

Article

Thermodynamic and Annual Performance Assessment of a Cascaded Air-Source Heat Pump for Industrial Steam Generation: a Mediterranean Case Study

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Abstract

Industrial heat decarbonization is a key challenge for the energy transition, particularly in applications requiring high-temperature steam when industrial waste heat is unavailable. This study presents a thermodynamic and annual performance assessment of a steam-generating heat pump system using ambient air as the external low-temperature heat source. The system combines cascaded low- and medium-temperature heat pumps with a mechanical vapor recompression unit. Steady-state thermodynamic models, parameterized using assumptions consistent with representative commercial components, are used to evaluate the system under nominal and annual operating conditions. The annual assessment is applied to the continuous steam demand of a real paper-production facility under Mediterranean climatic conditions. A grid-supported photovoltaic and battery energy storage system is included as the electricity supply framework for characterizing the operation of the steam-generating heat pump under time-varying ambient and renewable-generation conditions. Under the adopted modeling assumptions, the system produces saturated steam at 5–25 bar with design COP values of 1.35–1.49. As steam pressure increases, mechanical vapor recompression becomes the dominant electricity-consuming stage, resulting in a moderate reduction in COP while decreasing the sensitivity of overall system performance to ambient temperature variations. Full-year simulations show that the operating COP exceeds the design-point value for the investigated location because ambient temperatures are generally more favorable than the nominal design condition. The results of the preliminary economic analysis performed for three PV-BESS configurations show that the economic feasibility strongly depends on the electricity purchase price. None of the studied configurations recovers its additional investment at 50 €/MWh, whereas discounted payback periods range from 6.9 to 16.5 years at 100 €/MWh and from 4.2 to 7.3 years at 150 €/MWh. As the integrated system has not been experimentally validated, the results should be interpreted as model-based estimates.

Keywords: industrial heat; electrification; decarbonization; mechanical vapor recompression; photovoltaics; battery

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1. Introduction

According to the International Energy Agency (IEA) [1], industrial heat decarbonization is a critical component of climate mitigation. Indeed, industry accounts for roughly 30% of global final energy consumption (129 EJ in 2023) and 18% of direct en-

ergy-related CO₂ emissions (around 6 Gt/y). When including indirect energy-related CO₂ emissions from heat and electricity use, total industry energy-related CO₂ emissions reach approximately 12.5 Gt/y. Of this, low-temperature heat and steam in manufacturing alone are associated with roughly 3 Gt CO₂ per year.

The challenge of industrial heat decarbonization is especially relevant in sectors such as food and beverages, textiles, chemicals, paper, and wood products, where a large share of thermal demand lies within temperature ranges of 40–160 °C that are already commercially electrifiable [2]. Within this temperature range, commercial solutions based on vapor compression heat pumps (HP) are considered the most efficient electrification option [1]. Indeed, the IEA states that HPs alone could cover about 37% of the total process heat demand of European industry [3], with an even higher potential when considering the still underutilized potential of waste heat, which could be used as a heat source [2].

Based on their output temperature, Arpagaus et al. [4] classify heat pumps into low-temperature (LTHP) (0–60 °C), medium-temperature (MTHP) (60–100 °C), and high-temperature (HTHP) (100–160 °C) systems. It is well known that the efficiency of HPs is strongly dependent on the temperature lift. Specifically, for temperature lifts up to 25 °C, coefficients of performance (COP) typically range between 4 and 10; for lifts between 25 and 50 °C, COPs generally fall within the range of 1.5–6; for lifts between 50 and 75 °C, they are usually between 1.5 and 3; while for temperature lifts between 75 and 150 °C, COP values rarely exceed 2 [3].

Steam remains the main industrial heat carrier for temperatures up to 300 °C [5] and its production is still largely non-decarbonized [6]. When used for steam generation, heat pumps are referred to as steam-generating heat pumps (SGHPs). SGHPs operate using a variety of different processes, typically cascaded or multistage, where lower-temperature stages recover and preheat energy carriers or feedwater, while higher-temperature stages complete the final lift toward steam delivery conditions [5]. SGHPs can be mainly classified into open-loop, closed-loop, and hybrid systems [5]. Among the open-loop systems, where the steam itself is upgraded in the final stage, thermal vapor recompression (TVR) and mechanical vapor recompression (MVR) are the most common solutions. The IEA [1] recognizes MVR as the most mature technology for the electrification of steam generation with temperatures above 160 °C.

Typically, studies on industrial steam decarbonization via heat pumps assume that waste heat is available to be recovered [7,8]. According to Ma et al. [9], studies that do not rely on this assumption are very limited. Among these, Yan et al. [10] proposed a system capable of generating saturated steam at 120 °C from ambient air, calculating a COP between 1.5 and 2.1 for air temperatures ranging from 5 to 35 °C, and subsequently studied it in relation to a specific application [11]. To the authors' knowledge, only Ma et al. [9] explored higher temperatures without relying on waste heat recovery. Acknowledging the IEA's recognition of MVR as the most mature technology, they proposed a system combining a HP and an MVR unit that generates steam at 170 °C from air with temperatures between −30 and 80 °C, achieving a COP of 1.42 under the worst-case condition. Against this background, the present work assesses a cascaded SGHP configuration that, starting from ambient air, can exceed this temperature level. Table 1 summarizes the air-source SGHP studies identified as directly relevant to the present work, comparing their scale, system architecture, air-source conditions, steam delivery conditions, and reported COP.

Table 1. Comparison of published air-source steam-generating heat pump systems and their operating conditions, performance boundaries, contributions, and limitations.

Reference	Study Type and Scale	System Architecture	Air Temperature, °C	Steam Condition	COP	Main Contribution	Main Limitation
Yan et al. (2020) [10]	Numerical and prototype study	Cascade heat pump, flash tank, and water-injected twin-screw water-vapor compressor	5–35	Saturated steam at 120 °C	1.50–2.10	Demonstrates distributed steam generation from ambient air without requiring an industrial waste-heat source	Experimental evidence is limited to selected conditions
Yan et al. (2021) [11]	Numerical optimization and prototype assessment; nominal steam capacity up to 0.5 t/h	Cascade air-source heat pump, flash tank at approximately 85 °C, and twin-screw water-vapor compressor	5–35	Saturated steam at 110 °C	Above 1.65 at 7 °C; above 1.80 at 15 °C;	Extends the air-source concept to sterilization and brewing applications	Detailed results primarily concern 110–120 °C steam
Odukamaiya et al. (2024) [12]	Experimental prototype	Cascaded low- and high-temperature refrigerant cycles with multistage centrifugal compressors	15	Saturated steam at 112–151 °C	1.85 at 112 °C; 1.51 at 150 °C; 1.49 at 151 °C	Experimentally demonstrates saturated-steam production up to 151 °C at approximately 0.5 MW scale	Tests used controlled glycol and high-temperature feedwater rather than the complete ambient-air-to-steam boundary
Ma et al. (2025) [9]	Numerical model, including environmental and economic analyses	Low-temperature R245fa/R1234yf autocascade and high-temperature multistage R718 water-vapor compression	−30–80	Saturated steam at 170 °C	1.42 at −30 °C; 1.76 at 20 °C;	Extends the numerical assessment of air-source steam generation to 170 °C and source temperatures as low as −30 °C	Numerical study without integrated experimental validation

The environmental performance of heat electrification depends on the carbon intensity of electricity supply: indeed, Zeilerbauer et al. [13] carried out a life-cycle assessment of industrial-scale SGHPs, finding that CO₂ emissions could be reduced by as much as 98% relative to fossil-boiler benchmarks, although this outcome is strongly dependent on the electricity supply conditions [13]. This dependency is evident when comparing regions with low-carbon electricity mixes to those still dominated by fossil generation. Indeed, while the EU grid declined to around 180 gCO₂/kWh in 2024, wind- or solar-powered LTHPs emit only 3–9 gCO₂/kWh of heat, and electric boilers emit only 12–34 gCO₂/kWh, corresponding to emission reductions of around 90–95% relative to fossil alternatives [1]. By contrast, in fossil-intensive grids such as China (560 gCO₂/kWh) or Southeast Asian nations (600 gCO₂/kWh), electric boilers can emit up to 23% more CO₂ than coal boilers, becoming cleaner than coal only when grid intensity falls below roughly 440 gCO₂/kWh. In conclusion, this dependence makes the quality of the electricity supply a first-order design variable for any electrified concept, especially in jurisdictions where average grid electricity is not yet consistently low-carbon. For these reasons, the IEA emphasizes that industrial heat electrification must be coupled with renewable electricity deployment, especially for large-scale decarbonization projects. A further advantage of renewable couplings is the reduced exposure to fuel-price volatility, combustion-related pollutant liabilities, and increasingly stringent carbon constraints [1]. Moreover, if the industrial process, as is the case in most situations, requires a quasi-continuous thermal demand, it is essential to pair the renewable-generation source with an energy storage system, either electrical or thermal, to ensure a low-carbon supply even during fluctuations or the absence of the renewable source. In this regard, the strategic importance of electrical batteries is increasing rapidly: the IEA [14] reports that battery energy storage system (BESS) was the fastest-growing commercially available energy technology in 2023, with deployment more than doubling year-on-year, while lower costs are making behind-the-meter battery systems increasingly attractive. The same report concludes that batteries paired with solar photovoltaic (PV) are becoming among the most competitive sources of new electricity supply, thereby strengthening the case for industrial facilities to consider local renewable-electricity ecosystems rather than treating electrified steam systems as isolated loads. Given this, the present work assesses the SGHP concept in combination with a PV-BESS.

