAB.

The generic S&T Heat Exchanger design is completely specified by a set of Independent Design Variables (IDV): the shell diameter, the tube outlet diameter, the baffle spacing and cut, the number of tube passage, the tube side stream (the other fluid stream is placed on shell side), the tube pattern. All the above quantities are determined by solving an optimization problem aimed at optimizing one of the following quantities: heat transfer area, total cost, heat transfer effectiveness, area footprint, volume footprint and pumping power.

The procedure starts with the specification of main quantities required to calculate the mass and energy balance at design point: inlet temperature, pressure and mass flow for both fluids and the outlet temperature of one of them. Then, basic information specifying the heat exchanger type (e.g. fixed tube sheet, U-tube, split ring floating head and so on) has to be provided. Finally, all the thermo-physical properties of both fluids have to be provided in the range of pressure and temperature of interest.

In the third step, the genetic algorithm (GA) selects an initial population of individuals. Each individual is identified by a set of values constituting the Independent Design Variables (IDV) array previously discussed.

IDV values are supplied to a design routine which adopts the well consolidated Kern approach to determine Dependent Design Variables (DDV), such as the number and length of tubes, the number of baffles, the tube pitch and so on. Each tentative configuration is evaluated in a Performance Evaluation step, determining the expected value of pressure drop, operating cost capital cost, expected total cost, etc. Subsequently, the selected objective function is evaluated. A Constraint Violation Analysis aimed at establishing a penalty cost is considered if some constraint is violated, rejecting unsuitable designs, while each feasible design is saved. GA acts iteratively modifying IDV and supplying them to design routine looking for alternative design solutions. The adopted optimization tool, provided by MATLAB, uses traditional genetic algorithm rules – elite count, crossover, mutation and migration- to generate the new population. Procedure stops when the best current value is maintained for a defined number of iterations or when a maximum number of runs is reached.

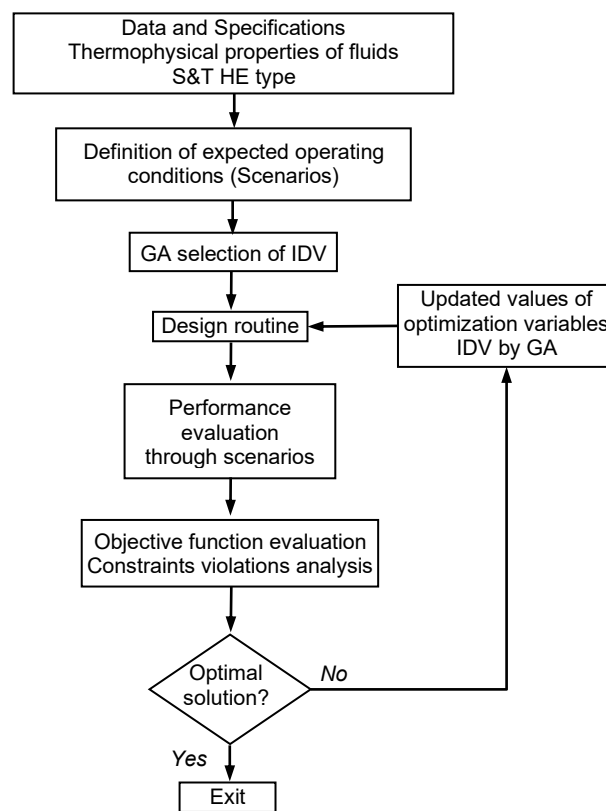


Figure 16: In-house developed code: flowchart of the optimized design procedure.

5.2.2 The CC-Therm code

CC-Therm is a tool for design and rating of heat exchangers integrated in CHEMCAD software package.

The Independent Design Variables (IDVs) array is basically similar to the one introduced when discussing the in-house developed code. A noticeable difference in respect to the MATLAB code is that only a subset of IDVs is established by an optimization procedure, because the others have to be directly assigned by the user.

The procedure starts with the definition of the design point specifications, i.e. the set of quantities which completely define the mass and energy balances.

In the second step, the user must specify the type of heat exchange equipment (e. g. S&T, plate,

double pipe type, etc.) and which fluid is allocated on tube side. Once the basic setup is completed, CC-THERM launches a guided design procedure composed of several interactive steps. This involves the selection of thermal and hydraulic calculation models, specification of design constraints (e.g., max tube and shell velocity, max allowable pressure drops) and input of IDVs directly assigned by the user: the tube outer diameter, the tube wall thickness, the tube pattern and pitch, the materials of the different parts constituting the heat transfer device. IDVs established by optimization are the number of tubes, the tube length, the number of tube passes, and the baffle configuration. Such quantities are varied in order to meet performance targets, to satisfy the design constraints and to minimize objective function, i.e. the overall heat transfer area.

A not specified metaheuristic optimization algorithm is adopted. CC-THERM allows the user to select the calculation approach. For example, for S&T design, Kern, Bell-Delaware and Stream Analysis approaches are available.

Preliminary analyses aimed at choosing the most appropriate design approach have been carried out. As an example, taking as reference the Small Size plant, 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. Sizing results obtained for the first intercooler are given in Figure 17. As an example, a compressed air mass flow rate of 1.5 kg/s has been assumed as reference. Results achieved by assuming atmospheric air mass flow rates of 1.5 and 2.2 kg/s are shown in the left side and right of the figure, respectively. Calculations have been carried out for different values of ΔT , i.e. the difference between the compressed air outlet temperature and the atmospheric air inlet temperature, assumed equal to 20°C.

In the first case, the temperature profile of the two streams involved in the heat transfer process is similar to that qualitatively shown in Figure 15 (right). Being the temperature lines parallel, the LMTD is practically equal to ΔT . In the second case, the assumption of a relatively large value of the atmospheric air mass flow rate implies a lower atmospheric air outlet temperature and consequently, a larger LMTD value for the same ΔT .

Calculations are carried out for different values of ΔT . When critical design specifications are assumed, e. g. a really reduced temperature approach between the two streams and/or a reduced value of the LMTD, the different calculation approaches can lead to quite different solutions. When LMTD increases to typical values, i.e. when ΔT assumes values in the order of 20 °C, the three methods tend to give results in a closer agreement. It can be noticed that the adoption of Kern's method leads always to the larger required heat transfer area. Results achieved by adopting the Bell-Delaware and the Stream Analysis methods are in a good agreement, being the Bell-Delaware approach slightly more conservative in respect to the second one. Such a trend has been confirmed by similar analyses carried out taking as a reference the Medium Size plant, not reported here for sake of brevity.

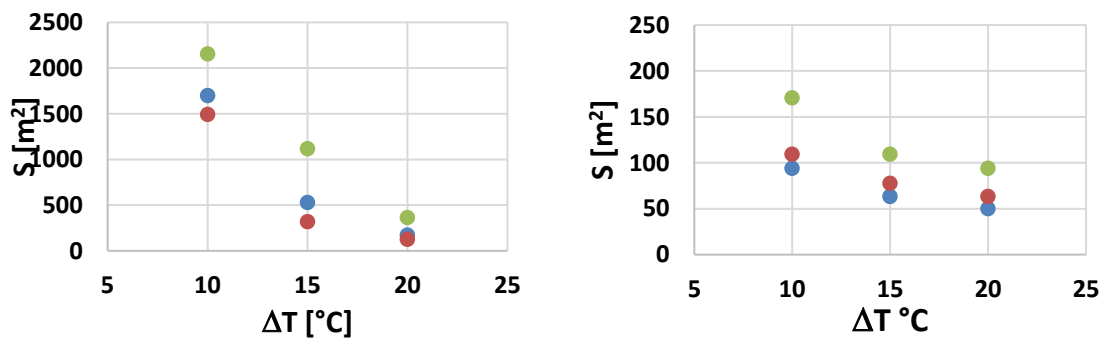


Figure 17: Comparison between different design methods: Kern (green), Bell-Delaware (red) and Stream Analysis (blue).

Table 8: Comparison between MATLAB and CC-Therm

	Matlab	CC-Therm
Optimised Variables	• Tube length • Shell diameter • Baffle spacing • Baffle cut • Number of tube passages • Tube diameter • Tube side fluid • Service fluid mass flow rate (optional)	• Tubes length • Shell diameter • Baffle spacing • Baffle cut • Number of tube passages
Constraints	• Maximum allowable pressure drop • Minimum/maximum fluids velocity • Length-to-diameter ratio	• Maximum allowable pressure drop • Minimum/maximum fluids velocity • Length-to-diameter ratio
Algorithm	• Genetic algorithm	• Not specified metaheuristic algorithm
Design Options	• Kern	• Kern • Bell-Delaware • Stream Analysis
Objective Functions	• Heat Transfer Area • Total cost • Effectiveness • Area footprint • Volume footprint • Pumping power	• Heat Transfer Area

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 [2, 21].

