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D3.3 – Process control challenges and recommended methods

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Executive Summary

The current deliverable (D3.3) is entitled “Process control challenges and recommended methods”. It is a public document produced within Task 3.3 “Process control challenges (system level)”.

The ASTERIX-CAESAR project aims to develop an innovative Compressed Air Energy Storage system with integrated thermal management. This deliverable, D3.3, provides a comprehensive evaluation of the process control challenges and recommends specific methods to ensure stable operation during both charging and discharging cycles.

The charging phase analysis identifies the need for precise temperature control at each compression stage to maximize storage density. The recommended solution involves a common air collector pressurized by a VFD-controlled blower, feeding four independent control valves. This architecture successfully manages the wide dynamic range of cooling required, especially for the final aftercooler which must accommodate high temperature fluctuations.

During the discharge phase, the report evaluates three different operation modes: Constant Pressure, Sliding Pressure-Mass Flow, and Sliding Pressure-Volumetric Flow. The summary concludes that these modes can be managed through a single, integrated control framework. By manipulating the atmospheric air mass flow, the system effectively stabilizes the turbine inlet temperatures, ensuring reliable power generation despite the evolving state of the air reservoir.

Technical and scale-related challenges were also assessed, noting that while single-blower systems suffice for small and medium-scale plants, large-scale applications will necessitate multi-blower arrays. The report emphasizes the importance of selecting "Blowing" configurations to minimize capital expenditures and ensure component longevity by avoiding exposure to high-temperature exhaust air.

In conclusion, the proposed control strategies provide a clear roadmap for the automation of the ASTERIX-CAESAR plant. By addressing the non-linearities of the heat transfer process and defining clear boundaries for out-of-scope elements like internal turbomachinery logic, this document establishes the foundation for the subsequent detailed design and implementation phases of the project's control systems.

Nomenclature

ACRONYM	DESCRIPTION
CAES	Compressed Air Energy Storage
CSP	Concentrated Solar Power
EU	European Union
HEX	Heat Exchanger
HHEX	High-Temperature Heat Exchanger
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
LHEX	Low-Temperature Heat Exchanger
LPT	Low Pressure Turbine
S&T	Shell and Tube
TES	Thermal Energy Storage
VFD	Variable Frequency Drive
VIGV	Variable Inlet Guide Vanes

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1 Introduction

1.1 Objectives

The ASTERIX-CAESar system operation imposes control challenges that will need to be correctly addressed. The difficulties are related to the heat exchange processes. The control design challenges are targeted for the best heat exchanger configuration of tasks T3.1 and T3.2, i.e. the classical recuperative design method. Realistic boundary conditions for all components and suitable plant control policies are assumed based on vast experience within the consortium. Potential process control challenges are identified, and suitable control methods are proposed to ensure robust and efficient operation. It has to be pointed out that, according to the proposal, rather than providing an optimised control design for a specific configuration, this task should identify control issues and shall propose possible solutions or methods, i.e. pave the way towards future commercial scale implementation.

1.2 Partners involved and role

According to the Grant Agreement, activities related to the identification of plant control issues and the definition of adequate control method have been carried out mainly by ROMA3, CENER, Alborg CSP, and USE.

1.3 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 28 months 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.

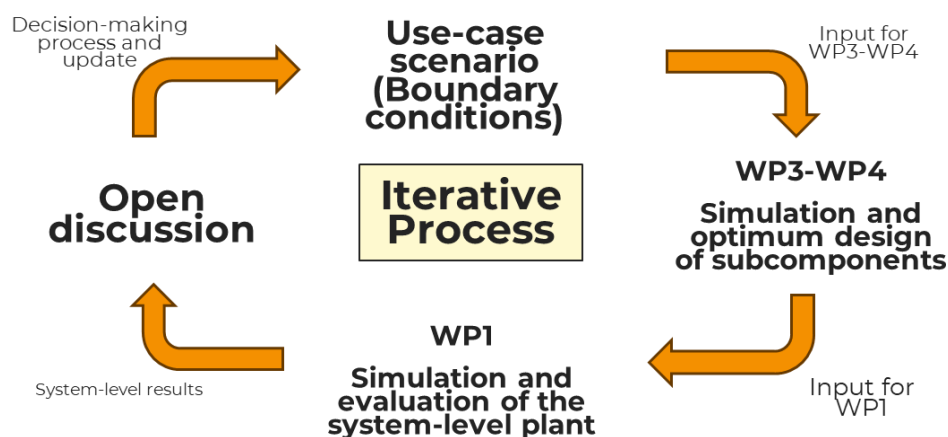


Figure 1. Iterative methodology carried out for interaction between WP1-WP3-WP4.

1.4 Deliverable outline and contents

The deliverable is structured as follows:

Section 2 recalls the ASTERIX-CAESar plant concept. In Section 3, aims of the deliverable are stated and boundaries and assumptions made to address control challenges clearly specified. Section 4 and Section 5, dedicated to the charging and discharging phase respectively, present a similar structure. In a first part, heat transfer equipment behaviour related to the plant management policies defined in WP1 are discussed and analysed. In a second part, control methods and strategies suitable to implement the above policies are proposed and discussed. Section 6 concludes and provides guidance for future developments.