In summary, and in contrast to most existing studies that assume the availability of industrial waste heat or focus on sub-160 °C applications, this work investigates the thermodynamic and annual operating performance of a high-temperature (up to 220 °C) SGHP system by (i) considering a thermal upgrade starting from ambient air temperature, (ii) adopting MVR as the final steam-upgrading stage, (iii) explicitly coupling the SGHP with PV generation and BESS to address electricity supply quality, and (iv) reporting system performance on an annual basis under a selected case study demand condition.

The remainder of this paper is organized as follows. Section 2 describes the methods adopted to simulate and evaluate the thermodynamic, annual, and preliminary economic performance of the system. Section 3 discusses the results, including both the standalone SGHP under nominal design conditions and the complete system over a full-year operation. Section 4 presents the conclusions.

2. Methods

The studied configuration is conceived to produce saturated steam in the range 5–25 bar (150–223 °C), using ambient air as the external low-temperature heat source.

The layout consists of a low-temperature heat pump (LTHP) and a medium-temper-

ature heat pump (MTHP), cascaded with a mechanical vapor recompression (MVR) unit, all primarily powered by a photovoltaic system coupled with a battery energy storage system (PV-BESS).

The energy flows of the described configuration are illustrated in Figure 1. The system satisfies the thermal load Q_L through the cascade of the two HPs and the MVR unit, with their respective electrical demands denoted as E_{LTHP} , E_{MTHP} and E_{MVR} . The total electrical load of the system (E_{SGHP}) is supplied directly by the PV system ($E_{PV,SGHP}$), by the BESS ($E_{B,SGHP}$), or by the grid ($E_{G,SGHP}$). Any surplus generation is fed into the grid (although including the electricity grid among the supply sources clearly makes steam production not fully decarbonized, assuming non-grid-supported solutions would require the arbitrary assumption of an unrealistically oversized PV-BESS) ($E_{PV,G}$).

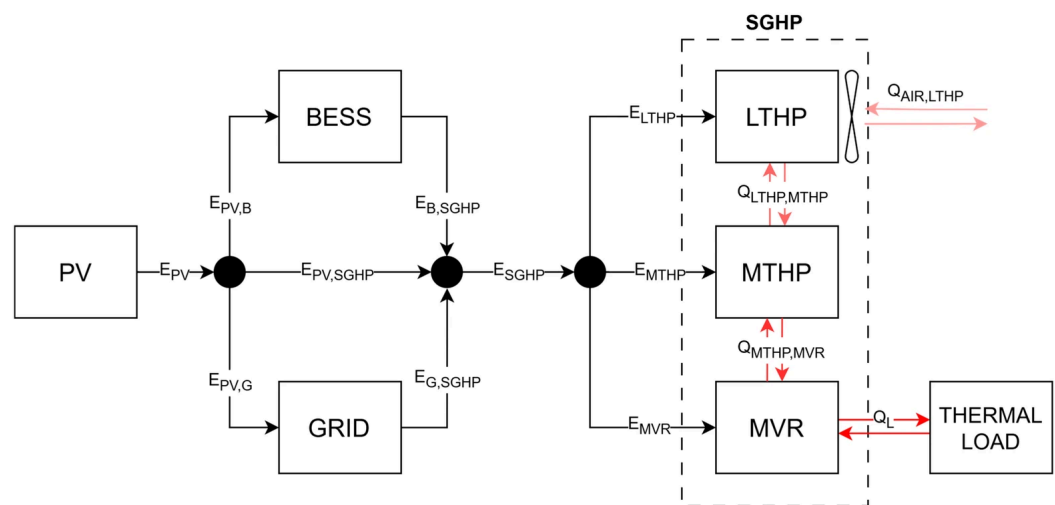


Figure 1. Energy flow diagram of the system.

The system management prioritizes energy self-sufficiency and, therefore, energy self-consumption over each period τ . Energy self-sufficiency (SS) and energy self-consumption (SC) are defined as follows:

$$SS(\tau) = \frac{\sum_{t=1}^{\tau} E_{PV,SGHP}(t) + \sum_{t=1}^{\tau} E_{B,SGHP}(t)}{\sum_{t=1}^{\tau} E_{SGHP}(t)} \quad (1)$$

$$SC(\tau) = \frac{\sum_{t=1}^{\tau} E_{PV,SGHP}(t) + \sum_{t=1}^{\tau} E_{PV,B}(t)}{\sum_{t=1}^{\tau} E_{PV}(t)} \quad (2)$$

In the following subsections, the mathematical models of each component of the system are presented. The models were implemented in MATLAB R2026a [15] and simulated using an hourly time step.

2.1. Photovoltaic and Battery Energy Storage Systems

The PV generation was estimated using PVGIS (ERA5-2023 solar radiation dataset) [16]. The simulation considered crystalline silicon modules with a Standard Test Conditions (STC) efficiency of 22.6% [17], accounted for system losses of 14%, and used tilt and azimuth angles optimized by the software for the selected location.

The BESS was modeled as a lithium-ion battery with a nominal capacity ($E_{B,N}$), a

unitary C-rate, and a round-trip efficiency (η_{RT}) of 90% [18]. The latter is given by

$$\eta_{RT} = \frac{\sum_{t=1}^{\tau} E_{B,SGHP}(t)}{\sum_{t=1}^{\tau} E_{PV,B}(t)} \quad (3)$$

The battery State of Charge (SOC), Equation (4), was constrained between 10% and 90% to prevent degradation from overcharging and deep discharging:

$$SOC(t) = \frac{E_B(t)}{E_{B,N}}; E_B(t) = E_B(t-1) + E_{PV,B}(t) - \frac{E_{B,SGHP}(t)}{\eta_{RT}} \quad (4)$$

where $E_B(t)$ represents the energy stored in the BESS at interval t . In addition, a daily self-discharge rate of 0.2% [18] was considered. The BESS usage over a given time interval τ was calculated by means of the Equivalent Full Cycles (EFC) indicator:

$$EFC_B(\tau) = \frac{\sum_{t=1}^{\tau} E_{B,SGHP}(t)}{E_{B,N}} \quad (5)$$

2.2. Steam-Generating Heat Pump

As mentioned above, the studied SGHP configuration consists of three cascaded components:

- A low-temperature air-to-water heat pump (LTHP), which provides the first temperature lift from 7 to 45 °C and operates with R134a;
- A medium-temperature water-to-water heat pump (MTHP), which delivers the second temperature lift from 45 to 85 °C and uses R1234ze(E);
- A mechanical vapor recompression unit (MVR), which accomplishes the final temperature and pressure increase generating saturated steam within the range 5–25 bar (150–223 °C).

The allocation of the temperature lift among the three components was selected to exploit closed-loop heat pumps within temperature ranges compatible with representative commercial refrigerants, while reserving the final high-temperature lift for direct water-vapor compression. This choice is also supported by the IEA [1], which identifies MVR as the most mature technology for the electrification of steam generation at temperatures above 160 °C. In this arrangement, the MVR operates as an open-loop steam-generating heat pump and produces the required process steam without introducing an additional refrigerant at the highest temperature level.

To increase the practical relevance of the assessment, the component assumptions were selected to be consistent with performance data from commercially available units and intermediate water loops were considered to couple the stages in a modular way. The thermodynamic cycle of each unit under nominal design conditions was reconstructed based on the main technical specifications and assumptions reported in Table 2. The model was checked at nominal conditions by comparing calculated COP values with manufacturer-reported data for the selected units (see Table 2, last row). This comparison provides a component-level consistency assessment under nominal operating conditions and does not constitute experimental validation of the complete cascaded configuration.