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 intercooler in different operating conditions arising during the charging phase.

5.2.3 Comparison between the MATLAB in house developed code and CC-Therm

Main features of the available codes are summarized in Table 8. It can be noticed that the in-house developed code constitutes a more flexible tool in respect to the commercial one. First of all, the MATLAB code allows the user to select the most appropriate objective function among various alternatives, while CC-Therm allows only the minimization the heat transfer area. Moreover, the number of variables optimized by the MATLAB code is higher than that of CC-Therm. Relevant design characteristics such as tube outlet diameter and which fluid stream is placed shell side or tube side are established by optimization. As a result, by using the MATLAB code, several tentative calculations to select the most suitable tube outlet diameter can be avoided.

5.2.4 The adopted design strategy

On the basis of the above considerations, a calculation strategy aimed at reducing the number of tentative designs has been established. To best exploit the different features of the two design tools, the MATLAB in-house developed code is first run to select the fluid to be placed on tube side and to individuate the most suitable value for the outlet tube diameter. Such inputs are then provided to CC-Therm to achieve a more refined and accurate design by adopting the Bell-Delaware method.

5.3 Design results (Small Scale, Medium Scale, Large Scale)

5.3.1 Preliminary considerations

According to WP1 activities, the same plant layout has been taken into consideration to size low temperature heat exchangers for Small, Medium and Large size plants.

At first, the original plant layout (shown in Figure 2) is addressed. Such a configuration is suitable for both the uses proposed for the thermal energy transferred from the compressed air stream to the atmospheric air: i) storage in a low temperature thermocline TES for compressed air pre-heating during the discharging phase and, ii) heat supply to an industrial process, as shown in Figure 4. Other cases involving the use of alternative heat carriers will be examined.

In the following, the four heat exchangers crossed in succession by the compressed air stream will be referred to as LHEX1, LHEX2, LHEX3 and LHEX4.

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 in the frame of WP3 activities.

In order to find a suitable compromise between heat transfer area requirements and overall plant performance, several parametric analyses have been carried out by varying design quantities such as the compressed air outlet temperature T_{HO} . As an example, Figure 18 shows how LHEX1 and LHEX2 heat transfer areas vary by increasing the compressed T_{HO} from 30 to 50°C. The analysis has been carried out by assuming as a reference a compressed air mass flow rate equal to 1.5 kg/s. The atmospheric air mass flow rate has been assumed to be slightly larger than that of compressed air, i.e. 1.55 kg/s for LHEX1 and 1.86 kg/s for LHEX2. Under this assumption, the lower the compressed air exit temperature, the higher the atmospheric air exit temperature T_{CO} is. As discussed in 5.1, the reduction of the compressed air exit temperature leads to a reduction of the mechanical power required for compression. Similarly, the increase of the atmospheric air exit temperature allows a higher storage temperature in the LTES, evidencing an improved recovery of the intercooling heat. On the other hand, such advantages are offset by a huge increase of the required heat transfer area, which implies a relevant increase in capital cost. It can be noticed that, by increasing T_{HO} from 30 to 40°C, the required heat transfer area is reduced to one third. Minor heat transfer area reductions are observed for further T_{HO} increases from 40 to 50°C. The atmospheric air exit temperature T_{CO} shows an opposite trend, reducing from 160 to some 140°C.

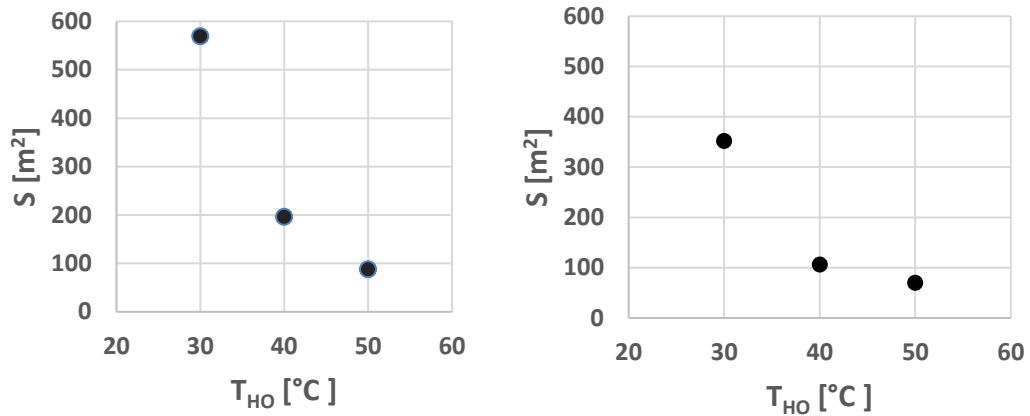


Figure 18: Heat transfer surface versus compressed air outlet temperature. Small size plant: LHEX1 (left), LHEX2 (right).

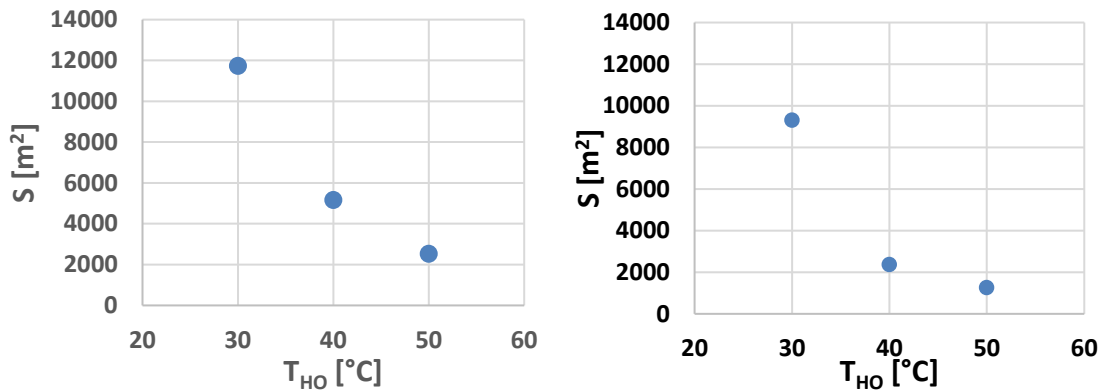


Figure 19: Heat transfer surface versus compressed air outlet temperature. Medium size plant: LHEX1 (left), LHEX2 (right).

Such a trend is substantially confirmed by a similar analysis carried out taking as a reference the Medium Size plant (Figure 19). Further analyses, not discussed here for sake of brevity, have been carried out by varying also the atmospheric air mass flow rate. Results have suggested the assumption for all LHEXs of a reasonable value of 45°C for the compressed air exit temperature.

5.3.2 Small scale plant

Flowsheet specifications required to design low temperature heat transfer equipment for Small Scale plant are reported in Table 9. Initial and final values of the compressed air inlet temperature and pressure refer to the beginning and at to the end of the charging phase.

To achieve the exit target temperature for both fluids, TEMA E shell 1-1 or TEMA F shell 2-2 type seem the most suitable arrangements. As previously stated, the MATLAB in-house developed code has been used to achieve a first tentative optimized design, assuming as objective function the heat transfer area. Subsequently, an improved and refined design is achieved by using the commercial tool ChemCad CC-Therm.

Table 9: Design specifications for low temperature heat exchangers. Small size plant.

Compressed air mass flow rate (kg/s)	0.8
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First intercooler (LHEX1)	
Compressed air inlet temperatures – initial and final - (°C)	170-176
Compressed air inlet pressure – initial and final - (bar)	3.5 – 3.5
Compressed air outlet temperature at the end of the charging phase (°C)	45
Second intercooler (LHEX2)	
Compressed air inlet temperatures – initial and final - (°C)	195 -205
Compressed air inlet pressure – initial and final - (bar)	10.6-12.3
Compressed air outlet temperature at the end of the charging phase (°C)	45
Third intercooler (LHEX3)	
Compressed air inlet temperatures – initial and final - (°C)	181 – 205
Compressed air inlet pressure – initial and final - (bar)	30-43
Compressed air outlet temperature at the end of the charging phase (°C)	45
Aftercooler (LHEX4)	
Compressed air inlet temperatures – initial and final - (°C)	83 -205
Compressed air inlet pressure – initial and final - (bar)	40 -150
Compressed air outlet temperature at the end of the charging phase (°C)	45

Water condensation on pressurized air side plays a relevant role in designing LHEX2. As an example, results achieved for LHEX2 by assuming i) dry ambient air, ii) 60% relative humidity, are compared in Figure 20. Calculations have been carried out by assuming for both streams the same inlet and outlet temperatures. It can be noticed as water condensation implies a not negligible thermal load increase in respect to dry air operation. In fact, the cooling of humid air requires a thermal power of some 146 kW versus 133 kW required by the same mass flow rate of dry air. Therefore, the increase is in the order of 10%.

