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High-Temperature Heat Pumps for Combined Heating and Cooling Processes: A study for milk Pasteurization decarbonization

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Abstract: High-temperature heat pumps for combined heating and cooling represent a concrete solution for the decarbonization of industrial processes. This thesis evaluates their integration into milk pasteurization plants, which simultaneously require high temperature heating (from 4 °C to above 72 °C) and subsequent cooling (at 4 °C). The proposed layout integrates a thermal energy storage system to recover excess heat from the condenser and exploit it for maintenance applications. A parametric analysis is conducted on three plant configurations, considering 20 low environmental impact refrigerants and three pasteurization temperatures, in order to identify the optimal solutions. For processes at 72 °C and 85 °C, the best performing configuration is a two-stage cycle using R1234ze(E) as refrigerant, achieving combined coefficients of performance of 4.03 and 3.30, respectively. These configurations lead to primary energy saving of 36 % and 23 % and emissions reductions of 56 % and 48 %, respectively, compared to conventional gas-fired systems. For the process at 125 °C, the optimal solution is a cascade cycle using Cyclopentane and Dimethylether, achieving a combined coefficient of performance of 2.58, primary energy saving of 4 %, and emissions reductions of 35 %. The economic analysis highlights investment pay back periods ranging from one to two years for all selected configurations, confirming the high profitability of the proposed system. Furthermore, the integration of a photovoltaic system is evaluated, allowing for the complete electrification of the thermal supply, increasing self-consumption and enabling an effective and structural decarbonization of the entire production process.

Key-words: High-Temperature Heat Pump, Industrial Processes Decarbonization, Combined Heating and Cooling, Milk Pasteurization

Nomenclature

AEP	Annual Electricity Production
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CAPEX	CAPital EXpenditure
CEPCI	Chemical Engineering Plant Cost Index
CFCs	Chlorofluorocarbons
CHC	Combined Heating and Cooling
CIP	Clean In Place
COP	Coefficient Of Performance
DME	DiMethylEther
FAO	Food and Agriculture Organization
GCR	Ground Coverage Ratio
GWP	Global Warming Potential
HCFC	Hydrochlorofluorocarbon
HCFO	Hydrochlorofluoroolefin
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HT	High Temperature
HTHP	High-Temperature Heat Pump
HTST	High-Temperature Short-Time
HX	Heat Exchanger
IHX	Internal Heat eXchanger
IT	Intermediate Temperature
KPI	Key Performance Indicator
LT	Low Temperature
LTLT	Low-Temperature Long-Time
ODP	Ozone Depletion Potential
OPEX	OPerational EXpenditure
PES	Primary Energy Saving
PHE	Plate Heat Exchanger
PV	Photovoltaic
RTE	Round Trip Efficiency
SAM	System Advisor Model
SHC	Simultaneous Heating and Cooling
SOC	State of Charge
SRC	Simple Real Cycle
TES	Thermal Energy Storage
TLV	Threshold Limit Value
TMY	Typical Meteorological Year
TWA	Time-Weighted Average
UHT	Ultra-High-Temperature
UT	Ultra-Temperature

1. Introduction

The global food demand is expected to increase 60 % by 2050 due to population growth. Consequently, for a sustainable food sector, energy use during the food processing stage needs to be reduced [1].

Liquid milk and dairy products (such as cheese, butter, yogurt, and milk powder) are produced and consumed almost worldwide. According to predictions made by the Food and Agricultural Organization (FAO), global milk production is expected to increase by 1.7 % per year over the current decade (2021–2030) reaching 1020 million tonnes by 2030 [2]. Industrial thermal processes are responsible for around 24 % of the total heating and cooling needs in Europe and they still largely rely on fossil fuels. The rising concerns for greenhouse gas emissions, together with the global growth of energy demand are leading to a necessary shift towards sustainable alternatives in the industrial heating sector [3]. As highlighted in Figure 1, the energy consumption during the pasteurization process is mainly related (91 %) to heating and cooling [4]. Combined Heating and Cooling (CHC) High-Temperature Heat Pumps (HTHPs) allow for the exploitation of both the heating capacity of the condenser and the cooling capacity of the evaporator as useful effects. The integration of HTHPs with large temperature lifts for CHC applications in dairy processing industry has the advantage of reducing primary energy consumption and decarbonizing the sector [1].

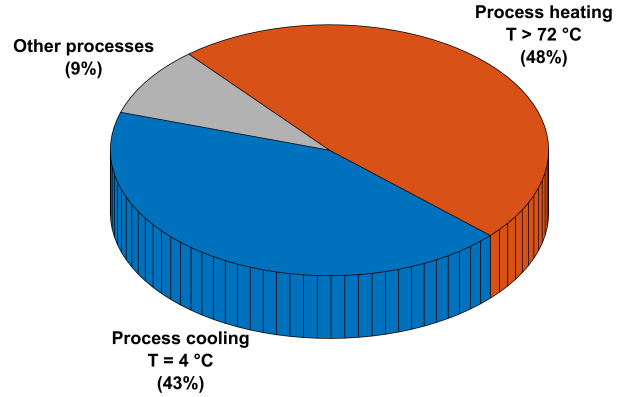


Figure 1: Energy use in milk pasteurization.

1.1. Literature Review

Due to the increasing prioritization of energy efficiency, CHC heat pumps have gained substantial attention within the research community over the past few years.

Numerous industrial processes require simultaneous heating and cooling, such as dairy pasteurization, building space conditioning, chocolate production, pharmaceutical processing, detergent production, and liquor distillation.

Figure 2, sourced from the Scopus database [5], illustrates the recent growth in publications concerning CHC HTHPs in the food industry, highlighting the increasing interest in these topics. China is by far the leading country in this research field, followed by Norway, Italy, Germany, and Switzerland.

Akbarzadeh et al. [6] conducted a comprehensive review of research on CHC heat pumps capable of providing simultaneous heating and cooling. The authors observed that current research focuses on several key areas, such as the development of advanced control and operational strategies, the analysis of environmental and operational factors, economic feasibility assessments, and integration with renewable energy sources. It is found that CHC heat pumps have great potential for cost reduction and sustainability, but there are several challenges to overcome, including promoting widespread adoption, validating applicability in real-world systems, and improving performance, which is often constrained by high temperature lifts.

Andersen et al. [7] evaluated the potential of CHC heat pumps compared to alternative solutions for the full electrification of four industrial applications (milk pasteurization, frozen food production, spray drying for powdered milk, and slaughterhouse operations). These applications are characterized by different operating conditions, delivering heat in the range between 23 °C and 210 °C and cooling in the range between -40 °C and 8 °C. A parametric analysis was conducted with 16 different working fluids and 10 different heat pump configurations. The authors demonstrated that it is possible to achieve significant Primary Energy Saving (PES) ranging from 12 % to 34 %. However, optimal performance is contingent upon specific operating conditions. In particular, it is crucial to maintain the proportion of heating demand relative to the total demand within a

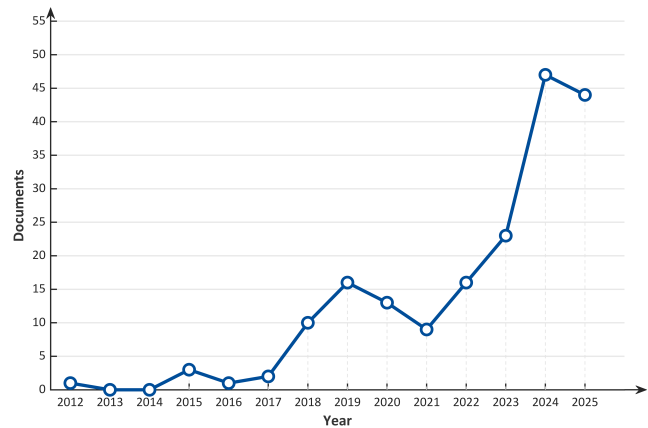


Figure 2: Growth of research interest.

certain range, specifically between 0.6 and 0.62. Furthermore, the study highlighted the importance of thermal storage capacity for stabilizing system operation.

Dong et al. [8] developed a prototype of a cascade HTHP, evaluating it under operating conditions typical of three different scenarios: dairy processing, liquor processing and deep dehumidification. R1234ze(E) was selected as the working fluid for the bottom cycle, while a blend of R1234ze(Z) and R1336mzz(E) was utilized for the top cycle. The experimental results indicated that the prototype was able to supply cooling as low as 2 °C and heating up to 120 °C, achieving a combined Coefficient Of Performance (COP) exceeding 2.58, 45.8 % higher than single heating systems. The study demonstrated that the overall performance of HTHP can surpass the conventional approach of using two separate heat pumps to provide cooling and heating respectively when the ratio of heating to cooling demand is high.

Ayou et al. [2] evaluated two distinct two-stage ammonia-based heat pump configurations designed for simultaneous heating (in the range between 65 °C and 95 °C) and cooling (up to 2 °C) in the dairy industry. The study compared a pure ammonia system with an ammonia/water system against conventional heating and cooling methods, composed by natural gas boilers and vapour-compression chillers. The results highlighted that the first system outperformed the mixture-based one in terms of thermodynamic efficiency, achieving a combined COP of up to 5.22 and PES of 54 %. However, the use of a mixture offered significant operational advantages, such as a considerably lower discharge pressure, leading to a more technically viable solution.

Another study by Ayou et al. [1] evaluated the performance of solar photovoltaic (PV) and solar thermal powered heat pump technologies for simultaneous heating (at 75 °C) and cooling (at 2 °C). Two heat pump cycles were investigated: an ammonia/water absorption heat pump and a two-stage vapour-compression heat pump, achieving COPs of 0.979 and 4.71, respectively.

Ahrens et al. [9] investigated an integrated heat pump system capable of meeting the heating and cooling demands of both a greenfield dairy plant and of an associated building. The dairy features a novel and innovative solution of a fully integrated energy system employing ammonia chillers, conventional ammonia HTHPs and a hybrid ammonia-water absorption compression heat pump to provide all temperature levels of heating and cooling demands. This system is capable of supplying cooling as low as -1.5 °C and heating up to 95 °C, achieving a combined COP of 4.1.

Schlemminger et al. [10] analyzed a 20 KW_{th} cascade HTHP capable of chilling water in the range between 4 °C and 10 °C and heating water in the range between 85 °C and 116 °C. Propane was selected as the working fluid for the bottom cycle, butane was utilized for the top cycle. A combined COP of 2.6 to 2.8 was achieved for a temperature lift of 110 K to 80 K, respectively. This performance gave a PES potential of 57 % and a potential CO_2 emission reduction of 94 %. The authors concluded that HTHPs can produce simultaneously chilled water and hot water efficiently for a lot of industrial processes, like drying, sterilization or pasteurization.

Urbanucci et al. [11] investigated the integration of a HTHP within a tri-generation system. The heat pump recovered the low-temperature heat from the condenser/absorber of an absorption chiller to produce hot water at around 90 °C. Ammonia, which achieved a COP of 3.9 at the expected working conditions, was found to be one of the most suitable fluids for this application. The studied system achieved promising performance, providing 10.3 %, 10.6 %, and 41.7 % economical savings in comparison to traditional tri-generation, cogeneration and separate production systems, respectively. Results showed that the proposed system provides the flexibility to cover variable energy demands and can achieve favorable performance in terms of annual primary energy consumption, also in stand-alone conditions.

Liu et al. [12] assessed the PES and the operating costs of an ammonia heat pump and two different types of carbon dioxide heat pumps, capable of simultaneously provide heating and cooling for seven different food processes. The analysis demonstrated that the proposed systems can achieve PES of up to 32 % in milk pasteurization, and 44 % in beer brewing. In general, the study concluded that thermal demand profiles, energy price structures, and geographic location are the most critical factors influencing both PES and operational costs.

1.2. Literature gap, Objective and Structure of the Thesis

The objective of this thesis is to assess whether High-Temperature Heat Pumps for Combined Heating and Cooling can be effectively integrated into a milk pasteurization plant. The ultimate goal is to enhance the environmental sustainability of the process, while reducing its operational costs. Most studies focus primarily on the thermodynamics of the cycle, such as assessing the impact of varying evaporation or condensation temperatures, but do not address how these systems can be effectively integrated into real industrial processes [13], such as building space conditioning, milk pasteurization, and pharmaceutical processing.

Section 2 is dedicated to the operating principles and main challenges of High-Temperature Heat Pumps, focusing on three different system configurations and paying particular attention to the selection of working fluids. Section 3 describes the milk pasteurization process and illustrates the proposed integration between the HTHP and the existing plant layout. Section 4 details the thermodynamic modeling of the proposed system, heat exchanger sizing, the parametric, PES, emission, and economic analyses, and the preliminary design of the PV system. Section 5 highlights the three optimal system configurations identified, analyzing their performance and feasibility from both technical and economic perspectives. This section also includes an assessment of the PES and avoided emissions.

2. High-Temperature Heat Pumps

This section aims to introduce the operating principles of High-Temperature Heat Pumps and to highlight the fundamental differences and challenges with respect to standard heat pumps.

2.1. Operating Principles and System Layouts of High-Temperature Heat Pumps

HTHPs are defined by their capability to supply heat at temperatures exceeding 80–100 °C [14][15] (Figure 3). The widespread adoption of this technology is crucial for the electrification and decarbonization of industrial heating processes. One of the main critical issues is related to the low performances, caused by the high temperature lifts between evaporation and condensation. Their combined use allows to exploit as a useful effect both the cooling capacity of the evaporator, and the heating capacity of the condenser. The COP is defined as the ratio between the useful thermal effect provided by the heat pump and the electricity required by the compressor. Depending on the specific application, three distinct COPs are computed (Equations 1). In terms of fundamental operating principles, HTHPs do not differ from conventional heat pumps designed for temperatures below 80 °C. They rely on the same thermodynamic cycles [16] and comprise the same four basic components: evaporator, compressor, condenser, and expansion valve.

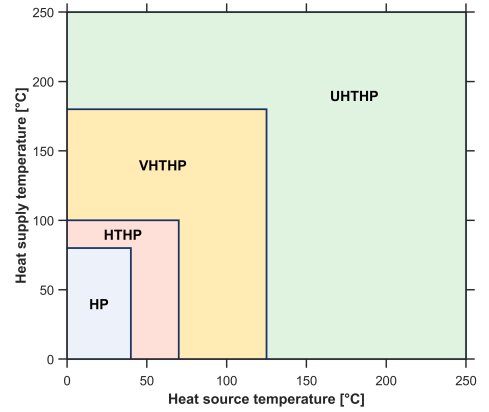


Figure 3: Classification of heat pumps.

$$COP_{heating} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \quad COP_{cooling} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \quad COP_{combined} = \frac{\dot{Q}_{cond} + \dot{Q}_{evap}}{\dot{W}_{comp}} \quad (1)$$

Nevertheless, some technical constraints must be addressed for high-temperature operation. First, the compressor is subjected to severe thermal stress. Current compressor technology is generally limited to a maximum discharge temperature of 160–180 °C [2], and necessitates the use of specific high-viscosity lubricants to prevent degradation. Second, the working fluid selection is crucial: its critical temperature must be sufficiently high to ensure that condensation occurs at the required heat delivery level. To mitigate these issues and keep the discharge temperature within safe limits, multi-stage compression cycles are usually employed.

The sources of irreversibility and entropy generation within a real heat pump cycle typically occur during the compression process, the isenthalpic expansion across the throttling valve, and the heat transfer in the heat exchangers.

In the thermodynamic cycle, the evaporator outlet (compressor inlet) is located close to the saturation curve. Therefore, superheating is necessary to prevent compressor damage caused by liquid droplet formation. A trade-off is required: if the superheating is too low, the risk of compressor failure increases; if it is too high, the system suffers a performance decrease due to increased heat transfer losses during the desuperheating section of the condenser [17].

Similarly, subcooling is required at the condenser outlet (expansion valve inlet) to prevent vapour flash (vapour formation due to pressure reduction) during the expansion process. Again, a trade-off must be considered: insufficient subcooling is ineffective, while excessive subcooling significantly lowers the inlet temperature of the secondary fluid in the condenser [17]. Superheating and subcooling are defined by the following equations:

$$\Delta T_{superheating} = T_{out_evap} - T_{sat_vapour} \quad \Delta T_{subcooling} = T_{sat_liquid} - T_{out_cond} \quad (2)$$

For the present study, three different cycle layouts are considered, which are further explored in the following sections.

2.1.1 Simple Real Cycle

The Simple Real Cycle (SRC) represents the most basic heat pump configuration (Figure 4). It consists of four primary components: an evaporator, a condenser, a compressor, and an isenthalpic expansion valve. This layout is widely adopted due to its cost-effectiveness and low complexity, which translate into reduced maintenance requirements. However, its main limitation lies in its inability to efficiently handle high pressure ratios; detrimental to the system's performance.

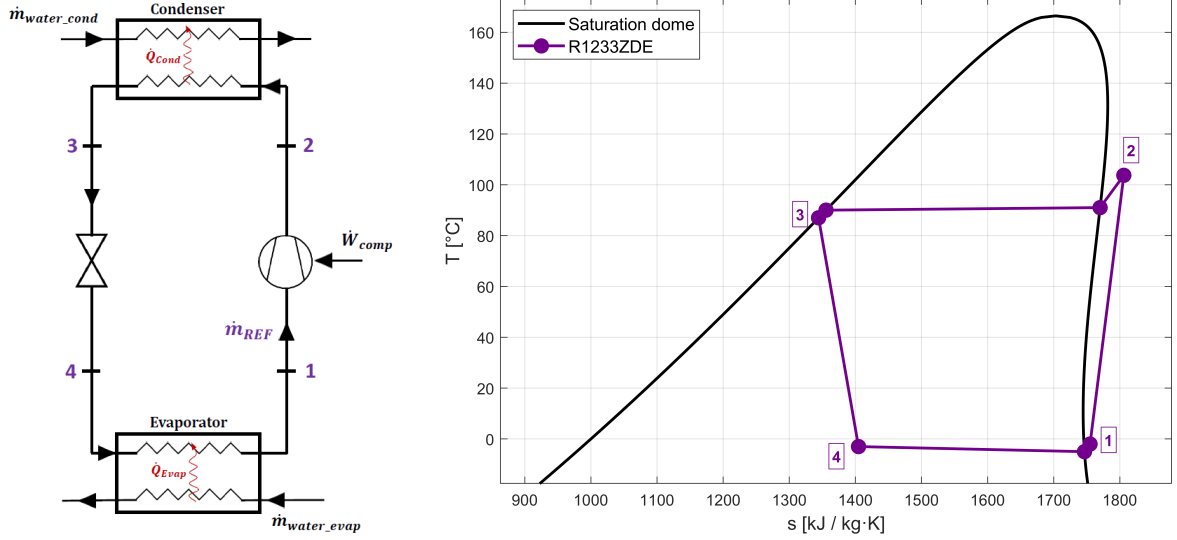


Figure 4: SRC plant layout and T-s diagram of a representative fluid: R1233zd(E).

2.1.2 Two-Stage Cycle

The Two-Stage Cycle (Figure 5) consists of two SRCs connected by a flash tank, where stratification occurs due to the density difference between phases. The overall pressure lift is distributed between two compressors. For each boundary condition, it is necessary to optimize the intermediate pressure (P_{int}). The optimal value is typically slightly higher than the geometric mean of the evaporation and condensation pressures [17]. Generally, the Two-Stage Cycle exhibits superior thermodynamic performance compared to the SRC; however, it entails higher Capital Expenditure (CAPEX) and system complexity, resulting in increased maintenance requirements.

