

Article

Enhancing the Performance of District Heating Networks Using a Low-Temperature Hybrid Heat Recovery System for Gas Cogeneration Units [†]

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[†] This paper is an extended version of our paper published in the 20th SDEWES Conference, Dubrovnik, Croatia, 5–10 October 2025.

Abstract

This study explores the selection of a heat recovery system for cogeneration units based on gas engines supplying the district heating system in Opole in order to enhance the efficiency and sustainability of the system. The proposed modifications focus on utilizing low-temperature (LT) waste heat from engine cooling circuits and improving exhaust heat recovery. The research examines retrofitting three cogeneration engines (total thermal capacity of 7.6 MW) by integrating water-to-water heat pumps to upgrade low-temperature waste heat (55–45 °C up to 700 kW), enhancing heat supply to the district heating network. Additionally, a second stage of economizers is evaluated to maximize condensation-based exhaust heat recovery from the existing 95–135 °C system. These system modifications increase the overall thermal capacity up to 9–9.1 MW. To maintain heat supply during cogeneration unit shutdowns (due to failures or electricity price fluctuations), an auxiliary air-to-water cascade heat pump provides an additional 0.8–1 MW. With increasing electricity price volatility, these system modifications provide crucial operational flexibility. Computational simulations confirm that the hybrid configuration successfully upgrades waste heat while strictly maintaining the existing engine return water safety limit. The evaluation demonstrates high economic profitability alongside stable emission reductions. This research presents a case study in optimizing heat recovery in cogeneration-based district heating networks, demonstrating practical and scalable applications for sustainable energy systems.



Academic Editors: Francesco Calise, Qiuwang Wang, Maria Vicidomini, Wenxiao Chu and Poul Alberg Østergaard

Received: 30 April 2026

Revised: 9 June 2026

Accepted: 15 June 2026

Published: 25 June 2026

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Keywords: district heating; cogeneration gas engine; decarbonization; waste heat; renewable energy sources

1. Introduction

Energy systems worldwide are currently undergoing a radical structural and organizational transformation aimed at achieving climate neutrality by 2050 [1]. The heating and cooling sector, which accounts for a significant portion of final energy consumption, is at the center of these changes. In the European Union, the revised Energy Efficiency Directive (EED) imposes rigorous criteria for district heating (DH) systems, necessitating a rapid shift away from fossil fuels. Concurrently, the decarbonization of the building sector through

electrification is becoming a global priority, which further emphasizes the need for synergy between district heating and power grids [2].

Despite these ambitious policy goals, progress in many member states remains insufficient. In Poland, the district heating sector is still heavily reliant on fossil fuels, with coal remaining the dominant primary energy source, followed by natural gas, the use of which is gradually increasing. The majority of small district heating systems in Poland are supplied with coal and do not have the status of efficient ones [3]. According to [4], a large number of them are considerably worn out, have low efficiency in the summer time, and will not comply with forthcoming regulations. While biomass is the primary renewable energy source (RES) currently utilized, achieving the required share of renewables and waste heat to meet the definition of an efficient district heating system by 2028 and 2035 requires far more complex technological interventions than simple fuel switching.

The transition from high-temperature second- and third-generation district heating networks to efficient, low-temperature fourth- and fifth-generation systems requires a holistic approach. This involves not only the modernization of heat sources but also the optimization of distribution networks and the integration of low-temperature heat sources that were previously technically unfeasible to utilize. The challenge is particularly acute for large, centralized systems where the inertia of existing infrastructure makes the integration of low-temperature sources, such as industrial waste heat or ambient heat via heat pumps, a complex hydraulic and thermodynamic problem.

The district heating system in Opole, which serves as the case study for this research, represents a typical large-scale, fossil fuel-fired centralized system undergoing this difficult transition. The system, supplying approximately 1.5 million GJ of heat annually, is currently based on a mix of coal-fired boilers and combined heat and power (CHP) units, as presented in Table 1. To meet future efficiency criteria, the system operator must implement a strategy that goes beyond conventional solutions, exploring hybrid configurations that combine cogeneration with waste heat recovery.

Table 1. Existing heat and power generation units of the ECO Opole plant (2026).

Unit		Fuel	Thermal Power MW	Electric Power MW
CHP 1	3 gas engines	NG	2.75	2.68
CHP 2	3 gas engines	NG	2.75	2.68
WP120	Pulverized coal boiler	HC	90.0	–
WR25 #1	Stoker hot water boiler	HC	32.6	–
WR25 #2	Stoker hot water boiler	HC	29.1	–
GTU	Gas turbine + HR boiler	NG	14.7	7.4
OR50	Stoker steam boiler + turbine	HC	31.0	10.9
	(as above, heat-only mode)		40.0	0
GB#1	Gas boiler	NG	25.0	–
GB#2	Gas boiler	NG	18.0	–

NG—natural gas, HC—hard coal.

Figure 1 illustrates the key operational parameters of the district heating system, specifically, the supply and return temperature profiles as a function of outdoor climatic conditions. As depicted, the return temperature exhibits significant variability, which is a critical boundary condition for the hydraulic integration of any waste heat recovery system.

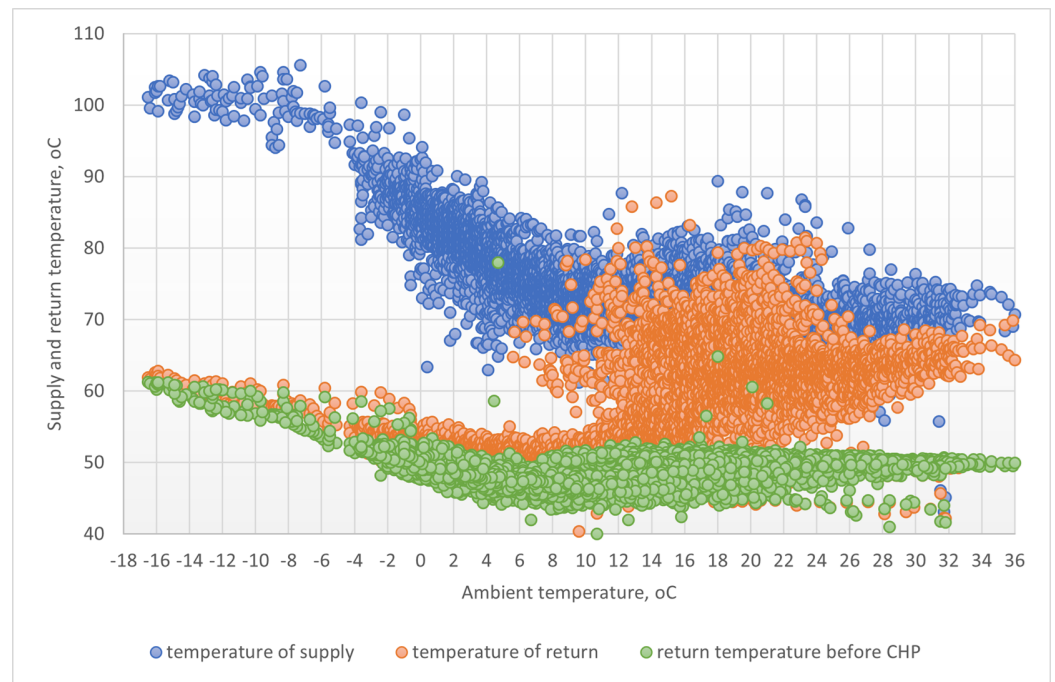


Figure 1. Temperature levels in the ECO Opole DH network in terms of the ambient conditions. Source: ECO Opole operator data.

The data points presented in Figure 1 are actual measurements from the 2023–2024 heating season extracted from the ECO Opole plant’s database. These measurements were collected using industrial-grade temperature sensors (Pt100) with an hourly sampling rate. The scatter plot format was selected to visualize the full range of temperature fluctuations.

This paper focuses on the technical feasibility and optimization of a specific retrofitting option: a low-temperature hybrid heat recovery system. This solution aims to maximize the utilization of waste heat from gas engine cooling circuits and exhaust gases, bridging the gap between high-temperature generation assets and the evolving requirements for sustainable energy systems.

State of the Art Review

The decarbonization of district heating systems is a subject of intensive research focusing on the integration of renewable energy sources (RES), waste heat utilization, and the deployment of large-scale heat pumps (HPs). The literature emphasizes that the future of DH lies in fifth-generation systems, also known as ultra-low-temperature district heating. As reviewed by Gong et al. [5], these systems reduce heat loss and significantly expand the range of available low-grade heat sources. However, for many existing networks, the concept of fourth-generation district heating (4GDH) remains the foundational framework for modernization. As highlighted by Lund et al. [6], this approach integrates smart thermal grids into future sustainable energy systems, emphasizing lower distribution temperatures. In this context, Rossi et al. [7] provided field evidence for this transition, demonstrating that medium-temperature heat pumps (MTHP) can effectively upgrade low-grade return flows (30–50 °C) to supply levels of 65–75 °C, achieving a stable weighted average COP of 3.96 while meeting fourth-generation requirements.

Furthermore, district heating infrastructures must be designed to cope with the intermittency of renewable energy. Lund [8] points out that district heating should be an integral part of smart energy systems, providing necessary flexibility to the electricity sector through thermal storage and power-to-heat technologies. For existing high-temperature networks, the transition requires intermediate steps and sophisticated integration strategies.

