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Research paper

Standardizing industrial high-temperature heat pumps: A techno-economic analysis of modular concepts for paper and spray drying ☆

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ABSTRACT

High-temperature heat pumps (HTHP) present a promising pathway for decarbonizing industrial process heat at temperatures up to 200 °C. However, widespread deployment remains constrained by high integration costs and a distinct lack of prior research on modular concepts for HTHPs and their direct impact on cost reduction. This study addresses this gap through a techno-economic analysis targeting two specific drying processes in the paper and food industries. Thermal demands derived from market data establish the sizing basis for simulating 1 MW and 2 MW base modules. These baseline configurations are systematically scaled to 10 MW. Thermodynamically derived compressor requirements are cross-referenced with market available technologies. Performance validation confirms technical feasibility at COPs of 1.84 and 2.54. Standardized piston and screw compressor units adequately cover capacities up to 5 MW. Techno-economic modeling demonstrates capital recovery in less than one year against electric heating and 6 years for the 2 MW system versus natural gas-fired systems at €350/kW specific investment costs. Component standardization with 11 % capital expenditure reduction further accelerates payback by roughly one year. These results show that modular standardization significantly improves cost-effectiveness and thus provides a solid framework for the accelerated industrial adoption of HTHP technologies. The proposed modular framework offers a scalable design methodology for future work.

1. Introduction

In recent years, the use of high-temperature heat pumps (HTHP) to provide process heat in industry has become increasingly important for achieving climate goals. Electrification of industrial process heat through the use of HTHPs can reduce the consumption of fossil raw materials and lower the generation of process-related carbon dioxide (CO_2) emissions. In 2022, 9 Gt of CO_2 were emitted in the industrial sector, which corresponds to about a quarter of emissions from the global energy system. Despite a slight decline in emissions in recent years, current measures are not sufficient to achieve the net zero emissions (NZE) by 2050 target, which aims a reduction in CO_2 emissions to 7 Gt by 2030. The industrial sector is responsible for around a quarter of energy-related CO_2 emissions from fossil raw materials worldwide (IEA, 2025). In Europe, around 60 % of the energy in industry is used to generate process heat (European Heat Pump Association, 2025a). In order to achieve further emission reductions and drive forward decarbonization, the use of HTHPs for process heat generation in industry is a crucial step. In the paper and food industry, process heat of up to 200 °C is required for drying processes and steam generation. By processing waste heat from existing industrial processes, market available heat pumps are able to generate temperatures of up to 160 °C. Accordingly, heat pumps can technically cover around 65 % of the process heat demand in the paper industry and around 40 % in the food industry (International Energy Agency (IEA), 2022). In addition to the existing potential, obstacles such as a lack of planning security or economic disadvantages due to current price structures are slowing down the integration of heat pumps in industry. The diversity and demands of industrial processes confront manufacturers with technological challenges. While heat pumps with a capacity of less than 1 MW are often standardized, heat pumps with a capacity between 1 MW and 10 MW are currently highly customized, non-standardized products, resulting in high investment costs and low production capacities. Although HTHPs are already a widely used, climate friendly solution for generating process heat in industry, and standardizing HTHPs in the 1 MW to 10 MW capacity range could reduce the specific costs per unit of installed capacity for

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Abbreviations

C	Compressor
CAPEX	Capital expenditures
COP	Coefficient of performance
CO_2	Carbon dioxide
DECHEMA	Society for Chemical Engineering and Biotechnology
DSM	Design Structure Matrix
E	Expansion device
EB	Electric boiler
EU	European Union
ETS	Emissions Trading System
GB	Gas boiler
GWP	Global warming potential
HC	Hydrocarbons
HCFO	Hydrochlorofluoroolefin
HFC	Hydrofluorocarbons
HFO	Hydrofluorolefin
HP	Heat pump
HTHP	High temperature heat pump
HTHX	High temperature heat exchanger
IAPWS-IF97	Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam
IEA	International Energy Agency
IHX	Internal heat exchanger
LTHX	Low temperature heat exchanger
METUS	Methodological support for system formation
MFD	Modular Function Deployment
MQB	Modular Transverse Matrix
NZE	Net zero emissions
O&M	Operating and Maintenance
ODP	Ozone depletion potential
OPEX	Operational expenditures
PN	Nominal pressure
R600	n-butane
R601	n-pentane
R717	Ammonia
R718	Water
R744	CO_2
TRL	Technology readiness level

Project Names

COMTA
CoPIRIDE
EEETHOS
ENPRO
FUSE
F^3-Factory
Heaten AS
SPIRIT
TASTE

Symbols

C	Costs [€]
CS	Cost share [%]
c	Costs [€/kWh]
e	Specific heat demand [MWh]
m	Production volume [t]
$\dot{m}$	Mass flow [kg/s]
N	Number of facilities [–]
P	Power [kW]
p	Pressure [bar]
Q	Energy [kWh]
$\dot{Q}$	Heat flow [kW]
R	Cost saving potential [%]
r_P	Electricity to gas price ratio [–]
S	Cost savings [€]
T	Temperature [°C, K]
t	Operating hours [h]

Greek Symbols

ϵ	Effectiveness [–]
η	Efficiency [–]
η_{is}	Isentropic efficiency [–]
Π	Pressure ratio [–]
Δ	Difference [–]

Indices

abs	Absolute
av	Average
el	Electric
gas	Gas
i	Index
in	Inlet
Lab	Labor
Main	Maintenance
th	Thermal
tot	Total
out	Outlet
R	Reduction
Ref	Reference
Rep	Replacement
sink	Heat sink
source	Heat source
spec	Specific

larger heat pump systems (Agora Energiewende and Fraunhofer IEG, 2023), there are currently no concepts or approaches in the technical literature for the modularization and standardization of HTHPs. This study examines existing methods and identifies opportunities for standardizing and modularizing heat pumps in the industry based on market opportunities. A modularization concept based on a 2 MW heat pump for a drying process in the paper industry and a 1 MW heat pump for a spray drying process in the food industry is presented. Based on an techno-economical analysis, the payback period for both systems is determined, and the potential cost savings resulting from standardized heat pump concepts are estimated, taking relevant manufacturer data into account. This work helps to overcome the existing gap regarding the lack of modular standardization concepts for HTHPs and highlights their technological and economic potential. The following introduction provides a literature overview of the current market and state of the art in the field of HTHPs, as well as existing approaches to modularization and standardization concepts. Finally, the basic structure and operation of a heat pump are explained.

1.1. Decarbonizing industry with high-temperature heat pumps

Industrial process heat generation in the European Union (EU-27) accounts for 47 % of industrial energy demand and about three-quarters of industrial CO_2 emissions. The main energy sources are fossil fuels such as natural gas and coal, which provide about three-quarters of the energy used for process heat (Fraunhofer ISI, 2024). Based on the study

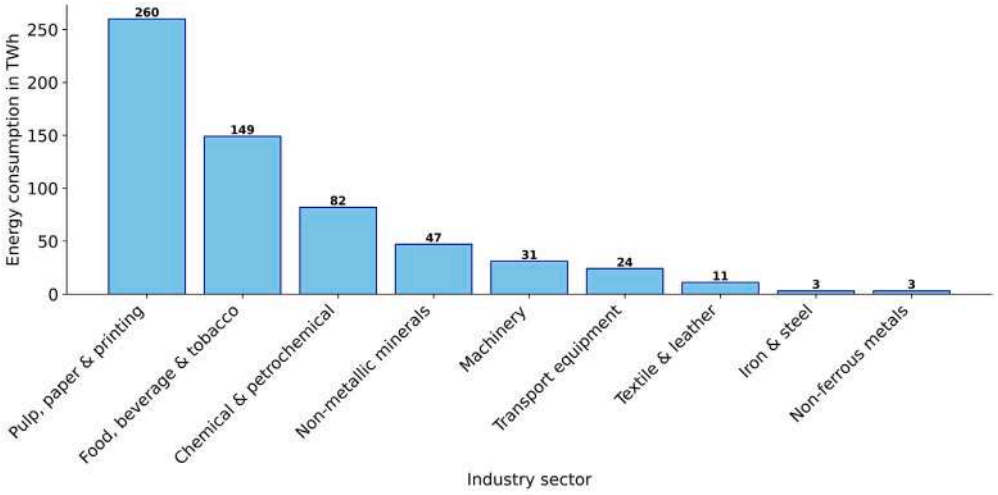


Fig. 1. EU industrial process heat below 200 °C energy consumption in TWh based on Marchetti (2026).

of the direct electrification of industrial process heat by Fraunhofer ISI (Fraunhofer ISI, 2024), electrifying process heat using climate-friendly technologies such as heat pumps would lead to a significant reduction in greenhouse gas and CO_2 emissions caused by fossil fuels and could thus contribute to the decarbonization of industry. Various studies (Pardo et al., 2013; Naegler et al., 2015; Rehfeldt et al., 2017; Marina et al., 2021) have examined the determination of process heat requirements for different temperature ranges across various industrial sectors. These studies concluded that the share of the various temperature levels in total energy demand varies considerably depending on the processes and technologies used in the individual industrial sectors. Process heat demand is generally divided into three temperature ranges: > 500 °C (e.g., for iron and steel production), 100 °C to 500 °C (e.g., in the chemical industry), and < 100 °C (e.g., in the food industry). Rehfeldt et al. (2017) and Marina et al. (2021) have primarily examined the heat demand of industry in the temperature range up to 200 °C and found a high demand in the paper, food, and chemical industries. The authors emphasize that applications in the low- to medium-temperature range up to 200 °C, which are particularly common in the food and paper industries, offer good opportunities for electrification using heat pumps. Given the existing potential, various studies have examined the market potential of HTHPs for the decarbonization of industrial process heat. Kosmadakis (2019) conducted a study on the potential of heat pumps in various industrial sectors in the EU and showed that industrial HTHPs with a coefficient of performance (COP) of 4 can cover approximately 15 % or 28.37 TWh/year of total industrial heat consumption in the temperature range between 100 °C and 200 °C. An industry-specific analysis of total heat consumption at different temperature ranges was carried out for individual industrial sectors. The author determined that for the temperature range between 100 °C and 150 °C, the total annual heat consumption in industry is 192.36 TWh/year. In this range, the food industry accounts for the highest heat consumption, with a share of around 35 %. For the temperature range between 150 °C and 200 °C, the total annual heat consumption was 80.11 TWh/year, of which the paper industry accounts for almost 50 %. In the overall temperature range between 100 °C and 200 °C, the paper industry accounts for around 50 % and the food industry for around 40 % of total heat consumption. In a recent project report by Marchetti (2026) as part of the SPIRIT project (SPIRIT Project, 2025), a comprehensive market analysis examined energy consumption for industrial process heat below 200 °C in the EU. The results show that a significant proportion of process heat below 200 °C is generated in the paper, pulp, and printing industries, as well as in the food and beverage industries. Fig. 1 shows the industrial process heat below 200 °C energy consumption in the EU in TWh/year.

The results of the sector comparison show that the paper, pulp, and printing industries, with an energy consumption of 260 TWh/year, are the largest consumers of low-temperature process heat up to 200 °C. The food and beverage industry is the second-largest consumer, with a consumption of 149 TWh/year. In third place, with a significantly lower share of 82 TWh/year, is the chemical and petrochemical sector. Energy consumption for industrial process heat below 200 °C amounts to 409 TWh/year in the paper and food industries alone, approximately 66.94 %, which is twice as much as the total consumption of all other sectors combined. The study also estimates the market potential for HTHP applications in the paper sector in the EU at approximately 2063 units associated with a weighted average unit size of 12 MW. The cumulative market potential for installed HTHP capacity in the food industry in the EU is estimated to 964 HTHP units at 675 kW capacity for milk powder spray drying applications. Marina et al. (2021) carried out an estimation of the EU industrial heat pump market potential especially for the paper and pulp sector, chemical and refining sector and food sector. The most common processes in these sectors were identified as drying processes e.g., in paper manufacturing and milk powder production, as well as distillation processes for chemical applications. The authors determined that the four sectors studied had a process heat demand of 1123 PJ/year and an available waste heat output of 1130 PJ/year. Based on their research, the authors concluded that the market potential for industrial heat pumps at process temperatures of up to 150 °C amounts to a cumulative heating capacity of 20 GW, distributed across 3229 heat pump units. At process temperatures of up to 200 °C, the cumulative heating capacity is 23 GW, distributed across 4174 heat pump systems. In the study, heat pumps with a median COP of 2.77 were able to cover a total of 641 PJ/year of process heat across the four sectors. Furthermore, it was concluded that around 50% of the total cumulative heating capacity of heat pumps could be covered by units smaller than 10 MW. The International Energy Agency (IEA) (2022) mentions the considerable potential of electric heat pumps for supplying process heat to industry in their world energy outlook special report. Suitable application processes in various temperature ranges and the current Technology Readiness Level (TRL) are presented. In the temperature range from 100 °C to 140 °C, there are already commercial applications with TRL 8 to 9. Typical areas of application in this temperature range are paper drying, evaporation in the food industry, and chemical processes. Process steam can also be generated in the range of 140 °C to 200 °C using heat pumps, whose pre-commercial demonstration and commercial applications are at TRL levels 6 up to 9. The report concludes that around 30 % of the total heat demand of the paper, food, and chemical industries could be met with current heat pump technologies and heat pumps with a total capacity of 15 GW could be used in around 3000 industrial plants in Europe. In summary, the studies examined indicate significant potential for the

electrification of industrial processes across various industrial sectors using heat pumps. The paper and food industries, in particular, have a high demand for process heat and steam for drying processes that require temperatures of up to 200 °C and high thermal capacities. This need can be effectively met by modern heat pump technologies, which help reduce CO_2 emissions caused by the use of fossil fuels.

1.2. State of the art in high-temperature heat pumps

Several studies provide an overview of the state of the art in heat pumps over the last years. The technologies report on Annex 58, Task 1 (Zühlsdorf, 2023) offers a general overview of HTHP technologies and applications. Currently available heat pump technologies have a capacity ranging from 30 kW to 70 kW and can deliver maximum heat sink temperatures between 100 °C and 280 °C. The TRL of the heat pump technologies presented ranges from 4, for high sink temperatures which have been generally validated in the laboratory, to 9, where testing was conducted in an operational environment. Common heat pump technologies include closed-cycle vapor compression systems and refrigerants such as CO_2 (R744), ammonia (R717), water (H_2O,) n-butane (R600) or n-pentane (R601). The report presents 18 case studies that demonstrate that steam-based drying applications are a common area of use for HTHPs in industry. Depending on the heat pump technology, investment costs are reported to range from €200/kW to €1,500/kW. In the study by Fraunhofer IEG and Agora Energiewende (Agora Energiewende and Fraunhofer IEG, 2023) an overview of market available heat pumps with TRL 9 is provided. The spectrum covers temperatures up to 250 °C and a heating capacity range up to 70 MW. The compressor technologies used vary depending on the heating capacity of the heat pump. The study shows that piston compressors or screw and turbo compressors cover a wide range of applications with temperatures of up to 120 °C and heating capacities of 10 MW. Sink temperatures above 200 °C are currently mainly achieved by open heat pump cycles using mechanical vapor compression. Arpagaus et al. (2018) conducted a market study on the state of the art of heat pumps with heat sink temperatures between 90 °C and 160 °C. The authors examined the potential areas of application and concluded that HTHPs offer great potential for use in the paper and food industries due to the temperature range required in these sectors. Drying, sterilization, and evaporation processes are cited as promising options for waste heat recovery. The heat pumps available on the market for providing the required temperatures had heating capacities between 20 kW and 20 MW with a COP between 2.4 and 5.8. Natural refrigerants and hydrocarbons (HC), hydrofluoroolefins (HFO), hydrochlorofluoroolefins (HCFO), and hydrofluorocarbons (HFC) are mentioned as common refrigerants for heat pumps. The authors point out that the choice of refrigerant should be based not only on technical and safety considerations, but also on ensuring that ozone depletion potential (ODP) and global warming potential (GWP) values are as low as possible to minimize harmful effects on the climate. Focusing on heat pumps for process steam generation, D'Alessandro et al. (2025) provide a general overview and different concepts for steam generating heat pumps used in industrial applications using environmentally friendly refrigerants such as HFOs, HCFOs, HCs, and natural refrigerants. The authors mention ways to improve thermodynamic performance through the use of an internal heat exchanger and refrigerant injection technology. Furthermore, they emphasize that many systems with a TRL of 9 are already in operation, but are rarely available for heating capacities exceeding 10 MW, requiring these systems to be scaled up. In addition to examining the current state of the art in heat pump technologies, other studies also address their technical applicability and feasibility in industrial settings. The Annex 58 Task 2 Integration Concepts report (Zühlsdorf et al., 2024) provides an overview of 12 different integration concepts with supply temperatures above 100 °C in industrial applications, including various drying processes. The report presents heat pump concepts for steam generation that utilize direct steam-generating compression heat pumps using HCs or HFOs, as well as direct steam-generating heat pumps in combination with a downstream steam compressor. Numerous studies (Zühlsdorf et al.; Bühler et al., 2019; Dumont et al., 2023; Schlosser et al., 2023; Vieren et al., 2023; Ciambellotti et al., 2024) have examined the use of industrial heat pumps in the food and paper industries and, through case studies, have demonstrated their suitability, particularly for drying processes involving paper or milk powder. Zühlsdorf et al. investigated the integration of high temperature heat pumps using zeotropic working fluids for spray drying facilities and demonstrated that working fluids with a 50/50 propane-isopentane mixture and a 80/20 propane-n-pentane mixture achieved the best economic results with a COP of around 3, outperforming the pure refrigerants. The authors also emphasized that the proposed solution could result in a 36 % reduction in natural gas consumption by using electricity instead of natural gas. Bühler et al. (2019) investigated strategies for the electrification of industrial processes using the example of milk powder production. The results showed possible energy savings of 65 % using a central heat pump system with an overall COP for supplying process heat of 1.95. The economic analysis showed that using electric boilers is not cost-effective compared to the proposed heat pump system, and that the heat pump achieved a positive net present value after 20 years. Schlosser et al. (2023) analyzed the economic break-even points of high-temperature heat pumps for spray drying of milk. The most promising result was achieved in a simulation using a trans critical CO_2 cascade cycle with a COP of 2.14. The calculated break-even point between electricity and gas prices was 1.56. A market-ready heat pump based on the Stirling cycle with a COP of 1.6 achieved a break-even price ratio of 1.2. The authors concluded that commercially available heat pumps can cover around 60 % of the heat demand of a spray dryer at a break-even price ratio of 1.41. The authors emphasize that the COP is the most important factor influencing the relationship between average heating costs in euro per kWh and the break-even price. Pellegrini et al. (2023) conducted a feasibility study on the use of a HTHP on a pulp and paper recycling plant in Italy. A 1.3 MW cascade HTHP was designed for the study to supply the thermal dryer with 120 °C hot heat. Under the given conditions, a COP of 2.4 and savings of approximately 1.613 t of CO_2 equivalent emissions per year were achieved. However, the authors mention that the integration of the heat pump is economically disadvantaged by excessively high electricity prices. Ciambellotti et al. (2024) simulated a two stage HTHP system producing 4 t/h of 9 bar saturated steam using data from a tissue paper mill. They investigated various refrigerants with a maximum achieved COP of 2.01. The results showed a 17 % reduction in energy consumption and a 40 % reduction in emissions with the use of the HTHP system. However, the authors also note that the investment costs for such a system may be prohibitive without public funding. The literature review on the current state of the art in industrial heat pumps shows that the technology and its potential applications have already been well researched and are largely ready for practical use, but it also highlights uncertainties regarding costs and economic viability. In general, the thermal output of the systems available on the market and used in industry ranges from 1 MW to 10 MW, with many of them being custom-designed solutions tailored to the specific process.