Moreover, the latent heat release in the colder zone of the heat exchanger leads to an increased value of the specific heat, which determines a reduction of the temperature approach between the two streams, as shown in Figure 20. It has to be pointed out that, in order to avoid the intersection between hot and cold fluid temperature lines, a suitable value of the atmospheric air mass flow rate has been assumed, as discussed in the following.

Condensed water is removed at the exit of each heat exchanger by installing moisture separators immediately downstream LHEX2. Different solutions based on gravitational deposition or inertial interception are available (e.g. [22]).

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 LHEX has been assumed.

Compressed air pressure losses (tube side) affect the compression 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.

At a first attempt, bare tubes have been adopted. Main design results are summarized in Table 10 and Table 11.

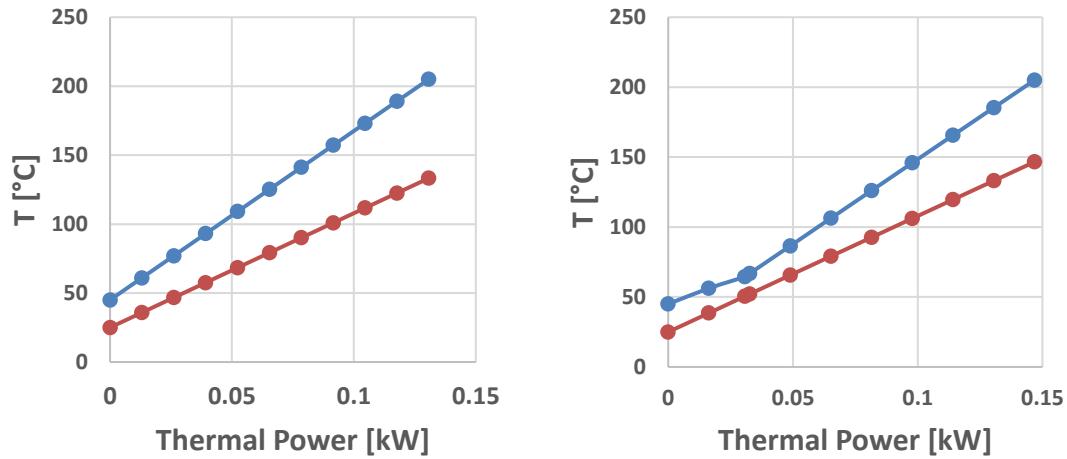


Figure 20: Stream temperatures vs exchanged thermal power: dry air (left), humid air (right).
Blu line: compressed air; Red line: atmospheric air.

Table 10: Bare tube design. Main process quantities.

HEX	$\dot{m}_H$ (kg/s)	$\dot{m}_C$ (kg/s)	T_{HI} ($^{\circ}\text{C}$)	T_{HO} ($^{\circ}\text{C}$)	T_{CI} ($^{\circ}\text{C}$)	T_{CO} ($^{\circ}\text{C}$)	Q (kW)	$\dot{m}_{COND}$ (g/s)
LHEX1	0.8	0.9	176.0	43.2	25	144.8	108	0.0
LHEX2	0.8	1.2	205.0	42.5	25	149.1	150	6.6
LHEX3	0.8	0.9	205.0	42.8	25	182.0	143	2.9
LHEX4	0.8	0.9	205.0	46.4	25	182.8	143	0.7

$\dot{m}_H$: compressed air mass flow rate; $\dot{m}_C$: atmospheric air mass flow rate; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; T_{CI} : atmospheric air inlet temperature; T_{CO} : atmospheric air outlet temperature; Q : exchanged thermal power; $\dot{m}_{COND}$: condensed water mass flow rate.

Table 11: Bare tube design. Main design results.

HEX	L (m)	D_s (m)	d_o (mm)	$n_s \times n_t$	N_t	S (m^2)	ΔP_s (kPa)	U ($\text{W}/\text{m}^2\text{K}$)
LHEX1	7.3	0.8	12	1x1	547	150	3.6	30.0
LHEX2	4.9	1.0	12	2x2	478	87	4.0	73.8
LHEX3	8.5	1.0	10	2x2	477	127	3.5	68.4
LHEX4	6.0	0.8	16	2x10	252	73	3.2	99.4

L : tube length; D_s : shell diameter; N_t : number of tubes; d_o : 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.

It can be noticed that important heat transfer areas (in the order of 70-150 m^2) are needed for all LHEXs. This is a consequence of the reduced design values of LMTD (in the range of 20-25 $^{\circ}\text{C}$) and of the relatively small value of the overall heat transfer coefficient [30-100 $\text{W}/\text{m}^2\text{K}$]. It is well known that heat transfer processes involving low pressure gases are intrinsically featured by a reduced convective heat transfer coefficient. The situation is further worsened by the fact that the fulfillment of the pressure drop constraint requires the adoption of reduced air velocities that penalize the shell side heat transfer coefficient.

According to the expected operating conditions, the maximum temperature that can be reached by the metal parts is of some 200 $^{\circ}\text{C}$. Therefore, carbon steel can be used for manufacturing. 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. Optimal solutions based on E shell type (one pass) or F shell type (two passes with longitudinal baffle) have been found by CC-Therm code. Segmental

baffles with 20-25% cut are selected.

The design led to a quite cumbersome heat transfer equipment. Anyway, both thermal duty and pressure drop constraint are fulfilled, demonstrating the feasibility of the proposed design.

In order to try to reduce sizes, weights and costs, a design based on the adoption of externally finned tubes has been carried out. Steel TruFin® finned tubes, whose main characteristics are shown in Figure 21, have been selected.

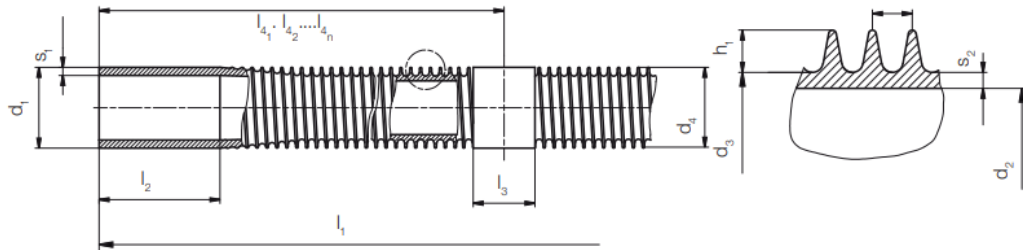


Figure 21: TruFin tube main features

Main design results are summarized in Table 12 and Table 13.

Table 12: Finned tube design. Main design process quantities.

HEX	m _H (kg/s)	m _C (kg/s)	T _{HI} (°C)	T _{HO} (°C)	T _{CI} (°C)	T _{CO} (°C)	Q (kW)	m _{COND} (g/s)
LHEX1	0.8	0.9	176.0	42.7	25	145.3	109	0.0
LHEX2	0.8	1.05	205.0	45.2	25	163.2	146	10.
LHEX3	0.7	0.7	205.0	44.4	25	193.7	119	0.0
LHEX4	0.7	0.78	206.0	44.9	25	187.2	128	0.0

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

Table 13: Finned tube design. Main design results.

HEX	L (m)	D _s (m)	d _o (mm)	n _s x n _t	N _t	S (m ²)	ΔP _s (kPa)	U (W/m ² K)
LHEX1	2.4	0.8	10	1x1	838	143.3	3.0	32.2
LHEX2	2.4	0.7	10	2x2	563	96.3	2.4	72.9
LHEX3	4.9	0.85	16	2x6	255	153.8	3.3	48.6
LHEX4	3.0	0.8	13	2x6	395	113.8	2.7	57.7

L: tube length; D_s: shell diameter; N_t: number of tubes; d_o: 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.

It has to be pointed out that the different heat exchangers have been designed by assuming different values of the compressed air mass flow rate, as reported in the second column of Table 12. The reason is related to the achievement of a satisfactory performance level on the entire expected operating range, as discussed at the end of the present chapter.

Steel TruFin® finned tubes with 10.0 mm OD, thickness of 0.8 mm, fin height of 1.6 mm and 750 fins per meter have been adopted for the first two heat exchangers. In order to be able to use LHEX3 and LHEX4 as compressed air pre-heaters during the discharge phase, they have been designed to operate at the maximum storage pressure, i.e. 150 bar. Therefore, tubes with a thickness of 1.25 mm have been selected.