Critical steps in this transition are the identification and mapping of potential heat sources. Lund and Persson [9] analyzed the potential for Denmark, demonstrating that low-temperature heat sources are present near almost all district heating areas, with sea water identified as a substantial future resource. Similarly, Fang et al. [10] proposed a holistic approach to utilizing industrial waste heat, defining an investment vs. carbon reduction (ICR) index to prove the cost efficiency of such solutions in Northern China. In their subsequent work, Fang et al. [11] expanded this concept to systems using multiple low-grade waste heat sources. They introduced ‘tangency technology’ to optimize the collection of heat from various grades and highlighted that lowering return water temperatures is crucial for the long-distance delivery of waste heat. Similar assessments for the integration of waste heat targeting carbon neutrality have been performed for other cities, such as Riga, where biomass and waste heat were identified as the main energy carriers [12].

Heat pumps are widely recognized as the key technology enabling sector coupling and the integration of low-grade heat. Volkova et al. [13] claim that large-scale HPs are key future technologies for the Baltic states, potentially generating more than half of their heat by 2050. Sayegh et al. [14] introduced a ‘technical triangle’ concept, comprising the heat source, heat pump technology, and heat requirements, to determine the optimal placement and connection modes of HPs in DH networks. Their work underscores that the correct integration strategy is essential for achieving high efficiency. Furthermore, Capone et al. [15] emphasized that the environmental and economic benefits of large-scale heat pump installations are strongly dependent on their physical location within the network structure, as suboptimal placement can significantly diminish the potential for CO₂ reduction. This is supported by Bach et al. [16], who analyzed the integration of large-scale HPs in the Greater Copenhagen DH system using the Balmorel model. They found that HPs connected to distribution networks could achieve more full load hours compared to those connected to transmission networks at elevated temperatures. Optimization strategies for 100% renewable systems, such as the one proposed for Aalborg by Yuan et al. [17], also confirm that an appropriate mix includes significant shares of industrial waste heat (e.g., 40%) and heat pumps (e.g., 20%).

Barco-Burgos et al. [18] provided a comprehensive review of high-temperature heat pump (HTHP) integration, identifying twelve generic configurations. They noted that, for individual HPs in DH networks, the coefficient of system performance (COSP) typically ranges between 3 and 4. The performance of heat pumps is strictly dependent on the combination of heat sources. Pieper et al. [19] assessed a combination of groundwater, seawater, and air as a heat source. Their optimization using linear programming revealed that a mix of heat sources can increase the seasonal coefficient of performance (SCOP) by up to 11% compared to using a single source individually. In multi-source systems, such as the one in Salaspils (Latvia) analyzed by Pakere et al. [20], it was proven possible to entirely phase out fossil fuels by combining solar thermal, biomass, and heat pumps, although increasing thermal storage capacity was crucial. Tańczuk et al. [21] proposed a waste heat recovery facility from slag based on a high-temperature R134a heat pump system. The results of this research indicate a possible recovery ratio of between 58.8% and 88.0% of energy slag potential.

Furthermore, the environmental impact of HPs versus combined heat and power (CHP) is subject to seasonal variations. Rinne and Syri [22] demonstrated that, while HPs are generally seen as emission-reducing technologies, their actual emission factors depend heavily on the seasonal carbon intensity of electricity production. In some scenarios, efficient CHP might still offer lower emissions than HPs during cold seasons if the power grid is coal-intensive. Moreover, electrification makes the DH system vulnerable to electricity price volatility. Sihvonen et al. [23] showed that, in such cases, thermal energy

storage and participation in reserve markets are essential to avoid cost increases and reduce investment risk. Expanding on sector coupling strategies, Massulli et al. [24] analyzed the synergistic coupling of waste heat from PEM electrolysis with district heating, showing that the temperatures generated during hydrogen production are highly compatible with modern urban heating systems.

The integration of high-temperature heat pumps (HTHPs) is particularly relevant for existing networks that cannot immediately lower their supply temperatures. Mateu-Royo et al. [25] illustrated the potential of HTHPs, showing that using the DH network as a heat sink for waste heat recovery can reduce operating costs by 50–100% and significantly lower CO₂ emissions compared to gas boilers. They also emphasized the importance of alternative low-GWP refrigerants in future-proofing these installations. Strategic planning must therefore be holistic, as proposed by Pettersson et al. [26] for Gothenburg, where industrial waste heat covers the base load while bio-feedstock and heat pumps cover medium and peak loads.

This study serves as a direct continuation and technical elaboration of the strategic analysis presented by Kalina et al. [27]. In their work, the authors addressed the broad planning aspects of energy transition and decarbonization for DH systems in Poland, identifying the critical constraints and the necessity of a holistic strategy involving technological, economic, and legal interventions. While the previous study provided the roadmap for achieving efficient district heating status by 2028 and 2035, it highlighted the need for specific technical solutions to manage dispatchable waste heat sources.

Recent studies have significantly expanded the scope of waste heat recovery, exploring not only industrial sources but also urban infrastructure. For instance, the integration of waste heat from supermarket refrigeration systems and power substations has been analyzed, demonstrating both technical feasibility and economic benefits for district heating networks [28]. This scope is further expanded by Verzino et al. [29], who investigated the smart reuse of waste heat from data centers combined with thermochemical energy storage, providing a scalable solution for seasonal decoupling in low-temperature networks. Furthermore, advanced techno-economic optimization methods for high-temperature heat pumps (HTHPs) have been developed to maximize waste heat recovery efficiency [30]. Comprehensive approaches now also include thermal-hydraulic modeling of flexible substation layouts, which is crucial for lowering return temperatures in modernizing grids [31].

One can see that the recent literature provides comprehensive and detailed analyses of large-scale heat pump (HP) performance, including variable coefficient of performance (COP) maps, source/sink temperature dynamics, and analysis of the use of low-GWP refrigerants. In terms of modeling accuracy, Brodal et al. [32] demonstrated that high-order spectral methods for approximating heat exchanger areas offer higher reliability and computational speed for heat pump optimization compared to traditional iterative LMTD methods. While these studies offer high technological resolution, a specific challenge remains in the hydraulic and thermal coupling of HPs with existing cogeneration assets. In many legacy systems, the primary constraint is not the HP efficiency, but the operational safety of the prime mover.

Despite the extensive body of literature on large-scale heat pumps, a distinct research gap persists regarding their practical integration into second- and third-generation district heating networks, where supply temperatures remain high. Current research often assumes theoretical optimization by connecting heat pumps to the network return line to maximize the coefficient of performance (COP). However, this purely thermodynamic approach overlooks the severe technological constraints encountered in legacy systems. Specifically, there is a lack of studies verifying the real-world operational boundaries of prime movers.

In high-temperature networks, uncontrolled temperature lifts on the return line can easily violate these limits, jeopardizing the uninterrupted operation of the cogeneration units.

To address this gap, the novelty of this study lies in proposing a hybrid integration framework that bridges the conflict between theoretical COP optimization and strict industrial safety limits. Rather than a standard connection, this work introduces a validated technological approach for legacy networks that dynamically manages the temperature lift to recover low-grade waste heat without compromising the prime mover's safety margins.

Therefore, this study aims to fill these gaps by:

1. Investigating the thermal safety margins to ensure that waste heat recovery does not violate the strict 70 °C engine inlet limit under variable network flow conditions.
2. Proposing a novel hybrid integration of components through two alternative configurations that serve as a low-temperature waste heat recovery system during engine operation while providing a renewable-based heat supply using additional air-source heat pumps (ASHPs) during engine downtimes, ensuring continuous thermal supply and the long-term stability of the primary gas engine.

This framework provides a clear understanding of how to optimize heat recovery within these narrow hydraulic boundaries, using professional tools such as the Oilon Selection Tool [33] to validate heat pump (HP) performance under real, variable network conditions.

This paper is an extended version of our paper published in the 20th SDEWES Conference, Dubrovnik, Croatia, 5–10 October 2025 [34].

2. Methodology

2.1. Technical Description: Reference and Low-Temperature Heat Recovery Variants

This study concerns low-temperature waste heat recovery from JENBACHER JMS 616 GS-N.L gas cogeneration units with the following design parameters provided by the manufacturer (INNIO Jenbacher): electric output 2678 kW, electrical efficiency 43.9%, thermal efficiency 39.9% for exhaust gas temperature 120 °C. For the design configuration, around 16% of the supplied chemical energy (LHV-based) is lost in the conversion process. To harvest additional low-temperature heat, a recovery system is proposed in two configurations, as shown in Figures 2 and 3.

In Variant 1, the LT engine heat recovery system (the C-LT exchanger) is hydraulically separated from the exhaust system (the ECO II economizer). The LT circuit receives heat from the engine and supplies it to the HP I heat pump. The ECO II economizer receives medium-temperature heat from exhaust gases and supplies it to the return pipeline of the district heating system. The expected advantages of this configuration are simpler hydraulic integration, a smaller required heat pump capacity (yielding lower investment costs), and a highly optimized coefficient of performance (COP) due to the independent management of heat sources.

In Variant 2, the LT system is integrated with the engine exhaust system (ECO II). In this variant, the ECO II economizer raises the temperature of the glycol–water mixture supplied to HP I, i.e., the source heat for HP I is increased. The expected advantage of this centralized setup is the deployment of larger heat pumps, which should significantly increase the total generation from renewable energy sources (RES) during engine downtimes, albeit at the expense of higher operational complexity and capital expenditure.

In the following subsections, elements of the HR system are described, along with their data and assumptions and model equations used for waste heat quantification.

2.2. Data Collection

The analysis is based on selected CHP unit historical operation data (Table 2), registered between 2023 and 2024 provided by the district heating operator, which were refined to hourly average values to ensure consistency and eliminate measurement noise.

Some nominal design values are provided by the manufacturer (INNIO Jenbacher, Jenbach, Austria) for the specific JMS 616 units commissioned at ECO Opole, representing operation at 100% nominal load under standard ISO 3046-1 [35] conditions.

