

Article

Analysis of Heat-Demand Coverage by a Hybrid PVT-Based System for a Small-Scale District-Heating Network Under the Climatic Conditions of Central Poland: A Case Study

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Abstract

This paper investigates the use of a hybrid renewable heat-supply system based on photovoltaic–thermal collectors, an industrial heat pump, thermal energy storage, and electrical energy storage for a small- to medium-scale district-heating network. A dynamic lumped-parameter model was developed and applied to hourly data from an existing network. The photovoltaic–thermal collector model accounts for low-temperature operation, wind effects, precipitation, and condensation-related heat gains, while the heat pump is represented using compressor performance characteristics under variable source and sink temperatures. The analysis focuses on whether the proposed configuration can meet summer heat demand and reduce reliance on a conventional peak or backup source during shoulder periods. The results show that, during an extended non-heating season, the system can supply approximately 90–100% of the district-heating demand while maintaining a daily mean coefficient of performance in the range of approximately 2.0–3.1. The photovoltaic–thermal field and electrical energy storage do not provide full electrical self-sufficiency, but they reduce grid electricity import; in July and August, the electricity autarky coefficient is approximately 48–49%. The results indicate that the proposed system can serve as a seasonal renewable heat source for district heating. Further refinement of the configuration, operating setpoints, and control strategy could improve its shoulder-season performance.

Keywords: PVT; heat pump; district heating; thermal energy storage (TES); energy management; mathematical modeling



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1. Introduction

Decarbonization of the heat supply is a central element of the European Union climate and energy policy. The European Green Deal set the objective of climate neutrality by 2050, while the Fit for 55 package translated this trajectory into legislative measures to reduce net greenhouse gas emissions by at least 55% by 2030 compared with 1990 levels [1,2]. This framework is particularly relevant to heating and cooling, which accounts for around half of EU energy use and remains linked to fossil-fuel combustion in many district-heating systems and combined heat-and-power plants.

The revised Renewable Energy Directive, RED III, introduces mechanisms directly affecting the heat sector [3]. Member States are obligated to increase the share of renewable

energy in heating and cooling by at least 0.8 percentage points on average annually for 2021–2025 and by at least 1.1 percentage points for 2026–2030, relative to the 2020 baseline. RED III also permits the inclusion of waste heat and cold within defined limits and sets an EU-level renewable-energy target of at least 42.5% of gross final energy consumption by 2030, with an endeavor to reach 45%. For district heating and cooling, Member States should endeavor to increase the share of renewable energy and waste heat and cold by an indicative 2.2 percentage points annually over 2021–2030. These provisions concern both new low-temperature networks and existing high-temperature systems.

The recast Energy Efficiency Directive further strengthens this direction [4]. It sets the EU target of reducing energy consumption by at least 11.7% in 2030 relative to the 2020 Reference Scenario and requires comprehensive heating and cooling assessments. It also requires local heating and cooling plans at least for municipalities with more than 45,000 inhabitants. These plans must identify the potential for energy-efficiency improvements, low-temperature district-heating readiness, high-efficiency cogeneration, waste-heat recovery, and renewable energy use. Consequently, district-heating and combined heat-and-power operators must consider system-level measures involving lower network temperatures, renewable heat sources, heat pumps, storage, and flexible operation.

In parallel, the EU Emissions Trading System creates a direct economic incentive to reduce fossil-fuel-based heat generation. Combustion installations with a total rated thermal input exceeding 20 MW are generally within the scope of the EU ETS, and the revised cap is intended to reduce emissions in the covered sectors by 62% by 2030 compared with 2005 levels. For heat plants and combined heat-and-power units, there is a link between decarbonization, operating costs, and investment risk. In this context, hybrid systems combining photovoltaic–thermal collectors, industrial heat pumps, thermal energy storage, and electrical energy storage may provide a technical pathway for increasing the renewable share in district-heating supply, especially during periods of reduced heat demand.