2 ASTERIX-CAESar plant description

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 (Figure 2). 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 (Figure 3). This process raises the air temperature, thereby enhancing the power output of the system.

In the originally proposed concept, 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.

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 carried out in WP3. 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.

More in detail, the charging section heat transfer equipment is constituted by four low temperature heat exchangers: three intercoolers (LHEX1, LHEX2, LHEX3) and one aftercooler (LHEX4). Atmospheric air is used as coolant medium and as thermal energy carrier to supply heat to industrial processes. The atmospheric air circulation through the heat exchangers is achieved by installing air blowers. The design of low temperature heat exchanger has been carried out in T3.1: Optimum low-temperature heat exchanger design (reuse heat of compression) and results presented in [1]. The design has been performed taking into account two relevant aspects:

- the achievement of a low compressed air exit temperature in order to reduce the specific compression work and to exploit more efficiently the compressed air storage volume;
- the achievement of a relatively high atmospheric air exit temperature in order to supply heat suitable for industrial users ($T=150-160^{\circ}\text{C}$).

The design of low temperature heat exchanger has been carried out by adopting the classical Shell & Tube approach and a less conventional approach based on the Heat Recovery Steam Generator technology (HRSG). For the Medium-Scale Plant as well as for the Large-Scale Plant, the HRSG

based design approach offers advantages in terms of cost and thermal performance. For the Small-Scale Plant, the classical Shell & Tube solution is fully sufficient as the needed flow cross sections are small enough.

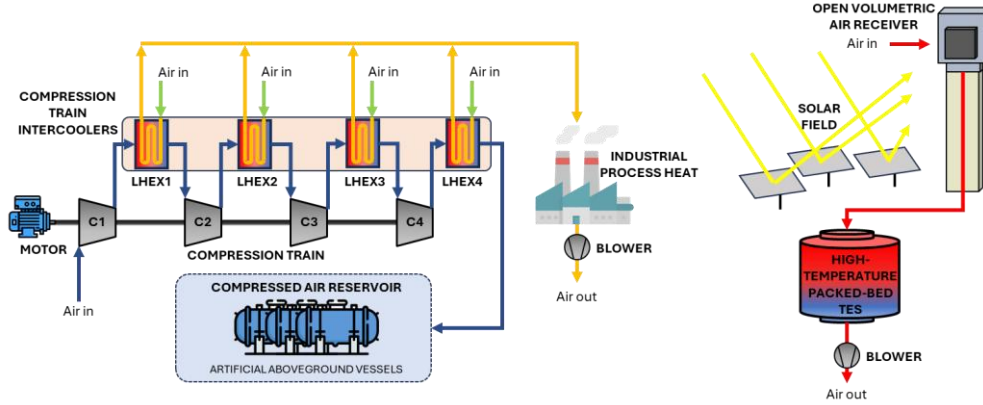


Figure 2: CAES plant charging section.

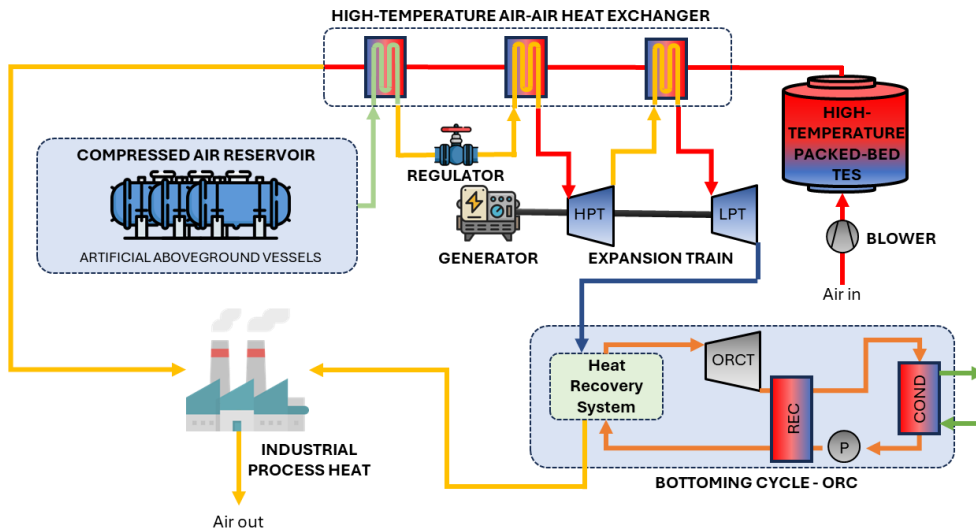


Figure 3: CAES plant discharging section.