The assumption of nominal parameters results from the fact that the engine is operating in an ON/OFF mode, i.e., it is either operating under standard conditions or not running.

Based on the given temperature values, and the known mass flow rates of flue gases and water, energy balance calculations are set for the exhaust gas system and heat pumps, as presented in the following subsections.

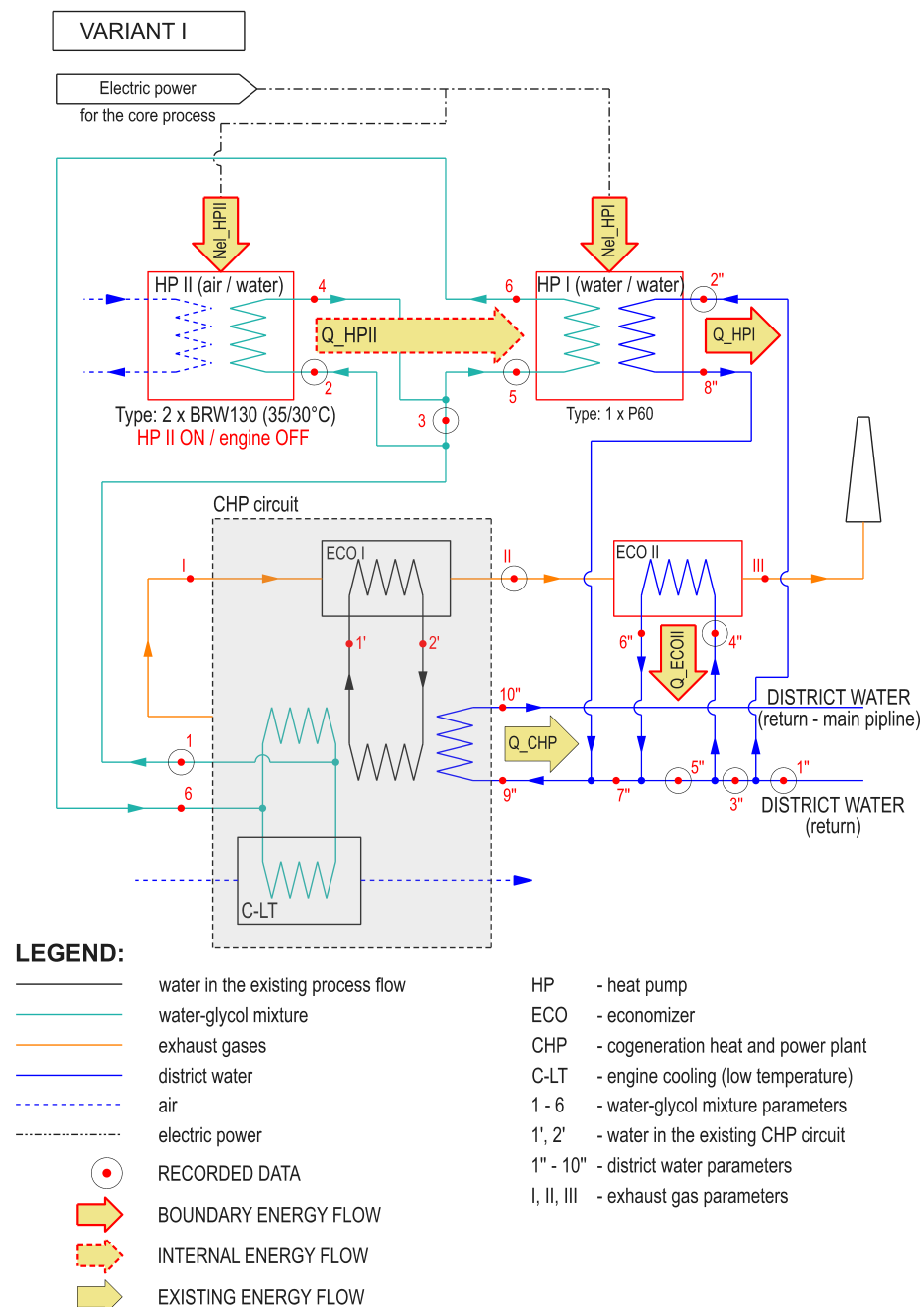


Figure 2. Variant 1 of LT heat recovery from a gas engine.

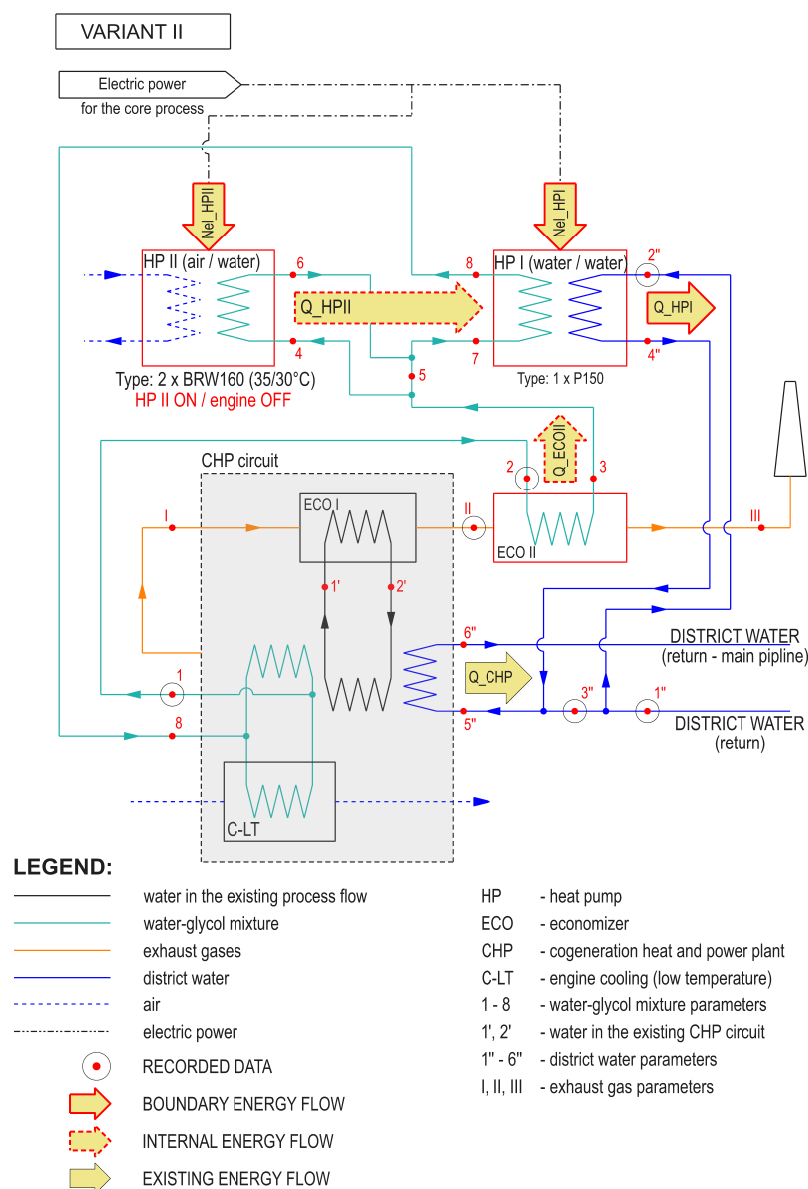


Figure 3. Variant 2 of LT heat recovery from a gas engine.

Table 2. Recorded example data from CHP unit.

Date and Time	Water–Glycol Mixture Temp. After Cooling CHP Unit	Return District Water Temp.	Exhaust Gas Temp. After ECO I
t, h	VAR.1: $T_1 \dots T_5, ^\circ C$ VAR.2: $T_1 = T_2, ^\circ C$	VAR.1: $T_1'' \dots T_5'', ^\circ C$ VAR.2: $T_1'' \dots T_3'', ^\circ C$	$T_{II}, ^\circ C$
15.12.2023 00:00	45.57	48.52	120.81
15.12.2023 01:00	45.56	48.42	121.22
15.12.2023 02:00	45.37	48.39	121.84
...	...	...	...
15.06.2024 11:00	48.23	50.67	124.72
15.06.2024 12:00	48.28	50.60	124.72
15.06.2024 13:00	48.31	50.83	124.91
...	...	...	...
14.12.2024 20:00	45.07	49.08	125.03
14.12.2024 21:00	44.95	49.23	124.72
14.12.2024 22:00	45.02	49.21	125.44

2.3. Gas Exhaust System

For the reference case, flue gas heat recovery is carried out in an economizer with a 120 °C design outlet temperature. In the varying operating conditions, this temperature T_{II} varies from approximately 100 to 130 °C.

For both Variants 1 and 2, an additional low-temperature heat exchanger ECO II is installed to harvest low-temperature heat. For Variant 1, the ECO II exchanger is connected directly to the return line, and a part of the return water $\dot{m}_{1''}$ is directed to this exchanger. The flow $\dot{m}_{1''}$ is calculated from the energy balance. For Variant 2, the ECO II exchanger is connected to the glycol mixture line.