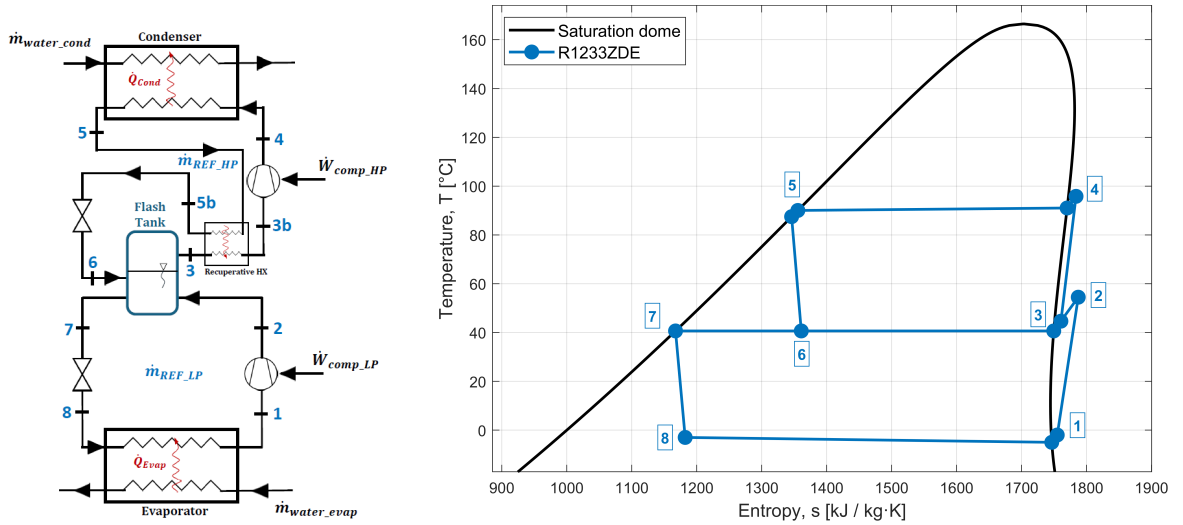


Figure 5: Two-stage plant layout and T-s diagram of a representative fluid: R1233zd(E).

2.1.3 Cascade Cycle

The Cascade Cycle (Figure 6) consists of two SRCs thermally coupled via an Internal Heat Exchanger (IHX), which operates simultaneously as the condenser for the low-temperature (bottom) cycle and as the evaporator for the high-temperature (top) cycle.

This configuration allows the use of two distinct working fluids, enabling specific fluid optimization for each temperature range and achieving high temperature lifts. However, the Cascade Cycle is penalized by heat transfer losses within the IHX, as well as higher CAPEX and maintenance costs due to increased system complexity.

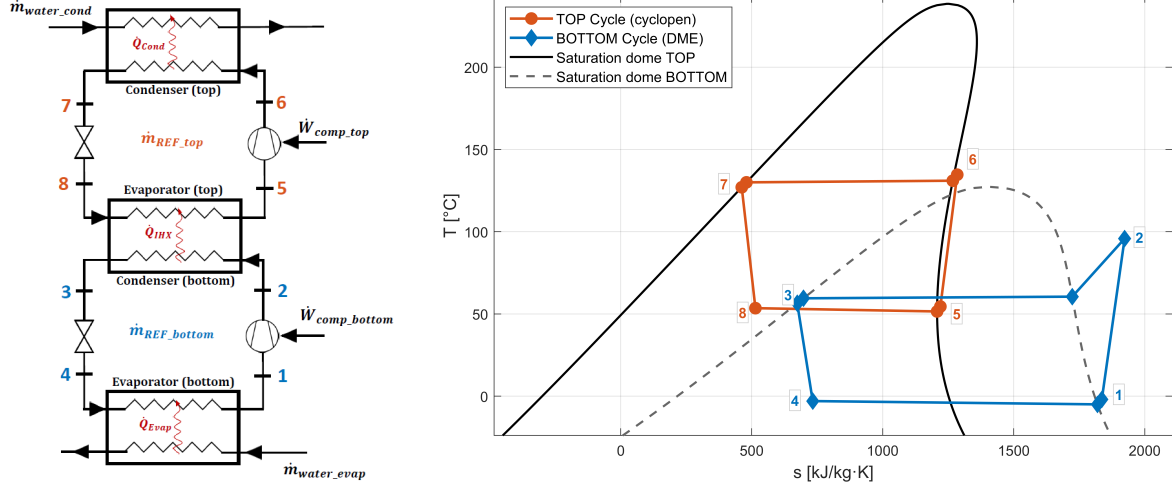


Figure 6: Cascade plant layout and T-s diagram of a representative fluid couple: Cyclohexane/DME

2.2. Working fluid and regulatory framework

The main focus of this Section is the working fluid, or refrigerant, which circulates through the components of the heat pump. The thermophysical properties of the working fluid have a significant impact on the COP, power consumption, heating and cooling capacity of the entire system [17][18].

The environmental impact of the working fluid is defined by the Global Warming Potential (GWP) and the Ozone Depletion Potential (ODP). Safety standards are defined by the American Society of Heating Refrigerating and Air conditioning Engineers (ASHRAE) Standard 34 [19], that classifies refrigerants based on toxicity and flammability. Toxicity is categorized into Class A (lower toxicity) and Class B (higher toxicity). Flammability ranges from Class 1 (non-flammable) to Classes 2/2L (slightly flammable) and Class 3 (highly flammable). These safety characteristics generally correlate with increased system complexity and costs [20]. Figure 7 groups the working fluids in terms of toxicity and flammability.

Over the past 200 years, refrigerant development has spanned four main stages as early substances were phased out due to safety and environmental concerns [21][22].

The first generation relied on available solvents and volatile fluids. The second generation introduced stable synthetic chemicals like Chlorofluorocarbons (CFCs) and Hydrochlorofluorocarbons (HCFCs). Following the discovery of ozone depletion, the third generation shifted towards ozone-benign Hydrofluorocarbons (HFCs). Today, strict climate policies demand fourth-generation refrigerants with zero Ozone Depletion Potential (ODP) and low Global Warming Potential (GWP), promoting Hydrofluoroolefins (HFOs), Hydrochlorofluoroolefins (HCFOs), and the revival of natural fluids [23]. This phase-out process is heavily driven by the EU F-gas regulation [24].

	Lower Toxicity	Higher Toxicity
Higher Flammability	A3	B3
	A2	B2
Lower Flammability	A2L	B2L
No Flame Propagation	A1	B1

Figure 7: Working fluids classification [25].

3. Milk Processing and Thermal Requirements

3.1. Pasteurization Process

Pasteurization is the heat treatment of milk which aims to reduce pathogenic microorganisms associated with milk to an acceptable level [26]. This process does not affect significantly chemical, physical and organoleptic properties of the milk.

The combination of temperature and holding time is very important, as it determines the intensity of the heat treatment [27]. Figure 8 illustrates the holding time required to eliminate various bacterial strains at different temperature levels. Although heat-resistant micrococci may be present in milk, they are generally considered non-pathogenic.

To ensure an effective pasteurization process, raw milk must be heated to a specific target temperature and maintained for a corresponding holding time. Subsequently, the product is cooled to 4 °C; consequently, Simultaneous Heating and Cooling (SHC) is required.

Selecting the optimal time/temperature combination requires balancing microbiological safety (greater at higher temperatures) with product quality (better taste at lower temperatures) [27].

Table 1 outlines four industrial pasteurization methods that reflect this trade-off and national legislative requirements.

Low Temperature Long Time (LTLT) and High-Temperature Short Time (HTST) produce fresh pasteurized milk, preserving authentic flavor and vitamins but limiting refrigerated shelf life to 5–7 days. The Ultra-Temperature (UT) process, primarily used for yogurt, denatures proteins to create a thicker texture with an 8–10 day refrigerated shelf life. Finally, Ultra-High Temperature (UHT) significantly extends shelf life but compromises the milk’s authentic taste.

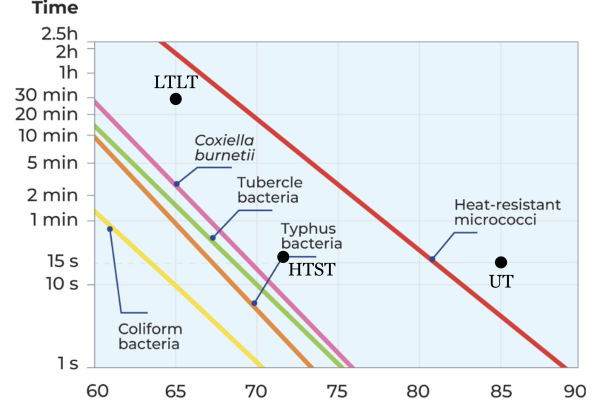


Figure 8: Lethal effect on bacteria [27].

Pasteurization Process	Temperature [°C]	Holding Time [s]
Low-Temperature Long-Time (LTLT)	63	1800
High-Temperature Short-Time (HTST)	72-75	15-20
Ultra-Temperature (UT)	85-90	8-15
Ultra-High-Temperature (UHT)	125-138	2-4

Table 1: Pasteurization methods temperature and holding time [1].

3.2. State of the Art of industrial Pasteurization Plant

The state of the art of industrial pasteurization plants is discussed in detail in the Dairy Processing Handbook [27]. The current heating and refrigeration systems, in the food processing industries, are mainly fossil-fuel-fired boilers and vapour-compression chillers [9]. The boiler systems are used to generate steam or hot water as a heating utility, and they are used for washing, concentration, evaporation, drying, pasteurization, Clean-in-Place (CIP), and other processes that need heating. The vapour-compression chillers consume electrical energy to provide chilled water as a cold utility for process cooling, product refrigeration, and storage [1].

Milk is taken from a balance tank (Point 1 in Figure 9) at a storage temperature of 4 °C [8] and it is delivered to the preheating section by a feed pump. The preheating section (Point 4 in Figure 9) is composed of a regenerative heat exchanger in which pasteurized milk preheats the raw milk.

Before reaching pasteurization temperature, the milk has to be clarified, because it often contains solid matter such as dirt particles, leucocytes and somatic cells that could compromise the effectiveness of the process. Clarification can be obtained in a filter or, more effectively, in a centrifugal clarifier (Point 5 in Figure 9).

In the heating section (Point 6 in Figure 9) milk reaches the pasteurization temperature receiving heat from either hot water or steam supplied by the utility central.

After the heating section, milk passes through a holding tube (Point 8 in Figure 9) properly dimensioned according to the required holding time and hourly capacity of the plant. The dimensioning of the holding tube is detailed in Appendix B. Typically, the holding tube is covered by a stainless steel hood to prevent temperature drop, and also to protect operators from getting burned. A temperature transmitter (Point A in Figure 9) sends a signal to the steam or hot water valve to provide the correct pasteurization temperature.

Figure 10 shows the proposed integration of a Simple Real Cycle HTHP that supplies combined heating and cooling with a pasteurization plant. Analogous schemes of the pasteurization plant integrated with Two-Stage and Cascade HTHPs can be found in the Appendix A of this work. The working principle is the same as a traditional plant, described in the previous section, but heating is provided by the condenser of the heat pump, while cooling is provided by the evaporator of the heat pump. The heating section of the pasteurization heat exchanger is called "Secondary Heat Exchanger 1", while the cooling section is called "Secondary Heat Exchanger 2". The plant is designed such that the heating capacity required to raise the milk to the pasteurization temperature equals the cooling capacity necessary to return it to the storage temperature (4 °C). This symmetry is dictated by the first law energy balance applied to the light blue section in Figure 10 and aligns with the findings of Dong et al. [8], who state that 'the heating and cooling demand in pasteurization is nearly 1:1'.

The heat pump is designed so that the cooling requested by the milk is equal to the thermal power of the evaporator: $\dot{Q}_{Evap} = \dot{Q}_{HX2}$. Due to the compressor work ($\dot{W}_{Comp}$), ($\dot{Q}_{Cond}$) will be higher than ($\dot{Q}_{Evap}$), so the heat pump will produce more than the thermal power required to reach the pasteurization temperature. Hence a three-way valve must be installed in the water circuit at the outlet of the condenser, to feed the Secondary Heat Exchanger 1 only with the necessary amount of water. The remaining water mass flow is delivered to a Thermal Energy Storage (TES), that can be used to supply heat during the Clean In Place process. CIP involves the circulation of a solution of alkaline or acid at hot temperatures (60-80 °C) through the pasteurization plant under turbulent flow conditions. The initial temperature of the cleaning solutions and water is 12 °C [28], so the Thermal Energy Storage charged with the thermal power supplied by the condenser of the heat pump can be exploited for this application. In the case of milk, the complex combination of heat sensitive proteins, minerals, fats, and lactose results in high fouling rates. In addition, milk contains a range of microorganisms that can grow either in the product or on a food processing surface, where they may subsequently contaminate the product and negatively impact food quality and safety. For these reasons, a pasteurization plant must be equipped with an appropriately sized CIP unit. This process significantly impacts the available production time.

The design milk volumetric flow rate is set to 25000 l/h, typical value of big commercial pasteurization plants [27]. Selecting a standard commercial tube diameter of 2.5 inches (0.0635 m), the resulting milk velocity is equal to 2.19 m/s. The obtained value is sufficiently below the critical threshold of 3 m/s, ensuring that the fluid's organoleptic and physical properties are preserved. The thermophysical milk properties, such as density (ρ) and specific heat capacity (c_p), are reported in Appendix C.

The auxiliary power consumption is related to the pumps and is computed as follows:

$$\dot{W}_{el,pump} = \frac{\dot{m} \cdot \Delta P_{pump}}{\rho \cdot \eta_{glob}} \quad (3)$$

Where η_{glob} is the product of the hydraulic, mechanical and motor efficiencies and is set equal to 0.65.

Table 2 outlines the criteria used to calculate the milk temperature within the plant.

The milk temperature at the holding tube inlet (Point 5 in Figure 10) is set considering the thermal losses in the holding tube, calculated in Appendix B. The goal is to ensure that the temperature at the outlet is at least equal to the pasteurization setpoint, preventing the fluid from dropping below the required limit during the holding period.

Milk Temperature	Notes
$T_{storage} = T_{milk}(1) = T_{milk}(2)$	$T_{storage} = 4^{\circ}\text{C}$
$T_{milk}(3) = T_{milk}(2) + \varepsilon_{Reg,HX} \cdot (T_{milk}(7) - T_{milk}(2))$	$\varepsilon_{Reg,HX} = 0.92$ [29]
$T_{milk}(4) = T_{milk}(3)$	
$T_{milk}(5) = T_{past} + \Delta T_{drop,holding,tube}$	See Appendix B for $\Delta T_{drop,holding,tube}$
$T_{milk}(6) = T_{past} = T_{milk}(7)$	See Table 1
$T_{milk}(8) = T_{milk}(7) - \varepsilon_{Reg,HX} \cdot (T_{milk}(7) - T_{milk}(2))$	$\varepsilon_{Reg,HX} = 0.92$ [29]
$T_{milk}(9) = T_{storage}$	$T_{storage} = 4^{\circ}\text{C}$

Table 2: Mathematical formulation of milk temperatures at different process stages.

Secondary Heat Exchangers are coupled with the evaporator and condenser water circuits according to the parameters defined in Table 3. The T-Q diagrams of these heat exchangers for the case with a pasteurization temperature of 85 °C are shown in Figure 11 and 12. Similar diagrams can be obtained for the other temperature cases.

Variable	Value
$\Delta T_{\text{pinch,Sec,HX1}} = T_{\text{w,out,cond}} - T_{\text{milk}}(5)$	2 °C [27]
$\Delta T_{\text{pinch,Sec,HX2}} = T_{\text{milk}}(9) - T_{\text{w,out,evap}}$	2 °C [27]
$\Delta T_{\text{w,cond}}$	5 °C [1]
$\Delta T_{\text{w,evap}}$	5 °C [1]

Table 3: Assumed Pinch Points and water temperature differences for Secondary Heat Exchangers.

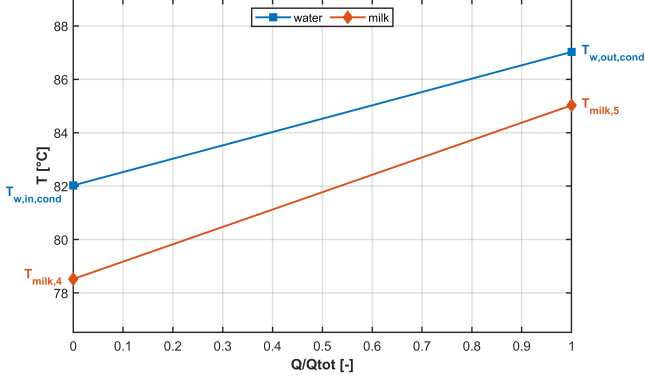


Figure 11: Secondary HX 1 T-Q diagram

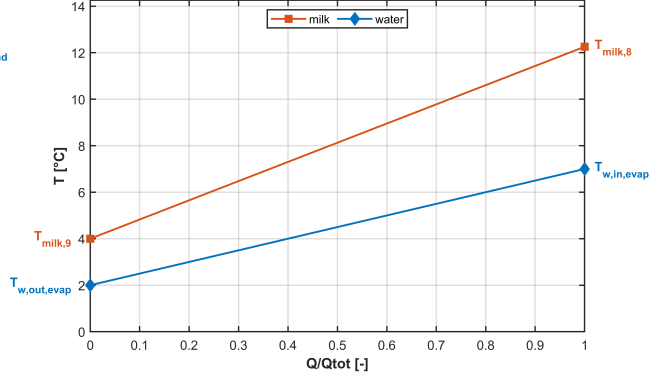


Figure 12: Secondary HX 2 T-Q diagram

4. Methodology: Heat Pump modelling

The methodology described in this section has been implemented in MATLAB, using the REFPROP 10 library [30] to compute the thermophysical properties of the working fluids.

The complete MATLAB source code developed for this study is available at [31].

4.0.1 Evaporator and Condenser modelling

The pinch-point temperature difference for evaporators with water as secondary fluid varies between 5 and 15 °C, depending on the thermodynamic properties of the working fluid. The pinch-point temperature difference for condensers with water as secondary fluid varies between 2 and 10 °C. The superheating at the evaporator outlet ranges from 3 and 10 °C, while the subcooling at the condenser outlet ranges from 3 to 7/10 °C [17]. To ensure a fair comparison between different working fluids, the pressure drops in the evaporator and condenser are defined by fixing the corresponding saturation temperature drop, rather than assuming a constant absolute pressure drop. For both evaporator and condenser, $\Delta T_{\text{sat},\Delta P}$ ranges from 0.5 to 2 °C [32]. As remarked in Tables 4 and 5, pressure losses related to the temperature drop in the evaporator are considered higher than the ones in the condenser. This is related to the fact that the pressure losses are inversely proportional to the density of the working fluid, which is expected to be much higher in the condenser.

The design parameters adopted for the evaporator and condenser are summarized in the following tables.

Variable	Value
$\Delta T_{\text{pinch,evap}} = T_{\text{w,out,evap}} - T_{\text{evap}}$	5 °C
$\Delta T_{\text{superheating,evap}}$	3 °C
$\Delta T_{\text{sat},\Delta P,\text{evap}}$	2 °C

Table 4: Assumed parameters for the Evaporator.