1.3. Modularization and standardization concepts in industry

While current heat pump solutions for industrial use consist mainly of custom-built units and prototypes, the concept of standardization and modularization is already being applied in various other industrial sectors. In the chemical industry, modularization is already being used to increase plant flexibility, resource efficiency, and cost-effectiveness. The modular production facilities are considered suitable for the manufacture of fine and specialty chemicals as well as active pharmaceutical ingredients due to their size and design. Modular solutions in the chemical industry are based on container-based solutions for small-scale production (Saulich, 2019). In a white paper by the Society

for Chemical Engineering and Biotechnology DECHEMA (DECHEMA | Gesellschaft für Chemische Technik und Biotechnologie e.V, 2020) the advantages and challenges of modular plants for flexible chemical production are discussed and publicly funded projects such as the F^3 Factory-project, CoPIRIDE or ENPRO are cited as demonstration projects for technical and economic feasibility. Based on the demonstration projects, capital expenditures (CAPEX) could be reduced by up to 40 %, energy consumption by 30 %, and time to market by 50 % through the use of a modular process structure. The article emphasized that the greatest challenges lie in establishing a common standardization approach at the device and system levels as existing solutions are often project-specific or limited to pilot-scale applications. An information sheet published by the German Bundestag (Wissenschaftliche Dienste des Deutschen Bundestages, 2023) on the topic of serial and modular construction in the building industry explains the concepts of serial and modular construction. In the construction industry serial construction refers to the mass production of building components or building sections according to a predefined standard and is frequently used in large-scale construction projects such as housing developments. Modular construction refers to the production of prefabricated modules in various sizes and configurations, such as fully equipped living units like kitchens and bathrooms, which can be combined as standalone units. Modular construction is primarily used for smaller projects such as single-family homes, thereby modular construction accounts for only about 5 % of the construction sector. The advantages of mass production and modular construction over conventional methods include shorter construction times, cost savings, and sustainability. The disadvantages include limited design options and the difficulty of making changes to prefabricated components, as well as a potential dependence on specialized manufacturers. Kudsk et al. (2013) examined the implementation of standardization and modularization in the construction industry using two case studies. The results showed that the modularization approach reduced short-term flexibility but improved long-term planning and efficiency of construction processes. The authors concluded that the implementation of modularization in the construction industry can bring significant benefits, especially when a bottom-up approach is used to optimize specific parts of a construction project. The automotive industry, where modular platform strategies are firmly established, is often cited as a model for systematic modularization. The Volkswagen Group's (Volkswagen, 2022) MQB modular transverse matrix modular vehicle platform uses a standardized subframe as its base module, which is independent of the specific product configuration. By defining uniform interfaces, a single platform module can be used for various drive variants, such as internal combustion engines, hybrid systems, or electric drives. This approach maximizes flexibility in terms of product variety and volume. A key factor here is that the compatibility of the interchangeable modules is ensured exclusively through standardized interfaces. Modularization can reduce the costs of replacement parts, training, and maintenance by up to 35% (Horst Wildemann and Hubert Waltl, 2019). In the energy sector, battery energy storage systems represent a certain degree of modularization. One example is the use of containerized storage solutions, such as the Tesla Megapack (Tesla, 2026). These solutions use standardized shipping containers as functional building blocks and a plug-and-play approach for the rapid assembly of prefabricated modules. Key advantages for system integration include easier transport and handling, the scalability of energy capacity through the addition of parallel units and shorter installation times, resulting in cost and time efficiency.

Modular concepts are already being successfully used in industries such as the chemical industry, the construction industry, the automotive industry, and the energy sector. However, the application of modularization and standardization in the context of industrial HTHPs has not yet been researched as extensively. The market for industrial HTHPs is currently dominated by customized, project specific solutions, while standardized mass produced products in the 1 MW to 10 MW power range are hardly available. Studies on standardization and modularization in this field remain scarce. However, a shift toward a modular design is seen as a key lever for reducing investment costs per kilowatt and enabling economically viable production volumes (Agora Energiewende and Fraunhofer IEG, 2023). For residential applications smaller 1 MW, there are already established standard solutions for heat pumps using cascading, such as those from Alpha Innotec (Alpha innotec, 2025), which can achieve an output of up to 180 kW by cascading up to four heat pump units. For industrial applications, however, approaches to modularization are still in the pilot and research phases. Table 1 provides a summary of relevant concepts and projects, based on the IEA Annex 58 Task 1 report (Zühlsdorf, 2023).

Current developments can be divided in thermoacoustic modularity, modular system architecture and component technology. Technologies based on thermoacoustic modularity leverage physical scalability to generate high power outputs. The TASTE project is developing a scalable system up to 30 MW based on piston compressors with a variable number of cylinders. The COMTA project employs compact modules for steam generation up to 200 °C by operating up to eight units in parallel. Modular system architectures focus on standardized design. The Skala Fabrikk and FUSE projects utilize modular configurations based on container solutions, which enable flexible configuration and scaling. In the Skala Fabrikk each module can simultaneously deliver up to 0.3 MW of heating capacity at 115 °C and 0.15 MW of cooling capacity at 0 °C. As part of the FUSE project, a full-scale industrial heat pump demonstrator with a capacity of 1 MW to 2 MW was developed for steam generation. Heaten AS developed the HeatBooster, a single-stage HTHP system with closed-cycle steam compression, featuring modules ranging from 1 MW to 10 MW cascadable up to 50 MW, that can be combined in parallel or in stages. The technological approaches focus on modular compressors. Spilling uses electrically driven, steam-powered piston compressors with a scalable number of cylinders. GEA uses an ammonia screw compressor for modular cascade configurations with one to four stages to achieve a capacity of up to 1.5 MW, which is scalable to up to 5 MW. Mayekawa uses modular cascade systems for piston and screw compressors to achieve thermal capacities ranging from 0.5 MW to 10 MW, which are scalable to 20 MW in cascades. In addition to the research and demonstration projects, a study by Preisinger et al. (2025) is noteworthy. The authors developed and investigated a modular heat pump system to replace the conventional gas boiler in a residential building. The study compares compressor technologies within modular racks. In the comparative study, the authors developed a rack system for up to eight modules connected in parallel or in series and used two different module types, piston and rotary compressors. The results were analyzed in terms of heat capacity and performance for various temperature levels and media, and showed that the module with a piston compressor performed better at nearly all delivery heads.

While standardization strategies are already well established in other industries, generic methods for modular HTHPs have hardly been researched to date. The existing literature is largely limited to application-specific pilot projects or manufacturer-oriented feasibility studies, resulting in a significant gap in systematic approaches to architectural modularity. In particular, there is a lack of scientific framework concepts that link hardware replication with adaptive process requirements, which hinders the systematic realization of cost-reduction potential. To close this gap, this study presents the first systematic framework for modular HTHP design and shifts the paradigm from custom-built plants to mass-producible, standardized architectures up to 10 MW. The methodological innovation lies in the derivation of transferable modularity rules that decouple the thermodynamic core from process-specific adaptations, thereby enabling the adaptation of components through parametric variation rather than custom redesign. By integrating country-specific market data and explicitly embedding parametric sensitivity analyses into a unified optimization framework, the methodology goes beyond descriptive case studies to a reproducible, parameter-driven design approach. The study

Table 1
Overview of HTHP systems based on IEA Annex 58 Task 1 and recent developments (Zühlsdorf, 2023).

Name	Modular Concept & Technology	Max. Temp.	Capacity Range
TASTE Project	Thermoacoustic scaling; Piston modules (2–8 cylinders)	180 °C	Up to 30 MW
COMTA Project	Thermoacoustic; Parallelizable modules (up to 8 units) for steam generation	200 °C	1–10 MW
Skala Fabrikk	Rack-based system inside a standard 10-foot container; Modular frames	115 °C	0.3 MW (heating), 0.15 MW (cooling)
Heaten AS	Standardized hardware platform; Parallel or cascade configuration	200 °C	1–10 MW (up to 50 MW)
FUSE Project	Modular approach with standardized industrial components	90 °C	1–2 MW
Spilling	Electrical steam piston compressor; Scalable cylinder stages (2–6)	180 °C	1–15 MW
GEA	Ammonia screw compressor; Modular cascade configuration	130 °C	0.1–1.5 MW (up to 5 MW)
Mayekawa	Cascaded systems using piston or screw technology	145 °C	0.5–10 MW (up to 20 MW)

makes several significant contributions. It quantifies how industrial heat demand varies nonlinearly between plants, thereby demonstrating the need for flexible rather than rigid system concepts. It is presented, how reduced investment costs can be achieved through standardized mass production. Furthermore, the analysis highlights that a successful market launch depends on structured collaboration between technology providers and industrial end users. Based on these findings, the study outlines the need for a standardized catalog of modular interfaces to accelerate technical pre selection, reduce development effort, and support reproducible system selection. Together, these elements bridge the gap between conceptual architecture and practical implementation, thereby paving the way for the theoretical and methodological developments presented in the following chapter.

2. Theoretical context

2.1. Structure and function of a heat pump

The following section describes the basics of the structure and operation of a heat pump. Fig. 2 shows a schematic diagram of a heat pump and its main components.

The heat pump shown consists of a compressor, three heat exchangers, and an expansion device. The state points in the system are numbered with Arabic numerals in the primary cycle and with Roman numerals in the secondary cycles at the heat sink and heat source. At state point (1), the working fluid is mechanically compressed by an electrically driven compressor (C), causing the pressure p and temperature T downstream of the compressor (2) to increase. The high-temperature heat exchanger (HTHX) located at the heat sink, transfers heat between the primary heat pump cycle and the secondary cycle. The heat flow at state point (II) at the heat sink can now be used in an industrial process, e.g., for process air. The working fluid in the primary cycle (3) flows through the internal heat exchanger (IHX). The temperature of the working fluid upstream (4) of the expansion device (E) is lowered even further by transferring residual heat to the working fluid downstream (7) of the IHX, which results in an increase in temperature upstream of the compressor (1). Behind the expansion device (5), the working fluid expands to its initial pressure, causing its temperature to drop. The low temperature heat exchanger (LTHX) at the heat source absorbs heat from the environment (III) and transfers it between the secondary and primary cycles, reheating the working fluid in the primary cycle (6). The cycle closes and starts again as soon as the heated working fluid re-enters the compressor. Thermodynamic cycles can be classified based on the phase change of the working fluid. While the Rankine cycle involves a change of state through evaporation and condensation, the working fluid remains in a gaseous state throughout the entire Brayton cycle. The reversed Rankine cycle is commonly

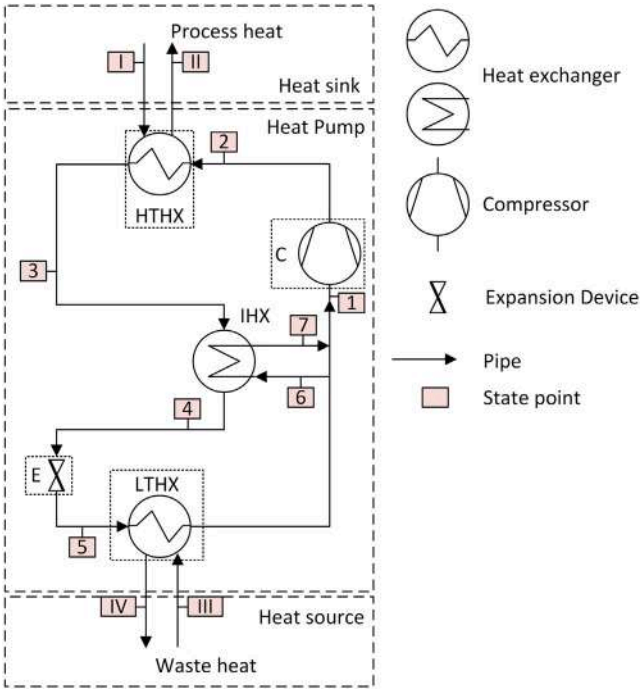


Fig. 2. Schematic of a heat pump cycle and its main components.

used in industrial and commercial heat pumps and well suited for applications with moderate temperature increases and a variety of refrigerants. Unlike the Brayton cycle, in which a turbine is used as an expansion device, the heat pump described here uses an expansion valve. From a thermodynamic perspective, this cycle can be regarded as a reverse Rankine cycle. At the heat source, the LTHX functions as an evaporator. Thermal energy is extracted from the secondary flow (III), causing the refrigerant in the primary flow to transition from a liquid (5) to a gaseous (6) state. At the heat sink, the HTHX functions as a condenser. Thermal energy is extracted from the refrigerant in the primary flow (2), causing it to transition from a gaseous (2) to a liquid (3) state, and the secondary flow (II) uses the released thermal energy to generate steam. An important parameter for assessing the efficiency of a heat pump is the COP. The COP is defined as the ratio of the heat delivered at the heat sink $\dot{Q}_{\mathrm{out,sink}}$ and the electrical input power P_{el} used for the compression of the working fluid (Arpagaus, 2019), see Eq. (1).

$$COP = \frac{\dot{Q}_{\mathrm{out,sink}}}{P_{\mathrm{el}}} \tag{1}$$

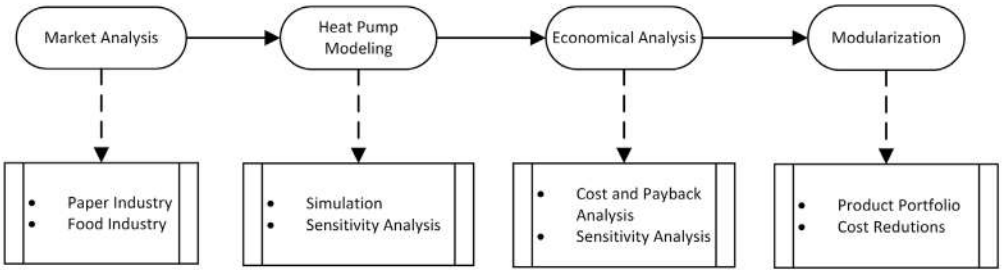


Fig. 3. Methodology flowchart.

2.2. Modularization and standardization

Modularization is an approach in which a complex system or product is broken down into functionally independent, interchangeable units with clearly defined interfaces (Dhillon, 1999). The modularization approach enables the definition of module groups based on the identification of product components and their dependencies. To define module boundaries and functional relationships, various methods are mentioned in the literature, such as DSM (Design Structure Matrix), METUS (methodological support for system formation), and MFD (Modular Function Deployment). These methods differ primarily in their objectives. While DSM minimizes technical interfaces and dependencies, METUS focuses on clear functional relationships, and MFD incorporates a holistic analysis that takes strategic corporate goals into account (Martin and Bögemann, 2025). The faster development processes and lower development costs achieved through the modularization of systems can further lower the barrier to new investments for potential users (V.D.I. Zentrum Ressourceneffizienz, 2025). Standardization, as part of modularization, unifies modules and components via interfaces as well as measurement and control technology. It enables flexible adaptation of systems to market changes and ensures a high degree of interchangeability without requiring entirely new developments (Ripperger and Nikolaus, 2020). This study does not follow a specific approach, but part of the research strategy involves breaking down complex systems into individual units based on their function, which is partly consistent with the METUS approach.

3. Methodology

This study presents an analytical approach to the development of modularization concepts for HTHPs. The methodology is divided into four main parts. First, market analyses are examined to define the requirements and framework conditions of the relevant industries for modular HTHP concepts. Based on this analysis, two representative industrial processes from the paper and food industries are selected to illustrate the modularization strategy using simulations. Two heat pump configurations, which serve as base modules, are modeled and simulated. The simulations are supplemented by sensitivity analyses to ensure the validity of the selected parameters. An economic analysis of both heat pump configurations is conducted to evaluate the efficiency of HTHPs compared to gas- and electric-powered systems using current electricity and gas price data. A modularization concept including modules and scaling factors is presented based on manufacturer data to highlight the potential cost savings that can be achieved through the proposed modular approaches. The methodological approach, which includes the main steps and sub steps, is illustrated in Fig. 3.

3.1. Industry needs and market opportunities

Based on the literature review, the food and paper industries were selected as promising application areas for HTHPs. Although the chemical industry also exhibits high process heat demand, its operations are often characterized by complex batch processes and stringent safety requirements, which impede the development of standardizable modules. In contrast, the selected paper and food sectors rely on continuous mass flows with relatively predictable thermal loads, making them ideal candidates for modularization concepts. The analysis for selecting example processes in these sectors focused on the market opportunities for various processes at temperatures up to 200 °C based on literature findings. In the paper industry, drying processes account for the largest share of the sector's energy consumption with 70 % and typically require temperatures between 125 °C and 180 °C (IEA, 2025; Marina et al., 2021). The pressure profile of the drying cylinders ranges from 0 to 8 bar, with 70 % below 5 bar (European Heat Pump Association, 2025b). As part of the SPIRIT project, a heat pump demonstrator is already being used for steam generation in paper drying, confirming the technical potential of integrating heat pumps in this sector (SPIRIT Project, 2025). Due to its characteristic thermal load as well as its specific pressure and temperature profiles, paper drying is an excellent reference case for the development of modular heat pumps and enables the efficient integration of heat pumps for the generation of process steam. Furthermore, mass processing plants which typically operate continuously offer ideal conditions for investigating scalable, modular system concepts. To enable a comprehensive evaluation of modular HTHP concepts across various industrial sectors, milk powder production in the dairy industry was selected as a second representative case study. Spray drying is the predominant industrial process for milk powder production and is characterized by a significant demand for process heat at temperatures between 150 °C and 200 °C (Agriculture Institute, 2024). This makes spray drying a demanding and representative application for demonstrating the performance of HTHP in the generation of high-pressure steam and process heat. Furthermore, the spray drying process generates significant waste heat with temperatures around 50 °C (Agroscope, 2025). By selecting these two processes, the study examines the integration of heat pumps into sectors with a high demand for high-temperature steam and enables a comparative analysis of modular scalability across different industrial sectors and thermal requirements. Ultimately, the focus on continuous, high-load processes ensures that the derived modularity rules are highly transferable to other industrial sectors operating under similar conditions.

3.2. Heat pump modeling and simulation

3.2.1. Thermal heat pump capacity

To determine the thermal capacity P_{th} required per heat pump system and to determine the corresponding module size, the product of the aggregated total annual production volume m and the specific heat demand e was distributed based on the number of facilities N and the annual operating hours t, according to Eq. (2):

$$P_{\mathrm{th}} = \frac{m \cdot e}{N \cdot t} \tag{2}$$

The production volume, the number of plants, and the resulting thermal heat pump capacity for paper and milk powder production in selected EU-27 countries are shown in Tables 2 and 3, respectively. For both processes, 8000 h per year are assumed as operating hours

Table 2
Production volumes, number of facilities and resulting thermal capacity of selected EU countries for the paper drying process.