While balancing the low-temperature exhaust heat, one should keep in mind that the conventional performance parameters of CHP units (based on combustion engines) are given with the *lower heating value* as the reference chemical energy of fuel. Therefore, if any latent heat due to vapor condensation is recovered, the overall engine efficiency may exceed unity, as happens in the case of condensing boilers. In general, the quantity of heat recoverable from an exhaust stream between two temperatures results from the decrement of the exhaust gas enthalpy and the latent heat of the condensed water [36]:

$$\dot{Q}_{\text{Exhaust}} = \left(\dot{n} \Delta \bar{h} \right)_{T_1}^{T_2} + \dot{m}_w r, \quad (1)$$

where r is the evaporation enthalpy. The formulation of the enthalpy function for the exhaust gas should account for the partial pressure of vapor, which must not exceed the saturation pressure:

$$\bar{h} = \sum_i x_i \bar{h}_i(p_i, T), \text{ where.} \quad (2)$$

$$p_i = x_i p \quad \cap \quad p_{\text{H}_2\text{O}} \leq p_{\text{sat}}(T). \quad (3)$$

The quantity of condensed water is calculated from the difference of the partial pressure of water in the exhaust gas and the maximum pressure at the final exhaust gas temperature following the heat recovery:

$$\dot{m}_w = M_{\text{H}_2\text{O}} \dot{n}_{\text{Exhaust}} [x_{\text{H}_2\text{O}} p - p_{\text{sat}}(T)], \quad (4)$$

where $M_{\text{H}_2\text{O}}$ is the molar mass of water.

This part of the study aims to select and simulate the operation of a heat exchanger in two previously defined configurations. The selection process was conducted in agreement with a local heat exchanger supplier using commercial heat exchanger selection software. This software determines the overall heat transfer coefficient (U) based on the specified physical geometry of the exchanger (plain tube bundles in a staggered arrangement) and the thermodynamic boundary conditions of the fluids (mass flow rates, inlet temperatures, and operating pressures). The calculation utilizes standard empirical correlations for cross-flow tube banks to derive the convective heat transfer coefficients while also accounting for the latent heat transfer component in the condensing section (wet part). Key parameters obtained from the selection process, particularly the overall heat transfer coefficient (U), were employed to develop a computational model within the TRNSYS 18 simulation environment. The model consists of a TYPE 5f cross-flow exchanger element supplemented with components designed for data extraction (TYPE 9). To verify the reliability of the simulation, the TRNSYS heat transfer rates and outlet temperatures were cross-checked with the guaranteed performance parameters from the heat exchanger software. The methodology is shown in Figure 4.

- Variant 1: Employs the C-LT cooling circuit as the sole heat source, supported by an Oilon P60 heat pump. The ECO II economizer supplements this by preheating the return water.
- Variant 2: Integrates both the C-LT circuit and flue gas heat recovery, using a larger Oilon P150 heat pump to supply the return line exclusively.

The selection of Oilon P60 and P150 units was based on an exhaustive verification of their operating envelopes using the Oilon Selection Tool. The operational boundaries and performance maps were validated using an industry-certified simulation environment to ensure high computational fidelity and cross-referenced with the manufacturer's certified COP maps. To ensure high computational fidelity, the models were directly calibrated using the actual hourly operational data from the ECO Opole plant. By mapping real-world water flow rates and variable temperature profiles into the software, we verified that the simulated off-design behavior matches actual network conditions. These specific models were chosen because they successfully cover the full range of hourly operational points throughout the simulated year, particularly under the unique conditions of a high-temperature heat source combined with a low temperature lift. Other models (e.g., P80, P40) were excluded as they could not maintain stable operation across the entire annual thermal map.

Both variants are designed to maintain return temperatures below 70 °C, ensuring compatibility with engine safety requirements. Variant 1 offers a simpler and more cost-effective solution, while Variant 2 provides higher integration potential and greater thermal output, albeit with increased investment and operational complexity.

For both heat recovery variants, all heat pumps were selected and modified using the Oilon Selection Tool provided by the manufacturer. The software contains design and off-design data and enables the analysis of single or cascade configurations using Oilon and Enerblue heat pumps.

Prior to describing the thermal selection of WSHPs, it is necessary to define the operating modes of the CHP unit:

CHP on: The CHP engine is running, waste heat is available;

CHP off: The CHP engine is not running. To ensure a continuous operation possibility for HP I, an additional air-source heat pump HP II is proposed, connected as a two-stage thermal cascade system.

The thermal selection and calculation for **Variant 1** is performed under the following assumptions:

For CHP on: It is assumed that the CHP unit operates in a constant power mode. The existing fan cooler capacity is 180 kW; however, based on the registered operational data, an Oilon P60 heat pump was selected with the nominal cooling capacity of $\dot{Q}_{LT} = 127$ kW. It is also assumed that the cooling stream is constant and equals 6 K ($T_1/T_3 = 50/44$ °C). However, the return water temperature T_1'' is variable and it is given as an input value $T_1'' \in (41, 61)$ °C. To keep the assumed source heat flux and the target temperature increment constant (as described), the operation of HP I is slightly adjusted by modifying the compressor input work N , which slightly changes the COP and the output thermal power.

For CHP off: The air-source heat pump HP II is used to provide the source heat for HP I. Here, the operation depends on the ambient temperature T_0 :

- (a) For $T_0 \in (-17; +3) ^\circ\text{C}$, the source heat flux for HP I results from the maximum possible heat generation in HP II:

$$\dot{Q}_{\text{HP I}}^{\text{source}} = \dot{Q}_{\text{HP II}}^{\text{prod, max}} \quad (5)$$

- (b) For $T_0 > +3 ^\circ\text{C}$, the source heat flux for HP I (generated in HP II) is kept constant:

$$\dot{Q}_{\text{HP I}}^{\text{source}} = 123 \text{ kW} \quad (6)$$

The calculation for **Variant 2** is performed under the following assumptions:

For CHP on: It is assumed that the CHP unit operates in a constant power mode. Heat generation in the LT system is constant and equals $\dot{Q}_{\text{LT}} = 127 \text{ kW}$, and the source temperature $T_{\text{LT}}^{\text{in/out}} = 50/44 ^\circ\text{C}$. Next, this stream is heated in the ECO II exchanger. Heat generation in the ECO II system depends on the exhaust gas temperature ($T_{\text{II}}^{\text{min}} = 80 ^\circ\text{C}$, $T_{\text{II}}^{\text{max}} = 140 ^\circ\text{C}$, $T_{\text{II}}^{\text{avg}} = 115 ^\circ\text{C}$) and it equals $\dot{Q}_{\text{ECO II}}^{\text{min}} = 180 \text{ kW}$, $\dot{Q}_{\text{ECO II}}^{\text{max}} = 390 \text{ kW}$, $\dot{Q}_{\text{ECO II}}^{\text{avg}} = 250 \text{ kW}$, respectively.

For CHP off: The air-source heat pump HP II is used to provide the source heat for HP I. Here, the operation depends on the ambient temperature T_0 :

- (a) For $T_0 \in (-17; +3) ^\circ\text{C}$, the source heat flux for HP I results from the maximum possible heat generation in HP II:

$$\dot{Q}_{\text{HP I}}^{\text{source}} = \dot{Q}_{\text{HP II}}^{\text{prod, max}} \quad (7)$$

- (b) For $T_0 > +3 ^\circ\text{C}$, the source heat flux for HP I (generated in HP II) is kept constant:

$$\dot{Q}_{\text{HP I}}^{\text{source}} = 287 \text{ kW} \quad (8)$$

Based on the calculated thermal streams, the Oilon P150 (R1234ze) heat pump was selected for Variant 2.

A summary of HR using heat pumps is given in Table 3.

Table 3. Summary of heat recovery approach per variant.

Variant	Lower Heat Source	Heat Pump
1	LT system	Oilon P60 (R1234ze)
2	Exhaust gases + C-LT system	Oilon P150 (R1234ze)

Rather than exploring new refrigerant properties, this work focuses on the technical feasibility of using existing low-GWP refrigerants to maximize recovery efficiency while ensuring the cogeneration unit's safe work. This study utilizes R1234ze within a heat pump system, where the operating envelope is dynamically limited by the $70 ^\circ\text{C}$ return temperature constraint. The use of R1234ze in the Oilon P60 and P150 units was verified using the Oilon Selection Tool to ensure that the compressor operating envelope is respected across the full range of variable district heating return temperatures.

2.5. Role and Configuration of Air-Source Heat Pumps

Air-source heat pumps (ASHPs) serve as renewable-based auxiliary sources, ensuring heat supply continuity during cogeneration engine downtimes caused by maintenance, failures, or economic constraints. Their integration into the Opole DH system enhances operational flexibility and supports decarbonization goals. Two configurations were analyzed:

- Variant 1: 1× Enerblue BRW 130 (operating point $35/30 ^\circ\text{C}$) + P60;

- Variant 2: 2× Enerblue BRW 160 (operating point 35/30 °C) + P150.

The selected ASHPs are high-temperature reversible units using R410A refrigerant, each equipped with two independent refrigeration circuits, scroll compressors, speed-regulated axial fans, and plate heat exchangers. They are capable of delivering hot water up to 62 °C and operate effectively in ambient temperatures from −18 °C to 35 °C.

The bivalent point was set at 3 °C, below which the efficiency of ASHPs drops below that of water-source systems. However, historical data show that only 20% of engine shutdowns occurred below 3 °C, meaning ASHPs mostly operate in favorable conditions. Maintenance and negative electricity prices predominantly affect operations from May to October, aligning with periods of higher ambient temperatures and optimal ASHP performance.

The heating capacity of the BRW130 and BRW160 units is matched to the cooling capacity of the P60 and P150 units, respectively, ensuring balanced operation. During engine operation, ASHPs can be powered by on-site generators; during downtimes, they switch to grid electricity, enabling flexible and cost-effective operation. This dual-mode capability allows the system to capitalize on negative electricity prices, converting surplus renewable electricity into thermal energy. The selection was based on a comprehensive analysis of the full annual operating map, covering both extreme winter and summer conditions. Furthermore, these units were selected due to the technical partnership between Oilon and Enerblue, which allows integration and verification of the complete heat pump cascade within a single, certified software environment. This ensured that our assumptions regarding hydraulic coupling and thermal performance were validated using industry-standard manufacturer algorithms.