Variable	Value
$\Delta T_{\text{pinch,cond}} = T_{\text{cond}} - T_{\text{w,out,cond}}$	4 °C
$\Delta T_{\text{subcooling,cond}}$	3 °C
$\Delta T_{\text{sat},\Delta P,\text{cond}}$	1 °C

Table 5: Assumed parameters for the Condenser.

4.0.2 Compressor modelling

HTHPs are characterized by high temperature lifts, necessitating high compressor pressure ratios, which complicate the component design. From a literature review it is possible to state that the two most common compressor types used in HTHPs are reciprocating piston and scroll compressors [33].

Reciprocating compressors are typically suitable for applications with a heating capacity ranging from 1 to 200 kW, whereas screw compressors are preferred for capacities between 150 and 1600 kW [34].

To guarantee a fair comparison during a parametric analysis involving various system configurations, working fluids, and temperature ranges, the isentropic efficiency must be kept constant. According to the literature, this parameter is assumed equal to 0.75 [1][3][13]. Both electrical-to-mechanical and mechanical efficiencies of the compressor have been taken into account to determine the gross electrical consumption [13].

Parameter	Value
$\eta_{\text{iso,comp}}$	0.75
$\eta_{\text{el/mech,comp}}$	0.94
$\eta_{\text{mech,comp}}$	0.96

Table 6: Compressor efficiency parameters.

4.1. Simple Real Cycle Modelling

The computation of thermodynamic properties for the SRC, detailed in Table 7 is performed referring to the scheme shown in Figure 4.

Point	Thermodynamic properties
Sat,vap (Low,Pressure)	$T_{\text{sat,vap,LP}} = T_{\text{evap}} - \Delta T_{\text{sat},\Delta P,\text{evap}}$ $P_{\text{sat,vap,LP}} = P(T_{\text{sat,vap,LP}}; X = 1)$
1	$P_{\text{ref,1}} = P_{\text{sat,vap,LP}}$ $T_{\text{ref,1}} = T_{\text{sat,vap,LP}} + \Delta T_{\text{superheating,evap}}$
2	$P_{\text{ref,2}} = P_{\text{cond}}$ $h_{\text{ref,2,iso}} = h(P_{\text{ref,2}}; s_{\text{ref,1}})$ $h_{\text{ref,2}} = h_{\text{ref,1}} + \frac{h_{\text{ref,2,iso}} - h_{\text{ref,1}}}{\eta_{\text{iso,comp}}}$
Sat,vap (High,Pressure)	$T_{\text{sat,vap,HP}} = T(P_{\text{cond}}; X = 1)$ $P_{\text{sat,vap,HP}} = P_{\text{cond}}$
Sat,liq	$T_{\text{sat,liq}} = T_{\text{sat,vap,HP}} - \Delta T_{\text{sat},\Delta P,\text{cond}}$ $P_{\text{sat,liq}} = P(T_{\text{sat,liq}}; X = 0)$
3	$P_{\text{ref,3}} = P_{\text{cond}} - \Delta P_{\text{cond}}$ $T_{\text{ref,3}} = T_{\text{sat,liq}} - \Delta T_{\text{subcooling,cond}}$
4	$P_{\text{ref,4}} = P_{\text{evap}}$ $h_{\text{ref,4}} = h_{\text{ref,3}}$

Table 7: Thermodynamic properties calculation for Simple Real Cycle.

4.2. Two-Stage Modelling

In the Two-Stage Cycle a recuperative Heat Exchanger is installed to synchronously superheat the vapour at the outlet of the flash tank and to subcool the liquid at the outlet of the condenser. $\Delta T_{\text{superheating,flash}}$ has been chosen between 2 and 10 °C, depending on the case. The computation of thermodynamic properties for the Two-Stage Cycle, detailed in Table 8 is performed referring to the scheme shown in Figure 5.

The optimal intermediate pressure is slightly higher than the geometric mean of the condensation and evaporation pressures [2][17], and it can be determined using the following equation:

$$P_{\text{int,2stage}} = \sqrt{P_{\text{cond}} P_{\text{evap}}} \cdot \sqrt{\frac{T_{\text{cond}}}{T_{\text{evap}}}} \quad (4)$$

Point	Thermodynamic properties
SatVap,LP	$T_{\text{sat,vap,LP}} = T_{\text{evap}} - \Delta T_{\Delta P, \text{evap}}$ $P_{\text{sat,vap,LP}} = P(T_{\text{sat,vap,LP}}; X = 1)$
1	$P_{\text{ref},1} = P_{\text{sat,vap,LP}}$ $T_{\text{ref},1} = T_{\text{sat,vap,LP}} + \Delta T_{\text{sh,evap}}$
2	$P_{\text{ref},2} = P_{\text{int}}$ $h_{\text{ref},2,\text{iso}} = h(P_{\text{ref},2}; s_{\text{ref},1})$ $h_{\text{ref},2} = h_{\text{ref},1} + \frac{h_{\text{ref},2,\text{iso}} - h_{\text{ref},1}}{\eta_{\text{iso,comp}}}$
3	$P_{\text{ref},3} = P_{\text{int}}$ $T_{\text{ref},3} = T(P_{\text{int}}; X = 1)$
3b	$T_{\text{ref},3b} = T_{\text{ref},3} + \Delta T_{\text{sh,flash}}$ $P_{\text{ref},3b} = P_{\text{int}}$
4	$P_{\text{ref},4} = P_{\text{cond}}$ $h_{\text{ref},4,\text{iso}} = h(P_{\text{ref},4}; s_{\text{ref},3b})$ $h_{\text{ref},4} = h_{\text{ref},3b} + \frac{h_{\text{ref},4,\text{iso}} - h_{\text{ref},3b}}{\eta_{\text{iso,comp}}}$

Point	Thermodynamic properties
SatVap,HP	$T_{\text{sat,vap,HP}} = T(P_{\text{cond}}; X = 1)$ $P_{\text{sat,vap,HP}} = P_{\text{cond}}$
5	$T_{\text{ref},5} = T_{\text{sat,vap,HP}} - \Delta T_{\Delta P, \text{cond}}$ $P_{\text{ref},5} = P(T_{\text{ref},5}; X = 0)$
5b	$P_{\text{ref},5b} = P_{\text{ref},5}$ $h_{\text{ref},5b} = h_{\text{ref},5} - h_{\text{ref},3b} + h_{\text{ref},3}$
6	$P_{\text{ref},6} = P_{\text{int}}$ $h_{\text{ref},6} = h_{\text{ref},5b}$
7	$P_{\text{ref},7} = P_{\text{int}}$ $T_{\text{ref},7} = T(P_{\text{int}}; X = 0)$
8	$P_{\text{ref},8} = P_{\text{evap}}$ $h_{\text{ref},8} = h_{\text{ref},7}$

Table 8: Thermodynamic properties calculation for Two-Stage Cycle.

4.3. Cascade Cycle Modelling

The optimal condensation temperature of the bottom cycle is determined through an iterative optimization process, with the aim of maximizing the combined COP, keeping constant $\Delta T_{\text{superheating}}$, $\Delta T_{\text{subcooling}}$ and $\Delta T_{\text{HT,IHX}}$. The top cycle is thermodynamically linked to the bottom cycle using the following equation:

$$\Delta T_{\text{HT,IHX}} = T_{\text{cond,bottom}} - T_{\text{evap,top}} = 7^\circ\text{C} \quad (5)$$

Bottom Cycle	
Point	Thermodynamic properties
SatVapLP	$T_{\text{sat,vap,LP}} = T_{\text{evap,bot}} - \Delta T_{\Delta P, \text{evap}}$ $P_{\text{sat,vap,LP}} = P(T_{\text{sat,vap,LP}}; X = 1)$
1	$P_{\text{ref},1} = P_{\text{sat,vap,LP}}$ $T_{\text{ref},1} = T_{\text{sat,vap,LP}} + \Delta T_{\text{sh,evap}}$
2	$P_{\text{ref},2} = P_{\text{cond,bot}}$ $h_{\text{ref},2,\text{iso}} = h(P_{\text{ref},2}; s_{\text{ref},1})$ $h_{\text{ref},2} = h_{\text{ref},1} + \frac{h_{\text{ref},2,\text{iso}} - h_{\text{ref},1}}{\eta_{\text{iso,comp}}}$
SatVapHP	$T_{\text{sat,vap,HP}} = T(P_{\text{cond,bot}}; X = 1)$ $P_{\text{sat,vap,HP}} = P_{\text{cond,bot}}$
Sat Liq	$T_{\text{sat,liq}} = T_{\text{sat,vap,HP}} - \Delta T_{\Delta P, \text{cond}}$ $P_{\text{sat,liq}} = P(T_{\text{sat,liq}}; X = 0)$
3	$P_{\text{ref},3} = P_{\text{cond,bot}} - \Delta P_{\text{cond}}$ $T_{\text{ref},3} = T_{\text{sat,liq}} - \Delta T_{\text{sc,cond}}$
4	$P_{\text{ref},4} = P_{\text{evap,bot}}$ $h_{\text{ref},4} = h_{\text{ref},3}$

Top Cycle	
Point	Thermodynamic properties
SatVapLP	$T_{\text{sat,vap,LP}} = T_{\text{evap,top}} - \Delta T_{\Delta P, \text{evap}}$ $P_{\text{sat,vap,LP}} = P(T_{\text{sat,vap,LP}}; X = 1)$
5	$P_{\text{ref},5} = P_{\text{sat,vap,LP}}$ $T_{\text{ref},5} = T_{\text{sat,vap,LP}} + \Delta T_{\text{sh,evap}}$
6	$P_{\text{ref},6} = P_{\text{cond,top}}$ $h_{\text{ref},6,\text{iso}} = h(P_{\text{ref},6}; s_{\text{ref},5})$ $h_{\text{ref},6} = h_{\text{ref},5} + \frac{h_{\text{ref},6,\text{iso}} - h_{\text{ref},5}}{\eta_{\text{iso,comp}}}$
SatVapHP	$T_{\text{sat,vap,HP}} = T(P_{\text{cond,top}}; X = 1)$ $P_{\text{sat,vap,HP}} = P_{\text{cond,top}}$
Sat Liq	$T_{\text{sat,liq}} = T_{\text{sat,vap,HP}} - \Delta T_{\Delta P, \text{cond}}$ $P_{\text{sat,liq}} = P(T_{\text{sat,liq}}; X = 0)$
7	$P_{\text{ref},7} = P_{\text{cond,top}} - \Delta P_{\text{cond}}$ $T_{\text{ref},7} = T_{\text{sat,liq}} - \Delta T_{\text{sc,cond}}$
8	$P_{\text{ref},8} = P_{\text{evap,top}}$ $h_{\text{ref},8} = h_{\text{ref},7}$

Table 9: Thermodynamic properties calculation for Cascade Cycle.

The computation of thermodynamic properties for the Cascade Cycle, detailed in Table 9 is performed referring to the scheme shown in Figure 6.

4.4. Equivalent Chiller and Boiler Modelling

To ensure a fair comparison, the chiller modelled for the conventional plant has the same cooling capacity as the heat pump. The chiller modelling is analogous to the heat pump modelling described in the previous sections, with the difference that heat is released to the ambient environment at a condensation temperature of 35 °C. It is characterized by a cooling COP of 3.84.

The gas boiler is sized by considering the total heat output leaving the condenser as the useful effect.

4.5. Heat Exchangers Sizing

By applying the methodology described, the T-Q diagrams for the evaporator and the condenser are plotted for a pasteurization temperature of 85 °C (Figures 13 and 14). Analogous profiles can be derived for the other evaluated temperatures.

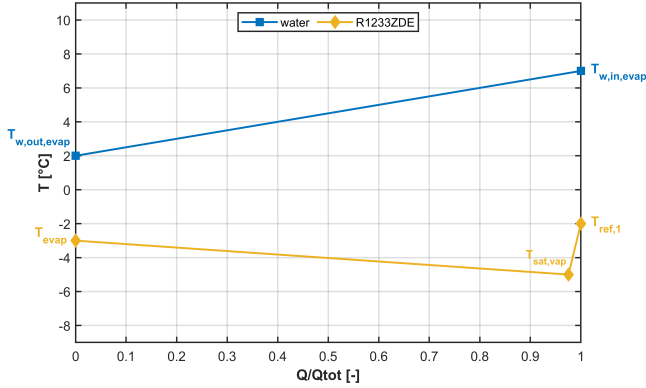


Figure 13: Evaporator T-Q diagram

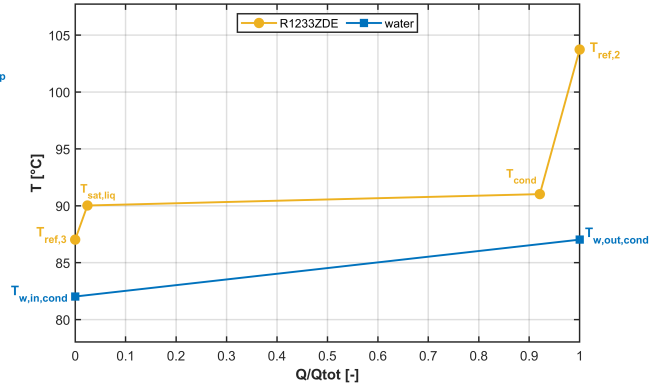


Figure 14: Condenser T-Q diagram

The evaporator cooling capacity is set exactly equal to the thermal load required to cool the milk down to the storage temperature of 4°C. The condenser heating capacity is then higher with respect to the thermal load necessary to heat the milk up to the pasteurization temperature. The excess thermal power is thus delivered to the TES.

$$\dot{Q}_{milk,HX,sec,1} = \dot{m}_{milk} \cdot c_{p,milk} \cdot (T_{milk}(5) - T_{milk}(4)) \quad (6)$$

$$\dot{Q}_{milk,HX,sec,2} = \dot{m}_{milk} \cdot c_{p,milk} \cdot (T_{milk}(8) - T_{milk}(9)) \quad (7)$$

$$\dot{Q}_{evap} = \dot{Q}_{milk,HX,sec,2} \quad (8)$$

$$\dot{m}_{ref} = \frac{\dot{Q}_{evap}}{\Delta h_{evap}} \quad (9)$$

$$W_{comp,gross} = \frac{\dot{m}_{ref} \cdot \Delta h_{comp}}{\eta_{el,mech,comp} \cdot \eta_{mech,comp}} \quad (10)$$

$$\dot{Q}_{cond} = \dot{m}_{ref} \cdot \Delta h_{cond} \quad (11)$$

$$\dot{Q}_{TES} = \dot{Q}_{cond} - \dot{Q}_{milk,HX,sec,1} \quad (12)$$

The Heat Exchangers sizing procedure is described in detail in Appendix C. The same methodology is applied to design the evaporator, condenser, IHX, regenerative and secondary heat exchangers.

The geometry of the selected Heat Exchangers, reported in Table 10, derives from technical data sheets of commercially available models. Commercial plates have commonly a surface enlargement factor φ between 1.15 and 1.25 [35].

Starting from the calculations described in this Section, it is possible to derive the value of the product of the overall heat transfer coefficient (U) and the heat transfer area (A) at design conditions:

$$(UA)_{design} = \frac{\dot{Q}}{LMTD} \quad (13)$$

By applying the heat transfer correlations detailed in Appendix C, the overall heat transfer coefficient and, consequently, the required total heat transfer area can be determined.

	Secondary HXs / Recuperative HX [29]	Condenser / Evaporator / IHX [36]
L_{plate} [mm]	1524	875
W_{plate} [mm]	803	386
b [mm]	4	4
φ [-]	1.2	1.2
L_c [mm]	$\frac{2b}{\varphi}$	$\frac{2b}{\varphi}$
t_{wall} [mm]	0,6	0,6
β [°]	60	60

Table 10: Characteristics of the selected Heat Exchangers.

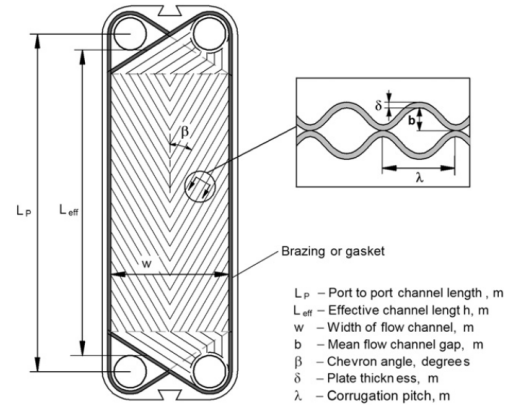


Figure 15: Plate Geometry [37]

The pressure drops within the single-phase heat exchangers (recuperative and secondary HXs) are evaluated as the sum of frictional losses, inlet and outlet losses, and gravitational losses [38]. The correlations used are listed in Appendix C. Calculating the pressure drops across the single-phase heat exchangers is necessary to control the pressure throughout the circuit and prevent contamination, as explained in Section 3.2.

4.6. Parametric Analysis and Key Performance Indicators

This work is a parametric analysis investigating the influence of:

- HTHP configuration (Simple Real Cycle, Two-Stage Cycle and Cascade Cycle);
- Pasteurization temperature (72 °C, 85 °C and 125 °C) and corresponding holding time (20 s, 15 s and 4 s, respectively);
- 20 distinct working fluids, listed in Table 11.

Only working fluids with GWP < 7 and negligible ODP have been considered.

Refrigerant	Chem. Form.	Type	T_{crit} [°C]	P_{crit} [bar]	GWP	ODP	Tox. Flamm.
R1270	C ₃ H ₆	HC	91,1	46,2	2	0	A3
R290	C ₃ H ₈	HC	96,7	42,48	3	0	A3
R600a	C ₄ H ₁₀	HC	134,7	36,6	3	0	A3
R711	C ₇ H ₈	HC	318,6	41,3	3	0	A3
R600	C ₄ H ₁₀	HC	152	38	4	0	A3
R601a	C ₅ H ₁₂	HC	187,8	33,8	4	0	A3
R601	C ₅ H ₁₂	HC	196,6	33,7	5	0	A3
R601c	C ₅ H ₁₀	HC	238,6	45,1	5	0	A3
R1224yd(Z)	CF ₃ CF=CHCl	HCFO	155,5	33,3	1	0	A1
R1233zd(E)	CF ₃ CH=CHCl	HCFO	166,5	36,2	1	0	A1
R1234yf	CF ₃ CF=CH ₂	HFO	94,7	33,8	1	0	A2L
R1234ze(Z)	CF ₃ CH=CHF	HFO	150,1	35,3	1	0	A2L
R1336mzz(Z)	CF ₃ CH=CFCF ₃	HFO	171,4	29	2	0	A1
R1234ze(E)	CF ₃ CH=CHF	HFO	109,4	36,4	6	0	A2L
R1243zf	C ₃ H ₃ F ₃	HFO	105	35,2	1	0	A2L
R717 (NH ₃)	NH ₃	Inorganic	132,3	113,3	0	0	B2L
Acetone	C ₃ H ₆ O	Organic	235	47	1	0	A3
Ethanol	C ₂ H ₆ O	Organic	241	61,5	1	0	A3
DME	CH ₃ OCH ₃	Organic	127	52,4	1	0	A3
Methanol	CH ₃ OH	Organic	240	80,9	2,8	0	B2

Table 11: Properties of selected refrigerants.

In order to compare the results and evaluate the best configuration for each pasteurization temperature, the following Key Performance Indicators (KPI), listed by importance, have been taken into account:

- Stability of the working fluid, in terms of:

- Dry compression, to avoid compressor mechanical failure;
- Minimum pressure ($P_{min} > P_{atm}$), to avoid vacuum conditions, which would cause infiltrations.
- COP_{comb} ;
- GWP, ODP, toxicity and flammability of the working fluid.