EU-Country	N (Austropapier, 2025)	m [1,000t/year] (Austropapier, 2025)	P_{th} [MW]
Germany	150	20,934	24
Sweden	66	19,139	50
Italy	153	8,237	9
Finland	53	16,060	52
Spain	79	8,257	18
France	83	7,946	16
Austria	31	5,843	32
The Netherlands	21	2,212	18
Arithmetic mean value	80	11,078.5	27

Table 3
Production volumes, number of facilities and resulting thermal capacity of selected EU countries for the spray drying process.

EU-Country	N	m [1,000t/year] (European Commission, Directorate-General for Agriculture and Rural Development, 2025)	P_{th} [MW]
Germany	37 (Bundesanstalt für Landwirtschaft und Ernährung, 2024)	517,240	3
France	81 (Bundesanstalt für Landwirtschaft und Ernährung, 2024)	508,400	2
The Netherlands	6 (P.B.L. Netherlands Environmental Assessment Agency, 2020)	150,900	6
Sweden	n.a	67,270	–
Spain	n.a	55,380	–
Belgien	n.a	214,820	–
Dänemark	n.a	88,200	–
Arithmetic mean value	41	228,887	4

t. The specific heat demand e for the paper drying process is 1.37 MWh per tonne of paper and cardboard products produced (Marchetti, 2026) and 2 MWh per tonne for the spray drying of milk powder (P.B.L. Netherlands Environmental Assessment Agency, 2019).

Based on the arithmetic mean of the country-specific thermal capacities, the thermal heat pump capacity is calculated to be approximately 27 MW for the paper drying process and approximately 4 MW for the spray-drying process. A heating capacity of 2 MW is assumed for the paper drying system, and a heating capacity of 1 MW is assumed for the spray drying system. With a view to investigating a modular concept and standardizing components, this study considers two additional heating capacities or module sizes for each process in addition to the base case, in order to provide the total required thermal heating capacity for the respective process step by combining different heat pump modules. The simulated reference case serves as the starting point for scaling additional capacities. Based on the reference case, the 2 MW capacity for paper drying was scaled up to 5 MW and 10 MW in order to meet the total demand, either fully or partially, by combining modular heat pumps of different sizes. For the spray-drying process, the 1 MW capacity was scaled up to 2 MW and 5 MW. The scaling of the simulation results is based on assuming a linear proportionality between mass flows and compressor capacity. This approach is based on the assumption of geometric and thermal similarity, in which compressor technology is the driving variable for scalability. Consequently, a constant isentropic compressor efficiency and, as a result, a constant COP, is assumed for all capacity classes. This assumption is supported by the specific operating conditions of the high-pressure range. As outlined in the IEA Heat Pump Technology Collaboration Program Project 68 (Zühlsdorf et al., 2025), the commercial market for high-pressure steam compressors is primarily limited to screw compressors and piston compressors. Those compressors are characterized by an approximately constant pressure ratio across the entire speed range. An increase in performance for this class of technology is therefore achieved primarily by scaling the geometric dimensions, e.g., increasing the compressor volume or connecting units in parallel, rather than by shifting the operating point. This approach allows for the maintenance of constant efficiency levels and for consistent analysis of the modular concept without requiring separate full simulations for each heat pump size.

3.2.2. Heat pump configuration

Based on the aggregated market data, specific boundary conditions for thermodynamic modeling were derived. A 2 MW heat pump for the integration into a paper drying process at a paper mill and a 1 MW heat pump for a spray drying process for milk powder production are modeled. Figs. B.12 and B.13 in Appendix B show the two schematic heat pump configurations for the respective processes. The heat pump configurations are designed to generate superheated process steam in three temperature ranges used in the respective industrial drying processes, utilizing waste heat from the existing industrial process. The configuration of the heat pump for the paper drying process was selected based on a publication by Kriese et al., which presented the integration of a heat pump into a paper drying process as part of the EEETHOS project (Eeethos project, 2025). As part of this study on modularity and standardization, the heat pump configuration for spray drying was derived from the paper drying model based on similar process conditions. In the 2 MW heat pump configuration for the paper drying process in Fig. B.12, R600 is used as the working fluid for the primary heat pump cycle. Moist exhaust air from the exhaust hood of the paper web's drying cylinders is used as the heat source of the heat pump cycle. A water-ethylene glycol mixture with a composition of 80:20 is used as the working fluid in the heat source cycle. The process heat required for the paper drying process is extracted via the HTHX at the heat sink in the form of superheated steam. Through a two-stage compression process, the temperature and pressure of the superheated steam are further increased to meet the required specifications, so that the process steam generated can be fed into the drying cylinders of the paper web for paper drying. A condensate injection between the compressor C1 and C2 serves to achieve lower process temperatures while maintaining constant pressure and enables the extraction of process steam at different temperature levels. The condensate generated in the paper web's drying cylinders is reused and returned to the process as a working fluid for the secondary side of the HTHX. The 1 MW heat pump configuration for the spray drying process in Fig. B.13 is based on a cascade heat pump configuration. R600 is used as the working fluid for the primary heat pump cycle or second stage of the cascade configuration and R717 is selected as the working fluid for the first

stage of the cascade configuration. Waste heat from humid air from the spray drying process serves as the heat source for the first heat pump cascade. The heat sink configuration of the 1 MW system is identical in design to that of the 2 MW system and consists of a two stage compression system with condensate injection. Because of its identical configuration, the steam compression component offers significant potential for standardization. The condensate formed on the plant side during the process is used as a working fluid on the secondary side of the HTHX in the second cascade stage. Working fluid assumptions are based on project-specific demonstration examples from the SPIRIT project (SPIRIT Project, 2025) as well as on standard recommendations from the literature (Arpagaus et al., 2018). Both fluids were selected based on their thermodynamic potential and environmental compatibility. R600 was selected for the primary cycle of both heat pump configurations due to its ODP value of zero and its extremely low GWP value of 4. In addition, at €1.80/kg, R600 is extremely cost-effective compared to other HC based refrigerants. With a critical temperature of 152 °C and a critical pressure of 38 bar, R600 efficiently covers the temperature range required for drying paper and milk powder, without entering the supercritical range, which would be necessary with CO_2 due to its very low critical temperature of 31 °C. The specified pressure and temperature ranges can be easily achieved using standard compressor technologies. R717 is used for the lower cascade stage in the spray drying process. As a natural refrigerant, the advantages of a high critical temperature of 132.3 °C, critical pressure of 113.3 bar and ODP and GWP values of 0 outweigh the disadvantages related to its toxicity. Due to its high specific enthalpy of vaporization and condensation, it enables high volumetric heating capacity, which has a positive effect on the size and therefore the cost of the compressor, even at low charge quantities. A cascade configuration was considered as another possible heat pump variant as part of the modularization concept. In single-stage configurations, suboptimal temperature gradients can occur within the heat exchangers because the refrigerant's heating curve does not match the heat source's cooling profile, resulting in very large heat exchanger surface areas. According to established engineering practices, the temperature gradient or minimum temperature difference should be at least 10 K for all gas-to-liquid heat exchangers and at least 5 K for liquid-to-liquid heat exchangers (ProcessConvert, 2026), (Jensen and Skogestad, 2008). For heat transfer from humid air to R600, a minimum gradient of 10 K is therefore recommended. The cascade configuration significantly smooths out these temperature profiles. The thermodynamic division of the process into two temperature stages allows the cascade system to gradually adapt the refrigerant curve to the cooling profile of the heat source. This results in a significant smoothing of the temperature profiles and prevents thermodynamic bottlenecks in the heat exchanger. The applicability of cascade configurations for overcoming large temperature differences is also supported by IEA Annex 58 (Zühlsdorf, 2023). The assumptions underlying the simulation of condensate injection between the two compression stages C1 and C2 are based on the mechanical requirements, taking into account the use of SPIRIT project-specific Spilling piston compressors. Since piston compressors require liquid injection for internal lubrication and to protect component parts, especially the piston rings, the injection must take place before or during the compression phase (Spilling, 2024). Furthermore, as the inlet temperature to the subsequent compression stage rises, the required compression work increases significantly, which consequently leads to a reduction in the COP. Therefore, the decision was made to use condensate injection to ensure efficient and safe process operation.

3.2.3. Numerical simulation

The thermodynamic modeling and simulation of the heat pump cycles presented were performed using the simulation software EBSILON Professional (Ebsilon, 2025), version 17.2. EBSILON features a customizable component library and allows for the visualization of energy and mass balances as well as the thermodynamic parameters to simulate and optimize complex energy conversion systems. The software uses the IAPWS-IF97 water vapor table and polynomial correlations to calculate the specific heat capacity for air and flue gas as its data basis. The refrigerants R600 and R717 are modeled using the software's Refrop property database. The calculation of thermodynamic state variables in EBSILON is performed at a reference temperature T_{Ref} of 0 °C and a reference pressure p_{Ref} of 1.013 25 bar. The simulation environment solves a nonlinear system of equations using an iterative approach until the calculation converges to a steady-state solution. The simulations conducted in this study were performed using a maximum number of iterations of 99,999 and a convergence criterion of 10^{-7}. The simulation is performed in design mode. In addition, the following simulation conditions are assumed. The simulations are performed under steady-state conditions. No pressure or heat losses are assumed in the piping. Expansion and compression processes are adiabatic with respect to the environment. The counter-flow heat exchangers at the heat sink and heat source are determined by a minimum temperature difference ΔT relative to the respective streams. The recuperator is calculated using a fixed effectiveness ε with common values based on the literature (Murshed and Lopes, 2017). The compressors are calculated using a fixed isentropic efficiency η_{is}. Possible transmission losses, electrical and mechanical losses of the motor are neglected. The hydraulic units are not included in the balance sheet. The individual heat pump components are configured for each cycle in accordance with the thermodynamic simulation parameters, presented in Table 4.

Table 4
Component part numbers and parameter assumptions for isentropic efficiency, effectiveness and temperature difference.

Component	Parameter	Value	EBSILON Part No.
C1	$\eta_{is,C1}$	0.75	24
C2	$\eta_{is,C2}$	0.7	24
C3	$\eta_{is,C3}$	0.78	24
C4	$\eta_{is,C4}$	0.78	24
IHX	ε	0.9	55
HTHX	ΔT	10 K	55
LTHX	ΔT	10 K	55

The values for isentropic efficiencies were selected based on project-specific manufacturer specifications from the SPIRIT project and data based on technical literature. Zühlsdorf et al. (2018) reports expected values of 74 % for steam turbo compressors. Both in the technical literature Mamaev et al. (2018), Eifler et al. (2009) and in manufacturer-specific specifications from the company Spilling (Spilling, 2024), isentropic efficiencies of approximately 80 % are reported for piston compressors. Since these values cannot been confirmed by experimental data, lower estimated values were assumed for this study. To account for the different compressor technologies depending on the load, different isentropic efficiencies were assumed for C1 through C4, based on the expected thermodynamic requirements. Each heat pump cycle is simulated with specified boundary conditions for the respective industrial process based on the analyzed industry needs and market opportunities. Table 5 presents the boundary conditions and process operating parameters for the 1 MW and 2 MW systems.

The selection of the parameters for the heat sink outlet (II), (IIa), (IIb), (IIc) is based on literature values from the analysis of process-specific temperatures and pressures in the paper and milk powder drying processes (IEA, 2025; European Heat Pump Association, 2025b; Kriese et al.; Agriculture Institute, 2024). Since the pressure and temperature specifications at the heat sink are very similar for both drying processes in industry, correspondingly similar but slightly varied values were assumed. For paper drying, 120 °C was assumed with pressures of up to 1.8 bar, 4.8 bar, and 10.1 bar bar to account for the drying of various paper types based on project specific and literature available data, see Table A.15. The operating parameters for spray drying are assumed to be sink temperatures starting at 110 °C with pressures of up to 1.2 bar, 3.5 bar, and 10.1 bar. The assumptions regarding heat sink inlet (I), (Ib) and the heat source inlet (III) temperatures and

Table 5
Boundary conditions and process operating parameter fo a 1 MW and a 2 MW system.

	1 MW	2 MW
Process boundary conditions		
Heat sink inlet T_I	95 °C	95 °C
Heat sink outlet T_{II}	110 °C	120 °C
Heat sink outlet p_{II}	1.2 bar	1.8 bar
Heat source inlet T_{III}	75 °C	70 °C
Heat source inlet p_{III}	1 bar	1 bar
Heat source inlet $\dot{m}_{III}$	11.5 kg/s	18.06 kg/s
Temperature and pressure specifications		
T_9	–	55 °C
p_2	27 bar	24.5 bar
p_5	5 bar	4 bar
p_8	–	2 bar
p_9	8 bar	–
p_{11}	–	43 bar
p_{Ib}	15 bar	15 bar
p_{IIa}	3.5 bar	4.8 bar
p_{IIc}	10.1 bar	10.1 bar

Table 6
Control parameters and settings for a 1 MW and 2 MW system.

Controller	Control Variable	Target Value	Actuating Variable	Limits
1 MW				
C1	$\dot{Q}_{IIc}$	1000 kW	m_{Ib}	0.4–2 [kg/s]
C2	T_3	131.4 °C	m_2	2–10 [kg/s]
C3	T_8	81.7 °C	m_{11}	0.5–10 [kg/s]
2 MW				
C1	$\dot{Q}_{IIc}$	2000 kW	m_{Ib}	0.4–2 [kg/s]
C2	T_3	125.7 °C	m_2	2–10 [kg/s]
C3	T_8	45 °C	m_{11}	2–50 [kg/s]

pressures are selected based on confidential measurement data from a paper mill that was collected as part of the EEETHOS project (Eeethos project, 2025). The pressure specifications at points (2), (5), (8), (9) and (11) and are based on assumptions derived from the publication by Kriese et al., taking into account a temperature difference between the condensation temperature of the refrigerant and the evaporation temperature of the condensate. The specified heat pump capacities $\dot{Q}_{IIc}$ and temperatures T_3 and T_8 are implemented by a controller using the component No. 39 to precisely adjust the operating conditions. The controller in the simulation models intervenes when a specific controlled variable in the system needs to be adjusted to a target variable. If the target value deviates from the actual value of the controlled variable, the controller intervenes by adjusting the so-called actuating variable. All controlled variables are regulated via mass flow. The target values for the temperatures are based on the refrigerant's condensation temperature. The refrigerant's condensation temperature must be higher than the saturation temperature of the medium on the secondary side to ensure a sufficient temperature difference for heat exchange with the secondary medium. The control variables, target values, actuating variables and control limits for each controller in the 1 MW and 2 MW systems are listed in Table 6.

3.3. Economical analysis

To enable a comprehensive economic assessment, this study is based on a detailed cost analysis that takes the payback period into account, as well as a comparison of the price ratios between industrial electricity and natural gas. The economic evaluation of the two heat pump systems is based on a comparison of total costs with standard reference technologies such as an electric boiler (EB) and a gas boiler (GB). The evaluation includes investment costs, energy costs, and annual maintenance costs. The annual heat demand Q_{th} is determined based on the thermal output $P_{\text{th,HP}}$ and the annual operating hours t, as defined in Eq. (3).

$$Q_{\text{th}} = P_{\text{th,HP}} \cdot t \tag{3}$$

Based on the heat demand, the heat pump's electricity consumption Q_{el} is calculated using the system's specific COP according to Eq. (4). For the reference systems, the electricity consumption of the EB corresponds directly to the heat demand, while the consumption of the GB is derived from its thermal efficiency η_{gas}.

$$Q_{\text{el}} = \frac{Q_{\text{th}}}{COP} \tag{4}$$

Operating costs are calculated by multiplying energy consumption by the respective prices. The electricity costs for the heat pump $C_{\text{el,HP}}$ and the electric boiler $C_{\text{el,EB}}$ are calculated based on the electricity price c_{el}, as shown in Eq. (5) and (6).

$$C_{\text{el,HP}} = Q_{\text{el,HP}} \cdot c_{\text{el}} \tag{5}$$

$$C_{\text{el,EB}} = Q_{\text{el,EB}} \cdot c_{el} \tag{6}$$

The operating costs for the gas boiler $C_{\text{gas,GB}}$ are calculated based on the gas price c_{gas} and the required heating capacity $Q_{\text{th,GB}}$.

$$C_{\text{gas,GB}} = Q_{\text{th,GB}} \cdot c_{gas} \tag{7}$$

The total operating and maintenance (O&M) costs $C_{\text{O\&M,HP}}$ for the heat pump consist of maintenance costs c_{Main}, repair and replacement costs c_{Rep}, and fixed labor costs c_{Lab} and are calculated according to Eq. (8). Fixed values are assumed for the maintenance and operating costs of the EB and GB.

$$C_{\text{O\&M,HP}} = (c_{\text{Main}} \cdot P_{\text{th,HP}}) + (c_{\text{Rep}} \cdot Q_{\text{th}}) + c_{\text{Lab}} \tag{8}$$

The total CAPEX for the initial year 0 is calculated by multiplying the plant's thermal output $P_{\text{th,HP}}$ by the specific investment costs $CAPEX_{\text{spec}}$, as shown in Eq. (9). For subsequent years, the cumulative costs are calculated by adding the annual operating costs C and maintenance costs $C_{\text{O\&M}}$ to the cumulative costs of the previous year. This iterative calculation allows for an assessment of the plant's economic viability and the determination of the payback period over its technical lifespan.

$$CAPEX = P_{\text{th,HP}} \cdot CAPEX_{\text{spec}} \tag{9}$$

The assumptions regarding the parameters for the cost analysis are presented in Table 7.

The parameter assumptions are based on literature values as specified. While the heat pump thermal capacity was calculated based on a nominal design capacity of 8000 h per year, the economic assessment assumes a reduced operating time of 5000 h per year. This reduction is justified by the assumption of necessary system downtime, including scheduled maintenance, repairs, and process outages, which are typical for industrial drying applications. The specific investment costs are based on the range for industrial heat pump investment costs reported by Marina et al. (2021), between €200/kW and €500/kW, and the mean value of €350/kW was used as the baseline for the reference case. In order to further evaluate the economic performance of the heat pump systems presented, the ratio of electricity to gas prices r_{p} is used. A COP that exceeds this price ratio favors the use of a HTHP in a operational comparison. While the achievable COP depends largely on the specific process conditions, the price ratio is determined exclusively by the geographic location. To derive r_{P}, industrial electricity prices and natural gas prices for commercial customers in Germany, Spain, France, Italy, the Netherlands, Austria, and Poland for the period from 2019 to 2023 are analyzed using the Eurostat database (eurostat, 2025). These countries were selected because they are among the leading producers in the European paper and food industries. The Eurostat classification divides industrial electricity prices c_{el} into four consumption groups,

Table 7
Parameter assumptions for the economic cost analysis.