The adoption of finned tube led to noticeable size reductions in respect to bare tube designs. The shell diameter of D_s shows reductions up to 30%. More noticeable decreases are achieved in terms

of tube lengths, which are approximatively halved by adopting externally finned tubes. Overall heat transfer coefficients are comparable with those obtained for bare tube, as well as heat transfer areas.

On the basis of the above considerations, externally finned tubes have been definitely chosen.

Table 14: LHEX1 off design behaviour

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	0.80	3.3	170.7	0.83	31.5	2.5	101
#5	0.71	3.5	176.3	0.75	29.1	2.1	95

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

Table 15: LHEX2 off design behaviour

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	0.80	10.6	195.7	1.03	71.8	2.3	137
#5	0.71	12.3	206.1	0.97	68.7	3.2	133

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

Table 16: LHEX3 off design behaviour

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	0.80	30.0	182.6	0.77	51.6	3.9	114
#5	0.71	43.0	206.8	0.71	49.0	3.4	121

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

Table 17: LHEX4 off design behaviour

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	0.80	40.0	82.6	0.54	47.5	1.1	31
#2	0.79	54.9	120.3	0.70	54.3	1.9	64
#3	0.79	69.2	146.2	0.76	56.9	2.4	86
#4	0.77	96.2	179.5	0.80	58.6	2.8	113
#5	0.71	150.0	208.3	0.79	58.2	2.8	131

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

LHEXs behavior has been checked over the entire expected operating range. A set of compression train working points (provided by WP1) ranging from the beginning (point #1) to the end (point #5) of the charging phase has been used to assess the off-design behavior of the heat transfer equipment. The system is managed in order that the compressed air temperature at the exit of each heat exchanger is kept equal to the target value of 45°C in all operating conditions. Such a result is achieved by regulating the atmospheric air mass flow rate entering the heat exchanger.

It can be noticed that the extent of the operating range during the charging phase is different for each heat exchanger. The compressed air mass flow rate m_c is determined by the matching among the intercooled compression stages in correspondence to each value of the compression train delivery pressure. In the present case, m_c decreases with the progress of the charging phase from 0.80 to 0.71 kg/s. Table 14, Table 15, Table 16 and Table 17 summarize the off-design performance of LHEX1, LHEX2, LHEX3 and LHEX4, respectively. It can be noticed as the operating range becomes progressively broader moving from LHEX1 to LHEX4. Changes in inlet pressure from the beginning to the end of charging for each heat exchanger are equal to 0.3, 2, 13 and 90 bar, respectively. The corresponding inlet temperature changes are 6, 10, 25 and 115 °C. LHEX1 operating conditions vary really little. Conversely, the major part of pressure and temperature variations are borne by LHEX4. Therefore, the off-design behavior of the latter has been checked more carefully by considering a greater number of operating conditions, unlike the other heat exchangers for which the off-design performance has been evaluated only at beginning and at the end of the charging phase.

In general, for the first three heat exchangers the required atmospheric air mass flow rate decreases as a consequence of the decrease of the compressed air flow rate during the charging phase. Conversely, LHEX4 operations are characterized by an opposite trend because the increase of the compressed air inlet temperature is prevailing on mass flow rate reduction.

LHEX1, LHEX2 and LHEX3 heat transfer coefficients change very little during charging, assuming values really close to the design ones reported in Table 22. LHEX4, conversely, shows a marked increase in the heat transfer coefficient mainly due to the relevant increase of the atmospheric air mass flow rate (from 0.54 to 0.79 kg/s), which causes a rise of the shell side heat transfer coefficient. At the beginning of the charging phase the overall heat transfer coefficient is equal to 47.5 W/m²°C, reaching a value close to the design one at the end (about 58.2 W/m²°C).

Shell side pressure losses show the same trend of the atmospheric air mass flow rate; for LHEX1, LHEX2 and LHEX3 pressure drop decreases as the charging phase progresses. On the contrary, LHEX4 shows a significant increase (from 1.1 to 2.8 kPa).

Therefore, it can be concluded that all heat exchangers show a satisfactory behavior in the expected operating range.

According to the ASTERIX-CAESAR Technical Annex, the possibility of using intercoolers also during the discharge phase for the e compressed air pre-heating has been considered. Since the compressed air mass flow during discharge approximatively doubles compared to the charging phase, two LHEXs in parallel arrangement are required. According to the reference discharge phase flowsheet, a compressed air mass flow rate of 1.54 kg/s has to be pre-heated from 25 to 125° C by hot atmospheric air at 180 °C coming from the LTES. The behavior as pre-heater has been checked in two extreme situations: at the beginning and at the end of the discharging phase, when the compressed air inlet pressure assumes values equal to 150 and 45 bar, respectively. Results for LHEX3 and LHEX4 are given in Table 18 and Table 19. The first line reports main results at the beginning of discharge phase, the second one at the end. Since operation as intercoolers implies a higher thermal duty (120-130 kW, according to Table 12), both heat exchangers, when used as preheaters, are capable to ensure the compressed air target temperature by using really reduced atmospheric air mass flow rates.

Table 18: Small Scale plant. LHEX3 used as preheater.

m_H (kg/s)	p_{HI} (bar)	T_{HI} (°C)	T_{HO} (°C)	m_c (kg/s)	T_{CI} (°C)	T_{CO} (°C)	U (W/m ² K)	ΔP_s (kPa)	ΔP_T (bar)	Q (kW)
0.77	150	25	125	0.58	180	25.8	46.8	2.0	0.3	90
0.77	45	25	125	0.53	180	25.7	44.1	1.7	1.0	82

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

Table 19: Small Scale plant. LHEX4 used as preheater.

$\dot{m}_H$ (kg/s)	p_H (bar)	T_{HI} (°C)	T_{HO} (°C)	$\dot{m}_C$ (kg/s)	T_{CI} (°C)	T_{CO} (°C)	U (W/m ² K)	ΔP_s (kPa)	ΔP_T (bar)	Q (kW)
0.77	150	25	125	0.58	180	26.6	51.9	1.3	0.4	90
0.77	45	25	125	0.53	180	26.5	48.9	1.1	1.4	82

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; T_{CI} : atmospheric air inlet temperature; T_{CO} : atmospheric air outlet temperature; U : overall heat transfer coefficient; ΔP_s : pressure drop shell side; ΔP_T : pressure drop tube side; Q : exchanged thermal power.

As reported in Chapter 2, a relevant modification to the original plant concept concerning an alternative reuse of the heat of compression has been introduced. In the updated plant flowsheet, the heat of compression is used to feed industrial processes, as shown in Figure 4. At the moment, air at atmospheric pressure has been selected as the thermal carrier. Therefore, LHEXs design here presented is not affected by the different end use of the intercooling heat. Such a consideration is applicable also to Medium Size and Large Size plants discussed in the following chapters.

5.3.3 Medium Scale Plant

Input data assumed to design LHEXs for medium scale plant are reported in Table 20.

On the basis of findings achieved for the Small Size plant, the design of heat transfer equipment has been performed by adopting externally finned tubes. Same limitations on tube side and shell side pressure drop have been imposed.

In this case also, the various heat exchangers have been sized by assuming different values of compressed air mass flow rate to achieve a satisfactory performance level over the entire expected operating range.

Table 20: Design specifications for low temperature heat exchangers. Medium Scale plant.

Compressed air mass flow rate (kg/s)	15.6
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First intercooler (LHEX1)	
Compressed air inlet temperatures – initial and final - (°C)	170-176
Compressed air inlet pressure – initial and final - (bar)	3.5 – 3.5
Compressed air outlet temperature at the end of the charging phase (°C)	45
Second intercooler (LHEX2)	
Compressed air inlet temperatures – initial and final - (°C)	195 -205
Compressed air inlet pressure – initial and final - (bar)	10.6-12.3
Compressed air outlet temperature at the end of the charging phase (°C)	45
Third intercooler (LHEX3)	
Compressed air inlet temperatures – initial and final - (°C)	181 – 205
Compressed air inlet pressure – initial and final - (bar)	30-43
Compressed air outlet temperature at the end of the charging phase (°C)	45
Aftercooler (LHEX4)	
Compressed air inlet temperatures – initial and final - (°C)	83 -205
Compressed air inlet pressure – initial and final - (bar)	40 -150
Compressed air outlet temperature at the end of the charging phase (°C)	45

Table 21: Main design process quantities.