Table 2. Main technical specifications and modeling assumptions of the LTHP, MTHP and MVR units under nominal design conditions.

	LTHP	MTHP	MVR
Type	Closed-loop	Closed-loop	Open-loop
Working fluid	R134a	R1234ze(E)	Water
Compressor	Screw (inverter)	Screw (on-off)	Centrifugal
Isentropic efficiency	0.70 [19]	0.70 [19]	0.85 [20]
Electromechanical efficiency	0.95	0.95	0.95
Heat source	Air (7 °C) Min. approach temp: 8 °C	Water (45/40 °C) Min. approach temp: 7 °C	Water (85/75 °C) Min. approach temp: 5 °C
Heat sink	Water (45/40 °C) Min. approach temp: 5 °C	Water (85/75 °C) Min. approach temp: 7 °C	Saturated Steam at 12.5 bar (return: saturated liquid at 1 bar)
IHX Effectiveness	0.70 [21]	0.70 [19]	-
COP calculated/real ($\Delta\%$)	3.34/3.18 (+5.0%)	3.46/3.44 (+0.6%)	2.63/2.80 (−6.0%)

A schematic of the SGHP components is presented in Figure 2, together with the corresponding thermodynamic cycles under design conditions in the T–s diagram. The thermodynamic state properties at key points are reported in Table 3. All data refer to a steam pressure of 15 bar, corresponding to the MVR intermediate operating pressure.

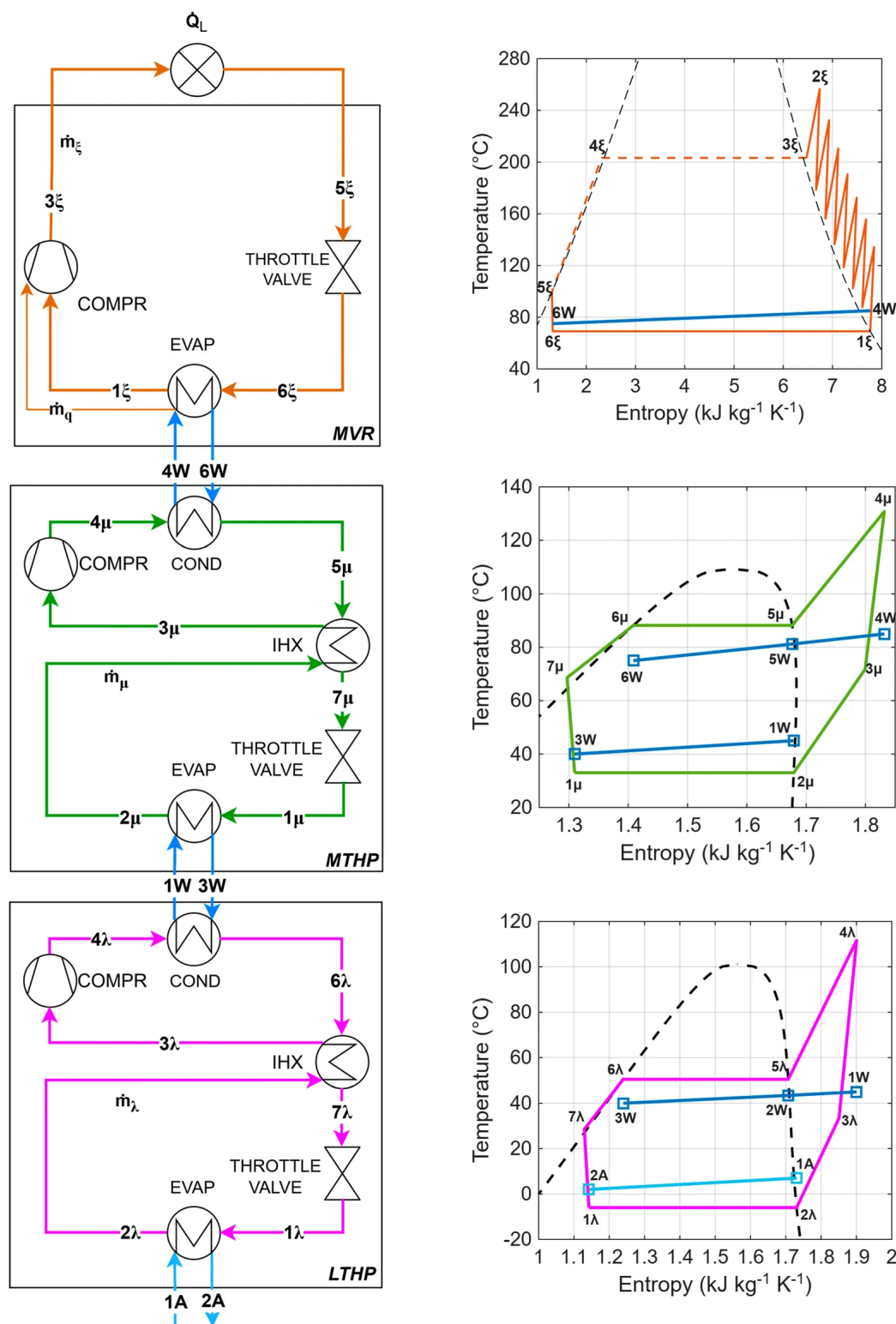


Figure 2. Schematic layout of the SGHP and LTHP, MTHP and MVR thermodynamic cycles under nominal design conditions (T-s representation).

Table 3. Thermodynamic state properties at key points of the SGHP system under nominal design conditions, for saturated steam production at 15 bar.

Point	Fluid	Pressure (p) (bar)	Temperature (T) (°C)	Specific Enthalpy (h) (kJ/kg)	Specific Entropy (s) (kJ/Kg K)
1A	Air	1.01	7	-	-
2A	Air	1.01	2	-	-
1λ	R134A	2.3	-6	238	1.1
2λ	R134A	2.3	-6	395	1.7
3λ	R134A	2.3	34	430	1.9
4λ	R134A	13.3	112	492	1.9
5λ	R134A	13.3	50	424	1.7
6λ	R134A	13.3	50	272	1.2
7λ	R134A	13.3	29	238	1.1
1W	Water	2.0	45	-	-
2W	Water	2.0	44	-	-
3W	Water	2.0	40	-	-
1μ	R1234ze(E)	6.3	33	293	1.3
2μ	R1234ze(E)	6.3	33	406	1.7
3μ	R1234ze(E)	6.3	72	445	1.8
4μ	R1234ze(E)	23.9	131	488	1.8
5μ	R1234ze(E)	23.9	88	429	1.7
6μ	R1234ze(E)	23.9	88	332	1.4
7μ	R1234ze(E)	23.9	69	293	1.3
4W	Water	2.0	85	-	-
5W	Water	2.0	83	-	-
6W	Water	2.0	75	-	-
1ξ	Water	0.3	69	2625	7.8
2ξ	Water	15	257	2940	6.7
3ξ	Water	15	203	2805	6.5
4ξ	Water	15	203	845	2.3
5ξ	Water	1	100	418	1.3
6ξ	Water	0.3	69	418	1.3

The LTHP, MTHP and MVR cycles were modeled under design (steady-state) conditions, with thermophysical properties evaluated using CoolProp vers. 6.1.0 [22].

The simulation models for the LTHP and the MTHP were driven by the imposed boundary conditions, namely the inlet temperature at the evaporator, the outlet temperature and mass flow rate at the condenser, the fixed minimum approach temperatures at each heat exchanger, and the IHX effectiveness. The energy balances of their heat exchangers (evaporator, condenser and internal) are reported in the following equations.

$$\dot{Q}_{EVAP,LTHP} = \dot{m}_{A,EVAP,LTHP} \cdot c_{p,A} \cdot (T_{1A} - T_{2A}) = \dot{m}_{\lambda} \cdot (h_{2\lambda} - h_{1\lambda}) \quad (6)$$

$$\dot{Q}_{COND, LTHP} = \dot{m}_{W, COND, LTHP} \cdot c_{p, W} \cdot (T_{1W} - T_{3W}) = \dot{m}_{\lambda} \cdot (h_{4\lambda} - h_{6\lambda}) \quad (7)$$

$$\dot{Q}_{IHX, LTHP} = \dot{m}_{\lambda} \cdot (h_{6\lambda} - h_{7\lambda}) = \dot{m}_{\lambda} \cdot (h_{3\lambda} - h_{2\lambda}) \quad (8)$$

$$\dot{Q}_{EVAP, MTHP} = \dot{m}_{W, EVAP, MTHP} \cdot c_{p, W} \cdot (T_{1W} - T_{3W}) = \dot{m}_{\mu} \cdot (h_{2\mu} - h_{1\mu}) \quad (9)$$

$$\dot{Q}_{COND, MTHP} = \dot{m}_{W, COND, MTHP} \cdot c_{p, W} \cdot (T_{4W} - T_{6W}) = \dot{m}_{\mu} \cdot (h_{4\mu} - h_{6\mu}) \quad (10)$$

$$\dot{Q}_{IHX, MTHP} = \dot{m}_{\mu} \cdot (h_{6\mu} - h_{7\mu}) = \dot{m}_{\mu} \cdot (h_{3\mu} - h_{2\mu}) \quad (11)$$

The annual case study adopts, as its reference operating condition, the continuous baseload steam demand of a real paper-production facility located in Cagliari, Italy: 2 t/h of saturated steam at 15 bar, with condensate returned as saturated liquid at 1 bar. It should be noticed that currently the reference facility meets this steam demand using a fuel-oil-fired boiler.