Parameter	Symbol	Value	Unit	Reference
Operation hours	t	5000	h/y	Own assumptions
Specific investment costs HP	$CAPEX_{spec,HP}$	350	€/kW	Marina et al. (2021)
Specific investment costs EB	$CAPEX_{spec,EB}$	175	€/kW	Münnich et al. (September 2022)
Specific investment costs GB	$CAPEX_{spec,GB}$	0	€/kW	Own assumptions
COP HP 1MW	COP_{HP}	1.84	–	Own calculation
COP HP 2MW	COP_{HP}	2.54	–	Own calculation
COP EB	COP_{EB}	1	–	Miara (2026)
Efficiency GB	η_{gas}	0.9	–	Miara (2026)
Electricity price	c_{el}	0.16	€/kWh	Statista (2026a)
Gas price	c_{gas}	0.07	€/kWh	Statista (2026b)
HP Maintenance costs	c_{Main}	3	€/kW	Jovet et al. (2022)
HP Replacement costs	c_{Rep}	0.0018	€/kWh	Jovet et al. (2022)
HP Labor costs	c_{Lab}	500	€/y	Own assumptions
Total O&M costs HP	$C_{O\&M,HP,1MW}$	12.5	k€/y	Dominkovic (2016); Agora Energiewende and Fraunhofer IEG (2023)
Total O&M costs HP	$C_{O\&M,HP,2MW}$	24.5	k€/y	Dominkovic (2016); Agora Energiewende and Fraunhofer IEG (2023)
Total O&M costs GB	$C_{O\&M,GB}$	400	€/y	Buderus (2026)
Total O&M costs EB	$C_{O\&M,EB}$	200	€/y	Dominkovic (2016)

based on annual consumption in GWh, while natural gas prices c_{gas} are differentiated into five groups, based on consumption in TJ. For each country, the price values for the first and second half-years were each arithmetically averaged, according to Eqs. (10) and (11).

$$c_{el} = \frac{\sum_{i=1}^{4} P_{kWh,i}}{4} \tag{10}$$

$$c_{gas} = \frac{\sum_{i=1}^{5} P_{TJ,i}}{5} \tag{11}$$

The resulting price ratio of electricity to gas r_P is calculated using Eq. (12). The calculated price ratio already includes taxes, surcharges, and fees. A lower r_p value is advantageous for the use of heat pumps, as it reduces the required COP needed to achieve cost parity with gas-fired boilers.

$$COP > r_p = \frac{c_{el}}{c_g} \tag{12}$$

3.4. Heat pump standardization and modularization concept

This study proposes a methodological framework for the development of standardized industrial heat pump systems, with a primary focus on thermodynamic performance and scalability. The general modularization approaches presented in Section 2 combine functional decomposition, interface standardization, and application-oriented configuration to create a repeatable design framework. Since a heat pump system consists of numerous subsystems, the first step was to perform a functional decomposition of the system. The heat pump system was divided into functional modules according to the thermodynamic functions within the cycle. Based on the four main components, this functional division results in four core modules: compressor unit, expansion unit, source or evaporator unit and sink or condenser unit. Auxiliary systems, such as hydraulic and control units, are categorized as functional support elements. While essential for operation, these auxiliary components offer lower potential for cost-driven standardization compared to the core thermodynamic components and are thus treated as standardized interfaces. Table 8 provides an overview of the functional breakdown of the heat pump into modules.

Building on this breakdown, the methodology focuses primarily on standardized compression technology and modular performance scaling, as these elements offer the greatest potential for efficiency and cost optimization and manufacturing economies of scale. The modular strategy relies on a clear interface concept ensuring both mechanical compatibility and thermodynamic consistency. Performance scaling is achieved exclusively through the parallelization or cascading of standard units, thereby avoiding custom designs while providing standardized solutions for specific thermal capacities. Acknowledging that experimental validation is not mandatory for a conceptual framework at this stage, this study employs a scenario-based simulation approach anchored in real-world reference cases. Operational data from an existing pulp and paper plant from the project EEETHOS (Eeethos project, 2025), covering documented thermal load profiles, flow temperatures, and pressure requirements, serve as the primary benchmarking baseline against existing deployed systems. This real-case verification is utilized to demonstrate that the modular concept's adaptability and thermodynamic performance are directly comparable to established conventional solutions. The analysis and presentation of a modular heat pump concept in this study is divided into two parts, starting with the matching of the product portfolio and the weighted cost analysis. In the portfolio matching, the study maps the required compressor capacities to the standard product lines of leading manufacturers GEA (GEA, 2025), Mayekawa (Mayekawa, 2025), and Spilling (Spilling, 2024). The product portfolios of these manufacturers are used to establish a correlation between the available compressor components and the compressor sizes calculated in the simulations. Products from these portfolios benefit from economies of scale and avoid the research and development costs associated with prototype development or custom design, which is typical for non-standardized industrial heat pumps. The weighted cost analysis is used to determine the impact on total system costs, using a weighted average. The calculation method is described below. To quantify the potential cost savings associated with standardization and mass production, this study uses a weighted cost estimation model based on detailed parts lists provided by the heat pump and compressor manufacturer Mayekawa (Mayekawa, 2025) and Spilling (Spilling, 2024). The manufacturers are selected according to their responsibility for the demo cases in the SPIRIT project. Since the specific absolute component costs are proprietary information, a normalization approach is used to ensure transparency while maintaining confidentiality. The absolute costs C_i of each component were available from detailed manufacturer tables. To normalize these values, the total system cost C_{tot} was calculated as the sum of all individual component costs. The cost of each individual component is then expressed as a relative cost share CS_i in percent:

$$CS_i = \frac{C_i}{C_{tot}} \cdot 100 \tag{13}$$

This ensures that the analysis focuses on the proportional impact of each component on the total system cost. The manufacturers provided data on potential cost reduction potential c_R of a component achievable through economies of scale, specifically for production volumes of 10 units and 100 units. This data reflects the difference between custom manufacturing and standardized series production. To determine the impact on the overall system, a weighted average cost reduction was calculated. This approach takes into account the fact that a percentage savings on a high-cost component contributes more to the total savings

Table 8
Modules and function of a heat pump system.

	Module	Function	Type
Required	Compressor unit	Increase in pressure and temperature level	Scroll, screw, piston compressors
	Expansion unit	Pressure reduction of the refrigerant	Turbine, expansion valve
	Source/Evaporator unit	Absorption of thermal energy from the heat source	Plate- or shell and tube heat exchanger
	Sink/Condenser unit	Heat transfer and process heat extraction	Plate- or shell and tube heat exchanger
Functional	Hydraulic unit	Fluid circulation within the auxiliary systems	Pumps, valves, safety valves, filters
	Control unit	Monitoring, control, and optimization of operation	Controller, sensors (temperature, pressure, flow)
Optional	Recuperator unit	Internal heat transfer (especially Brayton cycle)	Plate- or shell and tube heat exchanger
	Storage unit	Buffer storage	High-temperature storage

Table 9
Parameter variation with respect to the efficiency of isentropic compressors, heat losses, specific capital costs.

Parameter	Reference value	Parameter Variation	Increment
$\eta_{is,C1}$	0.75	0.7–0.8	5%
$\eta_{is,C2}$	0.7	0.65–0.75	5%
$\eta_{is,C3}$	0.78	0.73–0.83	5%
$\eta_{is,C4}$	0.78	0.73–0.83	5%
$\dot{Q}_{loss,th}$	0%	1%–5%	1%
$CAPEX_{spec}$	€350/kW	€200/kW–€500/kW	€50/kW

than the same percentage savings on a low-cost component. The average cost reduction $C_{R,av,i}$ for a given production volume is calculated according to Eq. (14):

$$C_{R,av,i} = \frac{CS_i \cdot c_{R,i}}{100} \tag{14}$$

The resulting $C_{R,av,i}$ value represents the potential reduction in total production costs achieved by switching from a custom solution to standardized mass production of 10 or 100 units. To determine the potential relative total cost savings of the heat pump, the individual weighted cost savings calculated are aggregated.

$$R_{HP} = \sum_{i=1}^{n} C_{R,av,i} \tag{15}$$

The sum of these weighted values corresponds to the relative total cost savings potential R_{HP} in percent of the heat pump assembly for a given production volume. This system-wide reduction percentage is then applied to the estimated CAPEX of the modular heat pump configurations of 1 MW and 2 MW. The absolute cost savings S_{abs} are calculated as follows:

$$S_{abs} = CAPEX \cdot R_{HP} \tag{16}$$

Here, CAPEX represents the base capital costs of the standardized heat pump configuration in year t=0, and R_{HP} represents the summed, weighted cost reduction potential derived from the component analysis. This approach enables a direct quantification of the economic benefits of standardization and mass production.

3.5. Sensitivity analysis

To ensure the robustness of the reported results, this section presents the assumptions for the sensitivity analyses conducted as part of this study to evaluate the effects of various key parameters on both the thermodynamic performance and the economic viability of the proposed system. To facilitate this analysis, a baseline scenario is defined, from which deviations for the individual parameters are introduced. The variation of the parameters with the reference values, the range of the parameter variation and the increment are presented in Tables 9 and 10.

3.5.1. Thermodynamic performance

The simulation models cannot be experimentally validated within the scope of this study. To compensate for the lack of experimental data and ensure that uncertainties are appropriately accounted for, sensitivity analyses were performed with respect to compressor performance and heat losses. In particular, the influence of different isentropic efficiencies of compressors, as well as he total heat losses in the system, on the COP was investigated. The isentropic efficiencies of the compressors C1, C2, C3 and C4 are varied in 5 % increments, starting from the initial reference value. The varied isentropic efficiencies and the resulting COP is shown in Table A.16 in Appendix A. The total heat loss of the system was varied in 1 % increments in the range of 0 % to 5 %. These losses were modeled in EBSILON using component No. 13 by assigning specific heat loss parameters to the piping components. It is assumed that the total heat loss per cycle is distributed evenly among three pipe components, which, as shown in Figs. B.12 and B.13, have been integrated into each of the heat pump's three sub cycles. Although the physical geometry typically leads to asymmetric heat losses, a uniform distribution was assumed for this study, and geometric details were neglected in the simulations. This simplification is justified by the fact that the focus is on the thermal losses of the entire system. Therefore, the specific heat loss parameter for each component was set to one-third of the total system heat loss. The calculated specific heat loss per pipe is applied simultaneously to all three piping components and is shown in Table A.17 in Appendix A. Parameters were varied independently, with all other values held constant.

3.5.2. Economic viability

The economic sensitivity analysis focuses on the effects of uncertainties regarding the assumed heat pump investment costs in €/kW. The specific investment costs of the heat pump system, which are based on the reference assumption of €350/kW, were varied within a defined range from a minimum of €200/kW to a maximum of €500/kW in increments of €50/kW to determine how these fluctuations affect the payback period. Furthermore, the calculated absolute cost savings are applied to investigate the influence of these cost reductions on the payback period.

3.5.3. Thermal capacity

To account for uncertainties in the assumptions regarding production volume, specific heat demand, number of plants, and operating hours, which are used to calculate the thermal capacity, parameter variations based on Table 10 are performed for the paper and dairy case. The parameters were varied independently of one another, while all other values were kept constant. The parameters m, e, N and t are varied in 10 % increments starting from the initial reference value. The arithmetic mean value of N and m, calculated as shown in Tables 2 and 3, serves as the respective reference value at the start of the sensitivity analysis. The resulting thermal capacity P_{th} is shown in Table A.18 in Appendix A.

Table 10
Parameter variation with respect to the production volume, specific heat demand, number of plants, and operating hours.

Parameter	Reference value Paper	Reference value Dairy	Parameter Variation	Increment
m [t/year)	11,078,500	228,887	−20%–+20%	10%
e [MWh]	1.37	2	−20%–+20%	10%
N [–]	80	41	−20%–+20%	10%
t [h]	8,000	8,000	−20%–+20%	10%

4. Results and discussion

4.1. Industry needs and market opportunities

Based on the literature review of the potential for integrating industrial HTHPs, the paper and food industries were identified as sectors with high potential for the use of HTHPs to provide process heat and steam. Since the current state of the art allows for heat pump systems with temperatures of up to 200 °C, the research focus was narrowed to industrial sectors in which a significant portion of the process heat demand lies below this temperature threshold. Accounting for about 67%, the paper and food industries are the most important sectors in terms of process heat demand up to 200 °C. These figures illustrate that the selected sectors are particularly well-suited for evaluating modular heat pump concepts, both due to the appropriate temperature range and the substantial scale of their process heat demand. Although other sectors, such as the chemical industry, also have considerable potential in terms of low-temperature process heat demand, a key argument for choosing the paper and food sectors over the chemical industry is the widespread use of drying processes and a constant thermal base load. While complex distillation and reaction processes are the focus in the chemical industry, drying processes are standard in the paper and food industries, e.g., paper drying, spray drying of milk powder. Drying processes typically exhibit stable thermal load profiles that are ideally suited for the modular design of heat pumps, as individual modules can be gradually brought into operation to balance out load peaks. The findings from this study can thus be applied to other industries with similar process characteristics. The principles of modularization for drying applications can be directly applied to other sectors, such as the non-metallic minerals industry or the textile and leather industry, where drying processes also account for the majority of energy consumption. Given these process characteristics, drying processes in the paper and food industries were specifically used as a reference in the development of modular heat pump concepts as part of this study. However, to demonstrate the need for modularization compared to conventional systems, it is crucial to place these technical requirements within the context of the global industrial landscape. The thermal capacity of a process step is not only determined by temperature limits but also depends directly on the local production volume per plant and the number of production facilities. Therefore, the required heat pump capacity can vary significantly from site to site even for the same process. A detailed, country-specific analysis of these sectors highlights the differences in production volume and plant density.

In Fig. 4, the total pulp and paper production volume in tonnes per year with the related number of pulp and paper mills and the resulting thermal heat pump capacity in MW is presented.

The bar charts represent the production volume of individual countries and the calculated thermal heat capacity while the line illustrates the respective density of mills. A comparison of country-specific production volumes shows that Germany is the leading market with the highest absolute production volume, closely followed by Sweden and Finland. However, the number of plants reveals interesting differences between the countries. Germany has a high density of plants combined with the highest total production volume. This points to a mixed market that includes both large integrated corporations and many specialized small and medium-sized enterprises. In contrast, Sweden has a significantly smaller number of plants, 66 compared to 150 in Germany, even though its production volume is nearly as high as Germany's. This suggests that the Swedish industry is heavily concentrated in a few, highly efficient large sites that achieve a high throughput per plant. The chart also shows that Italy, with 151 mills, has the highest number of systems, while its total production volume is lower than in the Nordic countries or in Germany. This is an indicator of a highly fragmented market structure with many small- to medium-sized, independent companies. Finland, on the other hand, lies between these extremes, it has a high production volume with a moderate number of systems, which in turn points to large sites with high individual capacity. These country-specific structures are crucial for the development of modular heat pumps. They demonstrate the need for several heat pump module sizes. In countries such as Italy, the decentralized structure results in many individual sites with limited individual heating loads. Here, small-scale modular heat pump concepts are often more economically and technically efficient. In contrast, large Swedish or German sites, which are characterized by an extreme concentration of production volume per plant, require a high MW load per plant. In this context, modularity is the key tool precisely because of this heterogeneity. Modularization makes it possible to tailor a system precisely to local needs. Through modularization, a system with many modules connected in parallel can be designed in Sweden to handle high peak loads, while smaller modules can be used in Italy, both based on the same scalable technology platform, without the need for custom individual solutions or prototypes. The specific sizing of the heat pump modules is illustrated by the calculated heat pump capacity. This shows that thermal requirements do not increase linearly with production volume. The range between 9 and 53 MWth underscores that a modular approach is essential for precisely adapting the systems to the specific process requirements of each country, thereby optimizing efficiency and investment costs. In this study, the arithmetic mean of 27 MW thermal heat pump capacity served as a reference value for the basic modular concept and goes well with the literature values researched in the review, which is consistent with the values identified in the literature review. It is therefore recommended that, in subsequent studies, the design of these concepts be tailored specifically to the individual performance requirements of each country in order to ensure a more precise adaptation and to facilitate comparisons with other countries. Furthermore, since the majority of the determined thermal loads exceed 10 MW, the development of modular systems should focus on large-scale base units of approx. 10 MW to achieve a high total thermal output through the parallel connection of multiple modules or their integration with supplementary technologies such as electric boilers.

To further validate the concepts for modular heat pumps, the milk powder production, a sub sector of the food industry, was examined due to the dominant process step of spray drying. Fig. 5 shows the production volume in tonnes per year with the related number of milk powder facilities and the resulting thermal heat pump capacity in MW.

However, analyzing the market potential for modularization in this sector proved difficult due to inconsistent data availability. The analysis is based on production data for the year 2025/26 and the number of processing plants of Germany in 2024 and of France and the Netherlands in 2019. It should be expressly noted that the lack of facility data for countries such as Sweden, Spain, Belgium, and Denmark, or other EU countries not listed, is due to the fact that this information is not publicly available. While total production volumes

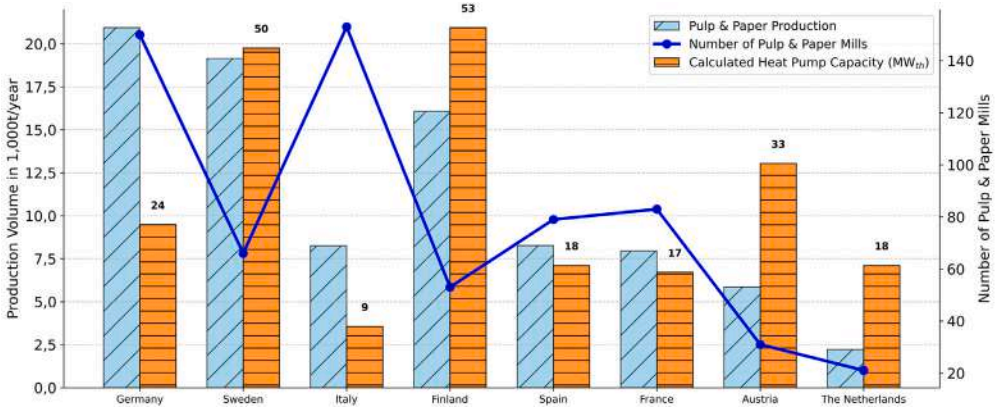


Fig. 4. Paper production volume, number of paper mills and calculated thermal heat pump capacity in MW for selected EU countries.