Based on the defined KPIs, three optimal configurations (one for each pasteurization temperature) are selected from the parametric analysis. Subsequently, a detailed assessment of PES, avoided emissions, and economic performance is conducted on these selected systems, comparing them with a conventional plant.

4.7. Primary Energy Saving Analysis

Primary Energy Saving is defined as the percentage difference between the primary energy consumption of a conventional system and the primary energy consumption of the proposed plant (Equation 14).

In this study the proposed configuration consists of a High-Temperature Heat Pump, while the traditional plant is composed of a vapour-compression chiller and a gas boiler (Figure 16).

$$PES = \frac{(PE_{chiller} + PE_{boiler}) - PE_{HP}}{PE_{chiller} + PE_{boiler}} \quad (14)$$

To ensure a fair comparison, the thermal loads are kept consistent between the two systems. The cooling effect of the conventional chiller ($\dot{Q}_{Evap,chiller}$) is set equal to the evaporator cooling capacity of the HTHP ($\dot{Q}_{Evap,HP}$). Similarly, the thermal output of the boiler ($\dot{Q}_{out,boiler}$) is equal to the condenser heating capacity of the HTHP ($\dot{Q}_{Cond,HP}$).

$$PE_{chiller} = \frac{\dot{W}_{Comp,chiller}}{\eta_{PE,el}} \cdot h_{working} \quad (15)$$

$$PE_{boiler} = \frac{\dot{Q}_{in,boiler}}{\eta_{PE,boiler}} = \frac{\dot{Q}_{out,boiler}}{\eta_{PE,boiler} \cdot \eta_{th,boiler}} \cdot h_{working} \quad (16)$$

$$PE_{HP} = \frac{\dot{W}_{Comp,HP}}{\eta_{PE,el}} \cdot h_{working} \quad (17)$$

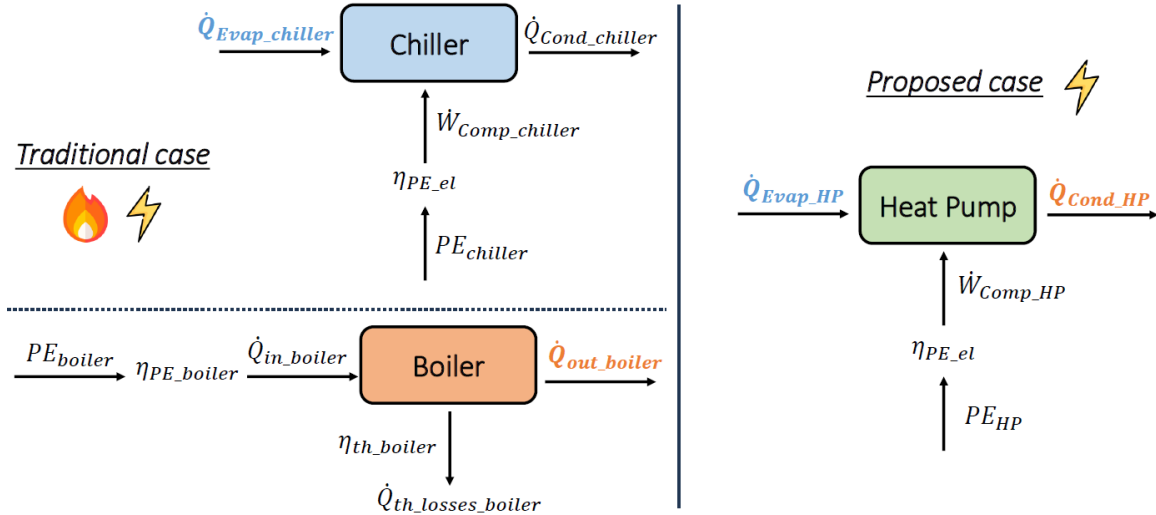


Figure 16: Primary Energy Saving: Traditional case vs Proposed case

The primary energy conversion efficiencies and boiler thermal efficiency are listed in the following table:

Parameter	Value
$\eta_{PE,el}$	0,413 [39]
$\eta_{PE,boiler}$	0,952 [39]
$\eta_{th,boiler}$	0,92 [17]

Table 12: Efficiency parameters for PES calculation.

Pasteurization plants typically operate for an average 20 hours per day [40]. The remaining time is allocated for Clean In Place or maintenance processes. Consequently, considering 365 days per year, the annual operating time ($h_{working}$) is set equal to 7300 hours. This value is consistent with findings by Cox et al. [41].

4.8. Emission Analysis

Emission sources are divided into direct emissions (related to gas boiler) and indirect emissions (related to the electricity required by the chiller and the heat pump). The emission factors utilized in the equations are detailed in Table 13.

$$E_{TOT}[kg_{co2eq}] = E_{el} + E_{gas} \quad (18)$$

$$E_{el}[kg_{co2eq}] = \dot{W}_{comp}[kW] \cdot h_{working}[h] \cdot Emission_{factor,el} \left[\frac{kg_{co2eq}}{kWh} \right] \quad (19)$$

$$E_{nat_{gas,i}}[kg_{co2eq}] = \dot{Q}_{in,boiler}[kW] \cdot h_{working}[h] \cdot Emission_{factor,i} \left[\frac{kg_{co2eq}}{kWh} \right] \quad (20)$$

$$\dot{Q}_{in,boiler} = \frac{\dot{Q}_{out,boiler}}{\eta_{boiler}} \quad (21)$$

Emission factor	Value
$Emission_{factor,el}$	$288 \times 10^{-3} \left[\frac{kg_{CO2eq}}{kWh} \right]$ [42]
$Emission_{factor,NG,CO_2}$	$202 \times 10^{-3} \left[\frac{kg_{CO_2}}{kWh} \right]$ [43]

Table 13: Emission factors for electricity and natural gas.

4.9. Economic Analysis

To ensure a fair comparison between the proposed plant and the existing plants, the economic analysis is based only on the High-Temperature Heat Pump, the chiller and the boiler. Other plant components (recuperative HX, secondary HXs, circulation pumps, and holding tube) are omitted as they remain identical in both cases. Table 14 shows the components considered for economic analysis of the different cases.

Component	Traditional plant	Proposed plant		
		SRC	Two-Stage	Cascade
Evaporator	Yes	Yes	Yes	Yes
Condenser	Yes	Yes	Yes	Yes
Compressor	Yes	Yes	Yes	Yes
Valve	Yes	Yes	Yes	Yes
Flash Tank	No	No	Yes	No
IHX	No	No	No	Yes
TES	No	Yes	Yes	Yes
Boiler	Yes	No	No	No

Table 14: Considered components for economic analysis.

CAPEX cost functions (shown in Table 15) are converted from United State Dollars to Euros dividing by 1.1313, the average conversion factor in 2025 [44].

Cost function (CAPEX)		reference
$C_{comp}[\$] = C_{comp,ref} \cdot \left(\frac{P_{comp}}{P_{comp,ref}} \right)^{0,89}$	$C_{comp,ref} = 722000 \$$ $P_{comp,ref} = 1000 kW$	[45]
$C_{boiler}[\$] = 170,61 \cdot \dot{Q}_{boiler}$		[41]
$C_{plate,HX}[\$] = C_{plate,HX,ref} \cdot \left(\frac{A_{plate,HX}}{A_{plate,HX,ref}} \right)^{0,6}$	$C_{plate,HX,ref} = 16000 \$$ $A_{plate,HX,ref} = 100 m^2$	[46]
$C_{flash,tank}[\$] = 280,3 \cdot \dot{m}_{ref}^{0,67}$		[47]
$C_{throttling,valve}[\$] = 114,5 \cdot \dot{m}_{ref}$		[47]
$C_{TES}[\$] = 20 \cdot TES_{capacity}$		[48]

Table 15: CAPEX Cost functions from literature.

It is necessary to update the cost functions using the Chemical Engineering Plant Cost Index (CEPCI). The most recent available value is from 2024.

$$C_{2024} = C_{by,year} \cdot \left(\frac{CEPCI_{2024}}{CEPCI_{by,year}} \right) \quad (22)$$

Both electricity and natural gas costs, part of Operational Expenditure (OPEX) shown in Table 16, refer to the italian market.

Cost function (OPEX)	reference
$c_{el} = 0,143 [\text{€}/kWh_{el}]$	[49]
$c_{natural,gas} = 0,124 [\text{€}/kWh_{th}]$	[50]
$c_{ETS} = 90,7 [\text{€}/ton_{CO_2}]$	[51]
$C_{O\&M} = 0,06 \cdot CAPEX \left[\frac{\text{€}}{year} \right]$	[46]

Table 16: OPEX Cost functions.

4.10. Preliminary Design of a PV Plant

This section is dedicated to the preliminary design of a photovoltaic system serving the pasteurization plant. The three selected configurations are referred to as the Low Temperature (LT), Intermediate Temperature (IT), and High Temperature (HT) cases.

Table 17 summarizes the electrical energy requested by the pasteurization plant, including the request of the auxiliaries.

	LT case ($T_{past} = 72^\circ\text{C}$)	IT case ($T_{past} = 85^\circ\text{C}$)	HT case ($T_{past} = 125^\circ\text{C}$)
Maximum load	151 kW_{el}	241 kW_{el}	672 kW_{el}
$E_{el,requested}$	1102 MWh/year	1753 MWh/year	4906 MWh/year

Table 17: Electrical load and energy requested by the pasteurization plant.

The analysis is carried out using the System Advisor Model (SAM) software on an hourly basis. As previously mentioned, the plant operates 20 hours a day and the electrical load can be assumed constant, since the heat pump always operates under the same boundary conditions. The plant's operation is divided into four time slots: two dedicated to production and two dedicated to the Clean In Place process or maintenance. Production is scheduled from 8:00 AM to 6:00 PM and from 8:00 PM to 6:00 AM. The remaining 4 hours are allocated to CIP and maintenance procedures. The electrical demand related to these last two processes is neglected.

The technical characteristics of the PV module (Trina Solar TSM-600 DE20 [52]) and the inverter (Huawei Sun 2000-100KTL-USH0 [53]) are reported in Tables 18 and 19.

PV module	
L_{module}	2,2 m
W_{module}	1,3 m
$P_{max,module}$	600 W
$V_{OC,PV}$	41,5 V
$V_{MPP,PV}$	34,4 V

Table 18: PV module characteristics.

Inverter	
P_{inv}	100 kW
$V_{max,inv}$	1200 V _{DC}
$V_{max,MPPT}$	1200 V _{DC}
$V_{min,MPPT}$	1200 V _{DC}

Table 19: Inverter characteristics.

The plant is located in the province of Parma, Italy, an area characterized by a high density of milk pasteurization plants. Assuming a specific location allows to obtain the data related to the Typical Meteorological Year (TMY) and the solar angles. The solar Zenith and Azimuth angles at design conditions are respectively 68.33° and 175.61° [54], calculated to avoid mutual shading between the panels at 12:00 PM on December 21st. The tilt angle is 25°, while the Ground Coverage Ratio (GCR) is 0.304 for all cases. For each case a different available area is hypothesized, equal to 1, 1.6 and 4 hectares, respectively. Figure 17 shows the actual footprint of the proposed photovoltaic systems, located near a milk pasteurization plant in the province of Parma.

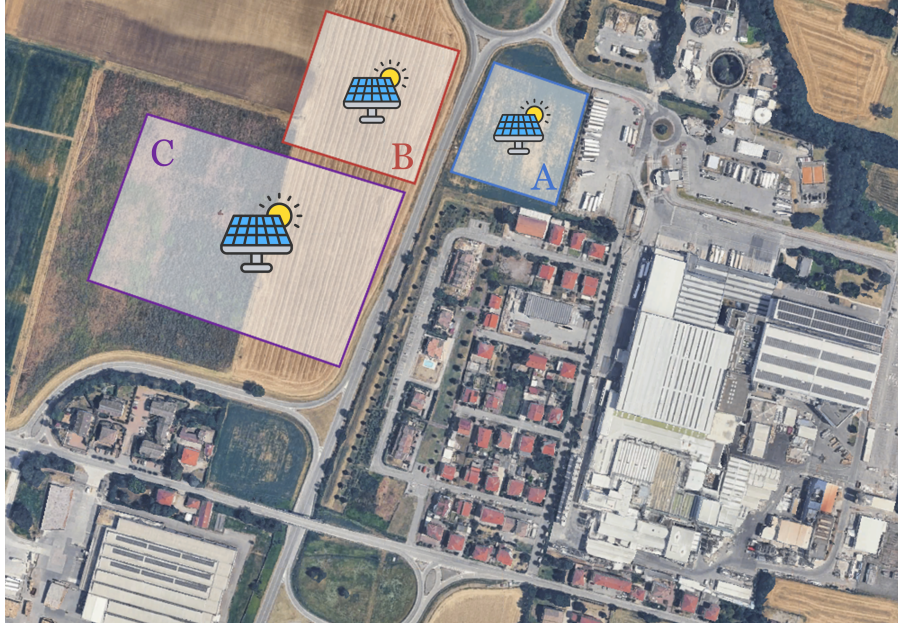


Figure 17: Actual footprint of the proposed PV systems.

5. Results and Discussion

This Chapter highlights the main results of the present work from technical, environmental and economic perspectives. The parametric analysis described in Section 4.6 yielded more than 170 distinct configurations to be compared. The graphs regarding the combined COP can be found in Appendix D. Three configurations are proposed as the best ones according to the KPIs discussed in Section 4.10.

5.1. Selected Optimal Configurations

5.1.1 Selected Optimal Configuration with $T_{past} = 72^\circ\text{C}$

Based on the KPIs, the best proposed configuration for both the case with $T_{past} = 72^\circ\text{C}$ and for the case with $T_{past} = 85^\circ\text{C}$ is the Two-Stage cycle with R1234ze(E). This working fluid is characterized by low GWP, ODP equal to zero, low toxicity and mild flammability. As can be noted from the graphs in Appendix D this configuration yields one of the highest combined COPs. The compression is dry and the compressor outlet temperature is well below the established limits. The evaporation pressure is above the atmospheric pressure. These features indicate that the system is both highly efficient and technically safe.

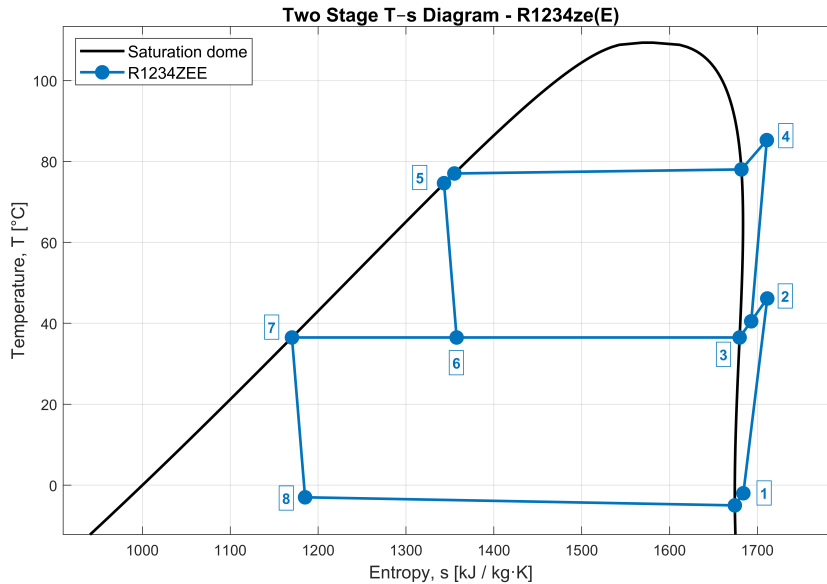
Other refrigerants with slightly higher COP have not been proposed as the best ones, because they are characterized by sub-atmospheric evaporation pressures, which involves the risk of air infiltration.

Ammonia-based configurations, although frequently investigated in the literature, have not been selected due to their excessively high compressor discharge temperatures, which introduce the risk of refrigerant thermal cracking.

For each proposed configuration, a comparison with the SRC baseline case is presented.

$T_{past} [^\circ\text{C}]$	HP configuration	working fluid	GWP	ODP	Tox/Flamm	COP_{comb}
72	Simple Real Cycle	R1234ze(E)	6	0	negligible/mild	3,23
72	Two-Stage	R1234ze(E)	6	0	negligible/mild	4,03

Table 20: Comparative analysis of the proposed optimal configuration vs. SRC at $T_{past} = 72^\circ\text{C}$.



Parameter	Value
P_{evap}	1,93 bar
P_{int}	6,96 bar
P_{cond}	19,2 bar
$\Delta T_{suph,flash}$	4 °C
A_{evap}	57,8 m ²
A_{cond}	62,6 m ²
$\dot{W}_{comp,LP}$	59,5 kW
$\dot{W}_{comp,HP}$	73,8 kW
$\dot{Q}_{evap} = \dot{Q}_{sec,HX2}$	208,9 kW
$\dot{Q}_{cond}$	329,2 kW
$\dot{Q}_{sec,HX1}$	179,6 kW
$\dot{Q}_{TES,CIP}$	149,6 kW

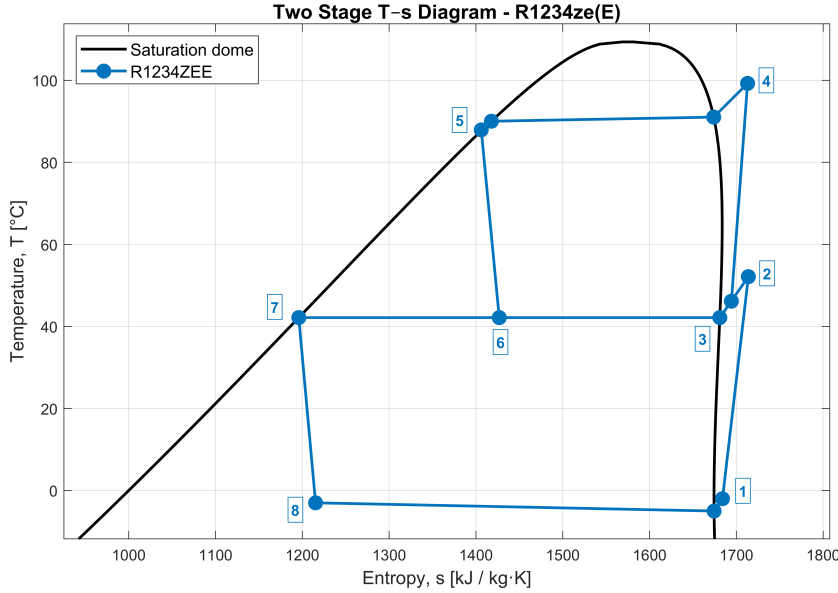
Table 21: Operating parameters.

Figure 18: Selected optimal configuration with $T_{past} = 72^\circ\text{C}$

5.1.2 Selected Optimal Configuration with $T_{past} = 85^{\circ}\text{C}$

$T_{past} [^{\circ}\text{C}]$	HP configuration	working fluid	GWP	ODP	Tox/Flamm	COP_{comb}
85	Simple Real Cycle	R1234ze(E)	6	0	negligible/mild	2,40
85	Two-Stage	R1234ze(E)	6	0	negligible/mild	3,30

Table 22: Comparative analysis of the proposed optimal configuration vs. SRC at $T_{past} = 85^{\circ}\text{C}$.