2.6. Preliminary Economic Evaluation

To enhance the overall understanding of the process, a raw estimation of the economic profitability of the proposed solutions is proposed under the following assumptions:

1. The analysis is performed on a differential basis, only accounting for additional equipment and energy flows;
2. Existing engine heat production Q_{CHP} , deriving from the jacket water and the economizer ECO I, is assumed to be invariant;
3. Capital and operational expenditures as well as prices are evaluated based on market inquiries as well as ECO S.A. internal assessments, as listed in Table 4. Uncertainty related to this evaluation is addressed by the sensitivity analysis.

Table 4. Assumptions for the economic analysis.

Item	Cost	Source
<i>Capital expenditures:</i>		
WSHP I type P60	81,000 EUR	Market inquiry
WSHP I type P150	138,000 EUR	Market inquiry
ASHP II type BRW130	60,000 EUR	Market inquiry
ASHP II type BRW160	105,000 EUR	Market inquiry
ECO II (both variants)	60,000 EUR	Market inquiry
Piping/ Automation/ Documentation/ Assembly	35,000 EUR	ECO S.A.
<i>Operational expenditures:</i>		
OPEX V1	3800 EUR/year	ECO S.A.
OPEX V2	5200 EUR/year	ECO S.A.
<i>Prices:</i>		
Electricity price	123 EUR/MWh	[37]
Heat price (ECO S.A.)	64.88 EUR/GJ	ECO S.A.

Investment cost J_0 is attributed to year 0, while yearly cash flow CF for years 1–15 is calculated as the sum of the generated heat production at the given price, subtracting the cost of electricity and other OPEX costs. Next, the net present value is calculated as follows:

$$\text{NPV} = -J_0 + \sum_{t=1}^{t=15} \text{CF}(t) \cdot \frac{1}{(1+r)^t} \quad (9)$$

The simple payback time is calculated as the ratio of the investment cost to the undiscounted yearly cash flow:

$$\text{SBP} = \frac{|J_0|}{\text{CF}} \quad (10)$$

Eventually, the internal rate of return (IRR) as well as two break-even points (BEPs) are calculated as follows:

IRR represents the discount rate for which the $\text{NPV} = 0$;

BEP (heat price) represents the heat price for which the $\text{NPV} = 0$;

BEP (electricity price) represents the electricity price for which the $\text{NPV} = 0$.

Results of the economic evaluation are presented in Section 3.6.

2.7. Emission Aspects

The application of the low-temperature waste heat recovery system enables a substantial reduction of heat generation in the conventional, fossil-based DH system. However, some electricity input is required which should be burdened with the current specific emission factor of the national electricity mix.

Accordingly, the avoided gross CO_2 emission is calculated as the product of the specific emissions in the heating sector and the reduction of heat generation ΔQ :

$$\Delta m_{\text{CO}_2}^{\text{gross}} = \varepsilon_Q \cdot \Delta Q \quad (11)$$

To obtain the net CO_2 emission reduction, it is required to calculate the increment of CO_2 emissions due to the electricity ΔE consumed for heat pump operation and to subtract it from the gross value:

$$\Delta m_{\text{CO}_2}^{\text{HP}} = \varepsilon_E \cdot \Delta E \quad (12)$$

$$\Delta m_{\text{CO}_2}^{\text{net}} = \Delta m_{\text{CO}_2}^{\text{gross}} - \Delta m_{\text{CO}_2}^{\text{HP}} \quad (13)$$

The following indicators are assumed:

1. The avoided emission factor for heat production in the existing DH sector is $\varepsilon_Q = 100$ kg CO_2 /GJ. It is assumed that the new input replaces the worst existing coal-based unit. Data source: ECO S.A.
2. The emission factor for electricity production in the national mix is $\varepsilon_E = 528$ g/kWh (2025). Source: Low Carbon Power [38].

3. Results and Discussion

3.1. Heat Exchangers

The selected system consists of two series-connected tube-type heat exchangers, designed to cool the flue gas below its dew point temperature. Based on the combustion of natural gas yielding a water vapor mole fraction of 15.9% and a realistic absolute flue gas pressure of 105.0 kPa, the partial pressure of water vapor is 16.70 kPa. This firmly establishes the physical flue gas dew point at 56.3 °C, which serves as the fundamental boundary condition for the condensation process.

Consequently, the simulation and selection processes were analyzed in two distinct thermal stages. In the first heat exchanger (dry section), the flue gas temperature is lowered from 130.0 °C down to 61.0 °C. Since this temperature remains safely above the calculated dew point of 56.3 °C, this stage operates entirely under dry sensible heat transfer conditions. In the second heat exchanger (wet section), the flue gas is further cooled from 61.0 °C to approximately 55.8 °C. As the fluid temperature crosses the 56.3 °C threshold within this unit, latent heat recovery via moisture condensation takes place.

After consultations with ECO Opole, stainless steel was selected as the construction material for both exchangers to ensure durability and resistance to corrosion. Both units share an identical physical geometry, forming a compact 2 × 2 m unit. Each is designed with plain pipes of 25 mm outer diameter and 2 m length arranged in a staggered pattern over 10 rows of 68 pipes, providing a bare heat transfer area of 100.5 m² per unit (201 m² total).

The thermal performance and boundary conditions for both operational variants are summarized in Table 5. Fouling factors for all streams were conservatively assumed to be 0 m²K/W (clean conditions). For Variant 1, the dry section achieves a capacity of 240 kW (heating water from 51.8 °C to 59.0 °C), while the wet section provides a thermal output of 56 kW (heating water from 49.0 °C to 51.8 °C). For Variant 2, the same heat exchanger geometry was evaluated using a 37% water–glycol solution at a mass flow rate of 8.3 kg/s, modifying the convective heat transfer coefficients and resulting in design capacities of 256 kW and 58 kW for the dry and wet sections, respectively.

Table 5. Thermal and operational parameters of the heat exchangers for Variant 1 and Variant 2.

Parameter	Variant 1 (Water)		Variant 2 (37% Glycol)	
	Dry Section	Wet Section	Dry Section	Wet Section
Cold-side mass flow (kg/s)	5.0		8.3	
Hot-side mass flow (kg/s)			4.2	
Flue gas inlet pressure (kPa)			105.0	
Cold medium inlet pressure (kPa)			500.0	
Fouling factor (m ² K/W)			0.0	
Flue gas temp. in/out (°C)			130.0/55.8	
Cold medium temp. in/out (°C)	49.0/59.0		49.0/57.0	
Total duty (kW)	240	56	256	58
Calculated <i>U</i> (W/m ² K)	89.74	87.18	89.09	85.91

3.2. Water-Source Heat Pumps

The performance of both configurations was validated through simulation and manufacturer data. The P60 unit achieves COPs above 7 under typical return conditions, while the P150 unit maintains high efficiency even at elevated return temperatures. These results support the feasibility of both variants, with the final choice depending on system priorities—cost minimization versus maximum renewable integration.

Maintaining return temperatures below 70 °C is critical for the safe and efficient operation of the cogeneration engines. Performance data from simulations and manufacturer charts show that the P60 and P150 units operate with high COPs under typical return temperature conditions, as shown in Table 6.

Table 6. Heat pump performance at various return temperatures.

Pump	Return Temp. (°C)	Heating Capacity (kW)	Power Demand (kW)	COP
P60	48	144	17	8.27
P60	60	154	28	5.57
P150	48	375	52	6.21
P150	60	392	78	5.04

3.3. Additional Air-Source Heat Pumps

The ASHPs are dimensioned to match the cooling capacity of the water-source units, ensuring optimal performance. The bivalent point is set at 0 °C, below which ASHPs are less efficient than their water-source counterparts. However, the majority of engine downtimes occur during warmer months, when ASHPs operate at higher efficiency.

Performance data (see Table 7) confirm that ASHP efficiency increases significantly above −3 °C, with COP values reaching up to 5.5 at 27 °C. The hybrid cascades (BRW130 + P60 and 2 × BRW160 + P150) ensure that the system can maintain thermal supply even during extended engine outages.

The increasing frequency of negative electricity prices in the Polish energy market presents a unique opportunity: ASHPs can be operated using low-cost or surplus renewable electricity, converting it into thermal energy. This not only reduces operational costs but also increases the share of renewable energy in the system's final energy mix.

Moreover, the deployment of larger ASHPs—despite higher investment costs—enables deeper integration of renewable electricity and waste heat. These systems can operate independently or in cascade with water-source units, depending on system needs and electricity market signals.

This hybrid configuration provides a replicable model for other DH systems, particularly those with high cogeneration shares. The greater the CHP share, the more flexible the system becomes in balancing renewable electricity and thermal demand, making ASHPs a key enabler of sector coupling and energy system resilience.

Table 7. ASHP performance chart.

Ambient Temperature (°C)	Heating Power (kW)	HP2, Variant 1 BRW 130		HP2, Variant 2 2 × BRW 160		
		COP	Power Demand (kW)	Heating Power (kW)	COP	Power Demand (kW)
−17	72.6	2.4	30.0	151.1	2.0	74.3
−15	77.6	2.6	30.2	176.2	2.4	74.4
−12	84.2	2.7	30.7	204.0	2.7	74.6
−9	91.0	2.9	31.1	219.0	2.9	75.0
−6	97.9	3.1	31.3	236.0	3.1	75.4
−3	106.0	3.4	31.5	254.0	3.4	75.8
0	114.0	3.6	31.8	274.0	3.6	76.2
3	123.0	3.8	32.1	287.0	3.8	75.2
6	123.0	4.1	30.3	287.0	4.1	70.3
9	123.0	4.3	28.5	287.0	4.3	66.7
12	123.0	4.6	27.1	287.0	4.5	63.2
15	123.0	4.8	25.7	287.0	4.8	60.0
18	123.0	5.1	24.1	287.0	5.0	57.5
21	123.0	5.3	23.1	287.0	5.2	55.0
24	123.0	5.5	22.5	287.0	5.4	53.0
27	123.0	5.6	22.2	287.0	5.6	51.1

Integration of the discussed ASHP (HP2) with the WSHP (HP1) operating in the analyzed system is shown in Figure 5.