HEX	m _H (kg/s)	m _C (kg/s)	T _{HI} (°C)	T _{HO} (°C)	T _{CI} (°C)	T _{CO} (°C)	Q (kW)
LHEX1	15.6	17.0	176	45	25	146	2080
LHEX2	15.6	17.0	205	48	25	171	2500
LHEX3	13.8	17.0	205	45	25	161	2330
LHEX4	13.8	18.0	205	45	25	163	2540

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

Table 22: 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)
LHEX1	6.1	1.7	16	1x1	3214	3111	2.9	23.2
LHEX2	8.5	1.4	16	1x1	2102	2854	4.0	31.2
LHEX3	7.3	1.4	16	1x1	2102	2444	3.6	30.2
LHEX4	8.5	1.4	16	1x1	2102	2854	4.1	31.5

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 features of the optimized design of low temperature heat exchangers are given in Table 21 and Table 22. Steel Trufin® finned tubes with 15.8 mm OD, fin height of 1.6 mm, 630 fins per meter have been adopted. A tube thickness of 0.8 mm has been assumed for LHEX1 and LHEX2. Similar to Small Size plant, in order to use LHEX3 and LHEX4 as compressed air pre-heaters, a tube thickness of 1.5 mm has been adopted. Such a value is considered appropriate to withstand the maximum expected value of the pressure on tube side (150 bar).

As for Small Size plant, carbon steel is chosen, and the same design solutions are assumed for the front and rear ends. In the present case, an E (one pass) shell type is selected for all heat exchangers.

It can be noticed that relevant heat transfer areas (in the order of 2500-3000 m²) are needed for all LHEXs. This is a consequence of the small values of the overall heat transfer coefficient, that are significantly lower than those evaluated for the Small Scale case. This occurs because pressure losses tend to increase with the size, as a consequence of the increasing length of the fluid paths inside the heat exchanger. Losses can be reduced by the adoption of reduced air velocities, that adversely influence the shell side heat transfer coefficient. Moreover, to contribute to limit the pressure drop, rod baffles have been selected and a large diameter for the inlet and outlet shell side nozzles have been adopted as evidenced in Figure 22, where the LHEX1 geometry is sketched.

The design led to a bulky equipment. Both thermal duty and pressure drop constraint are fulfilled, demonstrating, in this case also, the feasibility of the proposed design. The feasibility has been confirmed also by manufacturers questioned about it. Anyway, manufacturers have evidenced aspects requiring attention, such as the relevant dimensions of shell side nozzles and the expected high capital cost related to the need of large heat transfer surfaces.

The LHEXs behaviour has been checked over the entire expected operating range. The off-design analysis has been performed in the same way discussed for the Small Scale plant.

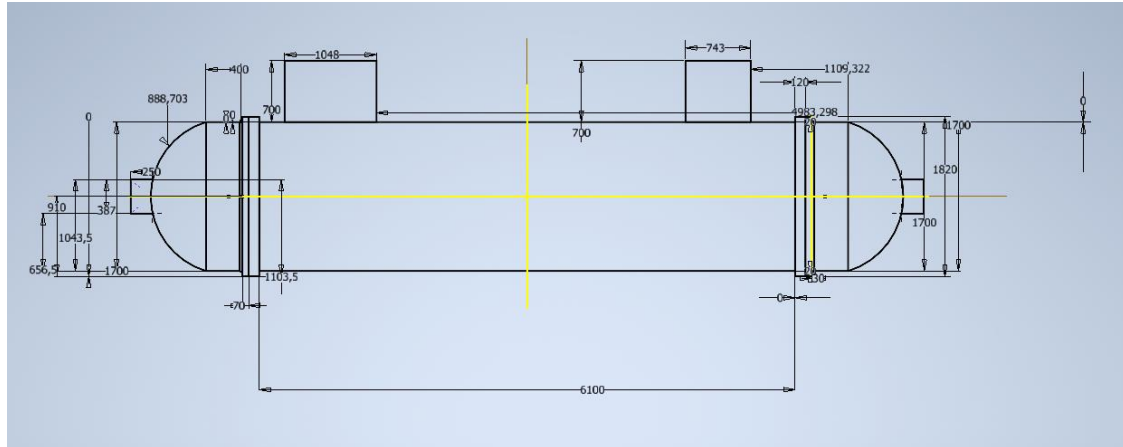


Figure 22: LHEX1 sketch and main dimensions

Table 23, Table 24, Table 25 and Table 26 summarize the off design performance of LHEX1, LHEX2, LHEX3 and LHEX4, respectively. As shown for the Small Size plant, the major part of pressure and temperature variations are absorbed by LHEX4.

Table 23: LHEX1 off design behaviour

Operating point	$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	$\dot{m}_C$ (kg/s)	U (W/m ² K)	ΔP_s (kPa)	Q (kW)
#1	15.6	3.3	170.7	18.0	23.6	2.98	2000
#5	13.8	3.5	176.3	16.0	21.5	2.53	1837

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

Table 24: LHX2 off design behaviour

Operating point	$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	$\dot{m}_C$ (kg/s)	U (W/m ² K)	ΔP_s (kPa)	Q (kW)
#1	15.6	10.6	195.7	17.6	31.5	4.19	2400
#5	13.8	12.3	206.1	15.7	28.8	3.37	2270

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

Table 25: LHX3 off design behaviour

Operating point	$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	$\dot{m}_C$ (kg/s)	U (W/m ² K)	ΔP_s (kPa)	Q (kW)
#1	15.6	30.0	182.6	18.6	32.6	4.18	2230
#5	13.8	43.0	206.8	17.4	30.5	3.73	2350

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

Table 26: LHX4 off design behaviour

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	15.6	40.00	82.6	11.2	26.6	1.43	610
#2	15.5	54.9	120.3	14.7	29.7	2.53	1230
#3	15.4	69.2	146.2	16.3	31.0	3.17	1670
#4	15.1	96.2	179.5	17.8	32.1	3.87	2270
#5	13.8	150.00	208.3	17.6	31.3	3.87	2550

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

In agreement with Small Size plant, LHEX1, LHEX2 and LHEX3 heat transfer coefficients change very little during charging, while LHEX4 denotes a marked increase from 26.6 to about 31 W/m²°C.

It can be noticed that in the worst cases, the shell side pressure drop reaches values slightly exceeding the maximum allowable ones assumed as a design requirement (4 kPa). Nevertheless, the off-design behavior of the Medium Size plant heat transfer equipment can be considered successfully assessed.

Table 27: Medium Scale plant. LHEX3 used as preheater.

m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	T _{HO} (°C)	m _C (kg/s)	T _{CI} (°C)	T _{CO} (°C)	U (W/m ² K)	ΔP _s (kPa)	ΔP _T (kPa)	Q (kW)
15.0	150	25	125	11.9	180	33.8	28.5	1.5	0.08	1755
15.0	45	25	125	10.8	180	33.4	26.5	1.3	0.2	1597

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

Table 28: Medium Scale plant. LHEX4 used as preheater.

m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	T _{HO} (°C)	m _C (kg/s)	T _{CI} (°C)	T _{CO} (°C)	U (W/m ² K)	ΔP _s (kPa)	ΔP _T (kPa)	Q (kW)
15.0	150	25	125	11.7	180	31.1	27.8	1.5	0.3	1757
15.0	45	25	125	10.6	180	30.7	25.8	1.2	1.0	1596

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

Finally, as made for the Small-Scale plant, the possibility of using intercoolers also during the discharge phase to preheat the compressed air pre-heating has been considered. Since the compressed air mass flow during discharge approximatively doubles in respect to the charging phase, two LHEXs in parallel arrangement are needed. A compressed air mass flow rate of 30 kg/s has to be pre-heated from 25 to 125° C by using hot atmospheric air at 180 °C The behavior as pre-heater has been assessed at the beginning and at the end of the discharging phase, when the inlet pressure is equal to 150 and 45 bar, respectively. Results for LHEX3 and LHEX4 are given in Table 27 and Table 28, where the first line refers the beginning of discharge phase, the second one at the end. The achieved results demonstrate that the use of Medium Scale intercooler as compressed air preheater during the discharging phase is fully satisfactory.

5.3.4 Large Scale Plant

Large Scale plant reference design quantities are given in Table 29. It can be noticed that, as

established in WP1, a different value of the storage pressure has been assumed in respect to Small and Medium size plants. 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 building 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 [23].