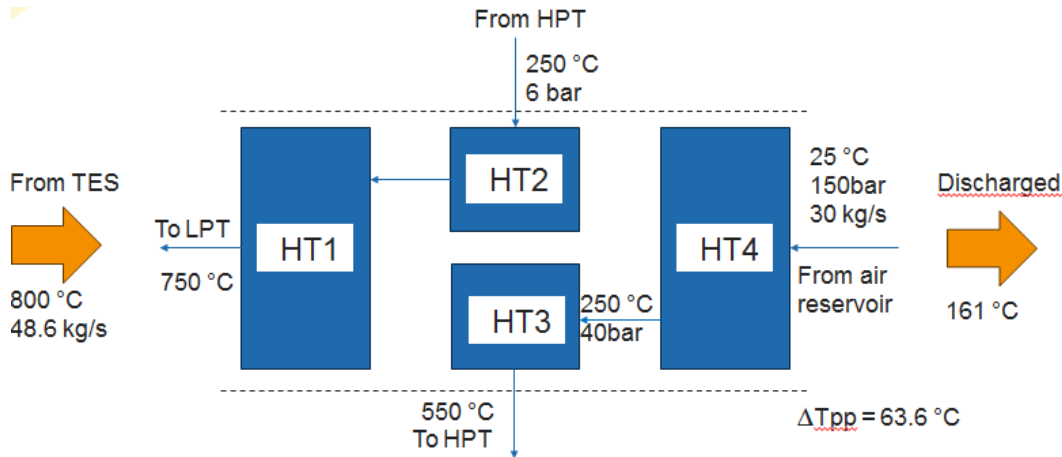


Figure 4: High-Temperature Air-Air Heat Exchanger according to HRSG design approach

Figure 3 presents the layout of the plant discharging section, where a stream of hot ambient air is used to heat and re-heat the pressurised air entering the High Pressure (HPT) and Low Pressure Turbine (LPT).

Also for the plant discharging section the design approach based on the HRSG technology brings advantages in terms of lesser capital costs and lesser pressure losses on both pressurized and atmospheric air sides in respect to the conventional one based on Shell & Tube heat exchangers.

The High-Temperature Air-Air Heat Exchanger of Figure 3 is arranged as sketched in Figure 4. Main design quantities are shown in the figure. Four tube bundles (HT1, HT2, HT3 and HT4) are placed inside a duct fed by hot air at atmospheric pressure coming from the High Temperature packed bed TES at 800°C. The hot stream crosses HT1, where the low-pressure air stream is heated up to the LPT inlet temperature of 750°C. Then, the atmospheric air stream is equally split to feed the parallel units HT2 and HT3, where high- and low-pressure air streams are heated up to 550°C, which represents the HPT inlet temperature. Finally, in HT4, high pressure air taken from the reservoir is preheated up to 250°C before entering HP3.

The four tube bundles have been sized according to the procedure described in [2], where design results are given and commented.

3 Aims and assumptions of the present analysis

The aim of D3.3 is the identification of the process control challenges and the assessment of suitable control methods to ensure robust and efficient plant operation.

As stated in Chapter 1.1, low and high temperature heat exchanger operation is closely linked with compressor, expander and pressure vessel charge/discharge operations. Therefore, the behaviour of the whole system is determined by the operating characteristics of each plant component and by the control device installed on them.

The present analysis is focused on the control of the heat transfer equipment to implement suitable plant management policies for charging and discharging phases.

Concerning the charging phase, the compressor train is operated at constant nominal speed and no compressor control devices (such as VIGV, Variable Inlet Guide Vanes) are foreseen. At design point, the nominal total pressure ratio (e. g. 150 bar in the case of Medium Scale plant)- is equally split among the four compressors during charging [3]. As the pressure ratio increases, the mass flow rate decreases and the isentropic efficiency of each compressor improves, according to compressor operating characteristic. In this way of operation, the pressure increase in the charging process is predominantly absorbed by the last compressor, as discussed in [3]. At the beginning of charging, the isentropic efficiency of the last compressor is some 15% lesser than the nominal

design value. Anyway, the lower efficiency is offset by the lower pressure ratio, resulting in slightly reduced energy consumption compared to the other compressors throughout the entire charging process. Consequently, the temperature of the air discharged by each compressor tends to increase as the charging phase proceeds. As a result, mass flow and temperature of air streams entering each heat exchanger vary continuously. In order to ensure stable and efficient operation of the compressor train, heat exchangers are handled to maintain the compressed air exit temperature close to the reference design value of 45°C.

Concerning the discharge phase, three operation modes have been taken into consideration:

- constant pressure and constant pressurized air mass flow rate;
- sliding pressure at constant pressurized air mass flow rate;
- sliding pressure at constant volumetric air mass flow rate.

In the first case, according to Figure 3, only the high-pressure air preheater placed upstream the pressure regulation valve operates under variable conditions. The remaining tube bundles operate at design conditions during the most part of the discharging phase.

In sliding pressure at constant mass flow rate operation, approximatively the first half of the discharge phase occurs at a constant HPT inlet pressure of 90 bar. In the remaining period of discharge, the HPT inlet pressure decreases from 90 to 40 bar.

The sliding pressure at constant volumetric flow rate represents the most efficient discharge option. Nevertheless, the marked decrease of the plant output during operation may hinder the plant integration with the electric grid and leads to a complicate system operation. For these reasons, such an option would not be considered for commercial implementation.

In the next chapters, the performance of heat transfer equipment will be investigated in reference to the different operation policies taken into consideration. On the basis of achieved results, suitable control methods will be proposed and discussed.