3.4. Overall Energy Balance and System Evaluation

The overall energy balance of the extended CHP system equipped with an additional exhaust gas heat exchanger and two kinds of heat pump units is presented in Table 8.

As can be seen in Table 8, the highest thermal power is obtained from the ECO2 heat exchanger installed in the exhaust gas system. The generation of heat in the water–water heat pump unit HP1 depends on the selected variant. In Variant 1, the HP1 unit produces about 1/3 of the total waste heat generation stream while 2/3 are covered by ECO2. In Variant 2, both units are connected in series, and all the heat generated in ECO2 is supplied to the HP1 unit, which further increases the heat generation. It can be seen that the sum of

ECO2 and HP1 generation for Variant 1 (2192 + 1083 MWh) is higher than the heat collected and upgraded by HP1 in Variant 2 (2779 MWh).

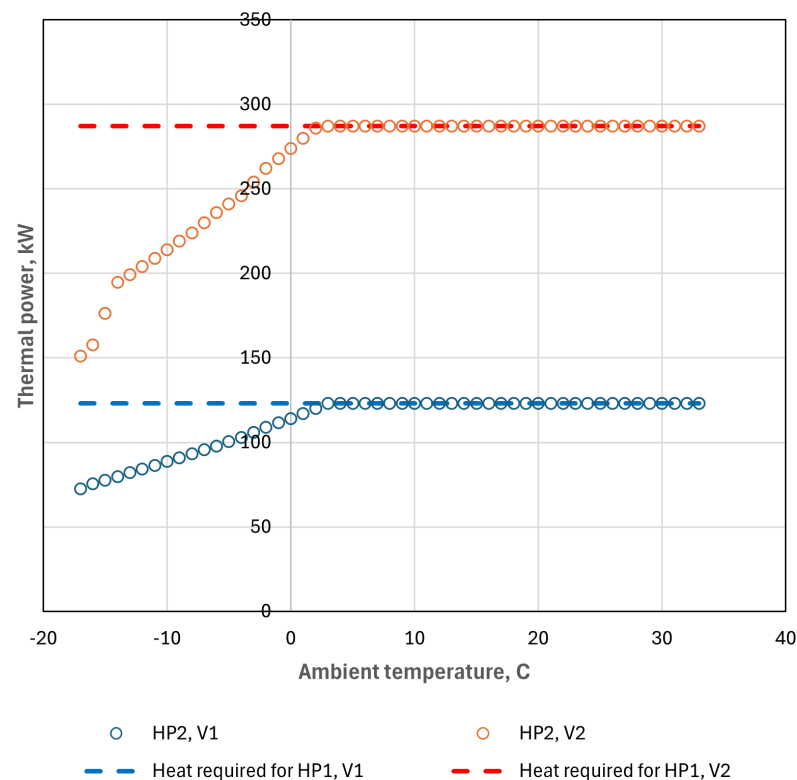


Figure 5. Matching the heat production in ASHP (HP2) with heat required for WSHP (HP1).

Table 8. Annual energy balance of the CHP system with the proposed waste heat recovery.

	Variant 1	Variant 2	
A. CHP ON: heat production in HP1 + ECO2 (7504 h):			
Existing power production		20,525	MWh
Existing heat production		18,655	MWh
Fuel consumption (LHV based)		46,755	kWh
Existing efficiency		0.838	–
MT heat production, ECO2	2192	2291 ^(a)	MWh
LT heat production, HP1	1083	2779	MWh
Electricity consumption for HP1	135	402	MWh
COP of HP1 system (waste heat)	8.03	6.90	–
System efficiency ^(b)	0.905	0.890	–
B. CHP OFF, heat production in HP1 + HP2 (1256 h):			
LT heat production, HP1 + HP2	178	408	MWh
Electricity consumption for HP1+HP2	65	163	MWh
COP of HP1+HP2 system	2.73	2.50	–
A&B. Overall system evaluation:			
Additional heat production	3453	3187	MWh
Additional electricity demand	200	566	MWh

^(a) In Variant 2: transferred to HP1, not directly to the heating network; ^(b) Simplified nomenclature, as a heat pump is included in the system.

Heat generation via the air-source heat pump HP2, operated in periods when the CHP unit is off, is higher for Variant 2. In both cases, this generation (of renewable type) is much smaller than the waste heat recovery described above.

Electricity consumption for heat pumps is much lower for Variant 1 for both types of pumps, which is due to their a smaller required size.

One can also evaluate the system by comparing its performance with the existing CHP efficiency. The efficiency of the upgraded system includes (as energy product streams) electricity production and all the managed heat generation, and, on the fuel side, it includes

the unvaried chemical energy consumption and, additionally, electricity consumption for heat pumps. Since heat pumps are included in the system, the proposed parameter ‘system efficiency’ is actually a hybrid parameter including both fuel conversion efficiency and the COP of the heat pump units; therefore, its interpretation is limited.

3.5. DH Return Temperatures

The proposed installations supply a certain quantity of heat in the form of increased return water temperature. A partial stream of return water passing through the studied installation (as per the variant) is heated in the installation, and water with an increased temperature then enters the conventional CHP system. The impact of the studied system on the temperature of the partial stream of DH return water is shown in Figure 6.

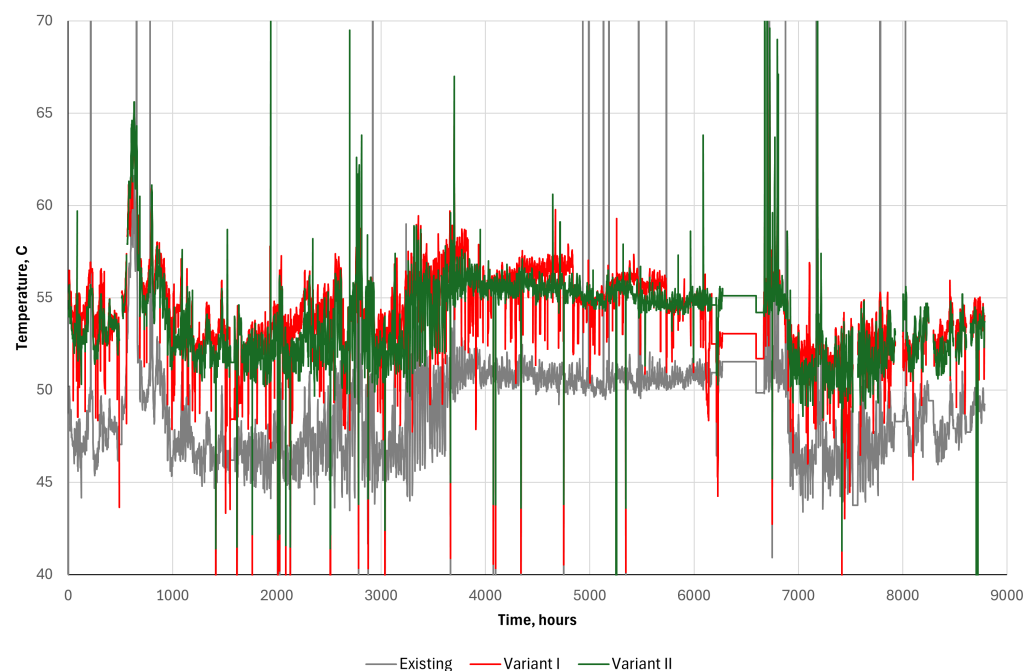


Figure 6. Impact of the designed low-temperature heat recovery system on network return temperature.

As can be seen in the figure, the average increment of the return temperature is about 6 K, and it ranges between 2 and 10 K. A key result of our comprehensive annual simulation is that Variant 1 mostly has a slightly higher temperature increment. One can also see that, during most of the operation time, the increased water temperature does not exceed 70 °C, which is the limit of undisturbed CHP operation. The simulation utilized real-world heat recovery streams correlated with verified hourly measurement data from the CHP.

3.6. Economic and Environmental Assessment

Both variants of low-temperature heat recovery provide clearly positive economic and environmental effects, as summarized in Table 9. As can be seen from the table, both variants yield highly attractive economic indicators which can be assessed, e.g., based on the short simple payback time. However, Variant 1 is more favorable in terms of all indicators, providing a much wider safety margin for the price of electricity and heat. Provided that Variant 1 also ensures higher quantity of recovered heat and higher reduction of CO₂ emissions while being burdened with a lower investment cost, the recommended choice between the two variants is straightforward.

Table 9 also includes additional scenarios related to the elongated CHP stand-by periods, which are related to the ongoing decarbonization process. Thanks to the proposed

hybrid configuration with a back-up air-source heat pump unit, the system still yields positive effects in terms of CO₂ reduction even for the reduced CHP operation time.

3.7. Sensitivity Analysis

An analysis of the sensitivity of the NPV and IRR to changes in electricity and heat prices and CapEx values was performed. The calculation results are presented in Tables 10–13 and Figures 7–10, respectively. Tables 10 and 11 present the NPV and IRR calculation results for Variant 1, while Tables 12 and 13 present the values of these economic indicators for Variant 2. Similarly, Figures 7 and 8 present the NPV and IRR calculation results for Variant 1, while Figures 9 and 10 present the values of these economic indicators for Variant 2.