Parameter	Value
P_{evap}	1,93 bar
P_{int}	8,12 bar
P_{cond}	25,3 bar
$\Delta T_{suph,flash}$	4 $^{\circ}\text{C}$
A_{evap}	52,9 m^2
A_{cond}	62,6 m^2
$\dot{W}_{comp,LP}$	87,2 kW
$\dot{W}_{comp,HP}$	128,2 kW
$\dot{Q}_{evap} = \dot{Q}_{sec,HX2}$	257,6 kW
$\dot{Q}_{cond}$	452,0 kW
$\dot{Q}_{sec,HX1}$	215,5 kW
$\dot{Q}_{TES,CIP}$	236,5 kW

Table 23: Operating parameters.

Figure 19: Selected optimal configuration with $T_{past} = 85^{\circ}\text{C}$

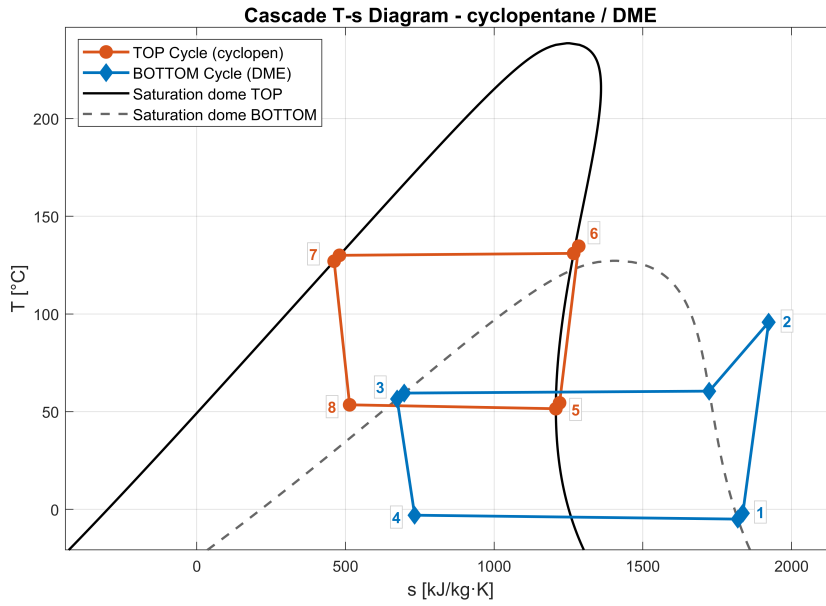
5.1.3 Selected Optimal Configuration with $T_{past} = 125^{\circ}\text{C}$

The best proposed configuration for the case with $T_{past} = 125^{\circ}\text{C}$ is the Cascade cycle with Cyclopentane as top cycle working fluid and Dimethylether as bottom cycle working fluid. These working fluids are characterized by low GWP, ODP equal to zero and low toxicity; however, due to their high flammability, stringent safety measures and specialized plant design are strictly required to mitigate fire and explosion risks.

$T_{past} [^{\circ}\text{C}]$	HP configuration	working fluid	GWP	ODP	Tox/Flamm	COP_{comb}
125	Simple Real Cycle	Cyclopen / DME	5/1	0/0	negligible/high	2,26
125	Cascade	Cyclopen / DME	5/1	0/0	negligible/high	2,58

Table 24: Comparative analysis of the proposed optimal configuration vs. SRC at $T_{past} = 125^{\circ}\text{C}$.

A physiological decrease in the combined COP is observed as the pasteurization temperature increases. It drops from 4.03 for the first case, to 3.30 for the second case, to 2.58 for the last case. This performance drop is directly related to extreme temperature lift required between the evaporator and the condenser, which causes a significant increase compressors electrical consumption. However, the calculated performance aligns with data from the literature.



Parameter	Value
$P_{evap,bottom}$	2,39 bar
$P_{cond,bottom}$	14,6 bar
$P_{evap,top}$	1,16 bar
$P_{cond,top}$	8,16 bar
$A_{evap,bottom}$	67,4 m ²
A_{IHX}	16,4 m ²
$A_{cond,top}$	211,8 m ²
$\dot{W}_{comp,bottom}$	201,3 kW
$\dot{W}_{comp,top}$	307,8 kW
$\dot{Q}_{evap,bot} = \dot{Q}_{sec,HX2}$	426,3 kW
$\dot{Q}_{cond,top}$	885,7 kW
$\dot{Q}_{sec,HX1}$	329,7 kW
$\dot{Q}_{TES,CIP}$	556,0 kW

Figure 20: Selected optimal configuration with $T_{past} = 125^{\circ}\text{C}$

Table 25: Operating parameters.

5.2. PES and Emission Analysis Results

The analysis of the results, shown in Figure 21, highlights a remarkable PES of 36 % achieved at 72 °C, a good PES of 23 % for the intermediate case, before dropping to 4 % in the high temperature process. This occurs because the electrical consumption required by the heat pump to cover such a demanding temperature lift increases disproportionately, rising from 133.3 kW to 509.1 kW. However, the environmental benefits remain substantial, with emission reductions of 56 %, 48 %, and 35 %, respectively, proving that thermal electrification is consistently advantageous from a climate perspective.

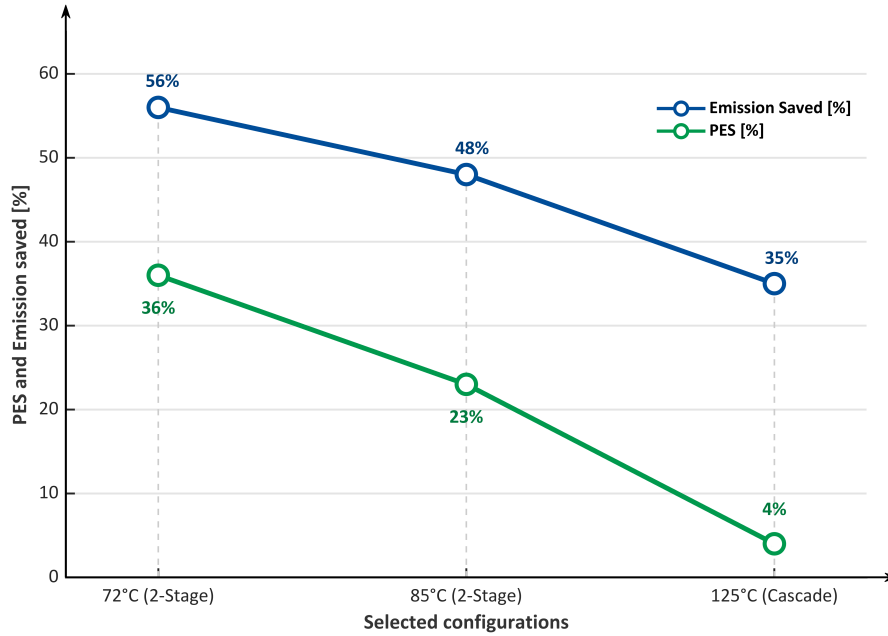


Figure 21: PES and emission analysis results.

5.3. Economic Analysis

The results of the economic analysis show that CAPEX is mainly composed of compressors, TES, and gas boilers, in that order. OPEX is primarily driven by the costs of electricity and natural gas.

The first proposed plant configuration (Figures 22a and 22b) features a CAPEX of 245 k€ and an OPEX of 198 k€/year, resulting in a PBT of approximately one year.

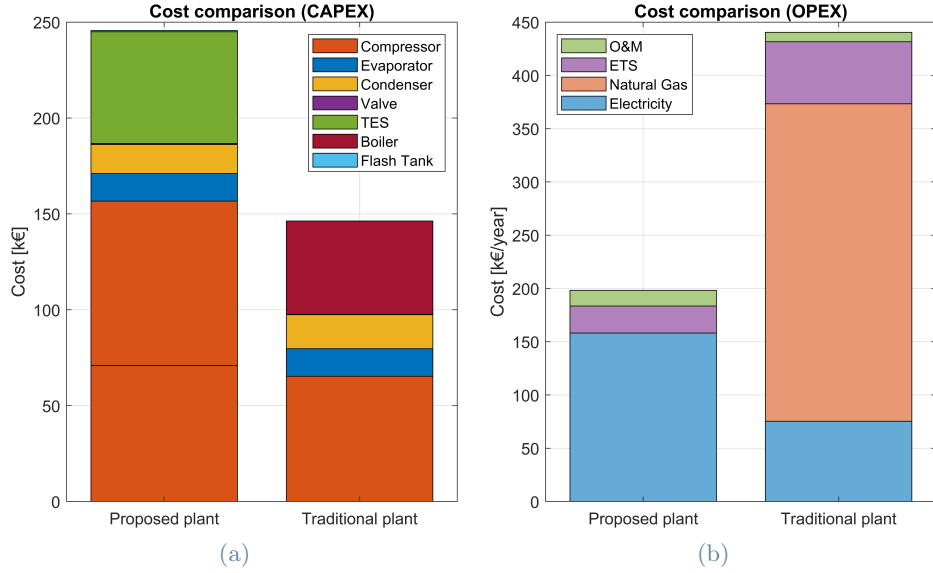


Figure 22: Economic analysis results for the selected configuration with $T_{past} = 72^\circ\text{C}$

The second proposed plant configuration (Figures 23a and 23b) features a CAPEX of 362 k€ and an OPEX of 314 k€/year, resulting in a PBT of one year and four months.

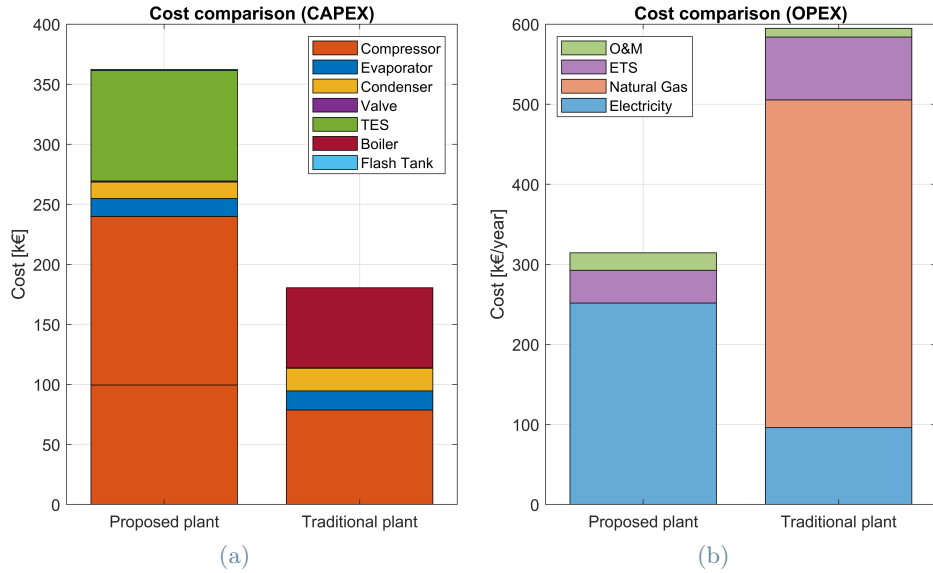


Figure 23: Economic analysis results for the selected configuration with $T_{past} = 85^\circ\text{C}$

The third proposed plant configuration (Figures 24a and 24b) features a CAPEX of 786 k€ and an OPEX of 848 k€/year, resulting in a PBT of slightly less than two years.

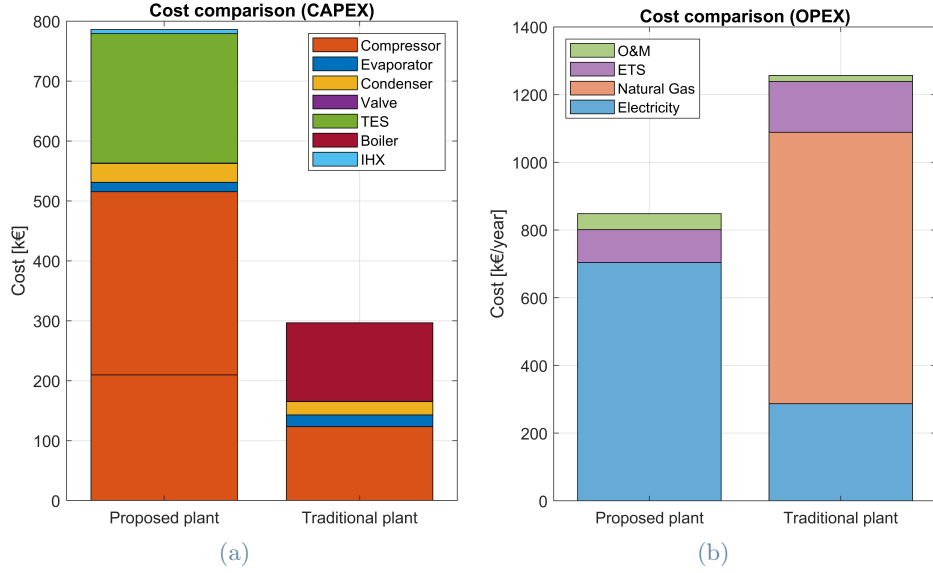


Figure 24: Economic analysis results for the selected configuration with $T_{past} = 125^{\circ}\text{C}$

All the proposed configurations show highly encouraging payback times, calculated considering the replacement of an existing plant. This result demonstrates that the analyzed setups are viable both for existing facilities and for newly constructed plants.

Looking at the initial investment cost charts, the financial gap between the proposed system and the traditional one is entirely attributable to the compressors. The cost of the TES effectively offsets the elimination of the gas boiler, while the costs of the heat exchangers remain largely equivalent. The drastic reduction in OPEX is mainly related to natural gas purchasing, now replaced by electricity.

5.4. Preliminary Design of a PV Plant

By knowing the Typical Meteorological Year, the solar angles, the available area, the technical specifications of the PV modules and of the inverters, it is possible to perform the design of a dedicated photovoltaic plant, obtaining the parameters highlighted in Table 26.

	<i>LT case</i> ($T_{past} = 72^{\circ}\text{C}$)	<i>IT case</i> ($T_{past} = 85^{\circ}\text{C}$)	<i>HT case</i> ($T_{past} = 125^{\circ}\text{C}$)
Land footprint	10000 m^2 (1 ha)	16000 m^2 (1.6 ha)	40000 m^2 (4 ha)
$N_{inverter}$	9	14	35
Module per string	25	25	26
String in parallel	66	110	267
DC/AC ratio	1,10	1,18	1,19

Table 26: PV plant design parameters.

The plant is equipped with batteries, allowing to store the excess power from the photovoltaic system. The operational logic (Figure 25) prioritizes self-consumption: the generated electricity is primarily utilized by the pasteurization plant, and any surplus is used to recharge the battery. Once the battery is fully charged, the excess electricity is injected into the grid. The Round Trip Efficiency (RTE) of the battery is of 91 %.

	<i>LT case</i> ($T_{past} = 72^{\circ}\text{C}$)	<i>IT case</i> ($T_{past} = 85^{\circ}\text{C}$)	<i>HT case</i> ($T_{past} = 125^{\circ}\text{C}$)
Maximum load	151 kW	241 kW	672 kW
P_{DC} installed	990 kW	1650 kW	4165 kW
Battery power	500 kW	800 kW	2200 kW
Battery capacity	2000 kWh	3200 kWh	9000 kWh

Table 27: System sizing and battery characteristics.

To prevent battery degradation, the minimum State of Charge (SOC) and maximum State of Charge limits are set to 15 % and 95 %, respectively. Furthermore, the minimum time at charge state is set to 10 minutes. The battery capacity is sized to ensure the operation of the pasteurization plant during the 10 nighttime hours, considering a discharge cycle from 95 % to 15 % SOC. This calculation accounts for both the operational charge/discharge limits and an additional 5 % safety margin. The characteristics of the selected batteries are listed in Table 27.

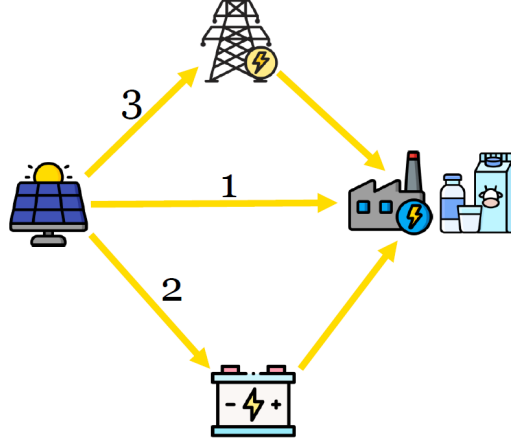


Figure 25: System Operational Logic.

5.4.1 Performance analysis of the PV system: LT case ($T_{past} = 72\text{ }^{\circ}\text{C}$)

The first proposed PV system has a net Annual Energy Production (AEP) of $1158\text{ MWh}_{el}/\text{year}$. Figure 26 illustrates both the monthly energy production of the PV plant and the electrical load of the pasteurization plant. With a load of 151 kW_{el} for 20 hours a day, the monthly load remains constant at around 90-95 MWh throughout the year. A clear energy surplus is observed during the spring and summer months (from March to August), with a production peak in May (of about 140 MWh). Conversely, during the winter months (November, December, January, and February), PV generation is insufficient to meet the demand, necessitating energy withdrawal from the grid.

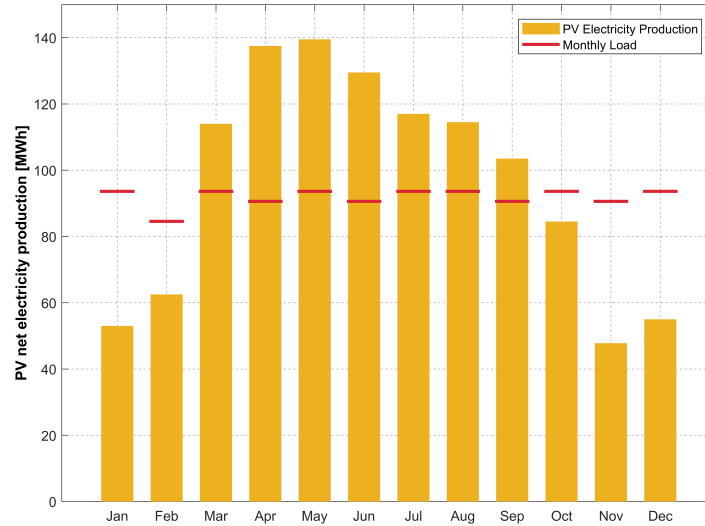


Figure 26: AEP by month (LT case)

Figure 27 illustrates the average coverage of the electricity demand. It can be observed that the majority of the load during the central hours of the day is directly met by the PV system, while the remainder is shared between the battery and the grid. During nighttime operation, a substantial portion of the load is satisfied by the energy stored during the day, with the rest being drawn directly from the grid. As expected, the battery discharges at night and recharges during daylight hours.