Table 29: Design specifications for low temperature heat exchangers. Large Scale plant

Compressed air mass flow rate (kg/s)	95
Ambient temperature (°C)	25
Ambient pressure (kPa)	101.3
Ambient relative humidity (%)	60%
First intercooler (LHEX1)	
Compressed air inlet temperatures – initial and final - (°C)	150-154
Compressed air inlet pressure – initial and final - (bar)	2.9 - 3.0
Compressed air outlet temperature at the end of the charging phase (°C)	45
Second intercooler (LHEX2)	
Compressed air inlet temperatures – initial and final - (°C)	175 -183
Compressed air inlet pressure – initial and final - (bar)	8.2 - 9.9
Compressed air outlet temperature at the end of the charging phase (°C)	45
Third intercooler (LHEX3)	
Compressed air inlet temperatures – initial and final - (°C)	167 -183
Compressed air inlet pressure – initial and final - (bar)	21.7 - 26.8
Compressed air outlet temperature at the end of the charging phase (°C)	45
Aftercooler (LHEX4)	
Compressed air inlet temperatures – initial and final - (°C)	132 -183*
Compressed air inlet pressure – initial and final - (bar)	40-80
Compressed air outlet temperature at the end of the charging phase (°C)	45

It can be noticed that the compressed air mass flow rate is 95 kg/s, equal to about 6 times that of the Medium Size plant. Recalling the issues affecting Medium Size LHEXs, it can be expected that the larger size can only worsen the subjects discussed in Chapter 5.3.3: the need of huge heat transfer areas, big shell side nozzles, relevant capital cost.

Moreover, the feasibility of the design can easily imply the adoption of several heat transfer units in parallel and/or in serial arrangement.

Table 30: Large Scale plant. LHEX1 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)
LHEX1	7.3	2.0	16	1x1	4585	5331	3.7	31.4

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.

As an example, a tentative design of LHEX1 has been performed on the basis of Table 29 data by using Chemcad CC-Therm. Results are given in Table 30. The overall heat duty can be fulfilled by installing three units in parallel arrangement. It can be noticed that each unit is bulky and that main dimensions (shell diameter and tube length) are close to the maximum allowable ones. The design of the single unit is characterized by strong similarities with the design of the heat transfer

equipment of the Middle Scale plant: one shell side pass, one tube side pass, rod type baffles. Consequently, the design is affected by same issues discussed in dealing the Middle Scale plant: relatively low values of overall heat transfer coefficient (about $30 \text{ W/m}^2\text{°C}$) and relevant dimension of inlet and outlet shell side nozzles.

Moreover, such an arrangement based on multiple units is expected to lead to noticeable cost augmentation and issues in managing the overall system integration.

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 discussed in the following chapter.

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

Basically, a HRSG is constituted by a rectangular duct containing the required number of tube bundles. 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 potential of avoiding problems connected with the large size of nozzles that can occur when a Shell & Tube design is addressed, as discussed in Chapter 5.3.3.

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

The code allows the definition of the arrangement of each tube bundle in terms of required tube rows and number of parallel serpentines, 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 23. In order to achieve an adequate pressurized air velocity, 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 Figure 23, where two tube rows are fed in parallel.

The following quantities have been assumed as degrees of freedom for the design and have to be assigned: free flow velocity at the inlet of the duct, duct height/width ratio, inlet and outlet tube diameter, fin geometry (length, thickness and number of fins per meter), bundle arrangement (in line or staggered tubes, tube pitch). Correlations available in literature [24, 25] are adopted to estimate the heat transfer coefficient and the pressure drop on tube and duct side.

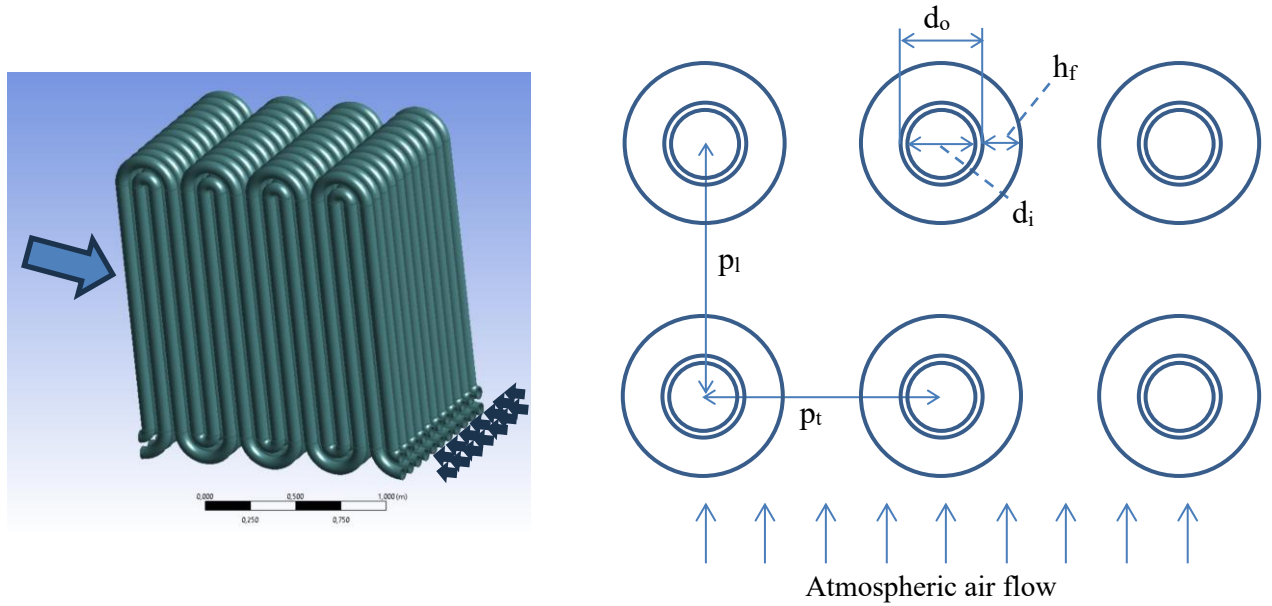


Figure 23: Main quantities characterizing the generic tube bundle

5.4.1 Medium Scale plant

The proposed approach has first been applied to the Medium Scale plant. Note that for the Small Scale Plant a conventional Shell & Tube design provides a satisfactory solution. The design specifications are given in Table 20. Table 31 contains the Degrees of Freedom (DOF), i.e. the quantities, assigned by the designer which fully define the design. Finally, in Table 32 main quantities evaluated by the computational code are reported.

Each LHEX unit has been designed as a “HRSG”, i. e. as a heat transfer device constituted by a rectangular duct containing a single tube bundle. The duct height/width ratio has been chosen to limit the number of parallel rows and to increase the number of turns per serpentine in order to assure a heat transfer path equivalent to the counter current one. Anyway, the correct allocation of the atmospheric air stream on the tube side led to a noticeable number of parallel serpentines. Geometrical layouts found for tube bundles are similar to those characterizing recuperative air pre-heaters often placed in the end section of the gas path of fired steam generators. A calculation procedure for such type of tube bundles is given in [25]. The method is based on the LMTD approach. A correction coefficient is introduced to account for parallel serpentines. Such a coefficient assumes values really close to unity in all four cases taken into consideration, confirming the correctness of the assumption of a pure counter current flow.

Table 31: Medium Scale plant LHEXs design. Degrees of Freedom.

	LHEX1	LHEX2	LHEX3	LHEX4
Free flow velocity [m/s]	5	5.32	5.17	5.5
Duct height/width ratio	1	1	1	1
Tube diameter [mm]	38	38	38	38
Tube thickness [mm]	1.0	1.2	1.5	1.5
Tube arrangement [Inline, Staggered]	Inline	Inline	Inline	Inline
Tube type [Bare, Finned]	Finned	Finned	Finned	Finned
Transversal pitch to diameter ratio	1.50	1.50	1.50	1.50
Longitudinal pitch to diameter ratio	1.50	1.50	1.50	1.50
Fin type [Solid, Serrated]	Solid	Solid	Solid	Solid
Fins height [mm]	8.0	8.0	8.0	8.0

Free flow velocity values have been assumed in order to have for all LHEXs the same 2×2 m cross flow section. It can be noticed as the duct length, the number of parallel rows, the number of turns and, consequently, the number of tubes reduce moving from LHEX1 to LHEX4. This can be explained by the increasing tube side values of pressure and density, which determine a reduction of the overall cross flow area required by the compressed air stream and an increase of the tube side heat transfer coefficient. For this reason, the overall heat transfer coefficient approximately doubles from the first to the fourth heat exchanger. Of course, an opposite trend is observed for the calculated heat transfer area and for the duct length. As expected, in analogy with actual HRSG, reduced values of duct side pressure drop in the order of 1.7-2.9 kPa are found. Tube side pressure drop are far below the allowable values.