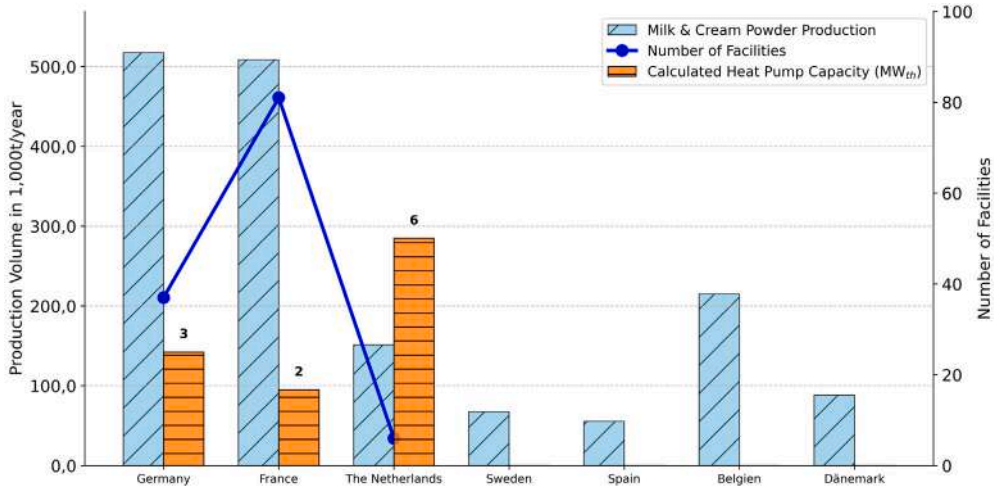


Fig. 5. Production volume of milk and cream powder (05/25–04/26), number of facilities and calculated thermal heat pump capacity in MW for selected EU countries.

can be reliably recorded statistically, the exact number of individual milk powder plants in these markets is not publicly available and is often not published in detail by industry associations. Nevertheless, the detailed data for the leading producing countries Germany, France, and the Netherlands enable a well-founded analysis of market structures. Both Germany and France are market-leading manufacturers, each with an annual production volume of approximately 510,000 tonnes. However, an analysis of the distribution of milk powder production facilities reveals significant structural differences. France has a high density of plants, with 81 facilities. This points to a decentralized production structure that requires a large number of medium-sized heat pump plants. Germany, on the other hand, achieves a comparable production volume with significantly fewer facilities of 37. This results in higher average production volumes per plant and, consequently, a higher thermal load per site. Despite a moderate production volume, the Netherlands has a very small number of facilities at 6. This is an indicator of an extremely high concentration of capacity. These few remaining plants must handle enormous production volumes, which points to extremely high, concentrated thermal loads in the megawatt range per plant. This heterogeneity in market structures, similar to that found in paper production, underscores the need for the modular heat pump concepts developed in this study. While in France a large number of modules can be designed for decentralized loads, modularization in Germany and the Netherlands allows for scaling to accommodate the few, but very high-capacity, large-scale plants. Thus, the milk powder industry provides further evidence of the transferability of these concepts to sectors with highly fluctuating plant density and heterogeneous heat demand. The analysis of specific heat output requirements reveals, similar to the paper industry, a nonlinear relationship between production volume and required heat output. Due to the limited availability of operating data, a calculated power range of 2 to 6 MWth was determined for micro-powder production. Based on that, an arithmetic mean of 4 MWth was derived as a reference value for the basic concept, which is consistent with the ranges of values identified in the literature review. Since all determined load profiles are consistently below the 10 MW threshold, the technical development of these systems focuses on small- to medium-sized modules. The low specific heat load suggests that the total demand can be met by one medium heat pump unit or a combination of small units. This means that, in contrast to the large-scale sites discussed previously, the use of hybrid supplementary technologies is not absolutely necessary in this application area.

A sensitivity analysis was conducted to evaluate the robustness of the proposed thermal capacity of 27 MW for the paper and 4 MW for the milk powder spray drying process to quantify the effects of uncertainties in the assumptions N, m, t, and e on the thermal capacity. It is important to note that the calculations of thermal capacities of 27 MW and 4 MW are based on the arithmetic mean of all values listed in Tables 2 and 3. In the sensitivity analysis, however, the reference state at 0% of P_{th} is calculated using Eq. (2) with the respective reference values of N, m, t, and e listed in Table 10. Consequently, there is a deviation between the averaged thermal capacity and the thermal capacity calculated at the 0% reference point. Fig. 6 shows the graphical results of a ±20% deviation for each parameter relative to its

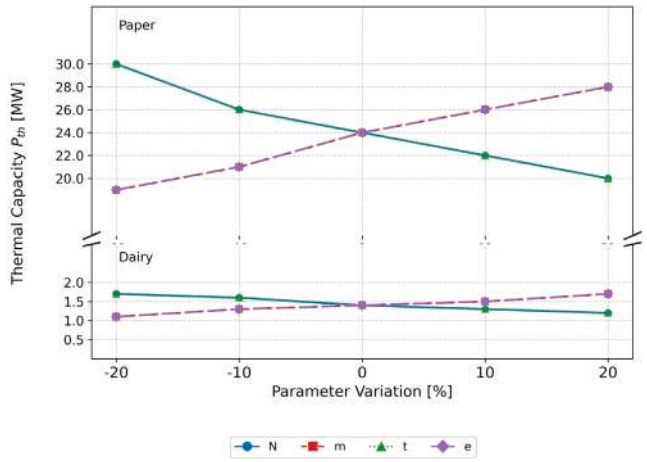


Fig. 6. Variation of the Parameters m, e, N, and t and their effect on the resulting thermal capacity for the paper and milk powder drying process.

reference values. The numerical results of P_{th} are shown in Table A.18 in Appendix A.

For the paper process, the thermal capacity calculated at the reference point with a 0% deviation is 24 MW, while in the dairy process it is 1.4 MW. Thus, both average values deviate by approximately 3 MW from the standard calculation. Although this sets a slightly lower threshold for the sensitivity analysis, it ensures that the model reflects stability under average conditions rather than under idealized peak design conditions. The sensitivity analysis shows that, due to the linear inverse relationship in the corresponding equation, the thermal capacity depends very strongly on the parameters N and t. When calculating the heat capacity of the paper drying process, a deviation of +20% in N or t results in a significant reduction in capacity from the calculated initial value of 24 MW to approximately 20 MW. Conversely, a deviation of −20% leads to a significant increase to 30 MW. This corresponds to a total deviation of 10 MW and approximately 41% of the base value. In contrast, the parameters m and e exhibit a direct positive correlation, accordingly, an increase of +20% leads to a rise in capacity from 24 MW to 28 MW. Although the absolute influence of m and e is somewhat smaller compared to N and t, they remain crucial factors for increasing system performance. The analysis of the dairy system shows identical structural behavior to that of the paper system, which confirms the model's scalability. The trends for all four parameters are linear and correspond to the paper scenario with higher capacity. A +20% deviation in the inverse parameters N and t reduces the output from the 0% reference value of approximately 1.4 MW to approximately 1.2 MW, while a +20% increase in the direct parameters m and e increases the output to approximately 1.7 MW. This sensitivity analysis confirms that the underlying thermal capacity assumptions of 27 MW for the paper drying process and 4 MW for the dairy system are sufficiently accurate for the preliminary development of modular heat pump configurations. However, the analysis underscores that the underlying input parameters N, m, t, and e have a profound impact on the calculated thermal capacity. The results show that P_{th} is most strongly influenced by uncertainties in the parameters the number of facilities and the operating time of the plant. This sensitivity stems primarily from their inverse proportionality in the relevant equations. Underestimating these values carries a significant risk of undersizing the modules, which could result in the required thermal load not being met. Conversely, an overestimation leads to the selection of oversized modules. While this provides a safety margin, it results in inefficient utilization due to unused capacity and excessive costs. Consequently, the realistic sizing of modules to meet heating demand must be based on extremely reliable and robust empirical data. For the robust design of heat pump modules, future validations must focus on the precise characterization of N and t. Their inherent uncertainty remains the dominant factor causing deviations in the predicted thermal output, which is why precise measurements are crucial for the economic and technical feasibility of such systems.

4.2. Simulation results

Based on the analysis of the numerical simulations, the proposed heat pump systems are evaluated, with a focus on their thermodynamic performance and technical feasibility. The results also provide insights into possibilities for a modular design and the standardization of components. The pressures, temperatures, specific enthalpies, and mass flows resulting at each state point in the heat pump model based on the simulation parameters are listed in Table A.19 for the paper drying 2 MW heat pump and in Table A.20 the spray drying 1 MW heat pump, respectively. The simulation of the 2 MW heat pump for paper drying shows that the proposed configuration is thermodynamically capable of meeting the requirements of the drying processes. The system successfully generates process steam at a target pressure of approximately 10 bar and a temperature of 273 °C. This results in a COP of 2.54. To achieve the target conditions for the process steam without exceeding material limits or efficiency thresholds, the steam compression at the heat sink is divided into two separate stages with intermediate cooling. The compression from 1.8 bar to 4.8 bar in C1 results in a pressure ratio Π of 2.7. The steam is cooled from 254 °C to 160 °C at constant pressure, thereby optimizing the enthalpy for the final compression step. In addition, the steam after condensate injection can be used to dry different paper grades. Final compression in stage C2 increases the pressure from 4.8 bar to 10.1 bar, resulting in a pressure ratio of 2 and a steam temperature of 273 °C. Steam at temperatures of up to 220 °C is typically required for paper drying. Nevertheless, this configuration with a higher steam temperature demonstrates that the system is capable of achieving various temperature ranges, including higher ones which could be relevant not only for drying processes in the paper industry but also for other processes that require higher steam temperatures. This bridges the gap between the presented modular concepts for drying processes in the paper and food industries and processes in other industrial sectors. A key finding of this simulation-based analysis is that the calculated pressure ratios are compatible with standard industrial compressor technology. In the process steam cycle, the low pressure ratios of 2.1 and 2.7 are characteristic of piston compressors. This confirms that commercially available systems can efficiently handle the final steam pressure increase. The refrigerant cycle operates at a pressure ratio of 6.1, which corresponds to the optimal operating range of commercially available screw compressors. While the thermodynamic principles remain consistent across all module sizes, the simulation results indicate a critical limitation regarding component availability as capacity increases. Since a single 2 MW heat pump module accounts for only a fraction of the typical thermal load of approximately 27 MW per paper mill, modules with higher capacity are required to ensure efficient process integration. Consequently, the thermodynamic performance and component sizing for 5 MW and 10 MW configurations were derived through theoretical linear scaling based on the detailed 2 MW baseline simulation. This scaling approach focuses specifically on extrapolating the electrical drive power of the compressors. These scaled values serve as primary indicators of market viability. By comparing the required rated power with standard manufacturing catalogs, this study determines whether specific module sizes can be realized using commercially available components or whether they fall into the category of custom-built prototypes. Table A.21 provides an overview of the simulation results for the component sizes of the 2 MW configuration, as well as the scaled results for the 5-, and 10 MW heat pump modules used for paper drying. For the 2 MW heat pump module, the required electrical drive power of C3 is approximately 474 kW, which is certainly compatible with commercially available options.

However, scaling this design up to a single-stage 10 MW configuration increases the power requirement to nearly 2370 kW. Such high-power individual units fall far outside the scope of typical manufacturing portfolios and would require the development of custom prototypes or specialized technical solutions. The steam compressors operate with a drive power of approximately 150–164 kW. Scaling up to the 5 MW module requires units with approximately 400 kW, which is still within market standards. For the 10 MW module, however, the required power increases sharply to approximately 815 kW and 755 kW, respectively. This nearly twofold increase in the size of standard components means that the system enters the realm of custom-built hardware, where procurement costs and lead times rise significantly. The analysis shows that directly scaling the components inevitably takes the system beyond the realm of standard components. To mitigate this, the proposed design advocates for standardized modular parallelization. Instead of relying on individual, oversize drives, higher thermal output is achieved by connecting multiple standard compressor modules in parallel. This strategy ensures that all core components remain within their optimal production series and eliminates the risks associated with developing custom prototypes.

As well as the 2 MW configuration for the paper drying process, the 1 MW heat pump for milk powder spray drying shows that the proposed cascade configuration is thermodynamically capable of meeting the process requirements. Process steam is successfully generated at the heat sink at a target pressure of 10 bar and a temperature of 313 °C. This results in a COP of 1.84. In stage C1, compression takes place from 1.2 bar to 3.5 bar, resulting in an outlet temperature of 255 °C. The steam is cooled to 149 °C in an intermediate cooling stage to prepare it for the final pressure increase. The final compression to 10.1 bar in stage C2 results in the desired steam state. Since the steam compression cycle of both systems is structured identically and differs only minimally in terms of parameters, the required compression work can be handled as well as in the paper drying system. The pressure ratio Π for the stage C1 and stage C2 is 2.9 in each case. These low-to-moderate pressure ratios are very characteristic of piston compressors and ensure that commercially available units can handle steam compression efficiently and without mechanical overload. At the same time, the stage C3 in the refrigerant cycle operates at a pressure ratio of 5.4 and 5.3 in stage C4 in the cascade ammonia cycle, which aligns perfectly with the optimal design range for screw compressors. Since a thermal capacity of 4 MW was derived, component sizing for a 2 MW and 5 MW heat pump configurations were derived through theoretical linear scaling based on the 1 MW baseline simulation. The calculated and scaled compressor power requirements are presented in Table A.22. The results reveals a market threshold with regard to the compressors C3 and C4. In the base scenario with a 1 MW heat pump, the compressor C3 requires a drive power of 222 kW, while C4 requires 130 kW. When scaled up to the 2 MW module, the power requirement for C3 increases to 555 kW. These power ratings fall well within the standard catalogs of industrial compressor manufacturers, ensuring the availability of off-the-shelf components. When scaled up to the 5 MW module, the power required for compressor C3 increases significantly to 1110 kW. This specific power output marks a critical threshold, as compressors exceeding 1 MW are often classified as custom-built units rather than standard industrial products. The analysis confirms that directly scaling up the module to higher capacities inevitably means that critical components, fall into the realm of custom-built prototypes. The development of such plants involves high costs and long lead times. Consequently, the results support the proposed strategy of modular parallelization as the optimal approach for industrial deployment. Condensate injection between the two piston compressor stages serves to meet mechanical requirements, e.g., internal lubrication and protection of the piston rings, while simultaneously optimizing thermodynamic performance. Based on simulations under steady-state operating conditions at the design point, the injection lowers the outlet temperature of the intermediate stage and reduces the specific power requirement, leading to an improvement in the cycle's COP. At full load, the mass flow of the injection is optimized to maximize cooling capacity and prevent component overheating. Partial-load conditions were not explicitly simulated in this study, as the methodological focus is on performance at the design point and modular scalability. However, given the thermodynamic behavior of multistage piston compressors and the injection process, it is expected that injection will remain beneficial even under partial load, even with a slightly reduced marginal COP gain attributable to lower refrigerant mass flows, shifted compressor operating points, and potential control-related efficiency losses during load transients. These dynamic effects are explicitly acknowledged as a modeling limitation and will be taken into account in future transient simulations.

The decision to use a cascade heat pump instead of a single-stage configuration for drying dairy products is based on a multi-criteria analysis that takes into account both thermodynamic performance and operational reliability. Both the cascade and single-stage configurations were evaluated in a preliminary simulation study. To focus the present study on the cascade configuration, the single-stage variant is not presented here but was simulated using the same routines for calculating thermodynamic properties and the same boundary conditions as the cascade system. The quantitative comparison presented here examines the two variants in terms of COP, system complexity and percentage costs. The simulations for all configurations are based on the assumption of a minimum temperature difference of at least 10 K for all gas-to-liquid heat exchangers. These values fall within the range of typical technical recommendations. Under the given boundary conditions, the simulation results for the single-stage 1 MW heat pump showed a COP of 2.19, a pressure ratio of 7.7 at C3, and a temperature difference of approximately 3 K at the air-to-liquid LTHX. The higher system efficiency of the single-stage variant is generally attributable to the smaller number of compressor components and the correspondingly lower efficiency-related losses. While the single-stage C3 variant has a compressor capacity of approximately 260 kW, the combined compressor capacity of C3 and C4 in the cascade configuration totals approximately 350 kW, which results in higher losses and a reduction in the system's COP. Although a COP of about 2 is acceptable, this configuration was rejected for two reasons. First, a temperature difference at the LTHX of less than 10 K is not technically feasible, result in very large transfer areas and are therefore usually not cost-effective. Second, a single-stage pressure ratio of 7.7 is at the upper limit of commercial market ready available compressor technology. Commercially available machines, such as screw compressors, cannot reliably achieve a pressure ratio of approximately 8 in a single stage. Therefore, a cascade configuration was simulated under the same boundary conditions. The simulation results for the cascade variant shows a COP of 1.84, which corresponds to a performance reduction of about 16% compared to the single-stage configuration. However, due to the distribution of the temperature rise from the source to the sink over two cycles and the two compression stages C3 and C4, the pressure ratio decreases from an initial 7.7 to approximately 5 for C3 and C4, respectively, which falls comfortably within the range of commercially available compressors. In addition, the minimum temperature difference at the LTHX is now 29 K, which is significantly higher than the required minimum value of 10 K and thus ensures design reliability. Another factor in favor of choosing a cascade configuration is system safety. Due to butane's low molecular weight and high flammability, classified as safety class A3, a single-stage system would require a larger refrigerant charge than the cascade configuration, which features a closed secondary cycle and a smaller charge. In terms of complexity, the two variants differ fundamentally in the number of main components and the associated piping, as well as in the measurement and control technology. In general, a cascade configuration is significantly more complex than a single-stage design, as it involves more components and, consequently, more connections and drive systems. For the analysis, the system without a mechanical vapor compression system is compared, since this

system is identical in both configurations. Compared to the single-stage variant, the cascade system includes one additional compressor, one expansion valve, and one heat exchanger, increasing the total number of main components from five in the single-stage system to eight in the cascade system. However, this increased complexity is offset by gains in performance and operational reliability. The increased complexity of the cascade configuration compared to the single-stage configuration has a direct impact on the total cost of the heat pump. To perform a rough quantitative cost comparison between the cascade and single-stage configuration, the percentage-based component costs listed in Table 14 are used. The costs listed in the table correspond to 100% of the cost of a pentane-ammonia cascade heat pump manufactured by Mayekawa. It is assumed that the unit costs of each component group scale linearly with the number of units, so that the cost share of a component in the single-stage configuration is calculated by scaling the unit costs from Table 14 by the number of units of that component in the single-stage system. Only the main component groups, the compressor, the motor system, and the heat exchanger, were taken into account. The quantities of the remaining components could not be reliably quantified for the single-stage system at the current stage of the simulation and were therefore excluded from this estimate. The three main component groups account for 24.5% of the total investment costs of the cascade system with 6%, 5%, and 13.5% for 2, 3, and 4 units, respectively. The corresponding percentages for the single-stage configuration with 1, 2, and 3 units are 16.4% 3%, 3.3%, and 10.1%, respectively, meaning that the single-stage configuration reduces the costs of these component groups by approximately 8% of the total investment costs of the cascade system. Given the neglect of the remaining components and costs, this value represents a lower bound on the direct savings in component costs. Additionally, in terms of operating costs, the higher COP of the single-stage configuration results in lower specific electricity consumption, while the cascade system requires higher compressor power. The economic analysis over the system's service life thus depends on the annual operating hours and the electricity price. Overall, the single-stage configuration offers a limited and, in some cases, uncertain direct cost advantage, while its feasibility is fundamentally compromised by the pressure ratio and the LTHX temperature difference. The cost comparison therefore does not refute the thermodynamic choice of the cascade configuration but confirms that the cascade is chosen primarily for reasons of technical feasibility, with a quantified and moderate cost increase. In summary, the minimal temperature difference at the LTHX and the compressor's pressure ratio are two key factors that favor the cascade configuration, despite its lower COP. For purely thermodynamic and application-related reasons, the cascade configuration was therefore selected for this study. For the compressor simulation, a general assumption based on the technical literature, assuming turbo compressors as the base technology, was initially selected. This choice is based on their superior specific speed and efficiency characteristics at high process temperatures and under the specific cycle configuration. The COP values and compressor power consumption determined in the simulation are in good agreement with the values from the studies cited in the literature review. The results also show that the assumed isentropic efficiencies correspond to the current state of the art. The robustness of these basic assumptions is confirmed by structured sensitivity analyses subsequently performed for the isentropic efficiency.