For sake of brevity, control issues and possible solutions will be discussed taking the Medium Scale plant as a reference keeping in mind that findings achieved are applicable regardless of the plant size.

For both charge and discharge phases, heat transfer equipment designed according the HRSG based approach is taken as a reference.

4 Charging phase

First, the behaviour of heat transfer equipment during the charging phase is analysed in order to define the requirements of the control system. In the second section of the chapter, suitable control loops are proposed and discussed.

4.1 Low temperature heat transfer equipment operations

In the following, the four heat exchangers crossed in succession by the compressed air stream will be referred as LHEX1, LHEX2, LHEX3 and LHEX4, as shown in Figure 2.

The design of each heat exchanger has been carried out by assuming an atmospheric air temperature of 25°C. The design has been performed in order to ensure the achievement of a compressed air exit temperature equal to 45°C on the entire expected operating range by modulating the atmospheric air mass flow rate. 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.

Table 1,

Table 2, Table 3 and Table 4 summarize the off-design performance of LHEX1, LHEX2, LHEX3 and LHEX4, respectively. As anticipated in the first part of Chapter 3, the operating range becomes progressively broader moving from LHEX1 to LHEX4. LHEX1 operating conditions

vary 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.

In order to achieve the target compressed air exit temperature, the control system should be able to vary the atmospheric air mass flow rate m_c according to values reported in Tables 1-4. It can be noticed that a small variation is required for the first three heat exchangers, while LHEX4 needs a huge modulation capacity.

The performance of LHEXs has been explored taking also into consideration the variation of the ambient temperature, which represents the temperature of the atmospheric air stream entering the heat exchangers. Further evaluations have been carried out by varying the inlet air temperature in the range 5-35°C. Results have shown the capability of LHEXs to cool the compressed air to the target temperature in the entire operating range. Of course, higher ambient temperatures require higher cooling mass flow rates. Therefore, the control system should be designed to ensure an atmospheric air mass flow rate variation from 14 to 18 kg/s for the first three heat exchangers. A wider range – from 11 to 22 kg/s-is required for the fourth one.

Table 1: 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	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 2: 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	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 3: 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	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 4: LHEX4 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	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

$\dot{m}_H$: 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} ; ΔP_s ; U : overall heat transfer coefficient; ΔP_s : shell side pressure drop; Q : exchanged thermal power.

4.2 Low temperature heat transfer equipment control system

The heat transfer equipment in the charging section consists of a train of four low-temperature heat exchangers (LHEX1, LHEX2, and LHEX3 acting as intercoolers, and LHEX4 as an aftercooler). The control system's primary function is to manage the thermal variations resulting from the compression process to ensure stable and efficient plant operation.

Control Objectives and Variables

The principal objective of the control system is to maintain the compressed air exit temperature from each of the four heat exchangers at a constant target of 45°C. Achieving this setpoint is critical for minimizing the specific work required for subsequent compression stages and optimizing the use of the compressed air storage volume.

The control architecture is defined by the following variables:

- Targeted Process Variable (PV): Compressed air outlet temperature for each stage (HEX1–HEX4).
- Setpoint (SP): 45°C, managed independently for each compressor stage.
- Manipulated Variable (MV): Atmospheric air volumetric flow, which is regulated by the pressure of the atmospheric air blower(s).
- Final Control Element (FCE): A combination of atmospheric air blowers and dedicated control valves.

Control Strategy Options

These configurations were evaluated for the atmospheric air circulation system. Three primary architectural options were evaluated for the atmospheric air circulation system, as shown in Figure 5:

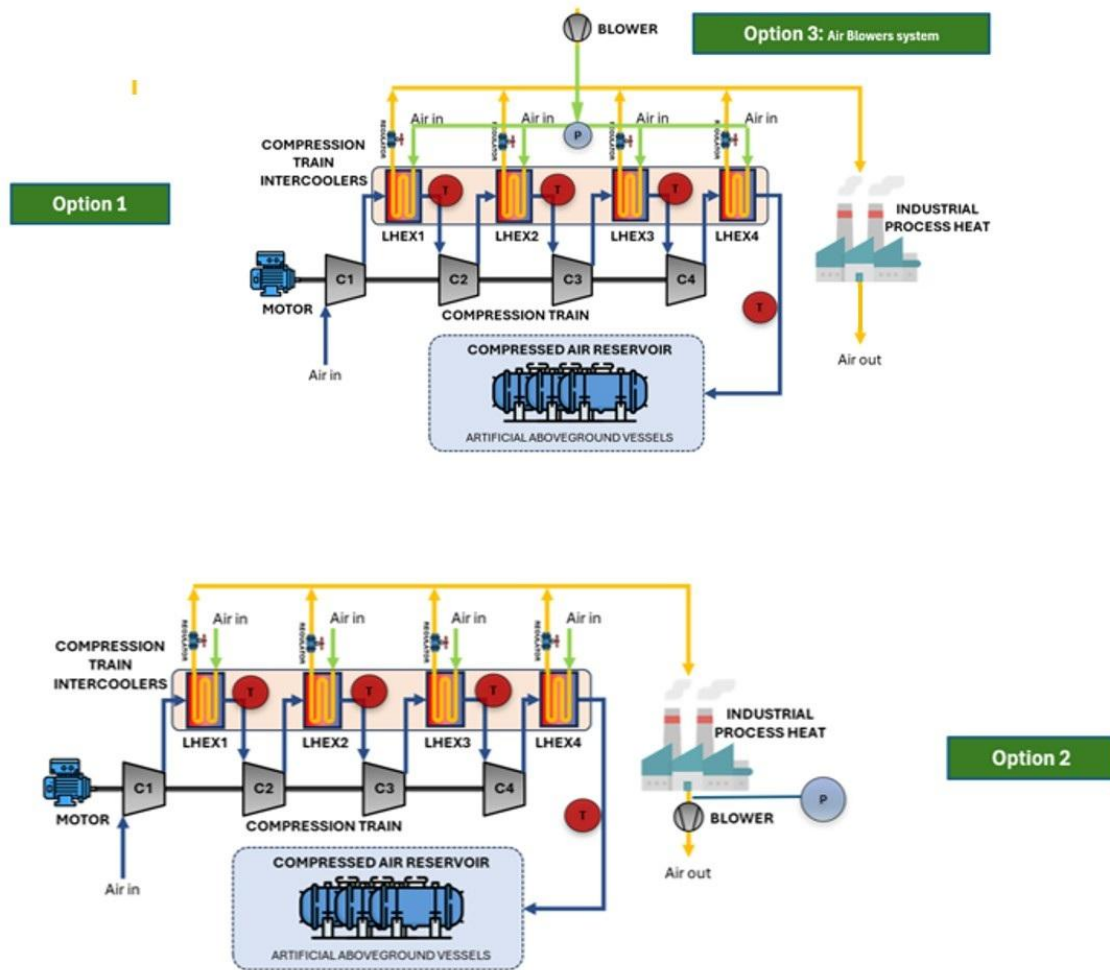


Figure 5: LT HEX Train Control Loops options

Option 1: Blowing Configuration: This setup employs an atmospheric air blower equipped with a Variable Frequency Drive (VFD) on the inlet side. The VFD regulates the pressure in a common distribution header. Downstream, four independent control valves modulate the specific flow to each heat exchanger stage based on real-time temperature feedback.

Option 2: Suctioning Configuration: This involves placing the blower downstream of the heat exchangers. While technically feasible, it is less desirable as it forces the blower to handle air temperatures that can reach up to 200°C, significantly increasing component costs and maintenance requirements.

Option 3: Scaled Independent Pairing: Specifically considered for large-scale plants, this option involves pairing individual blowers directly with independent control valves for each stream. In respect of option 1, the present option is expected to ease the proper management of the system. On the other hand, the adoption of multiple blowers leads to higher capital costs.

Out of this Analysis Scope

To define the boundaries of this control study, several interrelated systems are explicitly excluded from the LT HEX control elaboration:

- **Compression Train Internal Regulation:** The internal management of the compressor, including the motor VFD system for pressure regulation and the control of Variable Inlet Guide Vanes (VIGV), is handled within the scope of Work Package 4 (WP4).
- **Industrial Process Heat (IPH) End-User Side:** While the LT HEX train provides the thermal source, the secondary heat exchangers on the customer side—including the regulation of secondary fluids (water, steam, or oil) and the specific thermal load demand of the user—are outside the current scope.
- **High-Pressure Storage Management:** The logic governing the final charging pressure setpoints and the safety protocols for the air reservoir are treated as external boundary conditions.

Operational and implementation challenges

The control system must handle significant dynamic shifts throughout the charging cycle as the pressure in the storage reservoir rises from 40 bar to as much as 150 bar.

Asymmetric Modulation Requirements: While the first three heat exchangers (LHEX1–3) experience relatively stable conditions requiring minor flow modulations (approx. 14–18 kg/s), LHEX4 faces extreme variations. In LHEX4, the rapid increase in compressor discharge temperature outweighs the reduction in compressed air mass flow, requiring a massive modulation of cooling air from 11 to 22 kg/s.

Ambient Temperature Sensitivity: The system must compensate for seasonal ambient temperature fluctuations (ranging from 5°C to 35°C), which directly impact cooling capacity and require higher cooling mass flow rates during warmer periods.

Industrial Heat Integration: A secondary challenge is the management of the thermal load for industrial users. Maintaining a consistent atmospheric air exit temperature of 150–160°C is necessary to supply high-quality heat, but doing so without intermediate thermal storage is complex in real-world scenarios.

Scale-Dependent Configurations: For small and medium plants, a single air blower is typically sufficient. However, large-scale plants may require a multi-blower system where Option 1 is either scaled up or replaced by a configuration that pairs individual blowers with independent control valves for each stream (Option 3), provided blower flow limitations allow for such granularity.

5 Discharge phase

In the following, proposed plant management policies will be analysed to assess the plant performance and to address suitable control loops.