Under this profile, the MTHP and MVR were assumed to operate continuously at their nominal conditions. This assumption is consistent with the constant steam demand and the thermal decoupling provided by the intermediate water loops. Conversely, the LTHP was modeled under off-design conditions because it is directly coupled to the ambient air-source and its performance varies with outdoor temperature.

The off-design performance of the LTHP was simulated using the following COP–temperature lift correlation provided by the IEA [1], scaled with respect to the nominal COP of the selected unit, also represented in Figure 3:

$$\frac{COP(\Delta T_{LIFT}, t)}{COP_{LTHP, N}} = -0.962 \cdot \ln(\Delta T_{LIFT}(t)) + 4.5027; \quad \Delta T_{LIFT} = T_{1W} - T_{1A} \quad (12)$$

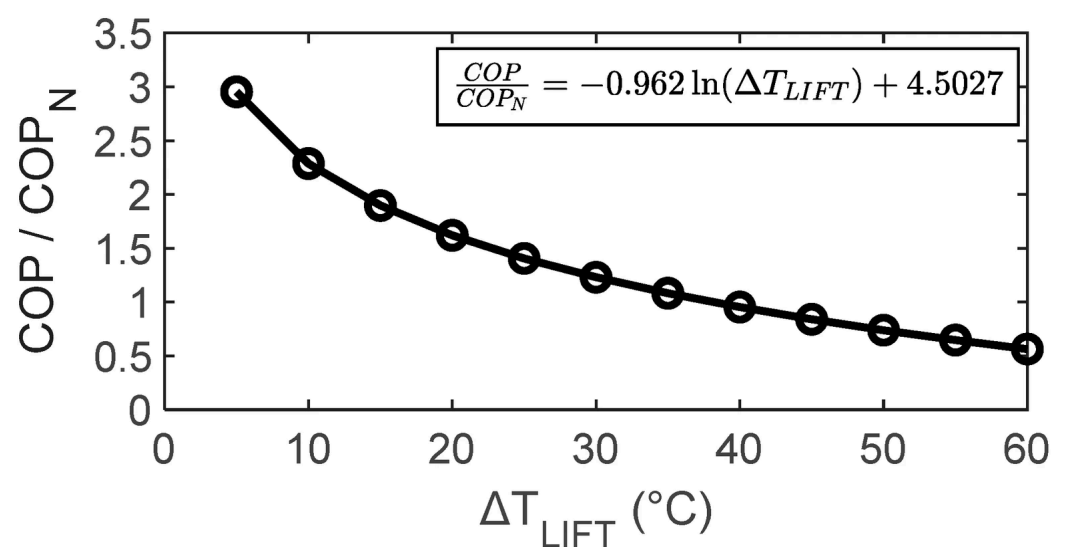


Figure 3. Normalized coefficient of performance of the low-temperature heat pump (LTHP) as a function of temperature lift, used for off-design performance modeling.

In addition, the increased LTHP electricity consumption associated with defrost cy-

cles was represented through a linear reduction in its COP, reaching a maximum of 20% [23] at its lower operating temperature ($-10\text{ }^{\circ}\text{C}$).

Regarding the MVR, the flow at the compressor inlet consists of evaporated vapor and flash vapor generated through the isenthalpic expansion of the condensate stream. The compression of the steam from the minimum pressure level ($p_{1\xi}$) to the required steam pressure ($p_{3\xi}$) was discretized into Z stages based on a maximum allowable pressure ratio per stage ($\beta_{\max}$) of 1.8.

$$z = \frac{\ln\left(\frac{p_{3\xi}}{p_{1\xi}}\right)}{\ln(\beta_{\max})}; \beta_z = \left(\frac{p_{3\xi}}{p_{1\xi}}\right)^{\frac{1}{z}} \quad (13)$$

After each compression stage, modeled using an isentropic efficiency approach, the steam is desuperheated by direct-contact interstage spray injection (quench) to a prescribed residual superheat margin above saturation, fixed at $5\text{ }^{\circ}\text{C}$. The quench flow rate is withdrawn from the saturated liquid phase accumulated at the bottom of the evaporator, in accordance with standard desuperheater practice [24–26]. The required quench

flow rate in each interstage desuperheater ($\dot{m}_{q,i}$) was determined from a stage-wise energy balance:

$$\dot{m}_{q,i} \cdot (h_i^* - h_{6\xi}) = \dot{m}_{\text{COMPR},i} \cdot (h_i - h_i^*); \quad i = 1, \dots, Z \quad (14)$$

where h_i denotes the actual vapor enthalpy at the compressor discharge of stage i (before desuperheating), h_i^* corresponds to the target state at $T_{\text{sat}}(p_i) + 5\text{ }^{\circ}\text{C}$, and $h_{6\xi}$ is the specific enthalpy of the saturated liquid extracted from the evaporator sump. Owing to cumulative interstage quench injection, the vapor mass flow rate increases progressively along the compressor train.

The electricity consumption of auxiliary equipment, including the air-side fan, circulation pumps, and control equipment, was estimated as 3% of the combined electricity consumption of the LTHP, MTHP, and MVR compressors. The resulting auxiliary consumption was included in the total SGHP electricity demand.

According to NREL formulation [7], the overall SGHP COP over an operational period τ was calculated as

$$\text{COP}_{\text{SGHP}}(\tau) = \frac{\sum_{t=1}^{\tau} Q_L(t)}{\sum_{t=1}^{\tau} E_{\text{SGHP}}(t)} \quad (15)$$

2.3. Economic Assessment

A preliminary economic assessment was performed over a 20-year period based on the Net Present Value (NPV) of the discounted cash flows and the discounted payback time (PBT). Electricity-related operating costs were evaluated for electricity prices of 50, 100, and 150 €/MWh and were assumed to increase by 1% annually to account jointly for electricity price escalation and the progressive increase in electricity consumption associated with system degradation. As conservative assumptions, the BESS lifetime was set to 10 years, requiring its complete replacement within the assessment period, while no residual value was assigned to the system components at the end of the analysis period.

Moreover, no economic remuneration was considered for electricity exported to the grid. The main economic assumptions are summarized in Table 4.

Table 4. Main assumptions adopted for the preliminary economic assessment.

Parameter	Value	Source
System Lifetime	20 years	
Discount rate	5%	[27]
SGHP CAPEX	3500 €/kW	[28]
SGHP OPEX	5% of CAPEX	[7,29]
PV CAPEX	600 €/kW	[27]
PV OPEX	2% of CAPEX	[27]
BESS CAPEX	150 €/kWh	[27]
BESS OPEX	2.5% of CAPEX	[27]

3. Results and Discussion

This section presents the thermodynamic, annual, and preliminary economic performance of the proposed system. Section 3.1 examines the performance of the standalone SGHP under nominal design conditions as a function of steam delivery pressure. Section 3.2 evaluates its full-year operation for the selected industrial case study within a grid-supported PV-BESS framework. Section 3.3 presents the preliminary economic assessment, and Section 3.4 discusses the main modeling assumptions and limitations.

3.1. SGHP Performance Under Nominal Design Conditions

This section presents the performance of the SGHP under nominal design conditions. These conditions comprise ambient air at 7 °C as the heat source, saturated steam delivery at pressures ranging from 5 to 25 bar (MVR operating range), and condensate return as saturated liquid at 1 bar.