4.2.1. Influence of isentropic compressor efficiency

In this study, the influence of the isentropic efficiency of the compressor on the COP of the heat pump system was systematically investigated to ensure the validation of the results and to determine the dominant role of individual compressor stages in the thermodynamic cycle. The dependence of the COP on the isentropic efficiency for the 1 MW and 2 MW systems is shown in Fig. 7. The points marked with a cross represent the initial values for the parameter variation. As can be seen in the figure, both system sizes show identical trends, with a

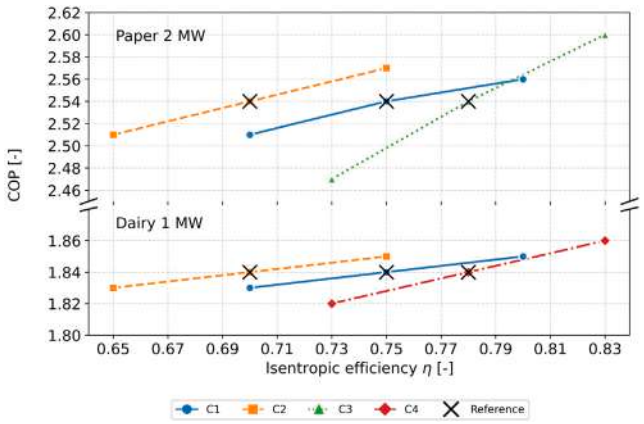


Fig. 7. Effect of variations in isentropic compressor efficiency on the COP for a 1 MW and 2 MW heat pump system.

clear positive correlation between increasing isentropic efficiency and increasing COP. Due to its system configuration, the 2 MW system consistently shows significantly higher COP values, approx. 2.46 to 2.60, than the 1 MW system, approx. 1.80 to 1.86. In both systems, compressor C3 has the most positive incline of the curve. This indicates that C3 is the component in the system that is most critical to overall efficiency. Thermodynamically, this can be explained by the fact that compressor C3 has the highest pressure ratio and makes the largest contribution to the specific compressor work in the cycle. Accordingly, an improvement or deterioration in isentropic efficiency at this point has the greatest leverage on the system's overall efficiency. In the 1 MW system, the curves for C3 and C4 are identical due to identical design parameters. Thus, the same conclusions apply to C4 as to C3. For the 2 MW system, it can be concluded that compressor C3 is the most critical component. In the 1 MW system, C3 and C4 are the primary factors. In general, C3 is more sensitive to losses under unfavorable operating conditions, but offers the highest theoretical performance potential when optimally designed. In contrast to C3, the curves for C1 and C2 are significantly flatter. Particularly in the 2 MW system, C2 exhibits a relatively stable performance over the investigated parameter range. The lower slope correlates with lower pressure ratios and as a result, the influence of isentropic efficiency on the system's overall performance is less pronounced compared to the other compressor stages. In conclusion, the results of this parametric analysis show that the qualitative trends and optimization results remain robust in terms of COP values even with significant deviations from the nominal design values. The isentropic efficiencies examined in the simulation, ranging from 0.65 to 0.83, can be directly correlated with current industrial compressor technologies. Values in the lower to middle range typically correspond to older piston compressors or less-optimized screw compressors. Higher isentropic efficiencies, on the other hand, reflects the state of the art in modern turbo compressors, screw compressors, or scroll compressor cascades. This technical classification is of central importance for the feasibility of modular heat pump systems.

4.2.2. Influence of heat losses

To quantify the impact of heat losses on the efficiency of the overall system, a sensitivity analysis was conducted with respect to system heat losses. Fig. 8 illustrates these results and shows the COP as a function of the percentage of total heat loss ranging from 0% to 5% for the two different configurations at 1 MW and 2 MW. The analysis shows a linear decrease in COP for both systems as total heat loss increases. For the 2 MW system, the baseline COP at 0% loss is approximately 2.54. Even under the maximum simulation condition of 5% total heat loss, the COP drops only to about 2.43. This corresponds to a relative efficiency loss of about 4.7% and demonstrates a high resistance to

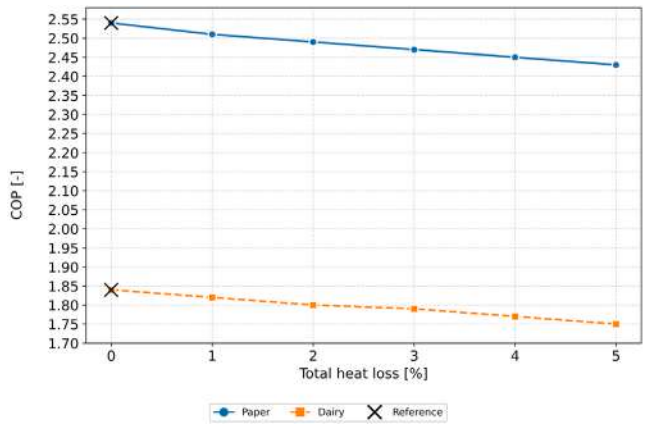


Fig. 8. Effect of total pipe heat losses on the COP for a 1 MW and a 2 MW heat pump system.

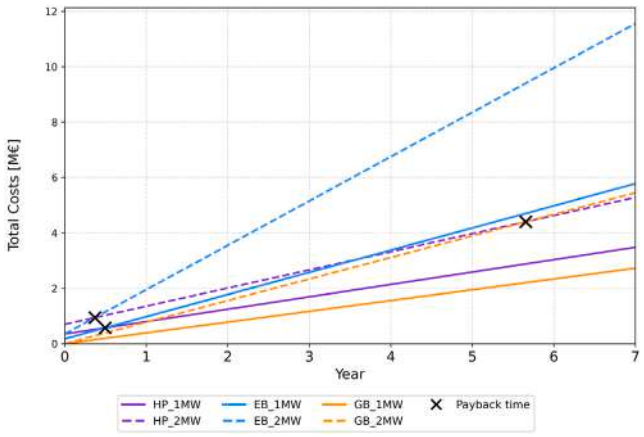


Fig. 9. Economic assessment of the total costs and payback period time for 1 MW and 2 MW heat pump compared to conventional electric and gas boiler solutions.

heat loss. The 1 MW system shows a similar trend. Starting from a COP of approximately 1.84 at 0% heat loss, the COP drops to about 1.77 at 5% heat loss. The observed correlation can be attributed to fundamental thermodynamic principles. Heat transfer from the piping leads to an enthalpy drop in the heat transfer fluid. To compensate for this energy loss and maintain the required flow temperature and capacity at the design point, the heat pump's compressors must operate under less favorable conditions, which typically results in a higher compression ratio. The parallel course of both trend lines suggests that the relative effects of piping losses are consistent regardless of the system's capacity class. The results show that while unavoidable heat losses in the piping affect efficiency, both proposed systems maintain relatively stable performance with a COP greater than 1. Nevertheless, when implementing such systems, priority should be given to thermal insulation as long as the costs are justified by the gains in efficiency and economic viability.

The results confirm that the proposed modular configurations maintain robust technical performance under these parameter variations, with COP deviations remaining within acceptable technical tolerances for industrial use.

4.3. Economical analysis

To evaluate the economic feasibility of the proposed modular heat pump systems for industrial applications and to use this as a basis for the modularization strategy, a comparative cost analysis was conducted over a operating life of 20 years and is presented in Fig. 9 for a 7-year period.

The total costs in million € including CAPEX and operational expenditures (OPEX), are compared between the two proposed 1 MW and 2 MW configurations and an electric boiler (EB) and a gas boiler (GB) as conventional alternatives. The graphs illustrates significant fluctuations in total costs depending on the underlying heating technology. The steepest cost curve is observed for electric boiler systems, particularly in the 2 MW configuration, where costs reach nearly 12 M€ in the seventh year. This rapid increase underscores the high operating costs associated with direct electric heating, which are attributable both to the system's lower efficiency compared to heat pumps and to current electricity prices. In contrast, both modular heat pump configurations exhibit significantly flatter cost curves, indicating superior operational efficiency due to their higher efficiency. Over the same period, the 2 MW system incurs costs of approximately 5.5 M€, representing a cost reduction of more than 50% compared to the EB. Although gas boiler systems have lower operating costs than the EB due to low fuel prices, they are economically disadvantaged in the long term compared to heat pump systems, as the heat pumps have a thermodynamic advantage due to a COP greater 1. Cost-effectiveness is quantified using the payback period, which is defined as the point in time at which the cumulative costs of the conventional system exceed those of the modular heat pump solution. The use of a small-scale system with a thermal output of 1 MW demonstrates exceptional cost-effectiveness. The break-even point occur very early, after just 0.92 years. This shows that, for smaller heat pump units, the investment in modular heat pumps pays for itself almost immediately due to significant energy savings. This is primarily attributable to the enormous difference in efficiency. With a COP of 1.84 compared to purely electric heating, the configuration already achieves energy savings of nearly 50%. The long operating hours typical in industrial settings therefore offset the investment costs almost immediately in this case. The short payback period for the 1 MW configuration is based on very favorable investment assumptions. It should be noted, however, that this is an optimistic scenario and the sensitivity analysis took fluctuations in investment costs into account. A key finding of the analysis is the comparison of the 1 MW heat pump system with the gas boiler. As the curve shows, the conventional gas system is more cost-effective than the heat pump over the entire period examined. This means that, for small industrial facilities, there is currently no purely economic incentive for electrification. Since the operating costs for gas remain below the savings achieved through the COP of 1.84 compared to electricity, the heat pump is economically inferior to the gas boiler without regulatory interventions such as CO_2 pricing or subsidies. For the 2 MW system, the payback period is 0.69 years compared to an electric boiler, due to its high efficiency of 2.54 and the assumed low investment costs. In contrast, the payback period compared to a gas boiler is about 6 years, which is considered very competitive. Since the technical service life of such heat pump systems is typically 15 to 20 years, there are still 9 to 14 years of pure operating profit from the fuel cost savings remaining after the payback period has elapsed. This underscores the potential of modular heat pumps not only as an environmental technology. For larger system configurations, e.g., 5 MW or 10 MW, however, it can be assumed that the payback period would be significantly longer compared to a gas boiler, which, due to higher acquisition costs, may mean that economic viability is not achieved until the second half of the system's technical service life. Therefore, it can be concluded that the focus for modular heat pumps is on smaller units rather than larger ones. Nevertheless, the modular design offers a key lever for mitigating this risk through economies of scale resulting from standardization. Since the systems consist of standardized components, particularly the compressors, synergies from mass production can be leveraged. For larger plant capacities, such as 5 MW or 10 MW, this means that the specific investment costs per kW could potentially decrease due to manufacturing efficiency. Should

these economies of scale materialize, investment costs for large-scale plants would become more affordable, and payback periods would be significantly shortened even at higher capacities. Thus, the modular approach serves not only as a concept for load balancing but also as a strategy for cost control on a large scale.

4.3.1. Impact of energy price ratios on heat pump economic viability

The economic feasibility of the proposed modular heat pump systems depends on the relative pricing of energy sources. Fig. 10 illustrates the ratio between industrial electricity and gas prices r_p including tax levies in selected EU countries between 2019 to 2023. The graph highlights significant fluctuations caused by macroeconomic factors, particularly the fourth phase of the EU Emissions Trading System (ETS) (European Commission, 2025) in 2021 and the energy crisis triggered by the conflict in Ukraine (Statistisches Bundesamt, 2023) beginning in 2022. The figure shows a clear trend of high ratios prior to 2021, with a peak of around 5.0 in Germany. Similar trends can also be observed in most of the other countries. This suggests that electricity was significantly more expensive than natural gas. After the energy crisis peaked, the ratio fell to a low of approximately 2.7 for Germany in mid-2022. Since then, prices have stabilized, and the ratios in the countries analyzed have returned to values between around 2.4 for France and 3.3 for the Netherlands. Data from Norway (Statbank Norway, 2025) and Germany (Deutsch-Norwegische Handelskammer, 2019) suggest that a ratio between 1.0 and 1.3 serves as a guideline for the food and paper industries, and it becomes clear that HTHPs can be operated very efficiently in markets with competitive electricity prices or high taxes on fossil fuels. Applying this benchmark to the simulation results the 2 MW heat pump for the paper drying process, with a COP of 2,54, is economically viable compared to gas boilers in countries such as France and Spain. In markets such as in Poland or Austria, with r_p values higher 2.8, however, a higher COP would be required. In the case of the 1 MW heat pump used for the spray drying of milk powder, the COP is 1.84. For this less efficient application, the current r_p in all analyzed EU countries poses an obstacle to economically viable operation when compared to gas boilers. In France alone, the minimum COP required under current price conditions is over 2.37. This analysis of the electricity-to-gas price ratio suggests that, while technological efficiency is of great importance, energy policy plays a decisive role. However, the modular concept for heat pumps can offer a strategic advantage that goes beyond purely thermodynamic performance. Modularity is based on standardized components, which can significantly reduce initial investment costs compared to custom-built large-scale systems. Lower CAPEX costs offset operational inefficiencies in markets with unfavorable energy conditions. Modular systems offer superior flexibility in part-load operation and can scale their output by turning individual modules on and off, thereby maintaining high efficiency even under variable loads. Therefore, the adoption of HTHP in industries such as paper and food processing depends not only on achieving higher COPs through technology, but also on leveraging the cost-efficiency of modular designs to offset market fluctuations between electricity and gas prices. If CO_2 prices continue to rise or electricity from renewable sources becomes cheaper, thereby lowering the r_p values, modular HTHP systems may offer advantages over custom-built systems.

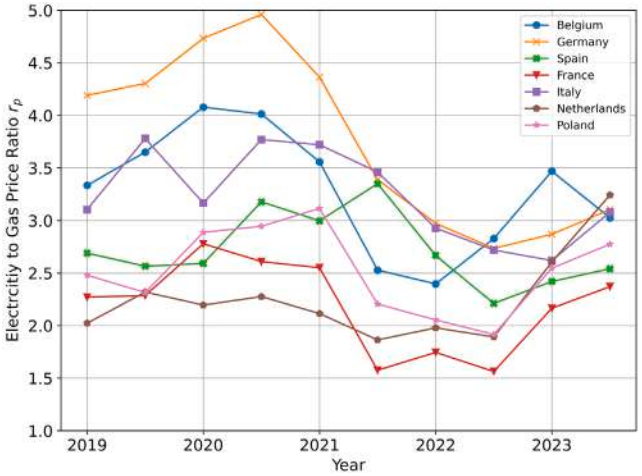


Fig. 10. Electricity to gas price ratio in selected EU countries including taxes based on eurostat (2025).

4.4. Modularization and standardization concepts for HTHPs

4.4.1. Modular configuration strategy

Based on the averaged, calculated steam demand for the respective process steps in drying processes in the paper and milk powder industries, a modular concept was developed that is based on the thermal capacities of the heat pumps and on standard production profiles. Based on the determined average heat demand of 27 MW for paper drying, three specific modularization concepts were developed. The choice of the optimal module configuration depends primarily on the respective production profile, load characteristics, and local conditions at the site. Table 11 shows the three derived scenarios using standardized modules of 2 MW, 5 MW, and 10 MW to achieve the total capacity through cascading. For production sites with a very stable base load profile of approximately 27 MW, installing three 10 MW modules to provide a total capacity of 30 MW with an approximately 11% reserve, offers the most cost-effective solution. This configuration is characterized by the lowest system complexity, and the number of components required is minimized, which simplifies integration. However, the system has limited flexibility in response to load changes. A failure of one module results in a loss of 33% of the total capacity, which can lead to production stoppages if the reserve is insufficient. In addition, the simulation results showed that this combination requires a large number of compressors to achieve the necessary output, since state-of-the-art compressors are not yet market-ready or are still in the prototype stage. For processes with dynamic load profiles and significant fluctuations in steam demand, the module cascade consisting of six 5 MW units proves to be technically superior. The key advantage lies in the staged operation. Due to the higher number of modules, individual units can be switched on or off as needed. This makes it possible to keep the remaining modules operating within the range of maximum COP at all times, even at lower total loads. In addition, the parallel configuration increases redundancy and a module failure reduces output by only about 17%. Furthermore, the simulation results show that most of the compressor technology currently available on the market can only be configured for 5 MW modules. For use in existing paper mills and retrofit scenarios, which are often characterized by limited space and complex existing infrastructure, a configuration consisting of fourteen 2 MW modules is recommended. This solution offers maximum operational flexibility and space optimization. The compact design of the modules allows for decentralized installation directly at the heat consumption points. The large number of modules results in increased control and installation costs. In addition, parasitic energy consumption, e.g., from pumps, increases due to the many parallel cycles, which can slightly reduce the system's overall efficiency compared to the other solutions.A comparison of the three scenarios shows that no single module configuration is universally superior. The 3 × 10 MW configuration is the best choice where CAPEX needs to be minimized and the load is stable. The 6 × 5 MW configuration is the technically optimal compromise for variable loads, maximizing efficiency and redundancy. 14 × 2 MW is the necessary solution for spatial constraints in existing facilities, even if this entails higher investment and control costs. The preceding sections demonstrated the thermodynamic feasibility of heat pump modules in the 1 MW to 10 MW power range. However, the

Table 11
Concept for the modularization of heat pumps for paper drying with a heat demand of 27 MW.

	3 × 10 MW	6 × 5 MW	14 × 2 MW
Total Capacity [MW]	30	30	28
Operating Profile	Constant base load	Load fluctuations	Existing infrastructure/space constraints
Redundancy	Low (max. −33%)	Medium (max. −17%)	High (max. −7%)

transition from simulation to industrial application requires a critical assessment of the risks associated with the procurement of compressor components. To make the 10 MW concept feasible without the use of custom-built giant compressors, each 10 MW module must utilize an internal arrangement of standard compressors connected in parallel, e.g., five units connected in parallel, each with a capacity of approximately 400–500 kW. This ensures component availability but increases the module's internal mechanical complexity. The 5 MW module option offers moderate flexibility. However, similar to the case of dairy products, it approaches the market threshold for standard steam compressors. To remain viable as a standard solution, internal parallel configuration is also essential here. The 2 MW configuration offers the highest degree of standardization and operational reliability. Since all compressor drives remain below the 500-kW threshold, procurement risks are negligible.

For the spray-drying processes for milk powder production, which require an calculated average thermal output of 4 MW, a modularization concept is proposed based on heat pump capacities of 1 MW, 2 MW, and 5 MW. The possible design scenarios, tailored to different production profiles, are shown in Table 12. For facilities with a stable and continuous demand, a single 5 MW module is sufficient. Since the average load is approximately 4 MW, this plant would operate at an optimal capacity utilization rate of around 80%, which often represents the point of highest efficiency for heat pump compressors. This configuration provides a reserve of 1 MW to accommodate short-term peaks without additional infrastructure. Redundancy is low and a failure of this single system would result in a complete loss of spray-drying capacity, necessitating a contingency plan. In dairy production, cyclic processes, e.g., filling or cleaning, frequently occur, leading to fluctuating thermal loads. Installing two 2 MW modules enables precise load staging. One unit can efficiently handle the base load, while the second is activated only during peak periods. Alternatively, a hybrid configuration consisting of one 2 MW and one 1 MW unit allows for finer load adjustment, precisely covering stages between 3 MW and 4 MW. Additionally, maintenance work can be performed on one unit while the other continues to operate. When retrofitting existing dairy plants, space constraints often prevent the installation of larger 5 MW units. Three compact 1 MW modules offer a tailor-made solution. The combination also ensures maximum flexibility and a high degree of redundancy where the failure of one module reduces output by only 33%, rather than bringing production to a complete standstill. This scenario can be very maintenance-intensive due to the large number of modules and components, which increases maintenance costs. Regarding the compressor technology, a single 5 MW module requires the C4 steam compressor with a drive power of over 1100 kW, which are generally classified as custom-built units. A configuration with two 2 MW units represents a strategic middle ground. The electrical drive power for steam compressors in a 2 MW module is approximately 550 kW, which falls well within the industrial standard range over 1 MW. This ensures high market availability and lower unit costs. The 1 MW configuration emphasizes supply chain security and operational resilience as all compressors are in the 100 to 200 kW range. All components are sourced exclusively from standard catalogs, however, the large number of units increases complexity.