In comparison with the Shell & Tube design (Table 22), relevant heat transfer area reductions - ranging from 30 to 20% - are found. Decreases of the same magnitude (from 6 to 60%) are observed in terms of atmospheric air pressure drop. Therefore, only taking technical aspects into account, the unconventional design based on the HRSG concept is certainly considered preferable in respect to the traditional Shell & Tube approach. Of course, an ultimate choice can be made taking also economic aspects into consideration.

The design is performed in order to achieve satisfactory operations along the entire charging phase. To check the off-design behavior of LHEXs designed adopting the HRSG technology, an analysis based on the same assumptions previously made for Shell & Tube heat exchanger type has been carried out. Results are given in Table 33, Table 34, Table 35, and Table 36.

Table 32: Medium Size plant LHEXs design output quantities.

	LHEX1	LHEX2	LHEX3	LHEX4
Duct width (m)	2.02	2.02	2.02	2.02
Duct height (m)	2.02	2.02	2.02	2.02
Duct length ^a (m)	7.20	7.20	3.42	3.42
Tubes per row x Parallel rows	32 x 11	32 x 6	32 x 2	32 x 2
Number of turns x Serpentine	12	21	30	30
Number of tubes	4224	4032	1920	1920
Total surface	2486	2058	1243	1380
Fin per meter	71	55	83	98
Overall heat transfer coefficient (W/m ² °C)	33.8	49.6	61.4	61.1
Shell side pressure drop (kPa)	3.03	2.72	1.52	1.83
Tube side pressure drop (kPa)	15.30	28.2	85.1	25.5
^a strictly required for tube allocation (inlet and discharge pieces excluded)				

Table 33: LHEX1 off design behaviour. HRSG design approach.

Operating point	m_H (kg/s)	p_{HI} (bar)	T_{HI} (°C)	m_C (kg/s)	U (W/m²K)	ΔP_s (kPa)	Q (kW)
#1	15.6	3.3	170.7	16.8	33.7	2.97	1986
#5	13.8	3.5	176.3	14.8	30.8	2.33	1832
m _H : compressed air mass flow rate; T _{HI} : compressed air inlet pressure; T _{HI} : compressed air inlet temperature; T _{HO} : compressed air outlet temperature; m _C : atmospheric air mass flow rate; U: overall heat transfer coefficient; ΔP _s : shell side pressure drop; Q: exchanged thermal power.							

Table 34: LHEX2 off design behaviour. HRSG design approach.

Operating point	m_H (kg/s)	p_{HI} (bar)	T_{HI} (°C)	m_C (kg/s)	U (W/m²K)	ΔP_s (kPa)	Q (kW)
#1	15.6	10.6	195.7	16.6	49.1	2.6	2394
#5	13.8	12.3	206.1	14.8	45.3	2.1	2270
m _H : compressed air mass flow rate; T _{HI} : compressed air inlet pressure; T _{HI} : compressed air inlet temperature; T _{HO} : compressed air outlet temperature; m _C : atmospheric air mass flow rate; U: overall heat transfer coefficient; ΔP _s : shell side pressure drop; Q: exchanged thermal power.							

Table 35: LHEX3 off design behaviour. HRSG design approach.

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	15.6	30.0	182.6	18.30	64.9	1.74	2226
#5	13.8	43.0	206.8	17.0	61.4	1.51	2344

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

Table 36: LHEX4 off design behaviour. HRSG design approach.

Operating point	m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	m _C (kg/s)	U (W/m ² K)	ΔP _s (kPa)	Q (kW)
#1	15.6	40.00	82.6	11.4	47.9	0.77	620
#2	15.5	54.9	120.3	15.0	56.0	1.30	1245
#3	15.4	69.2	146.2	16.7	59.4	1.59	1682
#4	15.1	96.2	179.5	18.2	62.2	1.87	2227
#5	13.8	150.00	208.3	18.0	61.1	1.83	2516

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

It can be observed as pressure side losses are always well below the allowed maximum value of 4 kPa. Moreover, values are significantly lesser than those achieved by adopting the Shell & Tube design, as can be seen from data reported in Table 23, Table 24, Table 25 and Table 26.

Concerning tube side pressure drop, values found taking into consideration the entire expected operating range remain below the allowed limit of 4% of the inlet pressure.

Finally, as reported in Table 37 and Table 38, the possibility of using LHEX3 and LHEX4 to preheat the pressurized air flow during discharge has been successfully assessed. Lesser shell and tube side pressure losses are found in respect to the Shell & Tube design (see for comparison Table 27 and Table 28), with positive effects on net electric output related both to a reduction of the power absorbed by the fans and to an increase of the power produced by the expansion train.

Table 37: Medium Scale plant. HRSG design approach. LHEX3 used as preheater.

m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	T _{HO} (°C)	m _C (kg/s)	T _{CI} (°C)	T _{CO} (°C)	U (W/m ² K)	ΔP _s (kPa)	ΔP _T (bar)	Q (kW)
15.0	150	25	125	12.1	180	36.3	51.3	0.81	0.18	1754
15.0	45	25	125	10.9	180	35.1	48.3	0.67	0.57	1592

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

Table 38: Medium Scale plant. HRSG design approach. LHEX4 used as preheater.

m _H (kg/s)	p _{HI} (bar)	T _{HI} (°C)	T _{HO} (°C)	m _C (kg/s)	T _{CI} (°C)	T _{CO} (°C)	U (W/m ² K)	ΔP _s (kPa)	ΔP _T (bar)	Q (kW)
15.0	150	25	125	11.9	180	34.4	48.9	0.86	0.18	1747
15.0	45	25	125	10.8	180	33.6	46.2	0.71	0.57	1594

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

5.4.2 Large Scale plant

As stated at the end of 5.3.4, the design based on HRSG concept is the preferred one for the Large Scale plant. Input process quantities are given in Table 29. Values of design Degrees of Freedom (DOFs) are reported in Table 39. It can be noticed that values are similar to those assumed for Medium Scale plant reported in Table 31. Design output quantities are given in Table 40.

Table 39: Large Scale plant LHEXs design. Degrees of Freedom.

	LHEX1	LHEX2	LHEX3	LHEX4
Free flow velocity [m/s]	4.9	5.26	4.97	5.2
Duct height/width ratio	1	1	1	1
Tube diameter [mm]	38	38	38	38
Tube thickness [mm]	1.0	1.2	1.5	1.5
Tube arrangement [Inline, Staggered]	Inline	Inline	Inline	Inline
Tube type [Bare, Finned]	Finned	Finned	Finned	Finned
Transversal pitch to diameter ratio	1.60	1.70	1.50	1.50
Longitudinal pitch to diameter ratio	1.50	1.50	1.50	1.50
Fin type [Solid, Serrated]	Solid	Solid	Solid	Solid
Fins height [mm]	8.0	8.0	8.0	8.0

All LHEXs are sized in order to have the same duct width and height (5×5 m). As hoped, the cooling of compressed air exiting each compressor stage can be accomplished by installing one single unit, thus avoiding the need of installing multiple heat transfer units in parallel arrangement.

As for the Medium Scale plant, the overall heat transfer coefficient increases moving from the first to the last cooler.

In respect to Medium Scale plant, a higher number of parallel paths (i.e. tube per row×parallel rows) is required to accommodate the huge mass flow rate (95 kg/s). Conversely, the required heat transfer surface can be arranged by using a lesser number of turns per serpentine.

Taking LHEX1 as reference, it can be noticed that the required heat transfer surface is some 17000 m². This value is similar to that resulting by adopting the Shell & Tube design approach of about 1600 m², equally shared on three heat transfer units in parallel arrangement.

Anyway, the design solution based on the HRSG concept is still preferred because it leads to a simpler plant layout and to reduced pressure drops on both tube and shell side.

Table 40: Large Size plant LHEXs design output quantities.

	LHEX1	LHEX2	LHEX3	LHEX4
Duct width (m)	5	5	5	5
Duct height (m)	5	5	5	5
Duct length (m)	9.7	10.8	5.6	4.9
Tubes per row x Parallel rows	79x34	75x19	85x7	85x5
Number of turns x Serpentine	5	10	14	17
Number of tubes	13430	14250	8330	7225
Total surface	16903	15248	9103	8779
Fin per meter	54	39	41	51
Overall heat transfer coefficient (W/m ² °C)	35.0	50.4	69.5	75.0
Shell side pressure drop (kPa)	3.23	2.75	2.21	2.26
Tube side pressure drop (kPa)	11.6	29.9	68.5	55.0
^a strictly required for tube allocation (inlet and discharge pieces excluded)				

Table 41: LHEX1 off design behaviour. HRSG design approach.