The concept of combining heat pumps with hybrid photovoltaic–thermal collectors (PVT-HP) is well recognized and extensively investigated in the literature. The reviews by Ma et al. [5] and Namdar [6] present numerous studies that highlight the benefits of such solutions which, depending on the system configuration, include a reduction in ground borehole length by 30–35% and an improvement in heat-pump COP by 20–35% thanks to the use of PVT as a stable low-temperature heat source. However, the discussed works focus on applications at the scale of individual buildings or local microgrids. Available field evidence supports the statement that PVT-HP coupling is technically mature mainly at building scale. The one-year operation of a direct-expansion PVT-assisted heat pump confirms that system performance depends not only on irradiance [7] but also on collector temperature control and matching between thermal and electrical outputs. In the context of district heating (DH), the review by Kang et al. [8] is the first to indicate the potential for integrating PVT with heat pumps and seasonal thermal energy storages (PTES—pit thermal energy storage/ATES—aquifer thermal energy storage), while at the same time emphasizing the lack of large-scale operational systems. Real-world examples in district heating—such as the systems in Cuenca and Ólvega (Spain) [9] and the district-heating plant in Lidzbark Warmiński (Poland) [10]—remain few and far, and PVT collector fields are on the order of 1000–2000 m², which, compared to large-scale solar collector fields (tens of thousands of m²), shows that this technology is still in its early stages of development in district heating.

Hybrid photovoltaic–thermal (PVT) collectors enable simultaneous electricity and heat generation from a single surface area, making them a potentially attractive solution for limited installation space and for hybrid systems that require different forms of energy. The Solar Heat Worldwide [11] report indicates that, globally, about 1.67 million m² of

PVT collectors (863.8 MW_{th} and 314.6 MW_{el}) have already been installed, mainly in building systems and small networks, with a growing share of installations coupled with heat pumps.

The IEA SHC Task 60 report “Status Quo of PVT Characterization” [12] shows that PVT applications depend strongly on operating temperature and PVT technology type. At temperatures above ambient (typically 40–70 °C), building applications dominate—domestic hot water and low-temperature space heating—in which PVT collectors replace or supplement solar thermal collectors. At the same time, the electrical part mainly supplies building loads. At temperatures close to ambient (approximately 0–30 °C), applications with heat pumps are increasingly important, where PVT collectors cooperate with or replace the low-temperature heat source, improving the evaporator inlet temperature and reducing the size of ground heat exchangers. For operation below dew-point temperature, which is primarily possible for WISC-type PVT (wind and infrared sensitive collector), there is an opportunity to harness additional heat gains from condensation and frosting. This makes the technology particularly interesting for the new generation of low-temperature networks, 4 and 5GDHC (fourth and fifth generation district heating and cooling), although so far mainly in research and demonstration projects. Without integration with heat pumps and energy storage, PVT has limited applicability in district heating because the operating temperatures of typical networks (60–130 °C) are too high for efficient operation of most commercially available PVT collectors. Their thermal yield per unit area is lower than that of solar thermal collectors, leading to larger required areas and higher investment costs. For this reason, PVT-based DH systems are almost always designed as hybrid solutions, in which PVT supplies heat at a lower temperature level. In contrast, heat pumps, seasonal storages and peak sources are responsible for matching the load profile and the network temperature requirements.

To model such hybrid systems, both detailed component models of PVT collectors (e.g., in TRNSYS, Modelica, COMSOL) [13] and simplified representations in integrated models of 4G/5G networks are used. The work of Chen et al. [14] shows that integrating a PVT installation with a PTES storage and a heat pump in a TRNSYS model enables the analysis of not only solar fraction and storage efficiency but also flow and temperature control strategies throughout the district-heating network. Similarly, the study by Madhusudhanan (North Macedonia) [15] uses a PVT model implemented in Polysun [16] to evaluate different configurations of PVT cooperating with a ground-source heat pump and thermal storage for an urban network, which enables a direct comparison of specific heat costs and CO₂ emissions with solutions based on solar thermal collectors. In 5GDHC systems, such as the Hilversum [17] project, PVT collectors are modeled as multi-energy sources coupled with ATEs and decentralized heat pumps at substations, which requires a simultaneous description of heat, cooling, and electricity balances. These studies indicate that PVT-HP concepts for district heating require system-level assessment. This is consistent with district-heating analyses showing that the effect of large heat pumps depends on network temperatures, hydraulic constraints, and the temporal distribution of heat demand [18].

Given the small number of studies addressing district-heating networks supplied by a PVT collector–heat pump system, it was decided to analyze the use of a system consisting of a large-area PVT field operating as the lower-temperature source for a high-capacity industrial heat pump intended to cover the energy demand of a medium-scale district-heating network. Within the proposed concept, two storage facilities were also planned for heat and electricity storage. In principle, such a solution should improve energy efficiency and extend the system’s operating period into nighttime hours. Considering that the PVT collectors may operate below dew-point temperature in the analyzed configuration and thus benefit from additional heat gains from condensation, unglazed, non-insulated collectors were selected. This approach requires including condensation in the PVT collector

model, which is not offered by commercial software. For this reason, an extended model with such functionality was implemented in the LabVIEW version 21.0 software (National Instruments, Austin, TX, USA) using the Control Design and Simulation Module. Although LabVIEW is not a typical primary environment for thermal system dynamics simulations, it was deliberately selected. The rationale was to develop a dynamic model that could be used not only for offline energy-system analysis but also in the supervisory control system of the planned demonstration plant. This enables subsequent use of the model for model-based control, setpoint refinement, and operational scheduling of the PVT field, heat pump, and storage units.