In addition to the purely modular design of heat pump technology, combining heat pump modules with conventional technologies or thermal storage systems generally makes sense for industrial applications but is not further investigated in the current study. The modular combining strategy enables an economically optimized design of the overall system by combining the advantages of high efficiency with operational reliability. A comparison of industrial strategies reveals a consistent trend and demonstrates that module size and component availability can only be decoupled from one another through internal parallelization.

4.4.2. Component standardization potential

The transition from thermodynamic simulation to practical design requires a critical assessment of whether the calculated component dimensions are consistent with commercially available technology. By comparing the simulated drive power requirements with the manufacturer specifications of leading industrial suppliers, this study evaluates the suitability of current technologies for decarbonization. Table A.23 provides an overview of existing screw and piston compressor models from manufacturers GEA, Mayekawa and Spilling and shows a high degree of alignment between the simulated requirements and the current manufacturer portfolios. The thermodynamic requirements can be adapted to the manufacturers' commercially available standard models for screw and piston compressors. The refrigerant cycle can be operated using proven screw compressor technologies, e.g., the Mayekawa UD/G series or GEA Grasso M, which typically offer an electric drive power in the range of 2 to 3 MW. For the steam compression cycle piston compressors from Spilling with standard cylinder configurations effectively cover the required pressure ratios. This market situation confirms that the technological requirements for the use of industrial heat pumps are already partly available in standard product catalogs. The availability of these standardized components reinforces the core premise of the proposed modular design strategy. The scalability of the system should be achieved through parallel integration rather than by enlarging individual components. As the simulation data show, directly scaling the thermal heat pump capacity results in the compressors falling into the custom-made category, which significantly increases procurement costs and lead times. The use of multiple identical, small standardized compressor units makes it possible to achieve high thermal capacities while remaining within the standard range of the market. This approach can offer advantages such as cost efficiency through the use of mass-produced standard modules, economies of scale, reduced technical development effort, and operational flexibility through modular parallelization. Although the simulation results provide a solid basis using linear scaling, real-world constraints that are not fully captured by thermodynamic models must be taken into account during practical implementation. Therefore, detailed design validation should always be carried out in consultation with equipment manufacturers. Ultimately, the transition to standardized heat pump systems depends on a collaborative approach between industrial end users and component manufacturers. Although the tools and components for modular heat pumps are currently available to a certain extent, further insights into the barriers to implementation are needed. The proposed strategy represents a simplified yet technically consistent approach to leveraging existing hardware for the decarbonization of the high-temperature industry.

4.4.3. Cost savings potential

The transition from theoretical thermodynamic modeling to practical industrial implementation requires a critical assessment of cost structures and market readiness. This section examines the economic viability of component standardization. The underlying hypothesis is that switching from custom-designed components to parallelized standard modules significantly reduces a system's capital expenditures.

Table 12
Concept for the modularization of heat pumps for milk powder spray drying with a heat demand of 4 MW.

	1 × 5 MW	2 × 2 MW	4 × 1 MW
Total Capacity [MW]	5	4	4
Operating Profile	Constant base load	Load fluctuations	Existing infrastructure/space constraints
Redundancy	Low (max. −100%)	Medium (max. −50%)	High (max. −25%)

Table 13
Potential cost savings for a 0.8 MW steam compression system specified by the compressor manufacturer Spilling.

Name	Quantity	S/P[a]	Cost [%]	Yes/NoP[b]	Reduction [10/100 pcs.]
Compressor	1	P	78.79	N	10%/25%
Motor	1	S	10.51	Y	8%/20%
Oil cooler	1	S	0.28	Y	10%/25%
Oil pump	1	S	0.23	Y	10%/25%
Oil filter	1	S	0.02	Y	10%/25%
Vessel	4	P	6.04	Y	30%/45%
Tank	1	P	0.53	Y	30%/45%
Pump	1	S	0.56	Y	10%/25%
Measurement Sensors	25	S	3.05	Y	10%/25%

[a] Standard/P: Prototype.
[b] Availability of the shelf.

The analysis is based on the manufacturer's data and its assessments regarding standardization and cost-saving potential. Table 13 shows the key components, cost breakdown, and potential cost savings of a 0.8-MW Spilling steam compression system. The breakdown includes the main components of the compressor unit as well as the peripheral equipment required for full operation.

The data highlights the central role of the compressor, accounting for 78.79% of total costs. It is currently classified as a prototype in this system, which means it is not yet fully standardized and is not available from stock. Despite this prototype status, the manufacturer's estimate indicates significant potential for cost reduction for this component in case of standardization. For a series production run of 10 units, this results in a 10% cost reduction for 100 units, the reduction is as high as 25%. In contrast, peripheral components such as the 160-kW motor, the complete oil system, and the measurement equipment are classified as standard. While most of these standard components offer cost savings of 10% for 10 units and 25% for 100 units, the motor follows a different pattern with initially only 8% for 10 units but rise to 28% for 100 units. The potential cost savings for pressure vessels and tanks amount to 30% for 10 units and 45% for 100 units. This is because, in mass production, these components are technically less complex to manufacture than the high-precision compressor. Based on Eq. (15), a relative total cost savings potential R_{HP} for the 0.8 MW system of 11% for a production volume of 10 units and 26% for a production volume of 100 units is calculated. This figure serves as an important benchmark for the concept's cost-effectiveness. Standardization, particularly of prototypes and cost-intensive key components, enables a drastic reduction in investment costs through economies of scale. This underscores the strategic advantage of modular heat pump concepts.

In addition to analyzing the vapor compression system, the cost-effectiveness of the HTHP manufactured by Mayekawa is examined. Table 14 compares the cost shares and economies of scale of the individual components. The data show that the cost dynamics of this system differ from those of conventional compressor systems, showing that the compressor and pump together account for only 6% of the total costs.

Mechanical and process-related components dominate the cost structure. Heat exchangers account for the largest single item at 13.5%, followed by manufacturing labor costs at 11.5% and valves at 11.5%. The control system is also a significant factor at 11%, as is measurement and control technology at 10%. When evaluating economies of scale, there are significant differences between scalable components and fixed costs. While the compressor does not offer any cost reduction when scaled up, as this component has already been optimized by the manufacturer itself, the potential savings for other components are significantly higher. The control system offers a 60% reduction, and the measurement technology a 40% reduction. Of particular note is the design labor costs, which accounts for 9.6% of total costs and has an extreme reduction potential of 95%. Savings can also be achieved through standardized manufacturing processes in production labor costs with 30% reduction and in valves with 15% reduction. In summary, the Mayekawa analysis shows that the modularity and standardization of components should focus also on other components than the compressor, such as heat exchangers or peripheral devices, which were not examined in detail in this study. A relative total cost savings potential R_{HP} for the 1 MW HP system of 31% for a production volume of 10 units is calculated.

4.4.4. Impact on the payback period

To verify the reliability of the economic results, a sensitivity analysis of the specific investment costs in €/kW for the 1 and 2 MW heat pumps was conducted. Fig. 11 illustrates the influence of the specific invest costs $CAPEX_{spec}$ on the payback period of the heat pump compared to a gas boiler and an electric boiler. Scenarios without cost reductions are presented with solid lines, scenarios with cost reductions are presented with dashed lines. Based on the specific investment costs in €/kW and the relative total cost savings potential R_{HP}, the absolute cost savings S_{abs} are calculated according to Eq. (16). The cost savings achieved relative to the investment costs in Year 0 influence the payback period of the heat pump compared to the conventional systems. To illustrate the impact of the absolute cost savings S_{abs} on the payback period, a savings potential R_{HP} of 11%, based on the 0.8-MW Spilling steam compression system with a production volume of 10 units, is used as a reference. This reference was selected because, unlike Mayekawa, the compressor offers proven potential for standardization and cost reduction. The x-axis shows the range of specific investment costs from a best case scenario at €200/kW to a worst case scenario at €500/kW. The results show a clear linear relationship between investment costs and the payback period. This becomes particularly evident when comparing the 2 MW heat pump with the gas boiler. Due to the significant difference in the magnitude of the costs between the two scenarios, a dashed Y-axis is used in the graph. The upper portion shows the scenario with the higher reference costs of the GB, in which the payback period is significantly longer. In the reference case of €350/kW, the payback period of a 2 MW heat pump compared to a GB is approximately 5.7 years. The payback period ranges from about 3.2 years with a CAPEX of €200/kW to over 8 years with €500/kW. Such a timeframe would mean that more than 40%–50% of the heat pumps technical lifespan would have elapsed before the investment had

Table 14
Potential cost savings for HTHP components specified by the heat pump manufacturer Mayekawa.

Name	Quantity	Cost [%]	Yes/No[a]	Reduction [10 pcs.%]
Compressor & pump	1 & 1	6	Y	0
Frame	1	3.5	N	25
Motors & VSD	2 & 1	5	Y	20
Control system[b]	1	11	N	60
Pressure vessels	6	5	N	10
Heat exchangers	4	13.5	N	10
Valves	150	11.5	Y	15
Instrumentation[b]	80	10	Y	40
Insulation & tracing	1	6.5	Y	40
Packing & transport	1	7	Y	0
Design labor cost[b]	1	9.6		95
Production labor cost	1	11.5		30

[a] Yes/No is interpreted as available from a catalog without engineering/design needs.
[b] These items are fixed costs and will take a variable % of HP total cost depending on increase/decrease of sink power output, others are scalable.

paid for itself. Implementing cost-reduction measures improves results and shortens the payback period across the entire CAPEX spectrum by approximately 0.35 to 0.89 years. In contrast, the chart below illustrates the scenario with lower reference costs for comparing the heat pump with an EB. In this case, the savings in operating costs are sufficient to keep the payback period under one year, even under the most unfavorable CAPEX assumptions. Due to the cost reductions, the payback period for the 1 MW plant in the worst-case scenario is 0.77 years. For the 2 MW system, the payback period in the worst-case scenario is reduced to 0.56 years. Although cost reductions also offer an advantage here, the investment remains economically viable across the entire range of initial investment amounts examined. If higher percentage cost savings are possible, for example, through larger production volumes or the use of other manufacturers, the payback period shortens even further in line with the reduction in investment costs, making the system more cost-effective. In general, considering the payback period of heat pump systems compared to conventional systems means that high capital costs for heat pumps only pose an economic risk when they must compete with very low-cost fossil fuels such as natural gas. Compared to conventional electric heating systems, the investment is always economically justified, even with high module prices. The analysis shows that manufacturer-specific standardized components have a significant impact on cost reduction and the overall payback period of the heat pump and are therefore important for the development of modular heat pump designs.

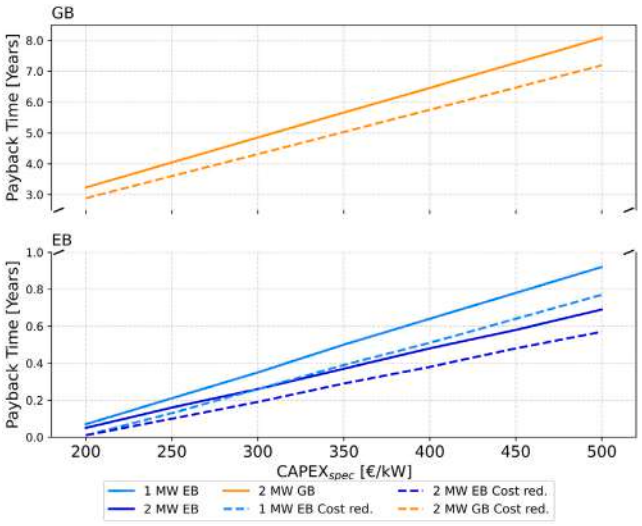


Fig. 11. Impact of specific investment costs and cost reductions on the payback period of a 1 MW and a 2 MW heat pump compared to an electric boiler and a gas boiler.

4.5. Barriers and limitations

The implementation of modular HTHPs is limited by several obstacles and technical constraints. Although the concept of a modular heat pump proposed in this study is designed to cover a broad spectrum of industrial applications in the low- and medium-temperature ranges, it is subject to thermodynamic and process-specific constraints, which are discussed below. From a thermodynamic perspective, the permissible temperature and pressure ranges determine the scope of application for modular HTHPs. Commercially available HTHPs can reach temperatures of up to approximately 160 °C. Temperatures of 200 °C have so far been achieved primarily in pilot and demonstration plants. Given the increasingly higher temperatures available on the market in recent years, the current technical temperature limit of approximately 200 °C at the secondary outlet of the heat sink heat exchanger is defined as the upper limit of the modularization concept without subsequent steam compression. The configurations simulated in this study increase the secondary outlet temperature to over 200 °C through two additional compression stages with condensate injection and illustrate how higher heat sink temperatures can be achieved through downstream steam compression, thereby representing a project-specific extension of the standardized module catalog. For a common standard solution using modules that include downstream steam compression, a solution for process steam supply up to 300 °C can be defined. For steam requirements above approximately 310 °C, the underlying studies suggest that the modular concept may be unsuitable due to a lack of standardized components. Based on the pressures prevailing in the system, conclusions can be drawn regarding the component-specific nominal pressures in terms of modularization. Standardized component families such as flanges, gaskets, valves, and piping are already available as catalog components well beyond the nominal pressure of PN 16. The interchangeability limit of modularization is defined as the range from PN 16, at which full interchangeability is guaranteed, up to PN 40 as the technically easily achievable upper limit. Although the technical feasibility of components above PN 40 is assured, pressures exceeding this nominal pressure complicate the design of components such as heat exchangers and compressors and require more extensive certification of pressure vessels as well as safety-critical design, which impairs interchangeability and thus the core element of modularization. Accordingly, the technically feasible limit for modularization is considered to be within the nominal pressure range up to PN 40. Depending on the type, standard compressors available on the market can achieve a pressure ratio of approximately 10 before further measures, such as specially designed single-stage or multi-stage systems, become necessary, thereby defining the technical limit for the pressure ratio of a modular unit. Another technical limitation arises from secondary media such as waste heat sources or

condensate streams. The technical suitability of the modularization concept is limited to non-corrosive media. Assuming a reference material for the standardized module, e.g., stainless steel 1.4404, on the surfaces in contact with the medium, this corresponds to a compatibility range of pH 4–10 and chloride concentrations of no more than 50 mg/l (Informationsstelle Edelstahl Rostfrei, 2012) which applies, for example, to treated process water, low-chloride flue gas condensates, as well as steam or distilled water. Media outside this range, with a pH < 4 or pH > 10, or chloride concentrations exceeding 50 mg/l, such as strongly acidic or corrosive media, or chlorine-containing vapors with correspondingly high-chloride condensates, require a customized material solution, e.g., high-alloy steels, titanium, coatings, or hermetically sealed secondary cycles, to prevent corrosion and material damage. The need for additional application-specific testing and the resulting use of special, more expensive materials lead to customer-specific adaptations that are not accounted for in the standard module components. Although this study generally examined the combination of different module sizes to achieve system output while taking load behavior into account, the technical parameters are limited to individual module sizes between 1 and 10 MW and a system output of 30 MW. This excludes small heat pump modules under 1 MW and large modules over 10 MW, for which customized solutions are required. For small modules, the cost of a complete modular unit, including controls, measurement technology, and housing, becomes disproportionately high relative to the low usable thermal output. For large modules exceeding 10 MW, hydraulic distribution losses and system complexity increase, thereby negating the efficiency benefits of modularization. Delivering a total output of 30 MW using a modular system is possible through various combinations of modules of different sizes, however, this pushes the limits of cost- and space-efficient solutions and is therefore considered the system output limit for the application of modular concepts. In general, modular architectures are also constrained by the operational dynamics of the underlying processes. The modular design requires that the individual units be operated close to their design point with limited load fluctuations. However, batch processes with significant load fluctuations require specialized control logic, thermal buffers, and start-stop sequences for each module, which can complicate the standardization of communication and control between modules and thus necessitate custom logic. Unstable or highly fluctuating waste heat sources also present a technical limitation in terms of operational dynamics. Modular HTHPs require a certain degree of stability in the heat source. Fluctuating temperatures or flow rates require customized control and buffer systems that undermine the modular approach. In general, site-specific integration constraints in retrofit scenarios can be mitigated through the decentralized design of the modular architecture, flexible piping, and optimized use of space. Nevertheless, the respective module sizes, which range from 1 to 10 MW, are critical for technical integration and require sufficient available space to accommodate the necessary modules. By using the module catalog, compliance with legal regulations and safety requirements can be significantly simplified through the use of certified, off-the-shelf compressor systems instead of custom-built prototypes. Nevertheless, broader market acceptance continues to be influenced by institutional factors. Fragmented incentives for decarbonization, fluctuating energy prices, supply chain bottlenecks for materials and components, a shortage of skilled workers, and the current lack of universal interface standards all pose concrete risks to expansion. Overcoming these obstacles requires a coordinated policy framework, standardized certification pathways, and closer collaboration between technology providers and users. In summary, the proposed modularization concept is based on the technical limitations described above. Applications outside the power range of the individual modules of 1–10 MW, as well as those with a total system capacity exceeding 30 MW, heat sink temperatures > 200 °C without subsequent steam compression and approximately > 310 °C with subsequent steam compression, nominal pressures > PN 40 and compressor pressure ratios > 10, as well as applications involving corrosive media with a pH value below 4 or above 10 and a chloride content exceeding 50 mg/l, are currently considered unsuitable for modularization. This study is furthermore subject to several methodological limitations that determine the scope and generalization of the results. First, the market analysis is based on aggregated production metrics from a selected group of EU countries, which may not fully capture regional differences or the profiles of smaller plants. The calculated heat capacities depend heavily on the underlying assumptions and may vary depending on the selection and availability of data for those assumptions. Furthermore, the thermodynamic modeling is based on steady-state simulations at the design point. While this allows for precise component sizing and the calculation of the COP, it inherently simplifies the dynamic realities of operation, including part-load behavior, transient startup phases, and efficiency losses caused by control systems. The scalability assessment is performed by linearly extrapolating 1 MW and 2 MW base modules to 5 MW and 10 MW configurations. This approach assumes proportional scaling of compressor volumes or the parallel operation of identical units while maintaining identical pressure ratios and isentropic efficiency curves. Consequently, secondary effects such as pressure losses in the piping, the scaling of the surface-to-volume ratio in heat exchangers, and complex cascade control logic are not explicitly modeled for each capacity step but are factored into the conservative base parameters. Furthermore, the economic assessment is based on theoretical CAPEX projections and manufacturer specifications. Actual costs may vary due to fluctuations in the supply chain, installation complexity, and regulatory requirements. Finally, the thermodynamic models have not yet been validated using direct experimental measurements from a prototype system. This limitation is explicitly acknowledged, as the primary goal of this work is to establish a transferable design methodology and a framework for parametric scalability, rather than to conduct a full-scale pilot validation.