Figure 4 shows a stacked bar chart of the specific power consumption of the SGHP (expressed in kW per t/h of produced steam) and the nominal COP of the SGHP as a function of the steam pressure at the load side. .

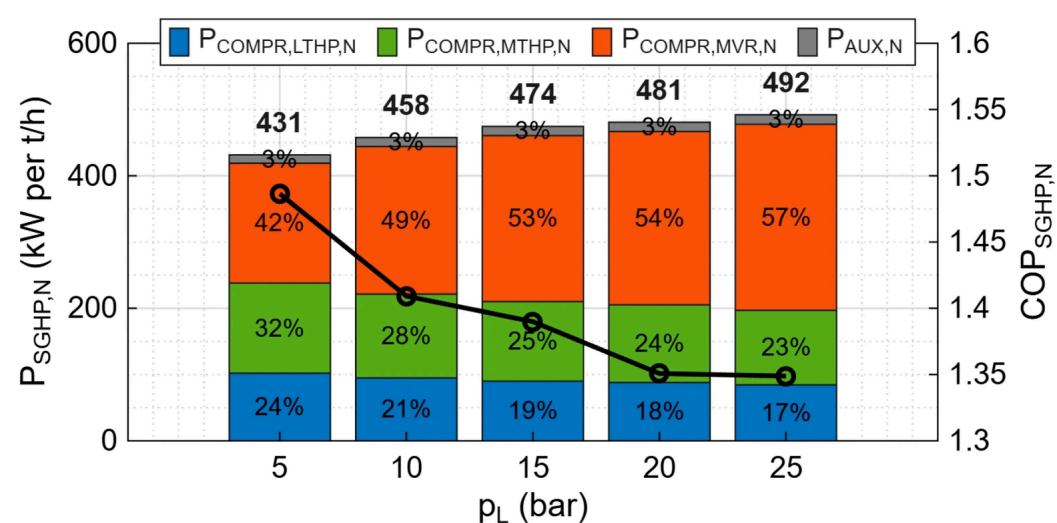


Figure 4. Specific electrical power consumption of the SGHP, decomposed into LTHP, MTHP, and MVR compressor contributions and aggregate auxiliary consumption, together with the resulting SGHP COP (black solid line) as a function of steam delivery pressure under nominal design conditions.

As the required steam pressure increases, the total specific power consumption of the SGHP rises, from about 431 kW per t/h at 5 bar to approximately 492 kW per t/h at 25 bar. This increase is mainly driven by the growing consumption of the MVR unit, whose absolute value and relative share increase steadily, from 42% at 5 bar up to 57% at 25 bar, reflecting the higher compression work needed to reach higher saturation pressures. In contrast, the consumption of both the LTHP and MTHP decreases with increasing steam pressure, both in absolute terms and as a percentage of the total. The LTHP share drops from 24% at 5 bar to 17% at 25 bar, while the MTHP contribution decreases from 32% to 23% over the same pressure range. This trend can be explained by the internal thermodynamic integration of the SGHP: when the required steam pressure increases, additional compression stages are needed, leading to a higher demand for interstage desuperheating flow. Since the latter is extracted as liquid from the evaporator bottom, the evaporator duty decreases. Therefore, the power required by the upstream heat pump stages (LTHP and MTHP) decreases as well.

For the same set of pressures, the $COP_{SGHP,N}$ exhibits a moderate but systematic decrease as the steam pressure increases (1.49 at 5 bar and 1.35 at 25 bar). This trend is due to the increasing share of $P_{MVR,N}$ on $P_{SGHP,N}$, given its lower COP compared to the LTHP and the MTHP. However, this shift toward MVR dominance reduces the sensitivity of the calculated COP to ambient-air temperature variations, because the MVR stage is not directly coupled to the outdoor air source.

3.2. SGHP Performance Under Full-Year Operating Conditions with PV-BESS Supply

This section evaluates the annual operation of the SGHP for the selected industrial case study within a grid-supported PV-BESS electricity supply framework. The case study considers the continuous baseload demand of a real paper-production facility located in Cagliari, Italy, corresponding to 2 t/h of saturated steam at 15 bar, with condensate returned as saturated liquid at 1 bar. The selected location provides Mediterranean climatic conditions with high solar availability. The PV system was modeled with south-facing modules with a tilt angle of 37°, resulting in an annual specific electricity yield of 1,671 kWh/kW_{PV,N}.

Figure 5 summarizes the annual performance in terms of self-sufficiency (SS) and self-consumption (SC) for varying PV and BESS sizes. $P_{PV,N}$ and $E_{B,N}$ are expressed in specific terms relative to the nominal electrical power of the SGHP ($P_{SGHP,N}$). As it can be observed, SS (Figure 5a) increases with the installed PV capacity for all battery sizes, rising from about 0.20 for $P_{PV,N}/P_{SGHP,N} = 1$ to values exceeding 0.9 for $P_{PV,N}/P_{SGHP,N} = 8$ and 16 h of BESS capacity related to load. The presence of the BESS increases SS markedly, by allowing excess daytime PV generation to be shifted toward nighttime operation of the SGHP. For the highest considered PV capacities ($P_{PV,N}/P_{SGHP,N} \geq 5$), each additional hour of BESS capacity increases SS by approximately 2.5 percentage points. Figure 5b reports the SC ratio. As PV capacity increases, SC decreases due to the growing mismatch between PV production and load. Battery storage significantly mitigates this reduction, increasing SC across the entire range of PV sizes.

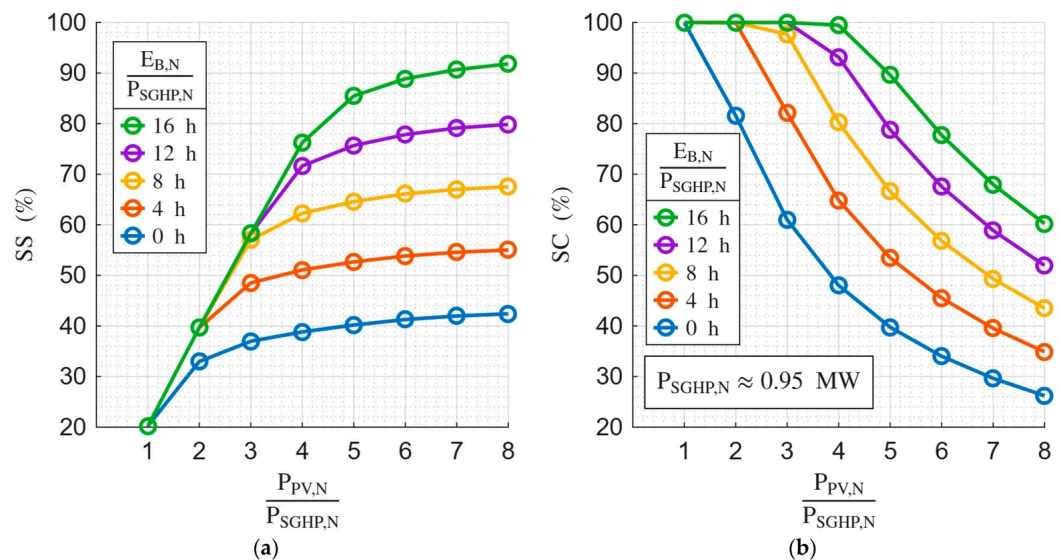


Figure 5. Annual self-sufficiency (a) and self-consumption (b) as a function of normalized PV capacity and battery size for a continuous steam load of 2 t/h at 15 bar and a PV located in Cagliari, Italy.

Figure 6 shows the monthly distribution of the SGHP electricity consumption among the three units (a) and the corresponding monthly SGHP COP (b). The dependence of LTHP electricity consumption on outdoor temperature is clearly visible. Indeed, its monthly electricity consumption is almost 40% lower in July than in January (72 MWh and 116 MWh, respectively). This variation is reflected in the monthly COP_{SGHP} , which increases from winter values close to 1.42 to summer values up to 1.53. Owing to the Mediterranean climate of the selected location, the monthly COP_{SGHP} remains consistently higher than its nominal design value ($COP_{SGHP,N}$), calculated for an ambient temperature of 7 °C.