5.1 High temperature heat transfer equipment operations

As stated in Chapter 3, two operation modes to accomplish the discharge phase are considered:

- constant pressure at HPT inlet and constant pressurized air mass flow rate;
- sliding pressure at HPT inlet and constant pressurized air mass flow rate;

The first one implies that the control system maintains a constant pressure at the inlet of HT3 (i.e. downstream the pressure regulator shown in Figure 3). Such a value corresponds to the HPT design inlet pressure (40 bar for Medium Scale plant) plus the tube side pressure loss in HT3 bundle. As a consequence, being constant the pressure at the exit of the expansion train, a constant air mass flow rate equal to the design value (30 kg/s) flows across the HPT and the LPT. In this mode of operation, provided that both HPT and LPT inlet temperatures are kept at their design values (550 and 750°C, respectively), all plant components, except the tube bundle HT4, operate at design conditions. HT4 operations, instead, are characterised by tube side decreasing pressure as a consequence of compressed air reservoir emptying. Anyway, calculations carried out by using an off-design model of the High Temperature Air-Air Heat Exchanger have shown that this does not affect the capability of HT4 to heat the high-pressure air stream up to the target temperature. As a consequence, the entire system has the potentiality of operate at design conditions during discharge phase.

Anyway, in the last part of the discharge phase the temperature of the atmospheric air stream coming from the high temperature TES shows a noticeable decrease. This, of course, implies that design values for both LPT and HPT cannot be reached. According to data provided by WP1, the effect of the reduction of the atmospheric air temperature on High Temperature Air-Air Heat Exchanger behaviour has been analysed. Reductions of LPT inlet temperature of the same order of the reduction of the temperature of the atmospheric air mass flow rate have been found. In the worst case (at the end of discharge phase), LPT inlet temperature reaches a value of some 700°C, in respect to the design value of 750°C. HPT inlet temperature shows a reduction in respect to the design value (550°C) of about 30°C.

As reported in [3], in sliding pressure at constant mass flow rate operation, approximatively the first half of the discharge phase occurs at a constant HPT inlet pressure of 90 bar. In the remaining period of discharge, the HPT inlet pressure decreases from 90 to 40 bar.

The compressed air mass flow rate is kept constant at the design value throughout the entire discharge phase (30 kg/s) by HPT and LPT regulating devices. In the first part of discharge, the system is operated as described in the constant pressure-constant mass flow case. When sliding pressure operations occur, the temperature at HPT outlet increases and, consequently, the thermal power amount required to heat the low-pressure air tends to decrease. In order to assess the behaviour of the Hot Temperature Air-Air Heat Exchanger, the knowledge of how the HPT outlet temperature varies with the inlet pressure is required. Table 5 shows values of HPT isentropic efficiency during the sliding pressure discharge phase provided by WP4 and used to evaluate the HPT outlet temperature. It can be noticed as the HPT efficiency increases when pressure at HPT inlet decreases, thus contributing to reduce the HPT outlet temperature variation during operations.

A control policy aimed at keeping constant the LPT turbine inlet temperature at 750 °C has been taken into consideration. This is expected to bring to high efficiency/high output power operations. The atmospheric air mass flow rate is modulated in order to achieve the above goal. The analysis has been performed using the High Temperature Air-Air Heat Exchanger off-design model developed in ChemCad environment. Results are given in Table 6. The first three lines refers to sliding pressure operations. For sake of completeness, results achieved for constant

pressure-constant mass flow are given in the last line.

Table 5: HPT efficiency for sliding pressure-constant mass flow rate discharge

HPT Inlet pressure [bar]	Constant pressure-constant mass flow isentropic efficiency [%]	Sliding pressure-constant mass flow isentropic efficiency [%]
40	90.3	89.1
50	-	89.0
60	-	85.5
70	-	82.6
80	-	80.9
90	-	80.7

Table 6: Results for sliding pressure-constant mass flow operation mode.

P_{IN-HPT} [bar]	m_{ATM} (kg/s)	T_{IN-HPT} (°C)	$T_{OUT-HPT}$ (°C)	T_{IN-LPT} (°C)	T_{DISCH} (°C)	P_{TH} (MW)	P_{EXP} (MW)
90	50.1	548	203	750	151	35.1	22.5
70	49.5	548	218	750	153	34.6	22.0
40	48.5	550	256	750	162	33.5	20.9
40 (const. p)	48.6	550	250	750	161	33.5	21.0

P_{IN-HPT} : high pressure turbine inlet pressure; m_{ATM} : atmospheric air stream mass flow rate; T_{IN-HPT} : high pressure turbine inlet temperature; $T_{OUT-HPT}$: high pressure turbine outlet temperature; T_{IN-LPT} : low pressure turbine inlet temperature; T_{DISCH} : atmospheric air stream exit temperature; P_{TH} : thermal power transferred in the heat transfer equipment; P_{EXP} : mechanical power produced in the expansion process.

It can be noticed as operations at nominal LPT inlet temperature can be achieved by moderate variations of the atmospheric air mass flow rate in the range 48.5-50.1 kg/s. Interestingly, the HPT inlet temperature remains really close to the nominal design value, confirming the validity of the proposed control policy in terms of effectiveness. In addition, no potentially dangerous operations related to the achievement of too high turbine inlet temperatures are to be feared. The narrow discharge temperature range found during sliding pressure operations (150-160°C) ensures an adequate quality and value of the heat supplied to industrial processes. In conclusion, for all the above reasons, the behaviour of the overall system during the whole discharge phase can be considered satisfactory.