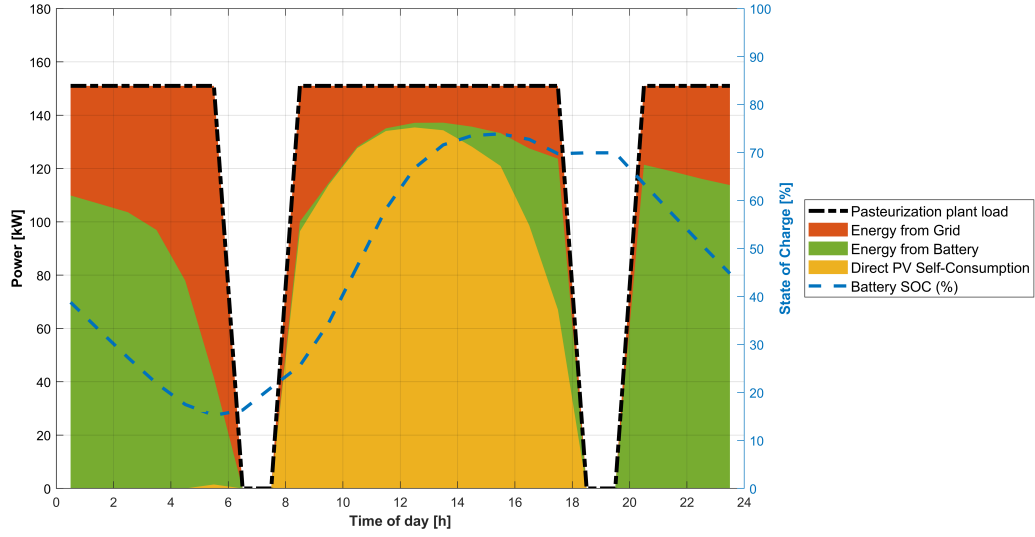


Figure 27: Annual average daily electricity demand coverage (LT case)

Figure 28 shows the operation of the plant during 3 sunny summer days. The daytime load is entirely satisfied by the PV plant, while the nighttime load is satisfied uniquely by the energy accumulated by the batteries during the day. As expected from the battery sizing, they are capable of autonomously powering the pasteurization plant when fully charged.

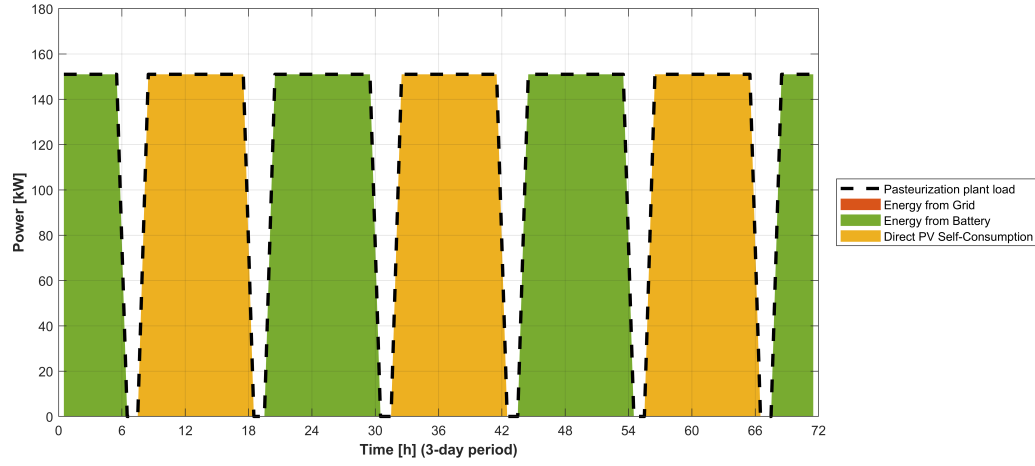


Figure 28: Analysis of hourly electricity demand coverage: July 28-30 (LT case)

Figure 29 shows the operations of the plant during three winter days characterized by low photovoltaic production. The pasteurization plant is powered almost entirely by the grid.

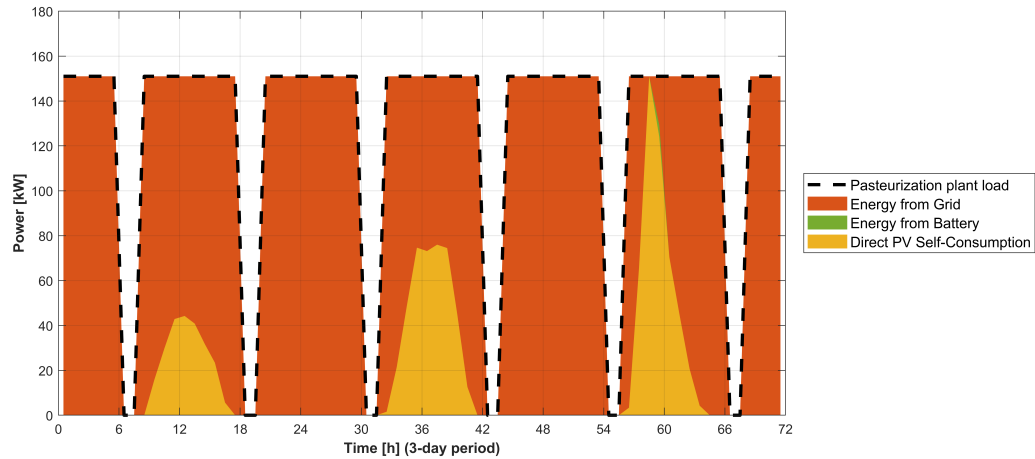


Figure 29: Analysis of hourly electricity demand coverage: January 14-16 (LT case)

Figure 30 illustrates a scenario characterized by intermediate PV generation, relative to summer and winter conditions.

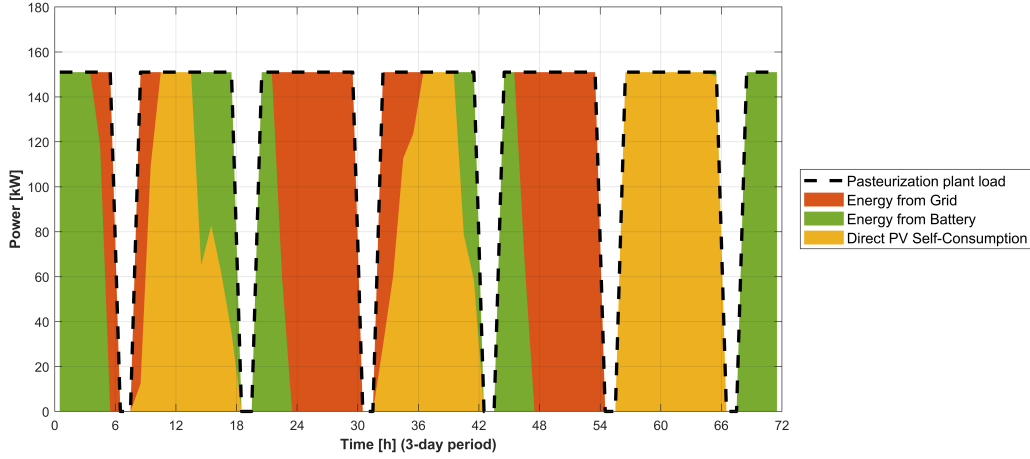


Figure 30: Analysis of hourly electricity demand coverage: March 25-27 (LT case)

5.4.2 Performance analysis of the PV system: IT case ($T_{past} = 85\text{ }^{\circ}\text{C}$)

The second proposed PV system has a net Annual Energy Production (AEP) of $1931\text{ }MWh_{el}/year$.

Figure 31 illustrates both the monthly energy production of the PV plant and the electrical load of the pasteurization plant.

This second case results in a proportionally scaled-up system, both in terms of PV plant size and required load (the latter settling around $145\text{--}150\text{ }MWh$ per month, with a request of $241\text{ }kW_{el}$ for 20 hours a day). The window of energy self-sufficiency is slightly wider, extending from March to September. The production peak occurs between April and May, exceeding $225\text{ }MWh$. However, even in this configuration, the colder period (from October to February) is characterized by a marked generation deficit compared to consumption.

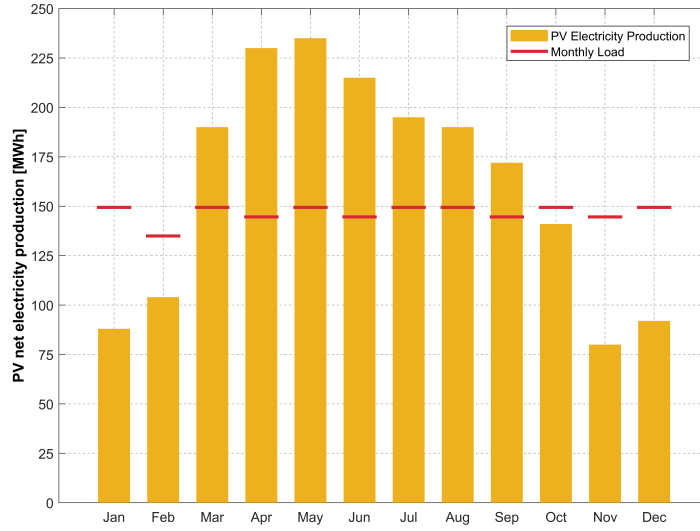


Figure 31: AEP by month (IT case)

With the IT case, analogous results to the LT case are obtained, therefore the same considerations made previously apply. In Appendix E it is possible to find the three-day graphs of the summer, winter and intermediate cases.

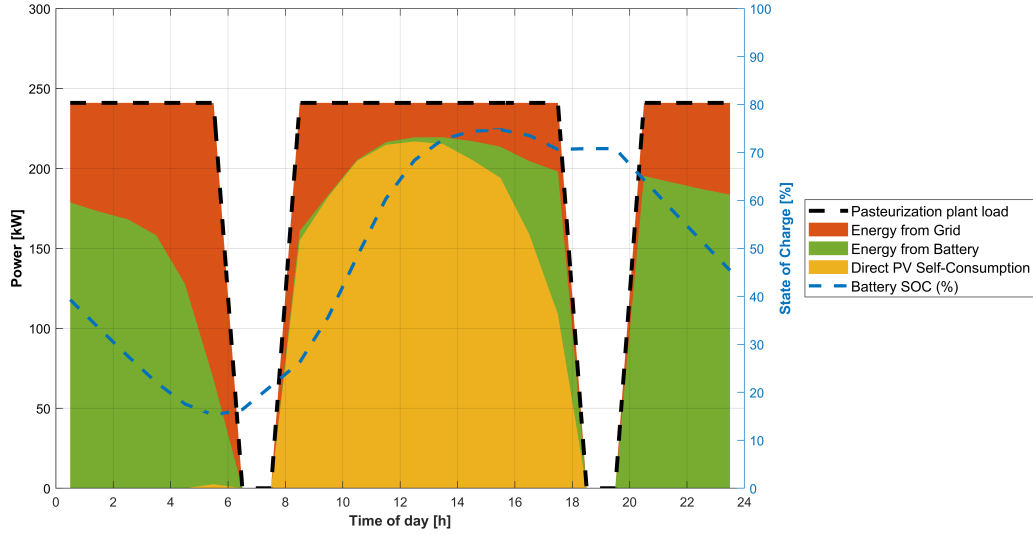


Figure 32: Annual average daily electricity demand coverage (IT case)

5.4.3 Performance analysis of the PV system: HT Case ($T_{past} = 125\text{ }^{\circ}\text{C}$)

The third proposed PV system has a net Annual Energy Production (AEP) of $5411\text{ MWh}_{el}/\text{year}$.

Figure 33 illustrates both the monthly energy production of the PV plant and the electrical load of the pasteurization plant.

The third case, with a load of 672 kW_{el} for 20 hours a day, is characterized by the strong seasonality already highlighted in the previous cases: a relevant mid-season and summer surplus (with monthly productions consistently above 600 MWh between April and July) contrasts with a deep winter deficit. In the critical months of November, December, and January, solar production manages to cover only about half of the monthly energy demand required by the system.

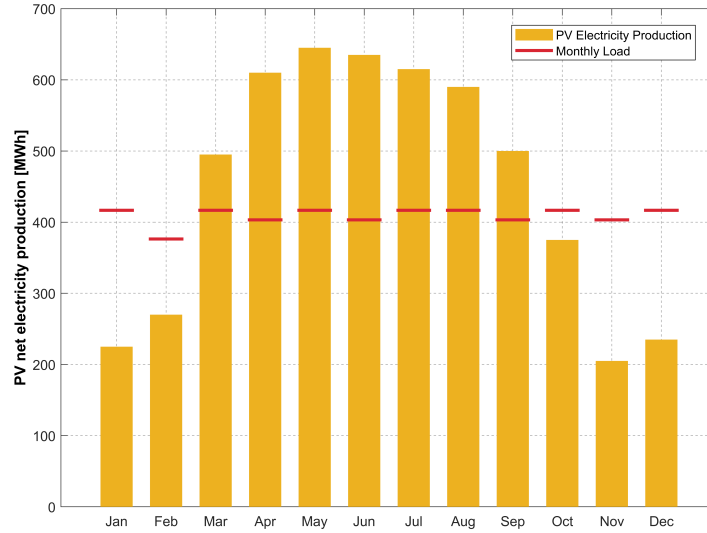


Figure 33: AEP by month (HT case)

With the HT case, analogous results to the LT and IT cases are obtained, therefore the same considerations made previously apply. In Appendix E it is possible to find the three-day graphs of the summer, winter and intermediate cases.

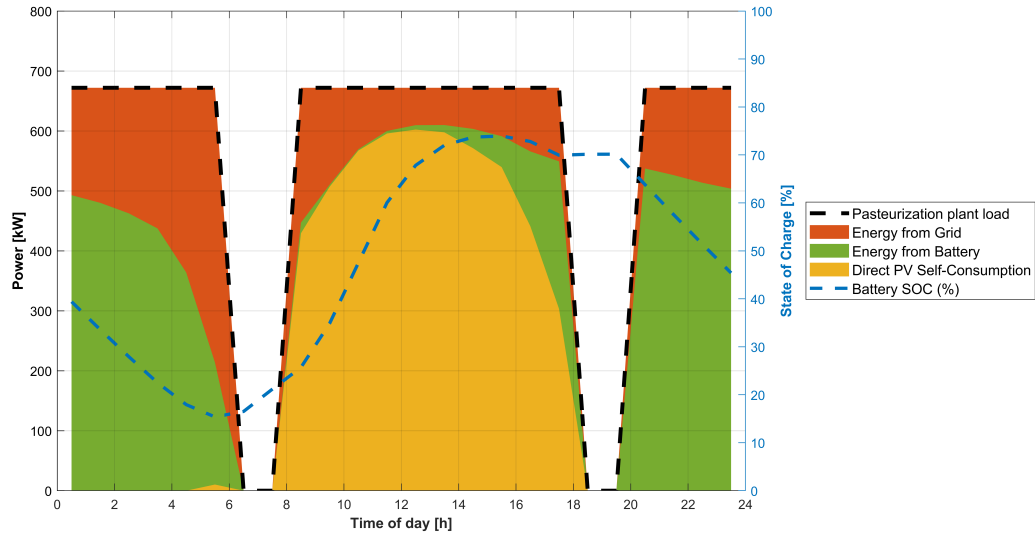


Figure 34: Annual average daily electricity demand coverage (HT Case)

5.4.4 Main Outcomes of the PV Analysis

The main strength of the PV analysis lies in its hourly modeling. The results show that the load can be almost entirely met by direct PV production during the day and by batteries at night, achieving full energy autonomy on sunny days.

A current limitation is that the available land areas are assumed, although they represent realistic scenarios (as shown in Figure 17). Finally, to provide a complete overview, economic considerations need to be integrated into the analysis.

6. Conclusions and Future Developments

The objective of this thesis is to investigate the integration of High-Temperature Heat Pumps with large temperature lifts for Combined Heating and Cooling in the dairy pasteurization industry, evaluating their techno-economic feasibility and the benefits in terms of Primary Energy Saving and avoided emissions.

The complete electrification of the thermal demand makes the coupling of the heat pump with a photovoltaic system both possible and highly advantageous, therefore for each proposed pasteurization plant configuration, a preliminary design of a PV plant has been conducted.

An extensive parametric analysis was carried out, analyzing 20 different low-environmental-impact working fluids, three heat pump configurations, and three distinct pasteurization temperatures. The aim was to identify the optimal configurations in terms of both technical feasibility and thermodynamic performance.

For the first and second proposed configurations (with pasteurization temperatures of 72°C and 85°C , respectively), the optimal layout is a two-stage cycle using R1234ze(E) as the working fluid. These two configurations achieve a combined Coefficient of Performance of 4.03 and 3.30, resulting in a PES of 36% and 23%, and avoided emissions of 56% and 48%, respectively.

For the process at 125°C , the best-performing solution is a cascade cycle using Cyclopentane and Dimethylether as working fluids. This system achieves a combined COP of 2.58, a PES of 4%, and a reduction in emissions of 35%.

From the economic analysis, it emerged that, in terms of CAPEX, the cost difference between the proposed heat pump and a traditional vapour-compression chiller is almost entirely attributable to the high-temperature compressors. Moreover, the investment required for the Thermal Energy Storage is comparable to the cost of a conventional gas boiler. The additional cost of the proposed system is quickly offset by the significant reduction in OPEX, mainly due to the replacement of natural gas with electricity. Consequently, all the selected configurations exhibit highly encouraging payback periods, ranging from approximately one to two years.

Assuming available areas of 1, 1.6, and 4 hectares, the photovoltaic systems proposed for the three cases have installed capacities of 990 kW_{DC} , 1650 kW_{DC} , and 4165 kW_{DC} , respectively. An hourly-based performance analysis revealed that, on an annual average, the energy demand during daylight hours can be met almost entirely (around 85 %) through direct PV self-consumption and battery discharging. During nighttime operations, a substantial portion of the load (around 60%) is covered by the energy stored in the batteries, while the remainder is drawn from the grid.

Overall, it can be concluded that Combined Heating and Cooling High-Temperature Heat Pumps can be effectively integrated into milk pasteurization plants because they perfectly match the simultaneous heating and cooling demands of the process. Furthermore, the short payback periods make the construction of the new plant economically viable. The integration of the PV system is highly effective in ensuring the actual decarbonization of the facility.

Future developments of this study could involve the investigation of alternative heat pump configurations, such as transcritical cycles, or the assessment of refrigerant mixtures' performance. Furthermore, future works could focus on the detailed economic analysis of the PV integration, or the techno-economic evaluation of coupling the system with other renewable energy sources, such as solar thermal collectors. Finally, it would be interesting to apply the same methodology to other applications which require CHC, including building space conditioning, chocolate production, pharmaceutical processing, detergent production, and liquor distillation.

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The diagram illustrates a two-stage heat pump system for milk pasteurization. The system consists of the following components and flow paths:

- Two Stage Heat Pump:** Composed of a Flash Tank and a Recuperative Heat Exchanger. It is driven by two compressors, $\dot{W}_{comp_LP}$ and $\dot{W}_{comp_HP}$.
- Heat Exchangers:**
 - Secondary HX 1:** Preheats milk (flow 5 to 6) using condenser water (flow 4 to 5).
 - Secondary HX 2:** Cools pasteurized milk (flow 9 to 8) using evaporator water (flow 8 to 9).
 - Regenerative section:** Recovers heat from the milk stream (flow 7 to 8) to preheat the feed milk (flow 2 to 3).
- Thermal Energy Storage (TES):** Stores heat from the condenser (flow 5 to 6) and releases it to the evaporator (flow 6 to 7) during off-peak hours.
- Milk Processing:** Milk enters at $\dot{m}_{milk}$ (4 °C) at point 1, passes through the regenerative section and Secondary HX 1, then through a Holding Tube (with thermal losses HT) to become pasteurized milk at point 9 (4 °C).
- Refrigerant Cycle:** The cycle includes an evaporator, a flash tank, a condenser, and a recuperative heat exchanger. Key points in the cycle are numbered 1 through 9b.
- Parameters:**
 - Mass flow rates: $\dot{m}_{water_evap}$, $\dot{m}_{REF_LP}$, $\dot{m}_{REF_HP}$, $\dot{m}_{water_cond}$, $\dot{m}_{water_TES}$, $\dot{m}_{water_HX1}$, $\dot{m}_{milk}$.
 - Temperatures: $T_{w_out_evap}$ (2 °C), $T_{w_in_evap}$, $T_{w_in_cond}$, $T_{w_out_cond}$, T_{past} .
 - Heat transfer rates: $\dot{Q}_{Evap}$, $\dot{Q}_{Cond}$, $\dot{Q}_{HX1}$, $\dot{Q}_{HX2}$, $\dot{Q}_{REG}$, $\dot{Q}_{TES}$.