Operating point	m_H (kg/s)	p_H (bar)	T_H (°C)	m_c (kg/s)	U (W/m²K)	ΔP_s (kPa)	Q (kW)
#1	105.0	2.91	150.0	103.3	34.8	3.13	11129
#5	95.0	3.02	155.0	94.1	32.5	2.61	10554
m _H : compressed air mass flow rate; T _H : compressed air inlet pressure; T _H : compressed air inlet temperature; T _{HO} : compressed air outlet temperature; m _c : atmospheric air mass flow rate; U: overall heat transfer coefficient; ΔP _s : shell side pressure drop; Q: exchanged thermal power.							

Table 42: LHEX2 off design behaviour. HRSG design approach.

Operating point	m_H (kg/s)	p_H (bar)	T_H (°C)	m_c (kg/s)	U (W/m²K)	ΔP_s (kPa)	Q (kW)
#1	105.0	8.18	175.0	103.2	50.0	2.66	13871
#5	95.0	9.00	183.4	94.2	46.8	2.22	13371
m _H : compressed air mass flow rate; T _H : compressed air inlet pressure; T _H : compressed air inlet temperature; T _{HO} : compressed air outlet temperature; m _c : atmospheric air mass flow rate; U: overall heat transfer coefficient; ΔP _s : shell side pressure drop; Q: exchanged thermal power.							

Table 43: LHEX3 off design behaviour. HRSG design approach.

Operating point	$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	$\dot{m}_C$ (kg/s)	U (W/m ² K)	ΔP_s (kPa)	Q (kW)
#1	105.0	21.73	167.0	96.0	68.1	2.04	11933
#5	95.0	26.82	183.7	100.0	69.5	2.20	13641

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

Table 44: LHEX4 off design behaviour. HRSG design approach.

Operating point	$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	$\dot{m}_C$ (kg/s)	U (W/m ² K)	ΔP_s (kPa)	Q (kW)
#1	105.0	40.0	132.0	99.0	73.6	2.01	9591
#2	104.0	50.0	152.0	105.1	75.9	2.26	11785
#3	102.5	60.0	167.0	108.2	77.0	2.40	13344
#4	100.0	70.0	177.0	108.5	76.9	2.41	14191
#5	95.0	80.0	184.2	105.0	75.0	2.26	14328

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; T_{CI} : atmospheric air inlet temperature; ΔP_s : shell side pressure drop; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

Also in this case the off-design behavior of each LHEX on the entire charging phase can be considered really satisfactory, as reported in Table 41, Table 42, Table 43 and Table 44.

Finally, based on Table 45 and Table 46, it can be concluded that the use of LHEX3 and LHEX4 to preheat the compressed air does not show any issue.

Table 45: Large Scale plant. HRSG design approach. LHEX3 used as preheater.

$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	T_{HO} (°C)	$\dot{m}_C$ (kg/s)	T_{CI} (°C)	T_{CO} (°C)	U (W/m ² K)	ΔP_s (kPa)	ΔP_T (bar)	Q (kW)
71.0	80	25	125	51.1	180	28.3	46.8	0.61	0.08	7817
71.0	45	25	125	49.2	180	28.1	45.9	0.56	0.14	7535

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; T_{CI} : atmospheric air inlet temperature; T_{CO} : atmospheric air outlet temperature; U : overall heat transfer coefficient; ΔP_s : pressure drop shell side; ΔP_T : pressure drop tube side; Q : exchanged thermal power.

Table 46: Large Scale plant. HRSG design approach. LHEX4 used as preheater.

$\dot{m}_H$ (kg/s)	p_{HI} (bar)	T_{HI} (°C)	T_{HO} (°C)	$\dot{m}_C$ (kg/s)	T_{CI} (°C)	T_{CO} (°C)	U (W/m ² K)	ΔP_s (kPa)	ΔP_T (bar)	Q (kW)
71.0	80	25	125	51.1	180	28.3	48.7	0.56	0.19	7819
71.0	45	25	125	49.1	180	28.1	47.6	0.52	0.34	7523

$\dot{m}_H$: compressed air mass flow rate; T_{HI} : compressed air inlet pressure; T_{HI} : compressed air inlet temperature; T_{HO} : compressed air outlet temperature; $\dot{m}_C$: atmospheric air mass flow rate; T_{CI} : atmospheric air inlet temperature; T_{CO} : atmospheric air outlet temperature; U : overall heat transfer coefficient; ΔP_s : pressure drop shell side; ΔP_T : pressure drop tube side; Q : exchanged thermal power.

5.5 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.

Shell & Tube heat exchangers costs have been calculated by adopting a formulation proposed by

[26] achieved by reworking and improving a correlation proposed by NETL [27]. 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 labor. The correlations applicable in a real wide range (25-5000 m²), and therefore, can cover Small, Medium and Large 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 [28] subsequently updated by other Authors [29, 30]. 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.

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

Costs evaluated for Small Size plant LHEXs are reported in Table 47. In this case only the Shell & Tube concept has been considered. The cost of each LHEX is in the order of 240-320,000 USD. The overall capital cost is of 1,130,000 USD, corresponding to an average cost per unit of heat transfer area of some 2,200 USD/m².

Table 47: Small Scale plant. Capital cost of Shell & Tube LHEXs.

	LHEX1	LHEX2	LHEX3	LHEX4	TOT.
Cost [\$]	270,000	240,000	300,000	320,000	1,130,000

Costs referring to both Shell & Tube and HRSG design approaches are reported in Table 48. It can be noticed as the overall cost estimated for Shell & Tube design is about 33% higher than the cost evaluated for the HRSG based solution. Such a result confirms and reinforces the preference expressed for the HRSG design approach in Chapter 5.4.1 taking only technical aspects into consideration.

Table 48: Medium Scale plant. Capital cost comparison between Shell & Tube and HRSG design approaches.

	Shell & Tube	HRSG
TOTAL COST [\$]	4,800,000	3,600,000

Finally, a complete design of Large-Scale plant heat transfer equipment has been carried out only by applying the HRSG design approach. Anyway, an indicative economic comparison between the two approaches can be attempted by considering cost figures found for LHEX1 reported in Table 49.

Table 49: Large Scale plant. LHEX1 capital cost comparison between Shell & Tube and HRSG approaches.

	Shell & Tube	HRSG
LHEX1 COST [\$]	5,400,000	5,000,000

Also in this case the Shell & Tube approach shows a higher cost, but with a reduced difference in respect to the Medium-Scale case. In fact, the Shell & Tube design is only 8% more costly than

the HRSG based one. Nevertheless, technical advantages such as the simpler plant layout and the better capacity to manage high volumetric flow rates make the HRSG based design the preferred one. The estimated overall cost is of some 18,500,000 USD.

In conclusion, it is of some interest to discuss how the specific cost per unit of heat transfer area varies with the size of the plant and with the adopted design approach (Table 50).

Table 50: Heat transfer equipment specific cost.

	Small Scale	Medium Scale	Large Scale^(*)
S&T Specific cost [\$/m²]	2200	430	340
HRSG Specific cost [\$/m²]	--	500	300
^(*) Calculated on the basis of only LHEX1 cost			

As expected, the specific cost diminishes as the size of the plant increases due to the scale effect. It can be noticed as costs for Shell & Tube heat transfer equipment decrease less rapidly than those evaluated for HRSG based design. The reason is likely connected with the need to split the overall heat transfer area into multiple parallel units, certainly more expensive compared to one single device capable of managing the overall thermal load.

6 Conclusions

This document considers the design of low-temperature heat exchangers needed for efficient intercooling of the compressed air between compressor stages. The heat exchangers are designed for 3 different power scales: Small Size Plant (1MW discharging power), Medium Size Plant (20 MW discharging power) and Large Size Plant (100+ MW discharging power).

In the case of the Small-Size Plant, the regenerative design as well as a Shell & Tube design is considered. In the case of the regenerative approach, 4 beds of 40 m length are required, having a total heat transfer area of about 1470 m² each. This results in a total heat transfer area of 5880 m². The total CAPEX of the regenerative system is estimated at 1.7 M USD. It must be noted that this cost already includes the cost of the low-temperature thermal energy storage unit. The Shell & Tube solution has a total heat transfer area of 510 m² and an estimated cost of about 1.13 M USD. In the latter case, the additional cost of the low-temperature TES needs to be considered, when the heat should be stored for later use. However, it is expected that the cost of the TES will be substantially below the resulting difference of 570 k USD, which makes the regenerative design economically not viable, in any case. Hence, the Shell & Tube solution is the better choice, also in terms of simplicity of process control and valve operation.

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 as well as thermodynamic performance. For the Small-Scale Plant, the classical Shell & Tube solution is fully sufficient as the needed flow cross sections are small enough.

In conclusion, it has to be pointed out that a 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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