5. Conclusions and outlook

This study breaks new ground in the investigation of modular concepts for high-temperature heat pumps for industrial applications, using drying processes in the paper and dairy industries as concrete examples. This work fills a significant gap in the current literature and goes beyond conventional system scaling by proposing a strategy based on standardized module configurations. A detailed market analysis of selected EU countries was conducted to quantify country-specific heat demand for the drying process steps. To validate the theoretical feasibility of the proposed concept, thermodynamic simulations were performed for modular heat pump configurations of 1 MW and 2 MW. These configurations were evaluated in comparison to conventional solutions, including electric boilers and gas boilers, to establish a basis for economic viability. In addition, the study extrapolated modular configuration options for larger plant capacities of 5 MW and 10 MW, assigning specific combinations based on different operating load profiles. A key focus of this analysis was the compressor, which was identified here as the primary cost-driving component. The feasibility of the modular approach was assessed by classifying compressors according to their performance classes and the technologies available on the market to ensure compatibility with off-the-shelf components. Using manufacturer-specific data, potential CAPEX savings resulting from component standardization and economies of scale were quantified for a representative 1 MW heat pump system. In addition, the impact of these cost reductions on the investment's payback period was determined. The key findings and contributions of this study can be summarized as follows:

- Market Potential and Demand: A specific market analysis across selected EU countries identified significant process steam demands in industrial drying processes. The study quantified an average requirement of 27 MW for paper drying and 4 MW per facility for milk powder spray drying, highlighting the scale of decarbonization potential.

- Technical Feasibility: Thermodynamic simulations demonstrated that modular concepts can effectively replace fossil steam generation. A 2 MW heat pump module for the paper industry achieved a COP of 2.54 at process conditions of 273 °C and 10 bar. Similarly, a 1 MW module for spray drying in the dairy sector reached a COP of 1.84 at 313 °C and 10 bar, confirming technical viability at high temperature levels.
- Scalability and Modular Configuration: The simulation results were extrapolated from the base modules to larger capacities, specifically extending the 1 MW results to 2 and 5 MW, and the 2 MW results to 5 and 10 MW configurations. The analysis of compression work and resulting compressor capacities revealed that for modular setups up to 5 MW, commercially available piston and screw compressor technologies are sufficient. This confirms that the proposed modular architecture avoids the need for custom-built prototypes while maintaining effective compression performance.
- Economic Viability: The economic assessment indicates a strong payback period of less than 1 year compared to conventional electric boilers. Competitiveness against gas boilers is highly sensitive to investment costs. With a specific capital expenditure of 350 €/kW, the 2 MW system pays for itself in approximately 6 years. Furthermore, leveraging economies of scale through component standardization, e.g., an 11% cost reduction, can shorten this payback period by nearly one year under the given assumptions in this study.
- Critical Thresholds: Analysis of the electricity-to-gas price ratio determined that a minimum COP of 2.37 is required for heat pumps to operate more economically than gas boilers under current market conditions.

In summary, the results show that the modularization of industrial heat pumps and the standardization of components represent a viable path toward the decarbonization of high-temperature industrial processes, provided that the specific investment costs are kept under control through mass production and standardization. Based on the results of this work, several avenues for future research remain open to address the gaps in the current study. For experimental validation, as part of the SPIRIT Demonstration Project 3, plans are in place to construct and test a HTHP demonstration plant at the Smurfit Kappa paper mill, a manufacturer of corrugated cardboard packaging based in the Czech Republic. Spilling will be the supplier of the compressor technology. A structured test campaign with a total duration of around ten months is planned, divided into nominal and part load operation. Part load operation is intended to document and quantify startup and shutdown scenarios, including the time required to reach operating conditions. In addition, the system's behavior during emergency shutdowns, e.g., the event of a paper tear, and in standby mode should be recorded. Validation of the system's COP within the vapor compression system is planned for a compressor inlet pressure range of 0.8 to 1.5 bar and an outlet pressure between 5 bar and 5.5 bar. The proposed measurement campaign will consist of nominal-load operation with a mass flow rate of approximately 600 kg/h over a total of 1000 operating hours, as well as partial-load operation with a mass flow rate between 250 and 1000 kg/h over a total of 600 operating hours. The measurement campaign includes an analysis of energy consumption and the COP as a function of steam flow rate and compression ratio, as well as the required condensate volume. Based on the measurement data collected, transient simulation using real operating data from the Smurfit Kappa paper mill will be validated and optimized in terms of system performance and COP. From an economic point of view, the CAPEX estimates submitted in connection with the economic feasibility study in this work will be verified using at least ten actual industry quotes from at least three European compressor and heat exchanger manufacturers in order to reliably identify deviations from the modeled costs per kW. Based on a breakdown of costs along the supply chain, the annual production volume at which the standardized module reaches the break-even point will be determined. Subsequently, in collaboration with at least two partners, a catalog of standardized modules will be developed, each featuring three thermal stages and two pressure stages. To facilitate market expansion and the transferability of the modular concept to other industries, in addition to the case studies from the paper and dairy industries, a feasibility study will be conducted for a 1 or 2-MW heat pump in a textile drying process, based on actual plant data from the last production year. The development of a screening matrix for industrial sectors, such as the chemical or textile industries, is intended to help identify suitable processes in at least two additional sectors. The experimental, ten month test series and the subsequent transient simulation based on real measurement data will play a key role in the future in further developing the framework concept presented here from a theoretical design methodology into a certified technology platform accepted by the market.

CRediT authorship contribution statement

Nancy Kabat: Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Software, Resources, Project administration, Methodology, Investigation, Funding acquisition, Formal analysis, Data curation, Conceptualization. **Maximilian Kriese:** Supervision, Software, Project administration, Funding acquisition, Conceptualization. **Riccardo Marchetti:** Funding acquisition, Data curation.

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Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Nancy kabat reports financial support was provided by European Union. Maximilian Kriese reports financial support was provided by European Union. Riccardo Marchetti reports financial support was provided by European Union. Funded by the European Union under Grant number 101069672 (SPIRIT). Views and opinions expressed are however those of the author(s) only and do not necessarily reflect those of the European Union or CINEA. Neither the European Union nor the granting authority can be held responsible for them. The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Nancy Kabat reports financial support was provided by Horizon Europe. Maximilian Kriese reports financial support was provided by Horizon Europe. Riccardo Marchetti reports financial support was provided by Horizon Europe. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix A. Tables

See Tables A.15–A.23.

Table A.15
Paper types with their corresponding drying temperatures and pressures based on P.B.L. Netherlands Environmental Assessment Agency (2019).

Category	Paper Type	T [°C]	p [bar]
1	Graphic paper	150	5
1	Graphic paper from recovered paper	150	5
1	Specialty paper	150	5
2	Corrugated board	180	10
2	Solid board	180	10
2	Folding boxboard	180	10
3	Sanitary paper	165	7
4	Molded fiber	160–220	–

Table A.16
Parameter variation of the compressor's isentropic efficiencies and their effects on the COP for a 1 MW and a 2 MW system.

Parameter	Paper 2 MW			Dairy 1 MW			
η_{is}	C1	C2	C3	C1	C2	C3	C4
0.65	–	2.51	–	–	1.83	–	–
0.7	2.51	2.54[a]	–	1.83	1.84[a]	–	–
0.73	–	–	2.47	–	–	1.82	1.82
0.75	2.54[a]	2.57	–	1.84[a]	1.85	–	–
0.78	–	–	2.54[a]	–	–	1.84[a]	1.84[a]
0.8	2.56	–	–	1.85	–	–	–
0.83	–	–	2.6	–	–	1.86	1.86

[a] Starting value. – Isentropic efficiency not simulated.

Table A.17
Parameter variation of total heat losses and their effects on the COP for a 1 MW and a 2 MW system.

Parameter		Paper 2 MW	Dairy 1 MW
$\dot{Q}_{loss,th}$ [%]	Value per pipe	COP	
0	0	2.54	1.84
1	0.33	2.52	1.82
2	0.67	2.49	1.8
3	1.00	2.47	1.79
4	1.33	2.45	1.77
5	1.67	2.43	1.75

Table A.18
Parameter variation of production volume, specific heat demand, number of plants, and operating hours and their effect on the thermal capacity for the paper and dairy industry.

Parameter	−20%	−10%	0%	+10%	+20%
P_{th} [MW] Paper					
m & e	19	21	24	26	28
N & t	30	26	24	22	20
P_{th} [MW] Dairy					
m & e	1.1	1.3	1.4	1.5	1.7
N & t	1.7	1.6	1.4	1.3	1.2

Table A.19
Resulting pressure, temperature, specific enthalpy, and mass flows at the state points in the system for a 2 MW heat pump.

State point	p [bar]	T [°C]	h [kJ/kg]	$\dot{m}$ [kg/s]
1	4.00	118	799.90	3.80
2	24.50	189	924.10	3.80
3	24.49	126	547.11	3.80
4	24.48	79	401.89	3.80
5	4.00	42	401.89	3.80
6	4.00	47	654.68	3.80
7	4.00	118	799.90	3.80
8	2.00	45	73.06	24.65
9	2.00	55	111.79	24.65
10	2.01	55	111.79	24.65
I	1.80	95	398.09	0.62
Ia	1.01	95	396.26	0.67
II	1.80	120	2708.05	0.62
IIa	4.80	254	2969.40	0.62
IIb	4.80	160	2769.68	0.67
IIc	10.10	273	2992.91	0.67
III	1.01	70	307.46	18.06
IV	1.00	50	254.26	18.06

(*continued on next page*)

Table A.19 (*continued*).

State point	p [bar]	T [°C]	h [kJ/kg]	$\dot{m}$ [kg/s]
1	5.00	124	810.40	1.91
2	27.00	191	925.39	1.91
3	26.99	131	567.88	1.91
4	26.98	87	424.19	1.91
5	5.00	50	424.19	1.91
6	5.00	55	666.72	1.91
7	5.00	124	810.40	1.91
8	42.99	82	755.72	0.38
9	8.00	18	755.72	0.38
10	8.00	23	1638.76	0.38
11	43.00	186	1978.35	0.38
I	1.20	95	398.05	0.30
Ia	1.01	95	396.23	0.32
II	1.20	110	2693.94	0.30
IIa	3.50	255	2976.83	0.30
IIb	3.50	149	2754.53	0.32
IIc	10.10	313	3078.81	0.32
III	1.01	75	178.13	11.50
IV	1.00	47	149.08	11.50

Table A.20
Resulting pressure, temperature, specific enthalpy, and mass flows at the state points in the system for a 1 MW heat pump.

State point	p [bar]	T [°C]	h [kJ/kg]	$\dot{m}$ [kg/s]
1	5.00	124	810.40	1.91
2	27.00	191	925.39	1.91
3	26.99	131	567.88	1.91
4	26.98	87	424.19	1.91
5	5.00	50	424.19	1.91
6	5.00	55	666.72	1.91
7	5.00	124	810.40	1.91
8	42.99	82	755.72	0.38
9	8.00	18	755.72	0.38
10	8.00	23	1638.76	0.38
11	43.00	186	1978.35	0.38
I	1.20	95	398.05	0.30
Ia	1.01	95	396.23	0.32
II	1.20	110	2693.94	0.30
IIa	3.50	255	2976.83	0.30
IIb	3.50	149	2754.53	0.32
IIc	10.10	313	3078.81	0.32
III	1.01	75	178.13	11.50
IV	1.00	47	149.08	11.50

Table A.21
Resulting performance of the individual components for 2-, 5-, and 10 MW heat pump modules for paper drying.

Component	Component Size [kW]		
	2 MW	5 MW[a]	10 MW[a]
C3	474	1185	2370
C1	163	408	815
C2	151	378	755
HTHX	1424	3560	7120
LTHX 1	955	2388	4775
LTHX 2	955	2388	4775
IHX	548	1534	2740

[a] Scaling factor 2.5 for 5 MW and 5 for 10 MW.

Table A.22
Resulting performance of the individual components for a 1, 2 and 5 MW heat pump module for spray drying.

Component	Component Size [kW]		
	1 MW	2 MW[a]	5 MW[a]
C4	130	325	650
C3	222	555	1110
C1	85	213	425
C2	107	268	535
HTHX	682	1705	3410
LTHX 1	463	1158	2315
LTHX 2	334	834	1670
IHX	274	548	1370

[a] Scaling factor 2 for 2 MW and 5 for 5 MW.

Table A.23
Combinations of available screw and piston compressor technologies from Manufacturers Mayekawa, GEA and Spilling for various thermal capacities.

Capacity [MW]	Screw Compressor		Piston Compressor
	Mayekawa	GEA	Spilling
1[a]	1 x SCV (4 MW)	C3/C4: 1 x Grasso M	C1 (1-stage): 2–3 cylinder C2 (2-stage): 3 cylinder
2[b]	1 x UD/G Series (2.6 MW)	Grasso LT	C1 (2- stage): 4–6 cylinder C2 (1-stage): 4–6 cylinder
2[a]	1 x SCV (4 MW)	C4: Grasso M C3: Grasso LT	C1 (1-stage): 3–4 cylinder C2 (2-stage): 4 cylinder
5[b]	1 x UD/G Series (2.6 MW)	Grasso LT	C1 (2- stage): 2 x 4–6 cylinder C2 (1-stage): 2 x 6 cylinder
5[a]	2 x SCV (4 MW)	C4: Grasso LT C3: Grasso LT	C1 (1-stage): 4–6 cylinder C2 (2-stage): 5–6 cylinder
10[b]	2 x UD/G Series (2.6 MW)	Grasso LT	

[a] Spray drying heat pump configuration.
[b] Paper drying heat pump configuration.

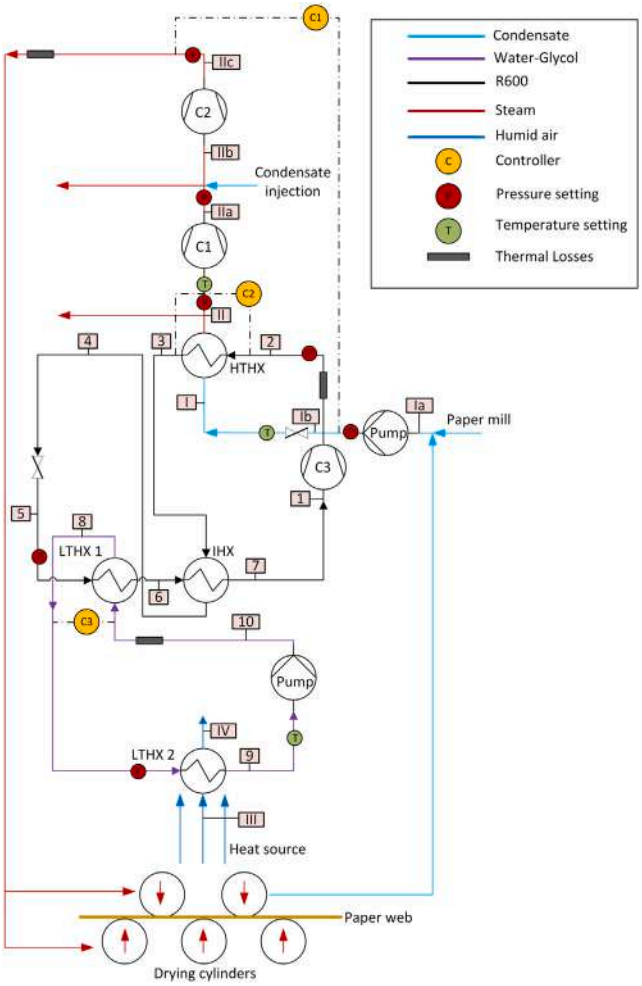


Fig. B.12. Flowchart diagram for a 2 MW heat pump configuration in the pulp and paper industry for paper drying.

Appendix B. Figures

See Figs. B.12 and B.13.

Data availability

Data will be made available on request.

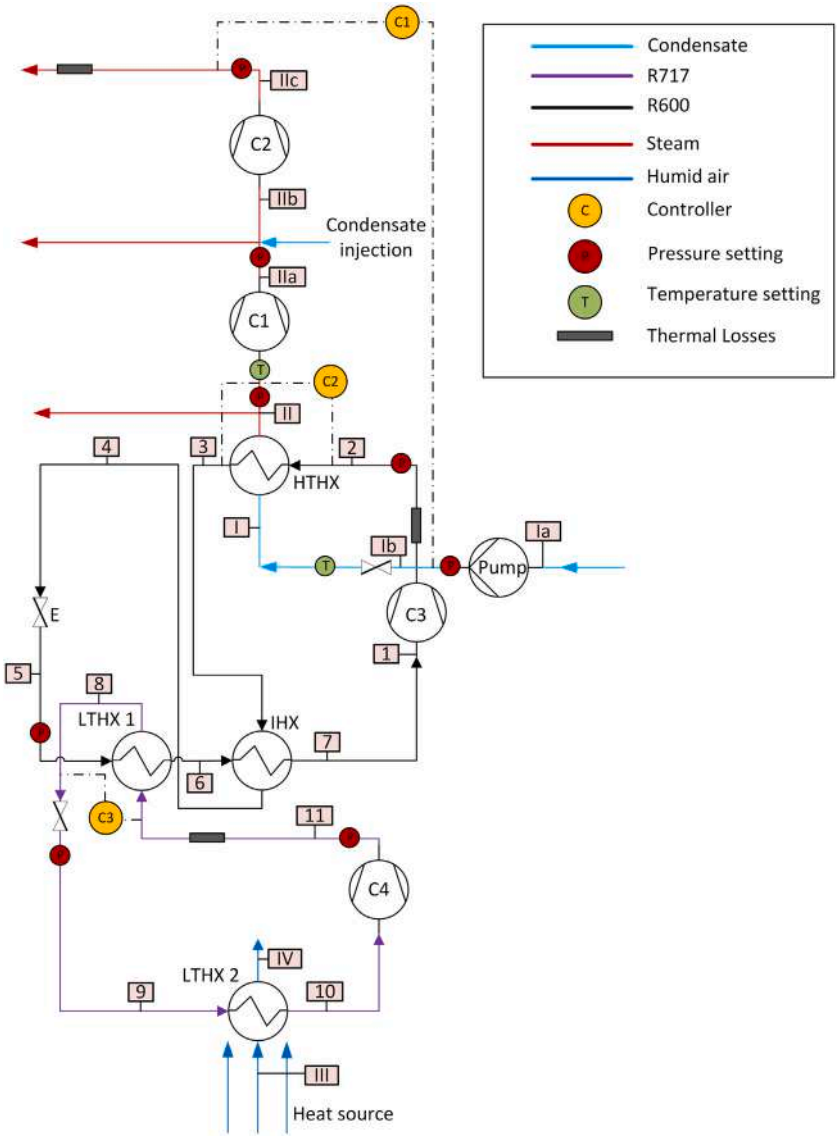


Fig. B.13. Flowchart diagram for a 1 MW cascade heat pump configuration in the dairy industry for milk powder spray drying.

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