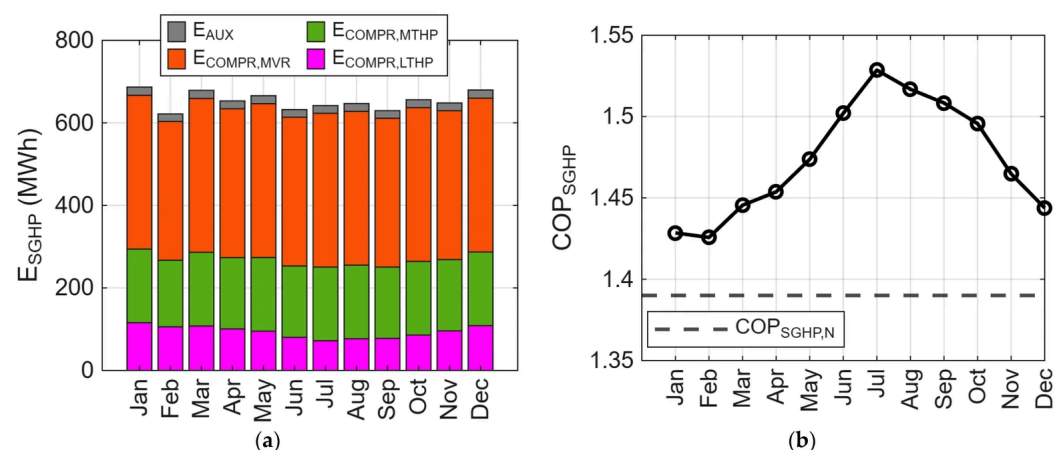


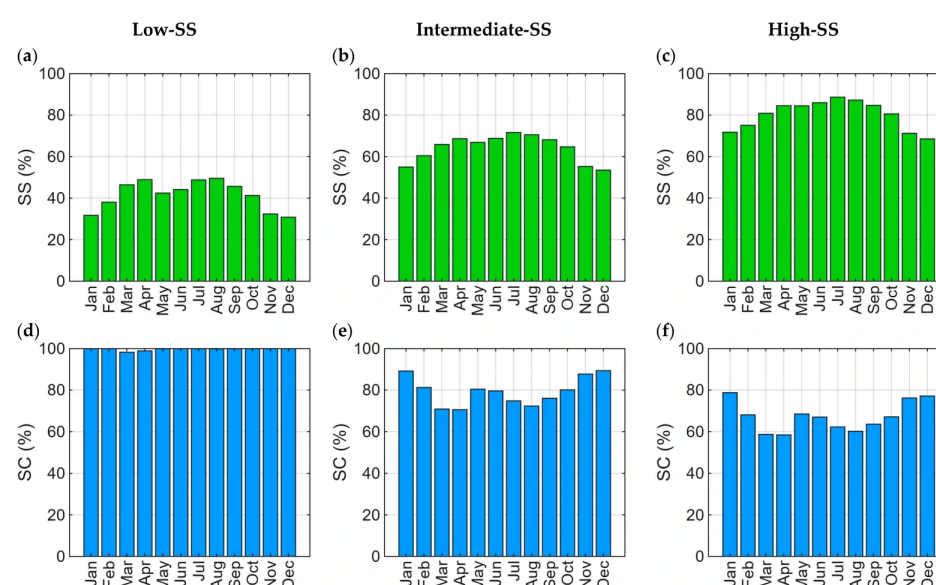
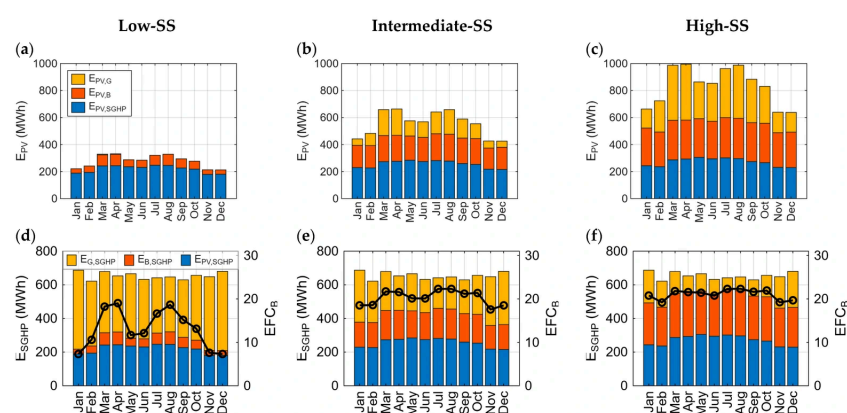
Figure 6. Monthly distribution of SGHP electrical consumption among the LTHP, MTHP and MVR units (a) and resulting monthly SGHP COP (b). Results refer to a continuous steam demand of 2 t/h at 15 bar and a system located in Cagliari, Italy.

In the following, system performance is presented through monthly results and average daily profiles for January and July for the three selected PV–BESS configurations reported in Table 5. Based on the resulting annual self-sufficiency, the configurations are referred to as the low-SS ($SS \approx 40\%$), intermediate-SS ($SS \approx 60\%$), and high-SS ($SS \approx 80\%$) configurations.

Table 5. PV–BESS sizing parameters of the three configurations selected for the monthly analysis and the comparison of average daily profiles for January and July.

	Low-SS	Intermediate-SS	High-SS
$\frac{P_{PV,N}}{P_{SGHP,N}} (-)$	2	4	6
$\frac{E_{B,N}}{P_{SGHP,N}} (h)$	4	8	12

Figures 7 and 8 provide a multi-layered view of monthly system operation for the three selected configurations. Figure 7 compares PV self-consumption and system self-sufficiency, whereas Figure 8 reports the allocation of PV electricity, the electricity supply of the SGHP, and the corresponding BESS usage.

**Figure 7.** Monthly self-sufficiency (top row) and PV self-consumption (bottom row) for the low-SS configuration (a,d), intermediate-SS configuration (b,e), and high-SS configuration (c,f). Results refer to a continuous steam demand of 2 t/h at 15 bar and a PV system located in Cagliari, Italy.**Figure 8.** Monthly allocation of PV electricity among direct SGHP supply ($E_{PV,SGHP}$), BESS charging ($E_{PV,B}$) and grid export ($E_{PV,G}$) (a,b,c) and monthly SGHP electricity supply from direct PV generation ($E_{PV,SGHP}$), battery discharge ($E_{B,SGHP}$), and grid electricity ($E_{G,SGHP}$), with battery usage expressed as equivalent full cycles (EFC_B , black solid line) (d,e,f) for the three PV–BESS sizing configurations: low-SS configuration (left), intermediate-SS configuration (middle), and high-SS con-

figuration (right). Results refer to a continuous steam demand of 2 t/h at 15 bar and a PV system located in Cagliari, Italy.

Figure 7 compares the monthly self-sufficiency (top row) and self-consumption (bottom row) of the three configurations. Even for the low-SS configuration, monthly SS never falls below approximately 30%. The minimum monthly SS increases to approximately 55% for the intermediate-SS configuration and 70% for the high-SS configuration. The maximum monthly values increase correspondingly from approximately 50% for the low-SS configuration to 72% and 89% for the intermediate- and high-SS configurations, respectively. All configurations show higher SS during summer because of the greater PV availability and the lower SGHP electricity demand resulting from the higher ambient temperatures.

The higher self-sufficiency achieved by the intermediate- and high-SS configurations results from the simultaneous increase in PV and BESS capacities. However, this improvement is accompanied by lower PV self-consumption, as shown in the bottom panels of Figure 7. Annual SC decreases from 99.7% for the low-SS configuration to 78.4% and 66.1% for the intermediate- and high-SS configurations, respectively. At the monthly level, SC remains close to 100% for the low-SS configuration, whereas it ranges from 70% to 90% for the intermediate-SS configuration and from 58% to 79% for the high-SS configuration. This aspect may be relevant in contexts where the system is connected to grids with high PV penetration and limited curtailment-mitigation capability, as surplus PV export may be constrained.

In this respect, the top panels of Figure 8 show that the amount of PV electricity exported to the grid, $E_{PV,G}$, and therefore potentially exposed to curtailment in constrained grids, increases for configurations with higher self-sufficiency and during months with greater PV generation. At the annual level, PV electricity exported to the grid increases from only 10 MWh for the low-SS configuration to 1.4 GWh and 3.4 GWh for the intermediate- and high-SS configurations, respectively.

The lower panels of Figure 8 show that the larger PV and BESS capacities progressively replace grid electricity by increasing direct PV supply and BESS usage. Consequently, annual grid electricity consumption decreases from 4.6 GWh for the low-SS configuration to 2.8 GWh for the intermediate-SS configuration and 1.6 GWh for the high-SS configuration.