5.2 Heat transfer equipment control system

Following the same logic applied to the charging phase, the control strategy for the heat transfer equipment during the discharge process is defined by the requirement to maintain high-quality thermal energy for both power generation and industrial reuse. This section focuses specifically on the "Constant pressure and constant pressurized air mass flow rate operation mode.

Control Architecture and Operational Logic

The primary objective during discharge is to ensure the Low-Pressure Turbine (LPT) Inlet Temperature is maintained at the design setpoint by managing the atmospheric air stream. This is critical to ensure the expansion train operates at its peak efficiency during the electricity generation phase.

The control logic utilizes the Atmospheric Air Mass Flow as the primary manipulated variable. Unlike the charging phase, which is driven by cooling needs, the discharge phase focuses on regulating the thermal power transferred to the air stream before it enters the low-pressure stage of the expansion train.

Control Strategy Options

Based on the identification matrix for constant pressure discharge, the following configurations were evaluated for the atmospheric air management:

Option 1: Air Blower VFD (Blowing): This setup utilizes a Variable Frequency Drive (VFD) on the atmospheric air blower to modulate the pressure and flow directly. This allows for precise control of the heat transfer rate by adjusting the mass flow rate of the cooling/heating medium.

Option 2: Air Blower with Control Valve: This alternative utilizes a blower operating at a minimum discharge pressure in combination with a dedicated atmospheric air control valve. The valve provides the fine-tuning necessary to reach the specific thermal setpoints required by the turbine.

Selection between these options depends on the plant scale. For most applications, Option 1 (VFD) is preferred for its superior energy efficiency, as it avoids the pressure drop losses associated with throttling valves. This configuration ensures that the thermal energy recovered from the storage system is used effectively to preheat the air, maintaining the High-Temperature Thermal Energy Storage (HT TES) output characteristics.

Out of this Analysis Scope

To maintain the focus on the heat transfer equipment (HT HEX) and its immediate loops, the following elements are considered outside the scope of this specific section:

- **Turbomachinery Internal Controls:** The regulation of the expansion train, including generator shaft speed, power electronics, and the High-Pressure Turbine (HPT) internal logic, is handled by separate turbomachinery work packages.
- **HT TES Temperature Stability:** While the system relies on a constant 800°C outlet temperature from the solid packed bed thermal storage, the internal management of the TES (how to keep that temperature as constant as possible throughout the entire discharge process) is treated as a given boundary condition and is out of the scope of this analysis.
- **HPT Admission Control:** The specific operation of the Pressure Admission Control Valve upstream of the High-Pressure Turbine is considered a separate mechanical control loop for the expansion train.

Operational and implementation challenges

TES Discharge Consistency: A major technical challenge is maintaining a reasonable constant TES outlet temperature with a solid packed bed storage system over the duration of the discharge process. Any decay in TES temperature would require the control loop to compensate by adjusting the expansion parameters.

Scale-Dependent Flow Management: Similar to the charging phase, the required mass flow rates (e.g., approximately 30 kg/s for a medium-scale plant) necessitate careful blower sizing to ensure that the control elements (VFD or valves) remain within their effective modulation ranges throughout the cycle.

Discharge phase control strategy options

Finally, the following analysis justifies why the various discharge operational modes—Sliding Pressure (Mass and Volumetric flow) and Constant Pressure (Mass flow)—can be consolidated into a single control strategy, as they share the same physical architecture and fundamental control logic. Figure 6 shows a comparison among different discharging operation strategies [3].