The diagram illustrates a milk pasteurization system. It features a **Cascade Heat Pump** with two stages: a bottom stage (compressor $\dot{W}_{comp_bottom}$, evaporator 1, condenser 2) and a top stage (compressor $\dot{W}_{comp_top}$, evaporator 5, condenser 6). The refrigerant flow is indicated by blue arrows and labels $\dot{m}_{REF_bottom}$ and $\dot{m}_{REF_top}$. The heat pump is integrated with three heat exchangers: **Secondary HX 2** (evaporator of the bottom stage), **Regenerative section** (condenser of the bottom stage and evaporator of the top stage), and **Secondary HX 1** (condenser of the top stage). The system includes **Thermal Energy Storage (TES)** and a **Holding Tube**. Water flows from TES through the condenser of the top stage, then through a valve and pump to Secondary HX 1, and finally to the holding tube. The holding tube has a temperature T_{past} and experiences thermal losses $\dot{Q}_{loss}$. The milk flow starts at $\dot{m}_{milk}$ at 4°C , passes through Secondary HX 2, then the regenerative section, and finally Secondary HX 1 before entering the holding tube. The pasteurized milk exits at 9°C . The water flow from the holding tube passes through Secondary HX 1, then a valve and pump to the condenser of the top stage, and finally to the condenser of the bottom stage, which is cooled by water at 2°C ($T_{w_out_evap}$). The water flow from the condenser of the bottom stage is labeled $\dot{m}_{water_evap}$. The heat transfer rates are labeled $\dot{Q}_{Evap}$, $\dot{Q}_{HX}$, $\dot{Q}_{Cond}$, $\dot{Q}_{TES}$, $\dot{Q}_{HX1}$, $\dot{Q}_{HX2}$, and $\dot{Q}_{REG}$.

V

B. Appendix B: Holding Tube

Correct heat treatment requires that the milk is held for a specific time at pasteurization temperature. This is done in an external holding cell that contains a holding tube. It consists of a pipe arranged in a spiral pattern covered by a metal shroud. The length of the pipe is calculated so that the time necessary to flow from inlet to outlet is equal to the holding time:

$$Volume_{holding_tube} = \frac{\dot{V} \cdot holding_time}{\eta} \quad (23)$$

$$L_{holding_tube} = \frac{4 \cdot Volume_{holding_tube}}{\pi \cdot D_{int}^2} \quad (24)$$

where η is equal to 0.85 [27] and represents the efficiency factor that takes into account the inhomogeneous milk velocity inside the tube. Table 28 shows the holding tube length corresponding to different pasteurization temperatures.

$T_{pasteurization}$	holding time	$L_{holding_tube}$
72 °C	20 s	51,6 m
85 °C	15 s	38,7 m
125 °C	4 s	10,4 m

Table 28: Holding tube length for different pasteurization processes.

The temperature is set to ensure that, at the holding tube outlet, it is at least equal to the pasteurization temperature, guaranteeing an effective process. Consequently, it is necessary to compute the thermal losses along the holding tube to properly set the operating temperatures.

The simplified model for calculating the heat losses across the holding tube is presented below.

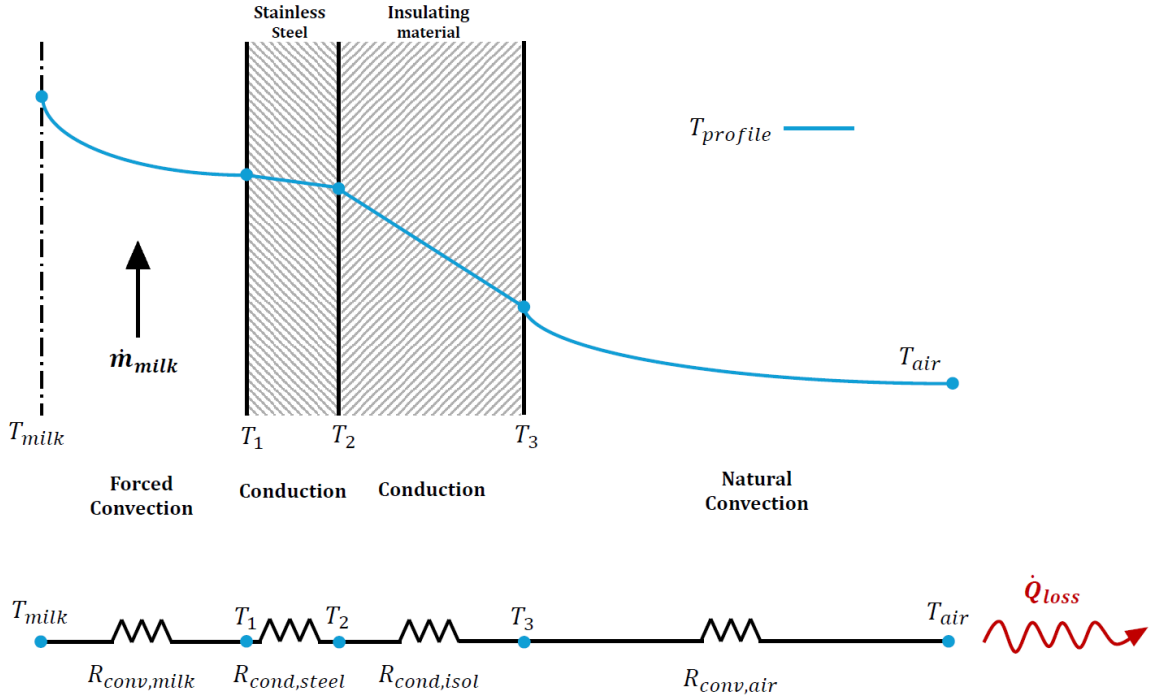


Figure 37: Thermal losses scheme for holding tube

$$\Delta T_{\text{drop,holding_tube}} = \frac{\dot{Q}_{\text{loss,holding_tube}}}{\dot{m}_{\text{milk}} \cdot c_{p,\text{milk}}(T_{\text{pasteurization}})} \quad (25)$$

$$\dot{Q}_{\text{loss,holding_tube}} = \frac{T_{\text{pasteurization}} - T_{\text{air}}}{R_{\text{tot}}} \quad (26)$$

$$R_{\text{tot}} = R_{\text{conv,milk}} + R_{k,\text{steel}} + R_{k,\text{isol}} + R_{\text{conv,air}} \quad (27)$$

$$R_{\text{conv,milk}} = \frac{1}{h_{\text{milk}} \cdot A_{\text{lat,int}}} \quad (28)$$

$$R_{k,\text{steel}} = \frac{\ln\left(\frac{D_{\text{est,steel}}}{D_{\text{int,tube,milk}}}\right)}{2 \cdot \pi \cdot k_{\text{wall}} \cdot L_{\text{holding_tube}}} \quad (29)$$

$$R_{k,\text{isol}} = \frac{\ln\left(\frac{D_{\text{est,tot}}}{D_{\text{est,steel}}}\right)}{2 \cdot \pi \cdot k_{\text{isol}} \cdot L_{\text{holding_tube}}} \quad (30)$$

$$R_{\text{conv,air}} = \frac{1}{h_{\text{air}} \cdot A_{\text{lat,est}}} \quad (31)$$

The fouling resistance is considered negligible, thanks to the frequent CIP process.

Steel thickness is equal to 2 mm, insulating material thermal conductivity is equal to 0.04 W/mK and its thickness is equal to 32 mm.

Thermophysical properties of milk and of stainless steel are specified in Appendix C.

The described calculations demonstrate that the temperature drop across the holding tube is consistently negligible (in the order of 10^{-2} °C). Nevertheless, developing a separate model was deemed necessary due to the critical nature of this variable.

C. Appendix C: Heat Exchangers Sizing

The methodology described in this section applies to all heat exchangers within the plant (evaporator, condenser, internal heat exchanger, recuperative heat exchanger, and secondary heat exchangers 1 and 2), which are all plate heat exchangers.

Munir et al. [55] reported milk properties (thermal conductivity λ , density ρ , viscosity μ and specific heat capacity c_p) as function of temperature and milk composition. These properties are necessary to calculate the overall heat transfer coefficients of the heat exchangers.

$$\lambda = (326.58 + 1.0412T - 3.37 \times 10^{-3}T^2) (4.6 \times 10^{-1} + 5.4 \times 10^{-1}X_{\text{Water}}) 1.73 \times 10^{-3} \quad (32)$$

$$\rho = 1,042.01 - 0.37T + 3.6 \times 10^{-4}T^2 \quad (33)$$

$$\mu = (0.9565 - 1.3004 \times 10^{-3}T + 1.9580 \times 10^{-4}T^2) + X_{\text{Fat}} (0.4766 - 1.144 \times 10^{-2}T + 7.2642 \times 10^{-5}T^2) \quad (34)$$

$$c_p = 3,744.48 + 1.15T + 3.93 \times 10^{-3}T^2 \quad (35)$$

Where $X_{\text{Water}} = 87.3\%$ and $X_{\text{Fat}} = 4.4\%$ for cow milk [27].

Stainless steel AISI 316 thermal conductivity k is calculated as follows [56]:

$$k_{\text{steel}} = 14.6 + 0.0127 \cdot T \quad (36)$$

Starting from the calculations described in Section 4, it is possible to derive the value of the product of the overall heat transfer coefficient (U) and the heat transfer area (A) at design conditions:

$$(UA)_{\text{design}} = \frac{\dot{Q}}{LMTD} \quad (37)$$

The Heat Exchanger is divided into sections with odd number of plates, to obtain an equal number of channels both for the hot and for the cold fluid. For each section, the main dimensionless numbers (Reynolds, Prandtl and Nusselt) are evaluated to determine the convective heat transfer coefficient (h):

Single Phase HX [57]

$$\text{Re} = \frac{\rho v_{\text{channel}} L_c}{\mu} \quad (38)$$

$$\text{Pr} = \frac{\mu c_p}{k} \quad (39)$$

$$\text{Nu} = 0.295 \text{Re}^{0.64} \text{Pr}^{0.32} \cdot \left(\frac{\pi}{2} - \beta \frac{\pi}{180} \right)^{0.09} \quad (40)$$

$$h = \frac{\text{Nu} k}{L_c} \quad (41)$$

Condenser [57]

$$G = \frac{m}{n_{ch} W b} \quad (42)$$

$$\text{Re}_{eq} = G \left[(1-x) + x \left(\frac{\rho_l}{\rho_v} \right)^{\frac{1}{2}} \right] \frac{D_h}{\mu_l} \quad (43)$$

$$h = 1.875 \phi \frac{\kappa_l}{D_h} \text{Re}^{0.445} \text{Pr}^{\frac{1}{3}} \quad (44)$$

Evaporator [58]

$$G_{eq} = G \left[(1-X_m) + X_m \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right] \quad (45)$$

$$\text{Re}_{eq} = \frac{G_{eq} D_h}{\mu_l} \quad (46)$$

$$\text{Bo}_{eq} = \frac{q_w''}{G_{eq} \cdot i_{fg}} \quad (47)$$

$$\text{Nu}_r = 1.926 \text{Pr}_l^{1/3} \text{Bo}_{eq}^{0.3} \text{Re}^{0.5} \cdot \left[(1-X_m) + X_m \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right] \quad (48)$$

$$h = \frac{\text{Nu}_r k_l}{D_h} \quad (49)$$

Single-phase HXs pressure drop calculations:

$$\Delta p_{\text{tot}} = \Delta p_{\text{port}} + \Delta p_{\text{grav}} + \Delta p_{\text{friction}} \quad (50)$$

$$\Delta p_{\text{port}} = 1.5 \cdot \left(\frac{\rho V^2}{2} \right) \quad (51)$$

$$\Delta p_{\text{grav}} = \begin{cases} N_{\text{sections}} \text{ even} \longrightarrow \Delta p_{\text{grav}} = 0 \\ N_{\text{sections}} \text{ odd} \longrightarrow \Delta p_{\text{grav}} = \pm \rho g L \end{cases} \quad (52)$$

$$\Delta p_{\text{friction}} = 4 \cdot f \cdot \left(\frac{L_{\text{plate}}}{D_h} \cdot \frac{\rho V^2}{2} \right) \times N_{\text{sections}} \quad (53)$$

$$f = \left(\frac{\beta}{30} \right)^{0.83} \cdot \left[\left(\frac{30.2}{\text{Re}} \right)^5 + \left(\frac{6.28}{\text{Re}^{0.5}} \right)^5 \right]^{0.2} \quad (54)$$

D. Appendix D: Combined COP results from parametric analysis

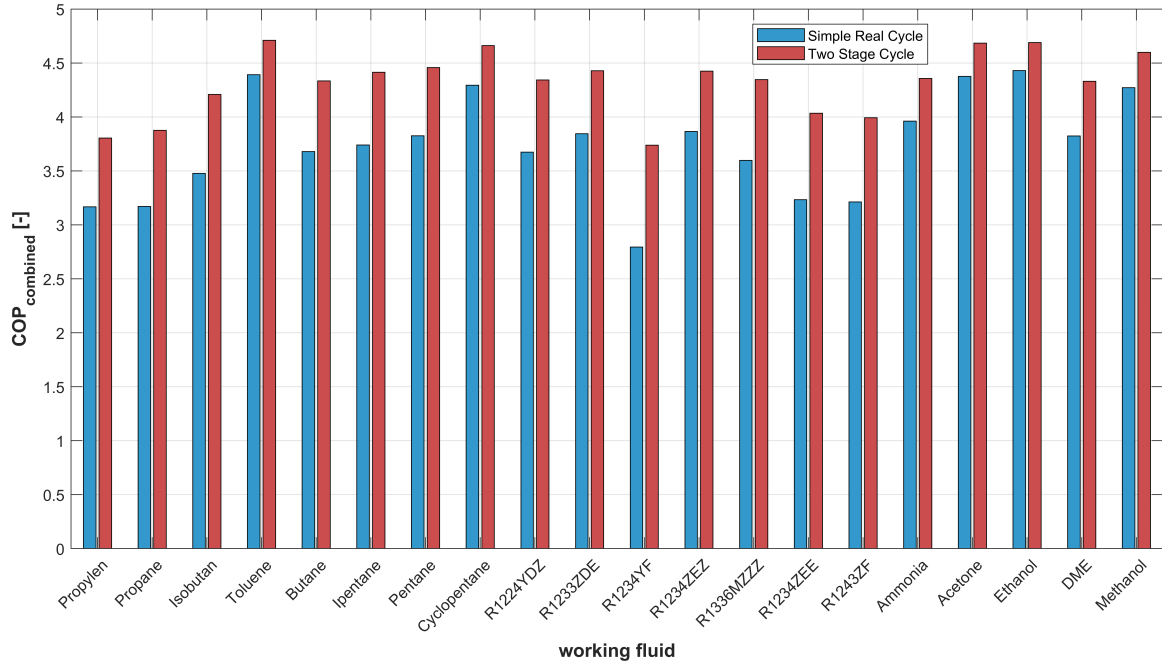


Figure 38: Simple Real Cycle vs Two Stage Cycle performance comparison with $T_{past} = 72^{\circ}\text{C}$

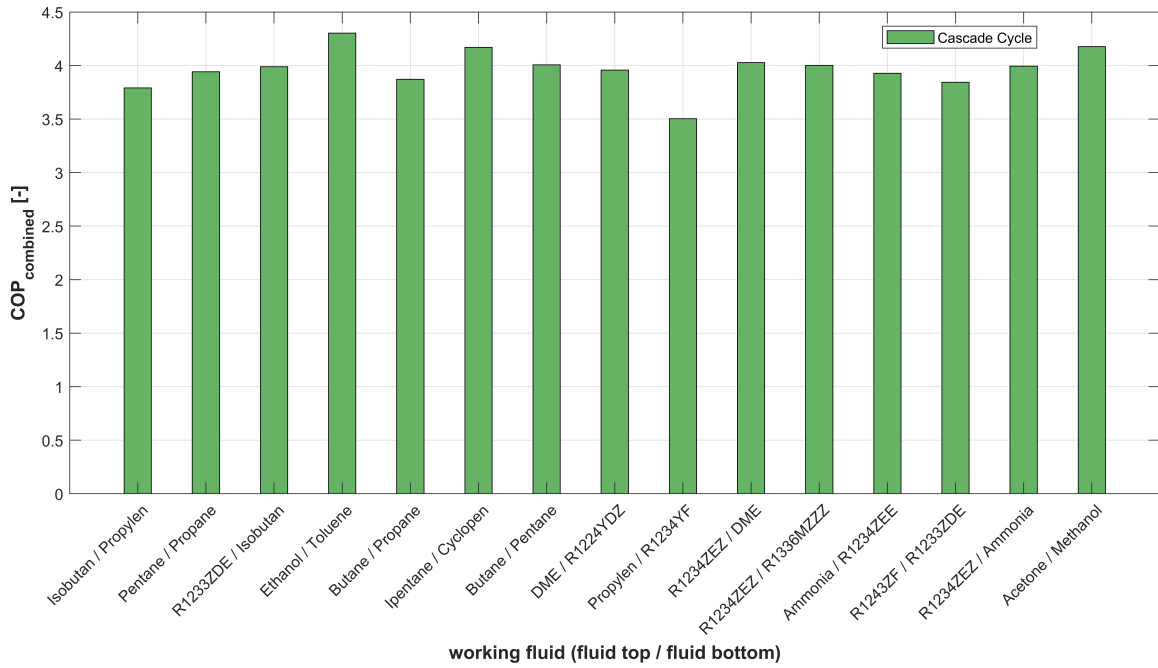


Figure 39: Cascade Cycle performance comparison with $T_{past} = 72^{\circ}\text{C}$