For the intermediate-SS configuration, BESS usage approaches 22 EFCs per month in summer and remains approximately 18 EFCs per month in winter. Battery use shows substantially greater monthly variability for the low-SS configuration, decreasing to approximately 7 EFCs in winter and approaching 19 EFCs during spring and summer. This variability becomes progressively smaller as PV and BESS capacities increase. In the high-SS configuration, monthly battery usage remains between approximately 19 and 22 EFCs throughout the year. At the annual level, battery usage increases from 157 EFCs for the low-SS configuration to 244 EFCs for the intermediate-SS configuration, but rises only marginally to 253 EFCs for the high-SS configuration.

Overall, the comparison highlights the trade-off between PV utilization and renewable coverage: the low-SS configuration exhibits greater grid dependence, whereas the high-SS configuration provides greater renewable coverage but generates more PV surplus.

Average daily energy balances for winter (January) and summer (July) are reported in Figure 9. PV generation values are shown as positive, while consumption values are shown as negative and represented using the same colors in darker shades. From the comparative analysis of the top and bottom panels, it can be observed that in winter the PV production curves are lower (lower power output) and narrower (fewer production

hours) than in summer. This leads to several considerations.

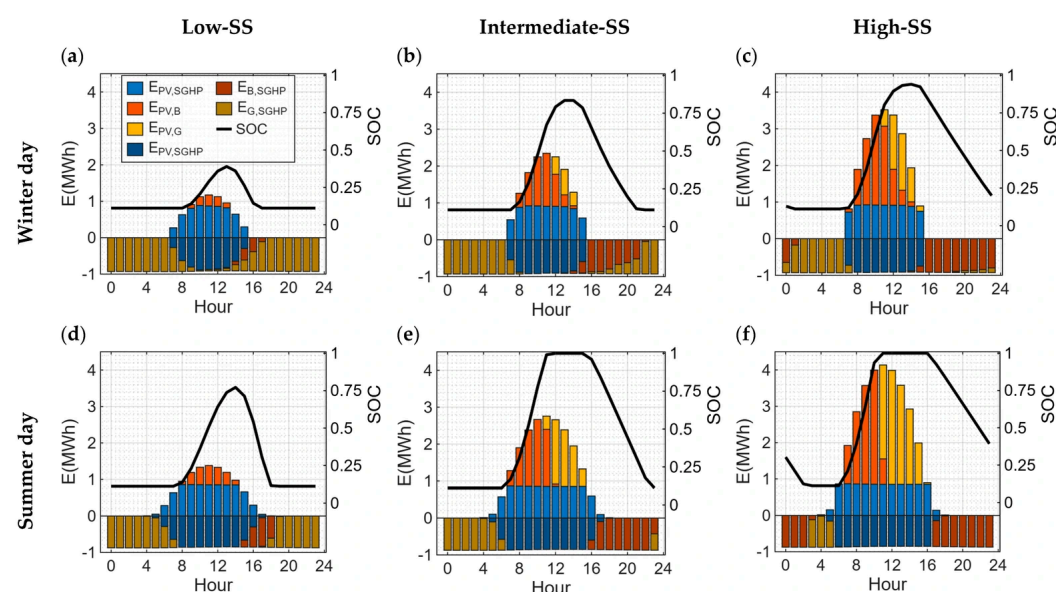


Figure 9. Average daily energy balances for January (a,b,c) and July (d,e,f), representing winter and summer operating conditions, respectively. The panels show PV electricity allocation, SGHP electricity supply, and battery state of charge for the low-SS configuration (left), intermediate-SS configuration (middle), and high-SS configuration (right). Results refer to a continuous steam demand of 2 t/h at 15 bar and a PV system located in Cagliari, Italy.

For all three configurations, the period during which PV generation directly and almost completely covers the SGHP electricity demand is approximately 2–3 h shorter in January than in July. The different PV and BESS sizes progressively extend both the period of direct PV supply and the duration of electricity supply from the BESS. In January, the lower PV availability also results in a lower daily maximum SOC and an earlier depletion of the battery than in July.

The larger batteries employed in the intermediate- and high-SS configurations extend SGHP supply after sunset. However, only the high-SS configuration provides grid-independent operation during at least part of the early morning period in both representative months. By contrast, the low- and intermediate-SS configurations still require grid electricity during the early morning, after the stored PV electricity has been depleted.

3.3. Preliminary Economic Assessment

The results of the preliminary economic assessment are summarized in Figure 10. The economic performance of the three selected PV–BESS configurations is evaluated relative to a grid-only configuration, in which the SGHP electricity demand is entirely supplied by the grid.

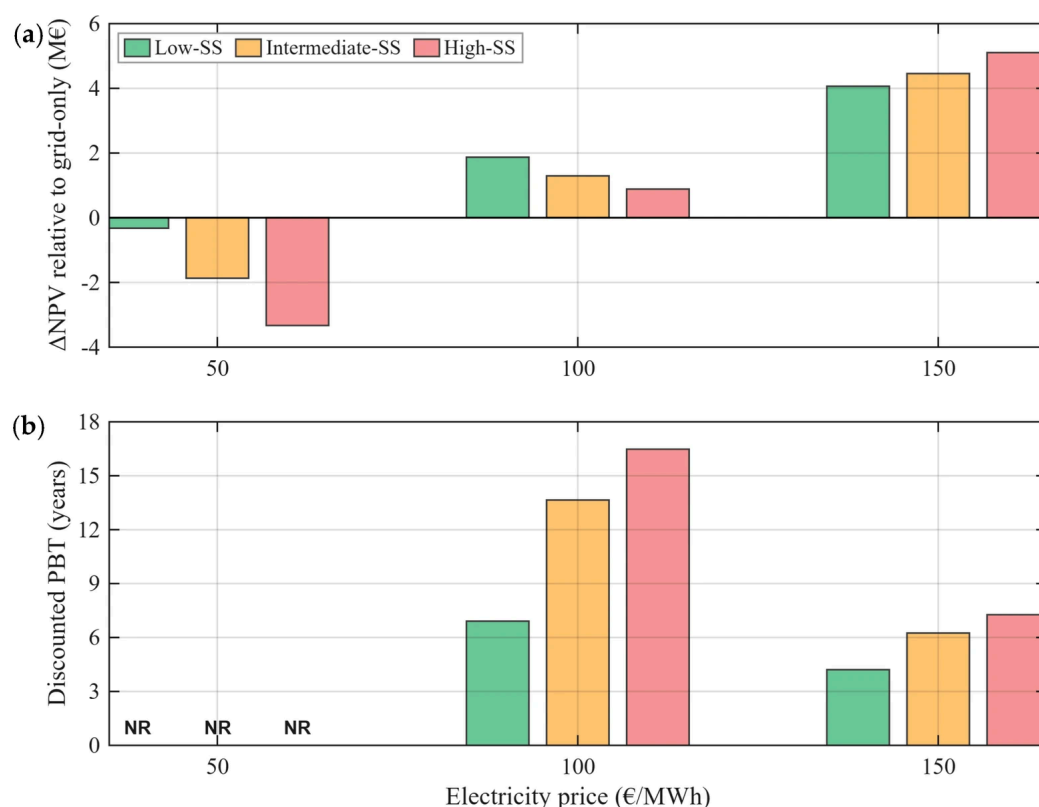


Figure 10. Difference in NPV relative to the grid-only case (a) and discounted payback time (b) for the selected PV-BESS configurations at different electricity purchase prices. Positive ΔNPV values indicate an economic advantage over the grid-only case, while NR indicates that the discounted payback time was not reached within 20 years.

At an electricity price of 50 €/MWh, the grid-only case exhibits the lowest discounted lifetime cost, around 11 M€. None of the PV-BESS configurations reaches the discounted payback time within the 20-year assessment period. Among them, the low-SS configuration shows the best economic performance, with an additional discounted cost of 0.32 M€ relative to the grid-only case. The corresponding additional costs increase to 1.87 M€ and 3.33 M€ for the intermediate- and high-SS configurations, respectively.

At an electricity price of 100 €/MWh, all PV-BESS configurations become economically preferable to the grid-only case. The low-SS configuration provides the greatest NPV improvement, equal to 1.87 M€, followed by the intermediate- and high-SS configurations, with improvements of 1.29 M€ and 0.89 M€, respectively. The corresponding discounted PBTs are approximately 6.9, 13.7, and 16.5 years.

At 150 €/MWh, the greater value of the avoided grid electricity expenditure substantially improves the economic performance of all PV-BESS configurations. In this scenario, the high-SS configuration achieves the highest NPV improvement relative to the grid-only case, equal to 5.10 M€, followed by the intermediate- and low-SS configurations, with improvements of 4.46 M€ and 4.06 M€, respectively. Their discounted PBTs decrease to approximately 7.3, 6.3, and 4.2 years, respectively. Therefore, unlike the results obtained at the lower electricity prices, the high-SS configuration exhibits the best economic performance over the complete assessment period at 150 €/MWh, although the low-SS configuration retains the shortest PBT.