Table 9. Differential evaluation of low-temperature heat recovery variants compared to the reference (existing) system. Reference prices: heat: 15.65 EUR/GJ, electricity: 123 EUR/MWh

Indicator	Value		Unit
	Variant 1	Variant 2	
Total investment cost	296	455	10 ³ EUR
Heat production value	194.5	179.5	10 ³ EUR/a
Electricity cost	24.6	69.6	10 ³ EUR/a
Other OPEX costs	3.8	5.2	10 ³ EUR/a
Yearly cash flow	166.1	104.7	10 ³ EUR/a
NPV ($t = 15$ years, $r = 0.05$)	1360	602	10 ³ EUR
SPB	1.8	4.3	a
IRR	0.56	0.218	–
BEP (heat price)	4.58	10.34	EUR/GJ
BEP (electricity price)	811	231	EUR/GJ
<i>CO₂ emission reduction scenario 1 (base) (1256 h CHP OFF):</i>			
Gross	1243	1147	Mg
Emission burdening heat pumps	106	299	Mg
Net	1137	848	Mg
<i>CO₂ emission reduction scenario 2 (1700 h CHP OFF):</i>			
Gross	1214	1203	Mg
Emission burdening heat pumps	143	404	Mg
Net	1071	798	Mg
<i>CO₂ emission reduction scenario 3 (3560 h CHP OFF):</i>			
Gross	1088	1435	Mg
Emission burdening heat pumps	300	845	Mg
Net	788	588	Mg

The obtained NPV sensitivity calculation results show that the heat price has the greatest impact on its value. A higher heat price value increases the NPV and vice versa, with the NPV in the assumed range of heat price variability having a positive value. This occurs in both the first and second calculation variants (Figures 7 and 9). In both variants, the electricity price and the CapEx value have a smaller impact on the NPV; however, an increase in these values causes a decrease in the NPV. In the case of the IRR sensitivity calculations for Variant 1, its value is significantly influenced, as in the case of the NPV, by the heat price and, in addition, the CapEx value. The influence of these values on the IRR index is opposite (Figure 8). In the case of the IRR value for Variant 2, the impact of the heat price clearly dominates (Figure 10).

Table 10. Sensitivity analysis results for the NPV for Variant 1.

NPV	−25%	−20%	−15%	−10%	−5%	0%	5%	10%	15%	20%	25%
Electricity price	1421	1409	1397	1385	1372	1360	1348	1336	1324	1312	1299
Heat price	880	976	1072	1168	1264	1360	1456	1553	1649	1745	1841
CapEx	1431	1417	1403	1388	1374	1360	1346	1332	1318	1304	1290

Table 11. Sensitivity analysis results for the IRR for Variant 1.

IRR	−25%	−20%	−15%	−10%	−5%	0%	5%	10%	15%	20%	25%
Electricity price	0.580	0.580	0.573	0.569	0.565	0.561	0.560	0.552	0.548	0.540	0.539
Heat price	0.390	0.430	0.461	0.494	0.527	0.561	0.590	0.626	0.659	0.690	0.725
CapEx	0.750	0.700	0.660	0.623	0.590	0.561	0.530	0.509	0.487	0.470	0.447

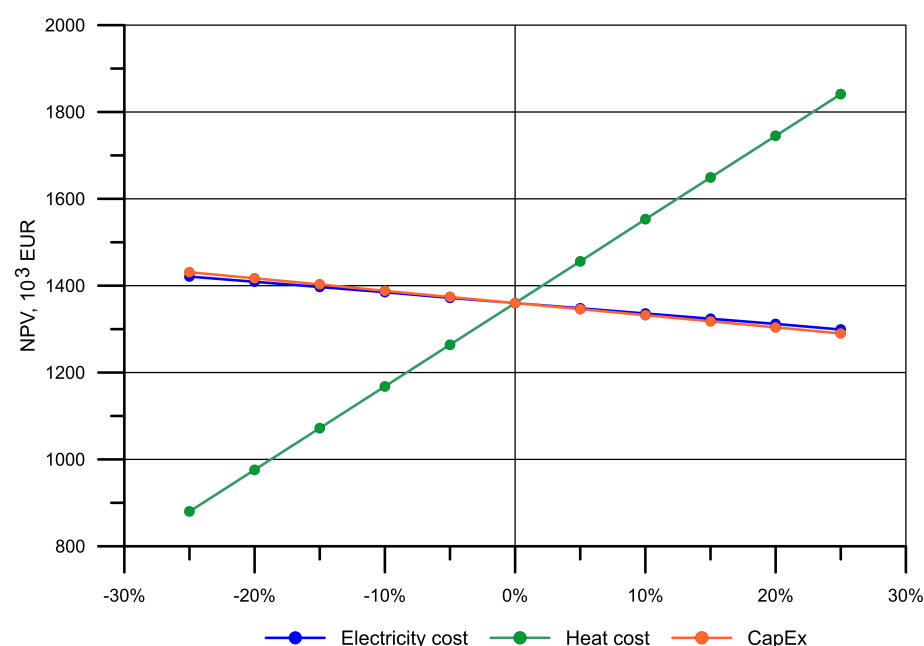
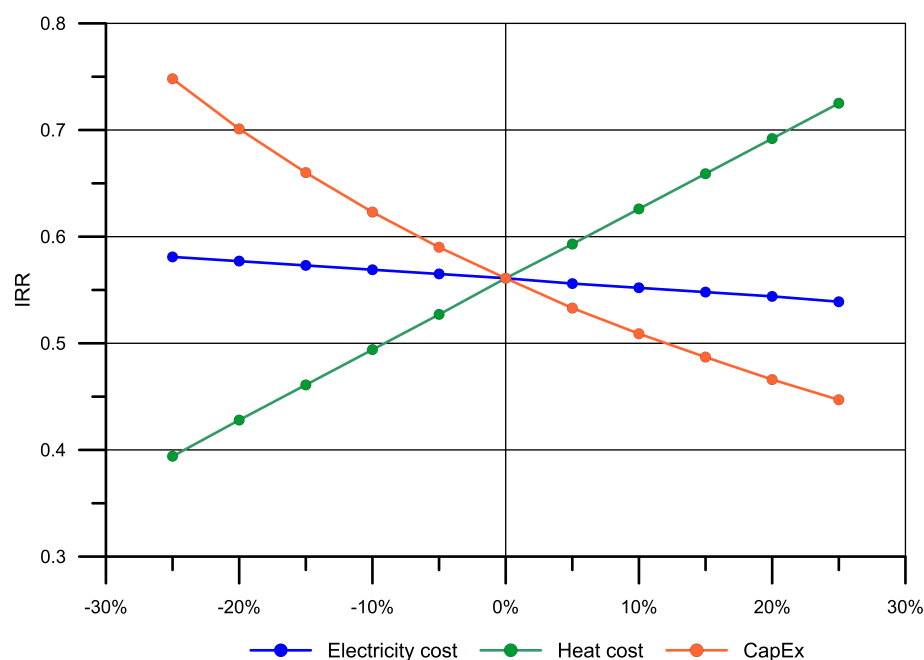
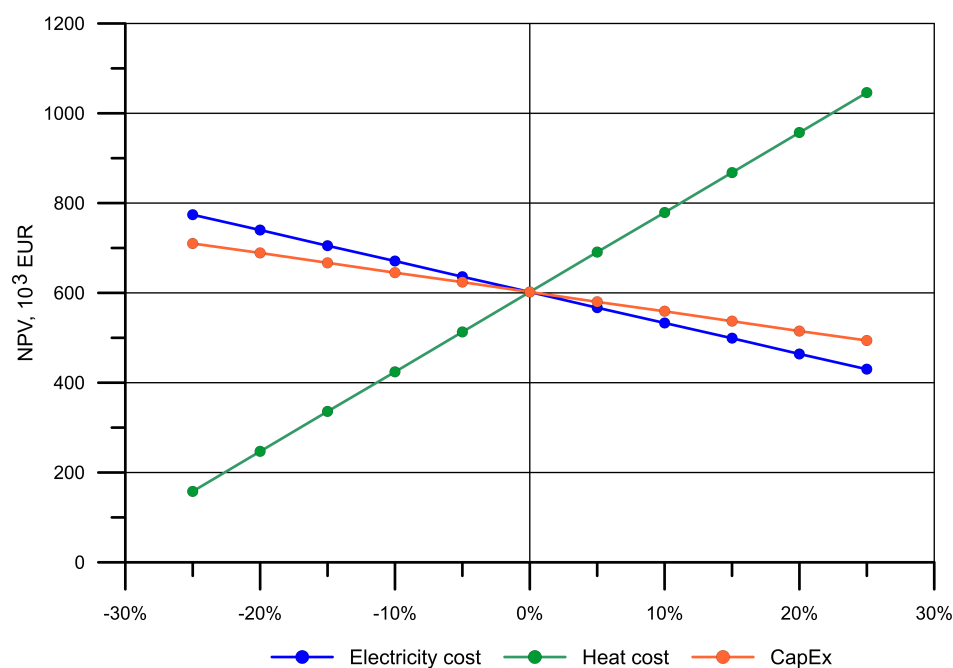
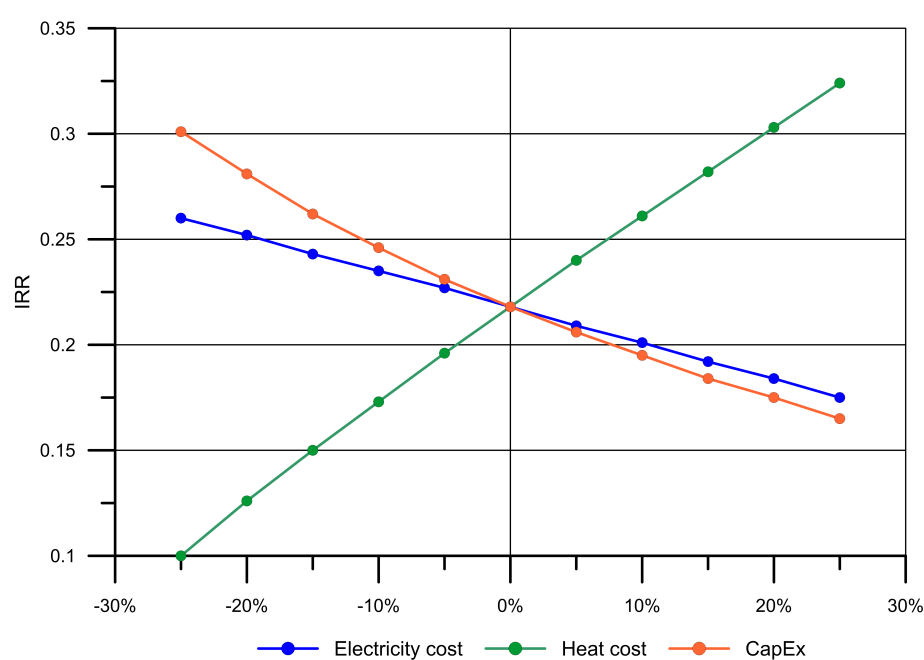
**Figure 7.** Diagram of the NPV sensitivity analysis results for Variant 1.**Figure 8.** Diagram of the IRR sensitivity analysis results for Variant 1.