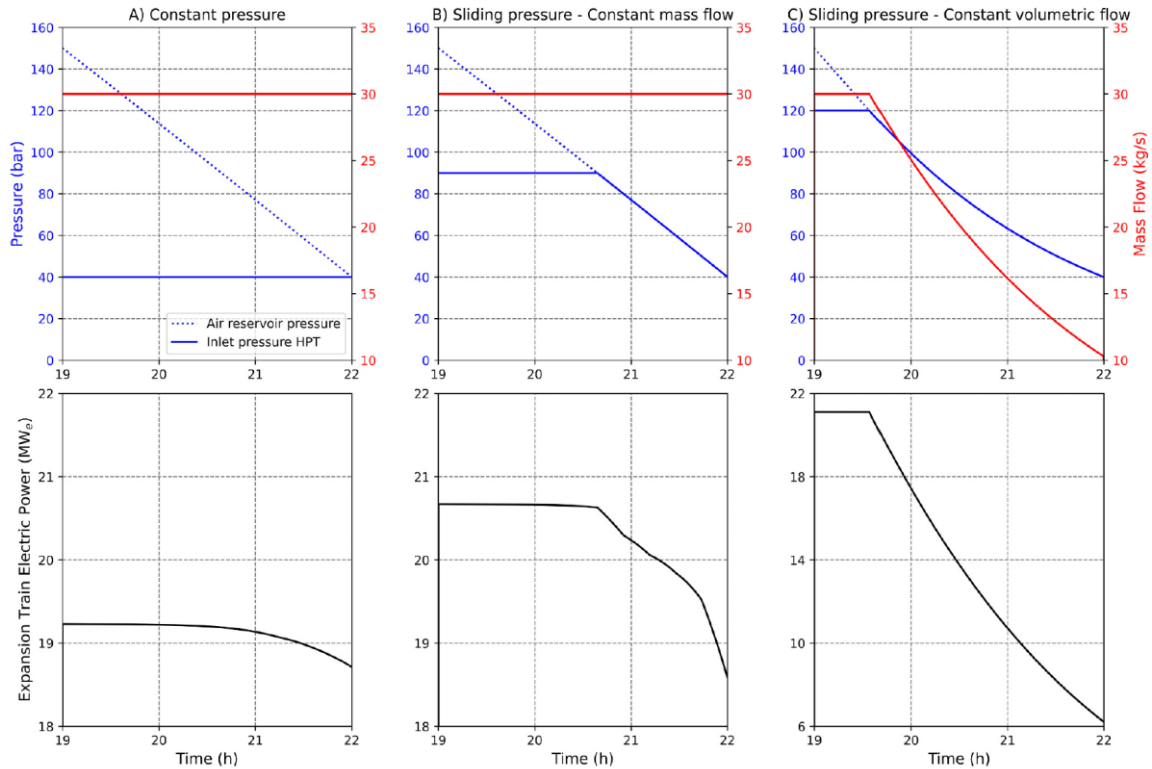


Figure 6: HT TEX Discharge operations strategies

The integration of these discharge scenarios into a single analysis is primarily justified by the commonality of the Manipulated Variables (MV) and the Final Control Elements (FCE). In all cases, the core thermal objective remains the regulation of the Low-Pressure Turbine (LPT) Inlet Temperature. Whether the air storage is at a constant pressure or decompressing (sliding), the primary way to stabilize this temperature is by modulating the Atmospheric Air Mass Flow. Since the mechanical equipment—specifically the atmospheric air blowers and regulation valves—is identical across all modes, the control architecture is treated as a versatile framework capable of handling different process dynamics.

The technical selection of the FCE remains consistent regardless of the discharge mode. The analysis identifies two main technological paths: Option 1 (Air Blower VFD/Blowing) and Option 2 (Air Blower with Control Valve). These hardware configurations represent the physical limit of the system's response. In a "Sliding Pressure" scenario, the VFD or the valve simply operates across a wider dynamic range to compensate for the continuous changes in pressure and flow coming from the CAES reservoir. By grouping these cases, the study focuses on the robust

performance of these mechanical elements under varying stress profiles rather than unnecessarily duplicating the control loop design for each specific pressure strategy.

Consolidating these modes allows for a comprehensive assessment of shared boundary conditions and technical risks. For instance, maintaining a reasonably constant 800°C outlet temperature from the High-Temperature Thermal Energy Storage (HT TES) is a critical challenge that persists across all discharge methods. By analyzing them together, it is demonstrated that a single, well-tuned control loop is resilient enough to handle different "Discharge Operation Modes" without requiring changes to the instrumentation. This unified approach not only simplifies the plant's automation architecture but also provides a clear, scalable roadmap for large-scale implementations where the atmospheric system must be flexible enough to adapt to any grid-dispatch requirement.

6 Conclusions

The primary conclusion regarding the system architecture is the preference for the "Blowing" configuration for atmospheric air management. Through Variable Frequency Drive (VFD) blowers and independent control valves on the cold side of the process, the plant avoids the excessive capital costs and maintenance risks associated with high-temperature suctioning components. This setup provides the necessary dynamic range to handle the significant thermal gradients observed during the final stages of the charging cycle.

Additionally, the analysis confirms that a unified control logic is applicable across all discharge modes—including Constant Pressure and Sliding Pressure scenarios. Since the manipulated variables and final control elements remain consistent, the system demonstrates high resilience to the varying pressure profiles of the air reservoir. This consolidation simplifies the automation architecture, ensuring that the plant can adapt to different grid-dispatch requirements without other final control elements.

A critical consideration remains the integration of the High-Temperature Thermal Energy Storage (HT TES). While the heat exchanger control loops provide a vital secondary regulation stage to stabilize turbine inlet temperatures, the inherent thermal decay of a solid packed bed during long discharge cycles represents a boundary condition that must be carefully monitored. Future efforts should focus on the fine-tuning of these loops to compensate for potential TES temperature drops toward the end of the discharge phase.

7 References

- [1] CENER e ROMA3, «D3.1 - Optimum low temperature heat exchanger design (reuse heat of compression),» 2025.
- [2] CENER e ROMA3, «D3.2 - Optimum high temperature heat exchanger design (solar input in power cycle),» 2025.
- [3] J. Baigorri, A. Federici, T. Kubikova, T. Du Toit, P. Rodriguez-deArriba, C. Salvini, D. Sanchez e F. Zaversky, «Modelling of an Innovative Integration of Compressed Air Energy Storage (CAES) with High-Temperature oncentrated Solar Power (CSP): a Comprehensive Use-Case Study,» *Journal of Energy Storage*, 2025.