Overall, the economically preferred configuration strongly depends on the electricity price. At 50 €/MWh, the additional investment required by the PV-BESSs is not recovered. At 100 €/MWh, the low-SS configuration represents the most favorable solution,

as its lower investment cost outweighs the additional electricity savings achieved by the larger configurations. At 150 €/MWh, the greater reduction in grid electricity purchases progressively compensates for the higher PV and BESS investment costs, making the high-SS configuration the most advantageous in terms of NPV. Nevertheless, the low-SS configuration consistently shows the shortest payback time because of its substantially lower initial investment. It should also be noted that no remuneration is assigned to the electricity exported to the grid. Assigning a positive value to the exported electricity would further improve the economic performance of the configurations characterized by lower self-consumption, particularly the intermediate- and high-SS configurations, which export larger amounts of PV electricity to the grid.

3.4. Model Limitations

The interpretation of the results is bounded by the following modeling assumptions. First, the proposed SGHP system was modeled using steady-state assumptions at the component level. While this approach is commonly adopted in assessments of industrial heat pump systems supplying constant thermal loads, it neglects transient effects associated with start-up, shut-down, and short-term load fluctuations. Second, the MTHP and MVR units were assumed to operate continuously at nominal design conditions. This assumption is supported by the relatively stable thermodynamic conditions of the intermediate loops feeding these components and by the continuous steam demand considered. Nevertheless, partial-load operation, off-design performance degradation, and control-related efficiency penalties were not explicitly modelled.

Third, defrosting cycles of the air-source LTHP were simplified. While literature reports seasonal COP penalties of 1.5–13% depending on climate severity [30], the Mediterranean conditions of the case study may lead to an impact closer to the lower part of this range. Fourth, the BESS model accounted for efficiency and cycling but did not explicitly consider aging or capacity degradation. In the economic assessment, this limitation was addressed through the simplified assumption of a 10-year BESS lifetime and its complete replacement at the end of this period. Fifth, the economic assessment should also be regarded as preliminary.

Finally, the complete SGHP configuration has not been experimentally validated.

Accordingly, the energy-performance results should be interpreted as upper-bound estimates rather than as guaranteed operating performance for a commercial installation, whereas the economic results provide a preliminary comparison under the assumed cost and electricity price scenarios.

4. Conclusions

This study numerically assessed the thermodynamic and annual potential performance of a cascaded steam-generating heat pump for producing high-temperature industrial steam using ambient air as the external low-temperature heat source, without relying on recoverable industrial waste heat. The results refer to steady-state component models and should therefore be interpreted within the assumptions and idealizations adopted in the analysis. The investigated configuration combines low- and medium-temperature heat pump stages with mechanical vapor recompression to address the large temperature lift required for steam generation. The annual assessment was applied to the continuous steam demand of a real paper-production facility under Mediterranean climatic conditions. A grid-supported PV-BESS was included as the electricity supply framework for characterizing system operation under time-varying ambient and renewable-generation conditions. A preliminary economic assessment was also conducted for three PV-BESS configurations.

Under the adopted modeling assumptions, the SGHP model indicates the technical possibility of producing saturated steam at pressures ranging from 5 to 25 bar, corresponding to approximately 150–223 °C, with calculated design COP values ranging from 1.35 to 1.49. These values represent upper-bound estimates and do not account for transient operation, part-load penalties, detailed frost dynamics, control interactions, or long-term component degradation. As steam delivery pressure increases, electricity consumption progressively shifts from the upstream heat pump stages to the mechanical vapor recompression unit. This shift results in a moderate reduction in the overall COP while decreasing the sensitivity of system performance to ambient-air temperature, owing to the growing contribution of the downstream stage.

Annual simulations for the selected Mediterranean case study indicated that when coupled with a PV-BESS, the SGHP can achieve high levels of self-consumption and meaningful self-sufficiency, while benefiting from improved summer performance due to higher ambient temperatures and solar availability. For the investigated case study, the operational COP exceeds that calculated at nominal design conditions, underscoring the relevance of climate and operating context in performance assessments.

The preliminary economic assessment indicates that the preferred PV-BESS configuration strongly depends on the electricity purchase price. At 50 €/MWh, none of the investigated configurations recovers its additional investment within the 20-year assessment period. At 100 €/MWh, the discounted payback periods range from approximately 6.9 to 16.5 years, whereas at 150 €/MWh they decrease to approximately 4.2–7.3 years. The low-self-sufficiency configuration consistently provides the shortest payback period, while the high-self-sufficiency configuration provides the greatest NPV improvement at the highest electricity price scenario.

These findings are consistent with, and extend, the existing literature on steam-generating heat pumps. While previous studies have largely focused on configurations relying on waste heat or limited to sub-170 °C applications, the present work indicates that MVR-dominated architectures could potentially be coupled with upstream heat pump stages to enable ambient-air-based steam generation at significantly higher temperatures. Experimental validation and more detailed off-design modeling are required before the predicted performance can be transferred to an industrial-scale implementation.

Although the annual assessment refers to a paper-production facility, the configuration addresses a broader class of industrial applications characterized by continuous high-temperature steam demand and limited availability of recoverable waste heat.

From a system-integration perspective, the considered configuration provides a potential pathway for the electrification of continuous industrial steam demand in applications where fossil-fueled boilers remain prevalent and recoverable waste heat is scarce or unavailable. The cascaded architecture, based on component parameters consistent with representative commercial equipment and thermally decoupled through intermediate water loops, also provides a modular arrangement for industrial integration.

The interpretation of the results is bounded by the steady-state component models, the nominal operation assumed for the MTHP and MVR units, and the exclusion of transient behavior, part-load penalties, and long-term component degradation. Accordingly, the reported performance indicators should be interpreted as upper-bound estimates under the adopted modeling assumptions.

Overall, the study provides a model-based characterization of the thermodynamic operating range, annual performance and a preliminary assessment of the economic potential of an ambient air-based system for high-temperature industrial steam generation, highlighting the effects of steam pressure, ambient conditions, PV-BESS sizing, and electricity prices on system performance.

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Abbreviations

The following abbreviations are used in this manuscript:

Symbols	
A	Air
c_p	Specific heat at constant pressure ($\text{J Kg}^{-1} \text{K}^{-1}$)
E	Electrical energy (kWh)
E_B	BESS Capacity (kWh)
$E_{B,SGHP}$	BESS electricity used to meet the SGHP load (kWh)
$E_{G,SGHP}$	Grid electricity used to meet the SGHP load (kWh)
E_{LTHP}	Electrical demand of the LTHP (kWh)
E_{MTHP}	Electrical demand of the MTHP (kWh)
E_{MVR}	Electrical demand of the MVR unit (kWh)
E_{SGHP}	Electrical demand of the SGHP (kWh)
$E_{PV,SGHP}$	PV electricity used to meet the SGHP load (kWh)
$E_{PV,B}$	PV electricity used to charge the BESS (kWh)
$E_{PV,G}$	PV electricity feed-ins (kWh)
G	Grid
h	Specific enthalpy (kJ/kg)
L	Load
$\dot{m}$	Mass flow (kg/s)
N	Nominal design conditions
p	Pressure (bar)
P	Electrical power (kW)
q	Quench
Q	Thermal energy (kWh)

$\dot{Q}$	Thermal power (kW)
s	Specific entropy (kJ Kg ⁻¹ K ⁻¹)
t	Time (hours)
T	Temperature (°C)
W	Water
β	Compression ratio
η_{RT}	BESS round-trip efficiency
λ	Related to the LTHP working fluid (R134a)
μ	Related to the MTHP working fluid (R1234ze(E))
ξ	Related to the MVR working fluid (steam)
τ	Period of time
Acronyms	
BESS	Battery Energy Storage System
COMPR	Compressor
COND	Condenser
COP	Coefficient of Performance
EFC	Equivalent Full Cycles
EVAP	Evaporator
HP	Heat Pump
IHX	Internal Heat Exchanger
LTHP	Low-Temperature Heat Pump
MTHP	Medium-Temperature Heat Pump
NPV	Net Present Value
PBT	Payback Time
SC	Self-Consumption
SGHP	Steam-Generating Heat Pump
SOC	State of Charge
SS	Self-Sufficiency
MVR	Mechanical Vapor Recompression
PV	Photovoltaic

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