Table 12. Sensitivity analysis results of the NPV for Variant 2.

NPV	−25%	−20%	−15%	−10%	−5%	0%	5%	10%	15%	20%	25%
Electricity price	774	740	705	671	636	602	567	533	499	464	430
Heat price	158	247	336	424	513	602	691	779	868	957	1046
CapEx	710	689	667	645	624	602	580	559	537	515	494

**Figure 9.** Diagram of the NPV sensitivity analysis results for Variant 2.**Table 13.** Sensitivity analysis results of the IRR for Variant 2.

IRR	−25%	−20%	−15%	−10%	−5%	0%	5%	10%	15%	20%	25%
Electricity price	0.260	0.250	0.243	0.235	0.227	0.218	0.210	0.201	0.192	0.180	0.175
Heat price	0.100	0.130	0.150	0.173	0.196	0.218	0.240	0.261	0.282	0.300	0.324
CapEx	0.300	0.280	0.262	0.246	0.231	0.218	0.210	0.195	0.184	0.180	0.165

**Figure 10.** Diagram of the IRR sensitivity analysis results for Variant 2.

4. Conclusions

This study evaluates the technical and economic feasibility of integrating hybrid waste heat recovery into an existing district heating network. By utilizing thermodynamic modeling and operational data, the research establishes a framework for enabling heat delivery from previously bypassed low-temperature sources. The findings confirm that retrofitting cogeneration units with heat pump cascades enhances system efficiency without compromising operational safety. This approach offers a practical strategy for balancing thermal supply and grid stability in increasingly volatile energy markets.

1. Single-engine heat recovery: For an individual JMS 616 unit, the proposed system successfully recovers 127 kW of low-temperature waste heat from the engine cooling circuit.
2. Variant performance comparison: Variant 1 represents a better configuration, achieving a system COP of 8.03 compared to 6.90 for Variant 2 while also providing a higher annual thermal yield of 3453 MWh. This means that a parallel configuration of heat recovery from ECO II and HP I is more favorable than organizing these devices into a cascade.
3. Economic viability: Indicators favor Variant 1 as the more effective investment, offering a simple payback period of 1.8 years (compared to 4.3 for Variant 2) and an internal rate of return (IRR) of 0.56 (compared to 0.22).
4. Environmental impact: Implementation of the recommended Variant 1 results in a net annual CO₂ emission reduction of 1137 Mg under baseline conditions involving 1256 h of cogeneration unit downtime. The analysis indicates that, in the future, the possible extended engine downtimes can reach 1700 h and 3560 h, reducing the net CO₂ savings to 1071 Mg and 788 Mg, respectively. This drop is driven by the increased operational time of air-source heat pumps powered by grid electricity, which carries a specific emission burden. Nevertheless, the hybrid system consistently supports the decarbonization of centralized heating assets in all evaluated operational scenarios.
5. Economic sensitivity: The sensitivity analysis reveals that project profitability, measured by NPV and IRR, is highly dependent on heat price fluctuations. While increases in electricity costs and CapEx negatively impact the financial parameters, their overall effect is secondary, allowing both analyzed variants to maintain a positive NPV across the entire assumed range of market variability.
6. Variant recommendation: While the specific quantitative results are tied to the site-specific operating parameters of the analyzed DH system, the proposed hybrid methodology is highly replicable across similar legacy networks in Central and Eastern Europe. Furthermore, our observations highlight that the final configuration choice is strictly limited by the prime mover's operational profile: a high frequency of unit downtimes shifts the recommendation towards Variant 2, whereas stable, continuous operation with fewer stops strongly favors the highly efficient Variant 1.

5. Outlook

The transition toward sustainable district heating requires the strategic integration of low-grade waste heat with power-to-heat technologies. The Opole case study demonstrates that hybrid systems combining cogeneration with heat pump cascades offer the necessary flexibility to manage increasing electricity price volatility. Since the commissioning of additional engines in 2025, the role of air-source heat pumps will expand, enabling the system to capitalize on negative electricity prices and balance the national power grid. Future development must prioritize the digitalization and predictive control of these hybrid assets to maximize the share of renewable energy. Ultimately, aligning local infrastructure

modernization with national energy goals will ensure the long-term resilience and climate neutrality of urban heating networks.

Author Contributions: Ł.J.: administration, conceptualization, methodology, and definition of system variants; analysis and simulation of the heat pumps (WSHP and ASHP). M.B.: thermal selection of the ECO II heat exchangers for exhaust gas heat recovery, TRNSYS modeling and simulations. A.B.: aggregation and processing of data for the simulations, analysis and simulation of the heat pumps (WSHP and ASHP). W.K.: supervision, critical review of the manuscript, and coordinated funding. M.S.: review of the manuscript in terms of sensitivity and economic analysis. All authors have read and agreed to the published version of the manuscript.

Funding: This work was co-financed by the GGrND Project (The GGrND Project at the Silesian University of Technology—support for doctoral students in the transition to a green and digital economy), which is part of the Territorial Plan for the Just Transformation of the Silesian Voivodeship 2030, with a contribution from European Funds project no. FESL.10.25-IZ.01-07E7/23. This work was also supported by the statutory research fund of the Faculty of Power and Environmental Engineering, project no. BK-240/RIE6/2026 08/060/BK_26/1169.

Data Availability Statement: The original contributions presented in this study are included in the article. Further inquiries can be directed to the corresponding author.

Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature

Latin symbols

CF	yearly cash flow
E	energy, kJ or kWh
$\bar{h}$	specific molar enthalpy, kJ/kmol
$\Delta\dot{H}$	enthalpy flux, kW
J_0	investment cost
M	molar mass, kg/kmol
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
N	compressor input work, kW
N_{el}	electric power, kW
$\dot{n}$	molar flow rate, kmol/s
p	pressure/partial pressure, kPa, bar or Pa
Q	heat production, GJ
$\dot{Q}$	heat flux/thermal power, kW
r	evaporation enthalpy, kJ/kg/discount rate, %
T	temperature, °C
ΔT	temperature drop, K
t	time, s
U	overall heat transfer coefficient, W/(m ² ·K)
x	molar share/mole fraction, –

Greek symbols

Δ	difference/macroscopic change, –
ε	emission factor, kg/GJ or kg/kWh
η	efficiency, –

Subscripts and superscripts

0	ambient conditions/reference year
1, 2, 3	state points/stream indices
"	return stream (e.g., water return)
a	ambient
avg	average

<i>d</i>	demand
<i>des</i>	design
<i>el</i>	electricity power/energy
<i>env</i>	environmental
<i>Exhaust</i>	exhaust gas stream
<i>f</i>	forward/fuel
<i>g</i>	generation
<i>HP</i>	heat pump
<i>i</i>	component/production unit index
<i>I, II</i>	stage or unit index (e.g., HP I, ECO II)
<i>in</i>	inlet
<i>max</i>	maximum
<i>min</i>	minimum
<i>out</i>	outlet
<i>prod</i>	production
<i>q</i>	heat
<i>r</i>	return
<i>sat</i>	saturation state
<i>source</i>	source (heat flux)
<i>t</i>	time step
<i>var</i>	variable
<i>w, H₂O</i>	water/condensed moisture
<i>y</i>	year

Abbreviations

<i>ASHP</i>	air-source heat pump
<i>BEP</i>	break-even point
<i>CHP</i>	combined heat and power
<i>COP</i>	coefficient of performance
<i>DH</i>	district heating
<i>DHN</i>	district heating network
<i>ECO</i>	flue gas heat exchanger/economizer
<i>ETS</i>	European Trading System
<i>HP</i>	heat pump
<i>IRR</i>	internal rate of return
<i>LT</i>	low temperature
<i>NPV</i>	net present value
<i>RES</i>	renewable energy source
<i>SBP</i>	simple payback time
<i>WSHP</i>	water-source heat pump

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