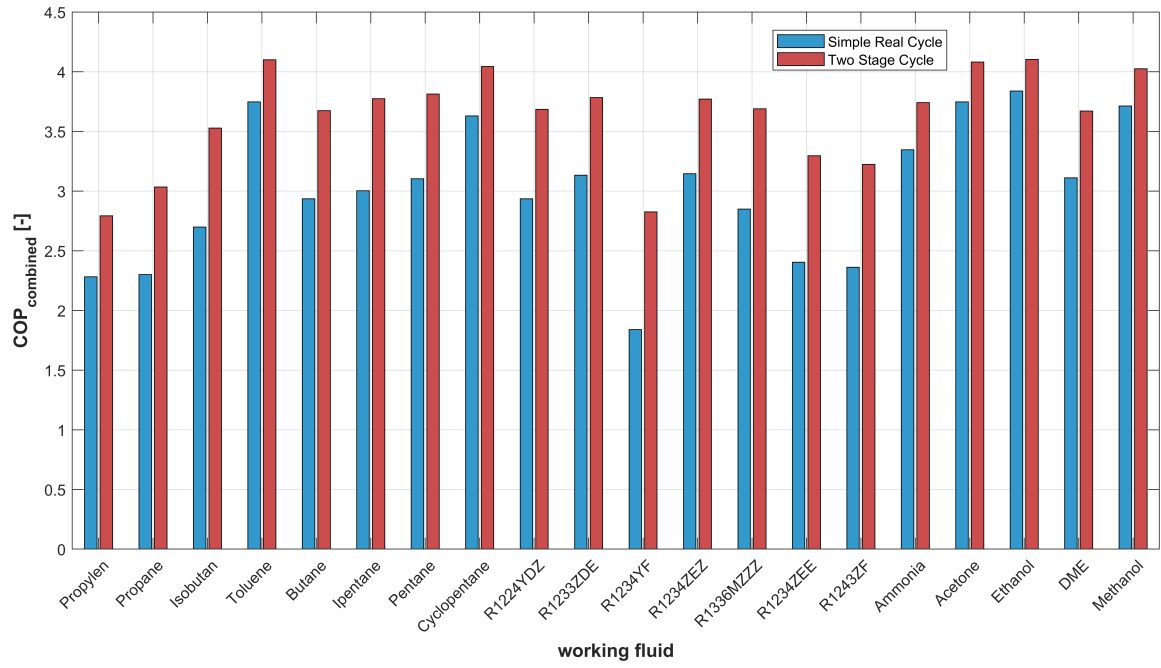


Figure 40: Simple Real Cycle vs Two Stage Cycle performance comparison with $T_{past} = 85^{\circ}\text{C}$

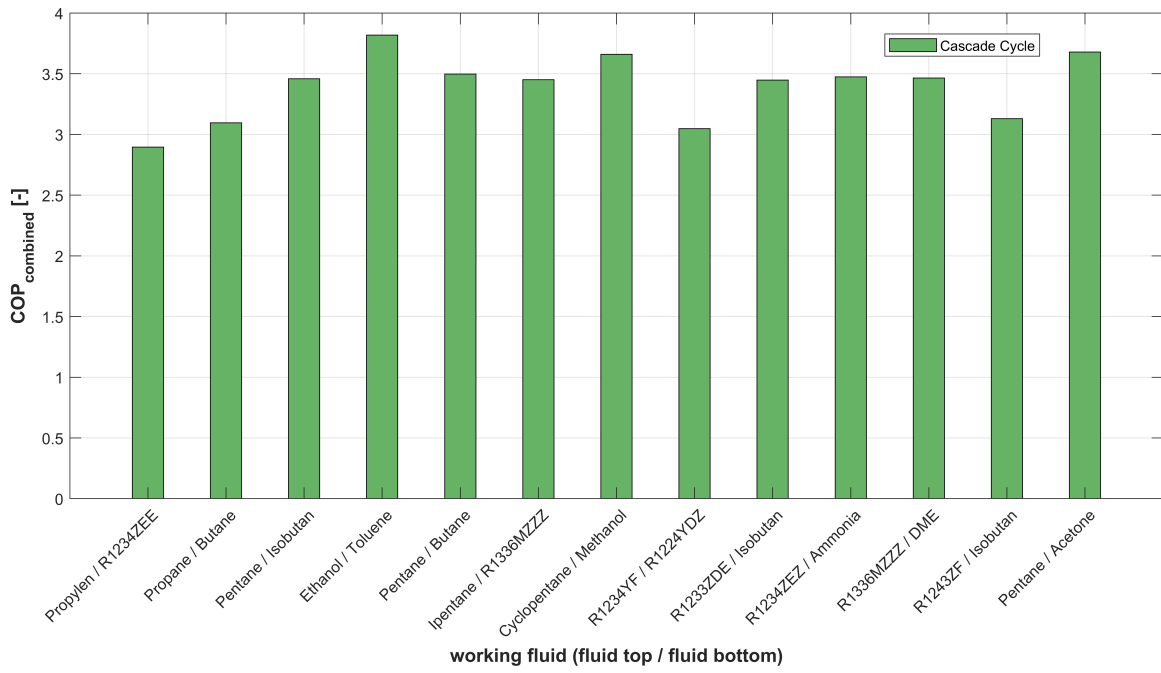


Figure 41: Cascade Cycle performance comparison with $T_{past} = 85^{\circ}\text{C}$

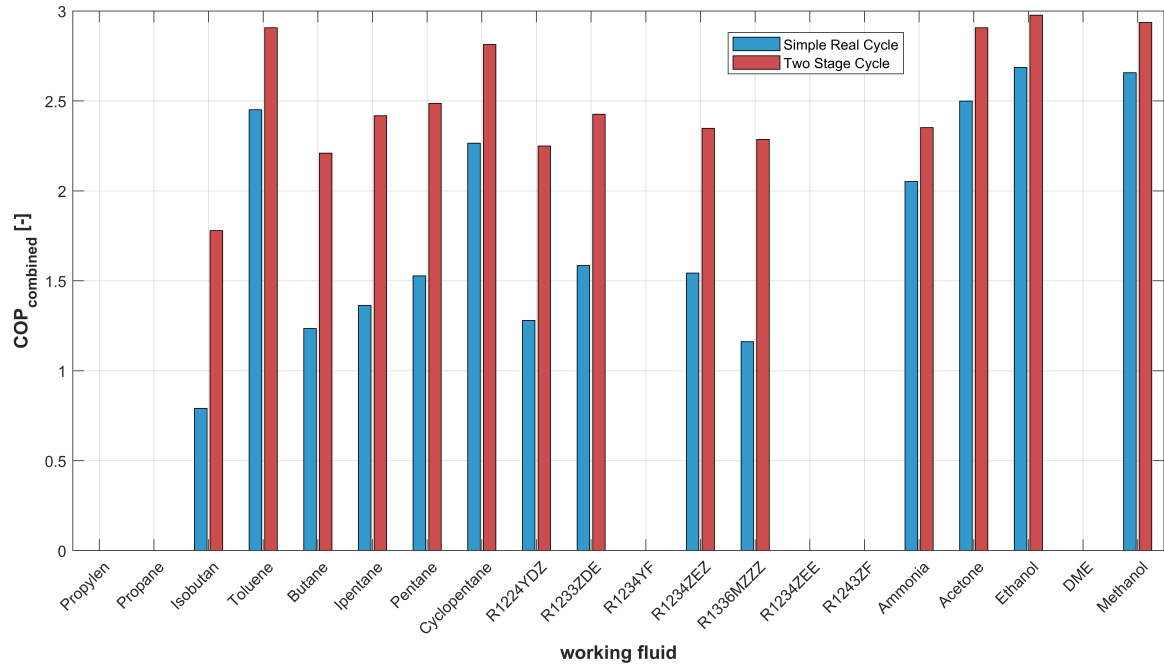


Figure 42: Simple Real Cycle vs Two Stage Cycle performance comparison with $T_{past} = 125^{\circ}\text{C}$

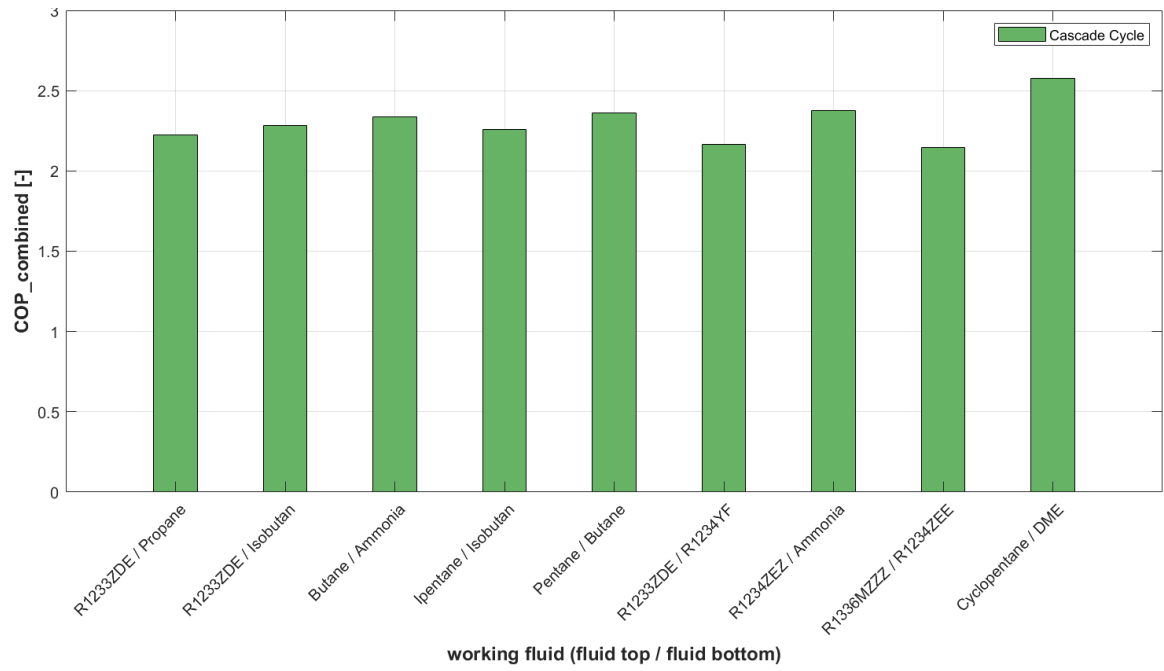


Figure 43: Cascade Cycle performance comparison with $T_{past} = 125^{\circ}\text{C}$

The configuration is not feasible where the corresponding column is absent.

E. Appendix E

Three examples (summer, winter, and intermediate cases) illustrating the hourly coverage of the plant's electricity demand for the case with a pasteurization temperature of 85 °C are presented below.

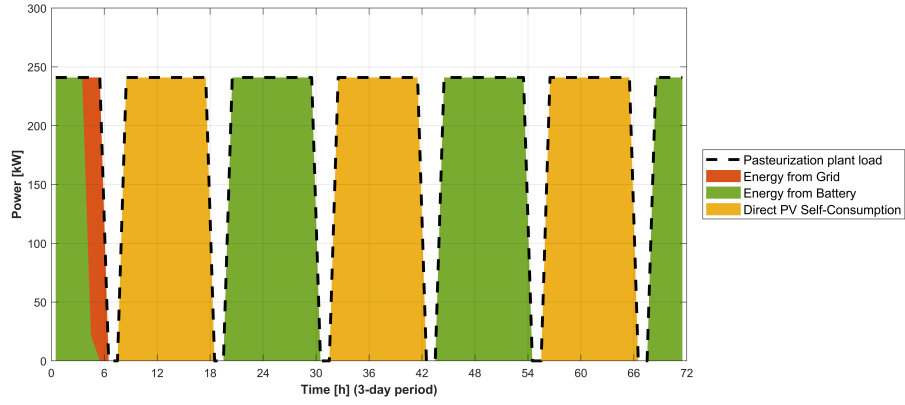


Figure 44: Analysis of hourly electricity demand coverage: July 28-30 (IT case)

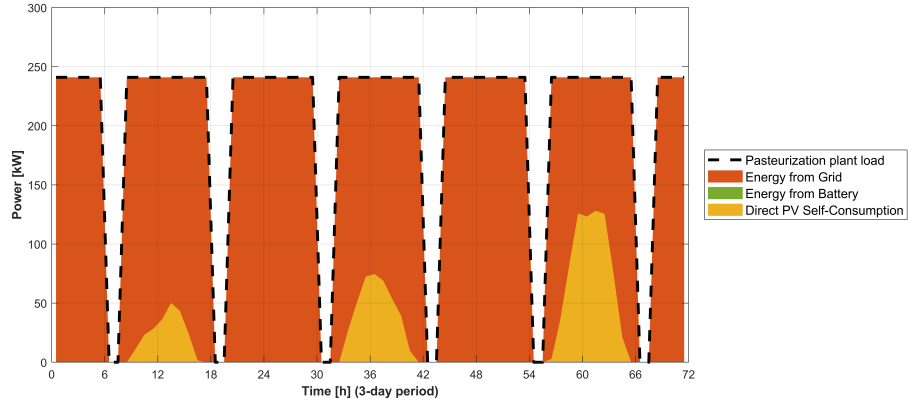


Figure 45: Analysis of hourly electricity demand coverage: January 14-16 (IT case)

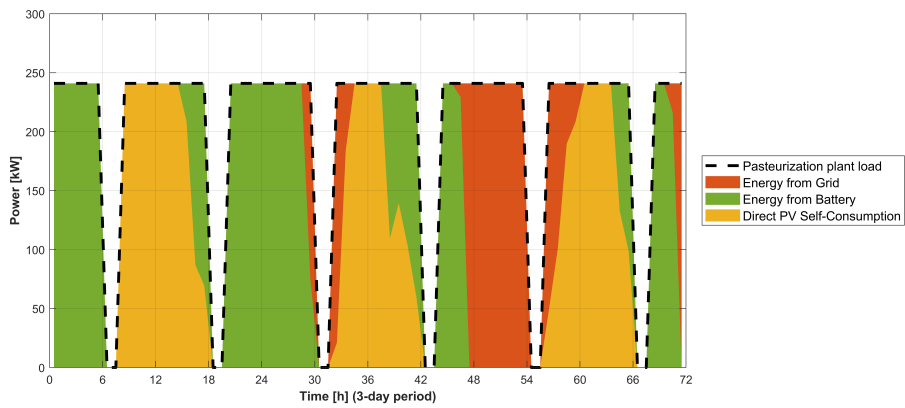


Figure 46: Analysis of hourly electricity demand coverage: March 25-27 (IT case)

Three examples (summer, winter, and intermediate cases) illustrating the hourly coverage of the plant's electricity demand for the case with a pasteurization temperature of 125 °C are presented below.

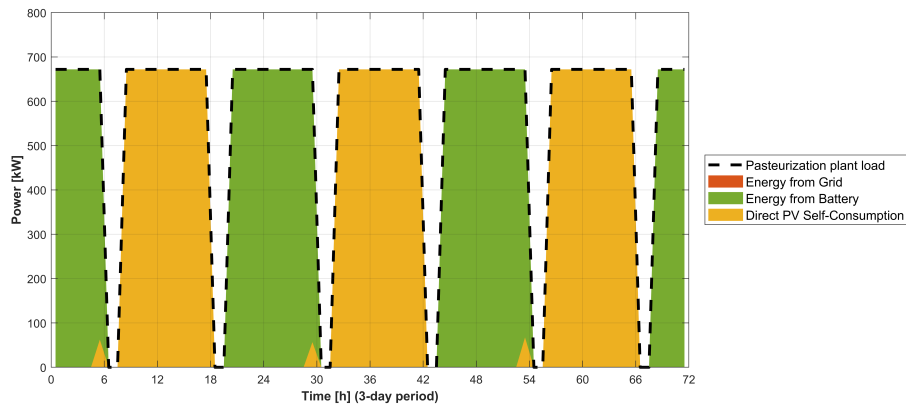


Figure 47: Analysis of hourly electricity demand coverage: July 28-30 (HT case)

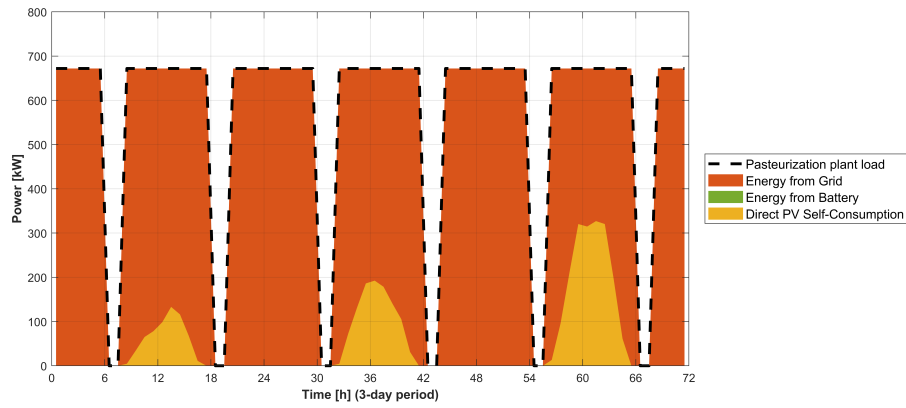


Figure 48: Analysis of hourly electricity demand coverage: January 14-16 (HT case)

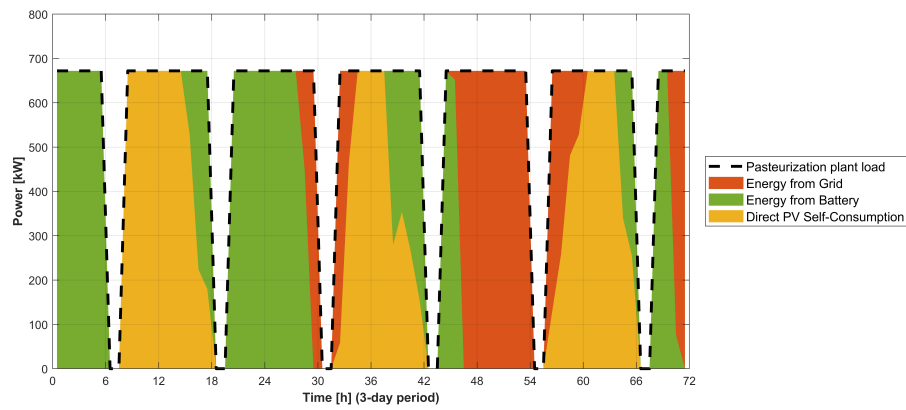


Figure 49: Analysis of hourly electricity demand coverage: March 25-27 (HT case)

Abstract in lingua italiana

Le pompe di calore ad alta temperatura, in grado di fornire contemporaneamente potenza termica e potenza frigorifera, rappresentano una soluzione concreta per la decarbonizzazione dei processi industriali. Questa tesi ne valuta l'integrazione all'interno degli impianti di pastorizzazione del latte, dove simultaneamente vengono richiesti riscaldamento ad alta temperatura (da 4 °C a più di 72 °C) e raffreddamento (dalla temperatura di pastorizzazione a 4°C). L'impianto proposto integra un sistema di accumulo termico che permette di recuperare il calore in eccesso fornito dal condensatore, per sfruttarlo in altre applicazioni in un momento successivo. È stata condotta un'analisi parametrica su diverse configurazioni impiantistiche, considerando 20 refrigeranti a basso impatto ambientale e tre temperature di pastorizzazione, al fine di individuare le soluzioni ottimali. Per i processi a 72 °C e 85 °C, la configurazione più performante è un ciclo a due stadi con refrigerante R1234ze(E), che raggiunge rispettivamente un coefficiente di performance combinato pari a 4.03 e 3.30. Queste soluzioni portano ad un risparmio di energia primaria del 36 % e 23 % e ad una riduzione delle emissioni del 56 % e 48 % rispetto ai sistemi convenzionali alimentati a gas. Per il caso di pastorizzazione a 125 °C, la soluzione ottimale è un ciclo a cascata con Ciclopentano e Dimetiletere, con coefficiente di performance combinato di 2.58, risparmio di energia primaria del 4 % e riduzione delle emissioni del 35 %. L'analisi economica evidenzia tempi di ritorno dell'investimento compresi tra uno e due anni per tutte le configurazioni selezionate, confermando l'elevata redditività della proposta. La completa elettrificazione della fornitura termica consente inoltre l'integrazione con impianti fotovoltaici, aumentando l'autoconsumo e abilitando una decarbonizzazione efficace e strutturale dell'intero processo produttivo.

Parole chiave: Pompe di calore ad alta temperatura, Decarbonizzazione dei processi industriali, Utilizzo combinato di potenza termica e frigorifera, Pastorizzazione del latte.