2. Materials and Methods

The analyzed configuration of the hybrid PVT-HP system is shown in Figure 1. The system consists of a large-area unglazed PVT collector field, a source-side buffer tank used as TES, an industrial heat pump, an electrical energy storage unit, and a hydraulic connection to the district-heating network. A 30% aqueous propylene glycol solution was used as the heat-transfer fluid on the low-temperature source side, including the PVT field, buffer tank, and heat-pump evaporator loop. Water was used on the condenser and district-heating side. The use of brine on the source side and the adopted minimum PVT-field inlet temperature of 4 °C were introduced to reduce the risk of freezing during low-temperature operation. At the same time, the PVT collectors were allowed to operate below the dew-point temperature, which made it possible to include condensation-related heat gains in the collector model. The TES shown in Figure 1 should be interpreted as a source-side buffer tank integrated into the PVT–evaporator brine loop, not as a seasonal or independently operated heat-storage unit. Therefore, no separate TES charging or discharging mode was assumed in the simulations. The buffer tank provides limited thermal inertia of the source-side loop and may support source-side mixing. The three-way control valve and bypass line were retained to keep the brine temperature at the heat-pump evaporator inlet within the compressor’s admissible operating range. In the analyzed case, this valve function was not activated because the heat pump remained within this range. The EES supplies the electrical demand of the PVT-HP subsystem, including the heat-pump compressor and circulation pumps. The heating plant shown in Figure 1 represents the existing conventional peak/backup source connected to the district-heating network. It covers the residual heat demand when the PVT-HP output is insufficient or unavailable.

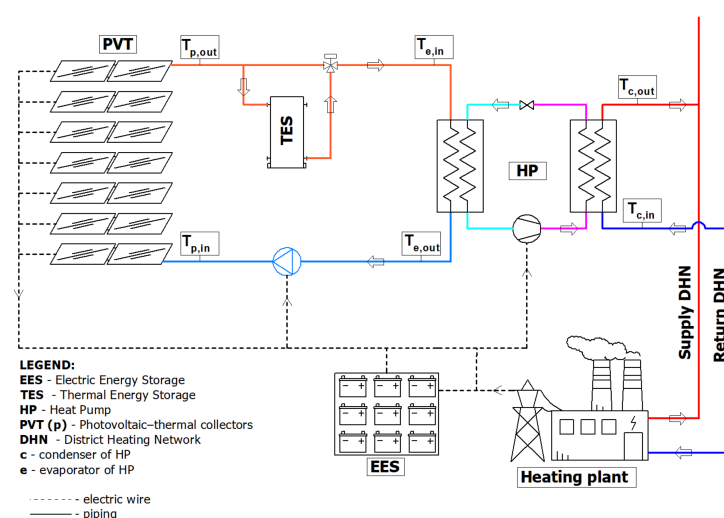


Figure 1. Schematic layout of the hybrid PVT-HP system with source-side TES, EES, and the existing conventional peak/backup source coupled to a small district-heating network.

The analysis of the proposed concept was based on full-year 2025 input data with an hourly temporal resolution. The operational data obtained from the district-heating operator included the network supply and return temperatures, heat demand, ambient temperature, and wind speed. Relative humidity, precipitation, and horizontal solar irradiance were generated using the CFSR 2.0 dataset implemented in energyPRO version 5.0 software (EMD International A/S, Aalborg, Denmark) [19]. All input series were used with a common hourly time index corresponding to the operator dataset. No missing or abnormal values were identified in the operator data; therefore, no filtering procedure was applied to these records. Detailed information on the measurement uncertainty of the operator data was not available. However, since the ambient temperature was used by the operator for heat-source control, it was assumed to have been verified within the operator's monitoring system. For numerical integration, the hourly input variables were represented by spline functions. In a few cases, spline interpolation produced slightly negative non-physical values, mainly for solar irradiance and precipitation. These values were clipped to zero before being used in the model. The resulting annual time series of the main parameters used in the calculations are presented below. The district-heating heat demand and ambient temperature are shown in Figure 2, the network supply and return temperatures in are shown in Figure 3, and the frequency distribution of heat demand is shown in Figure 4. The remaining weather boundary conditions used in the PVT model are presented in Figures 5–8.

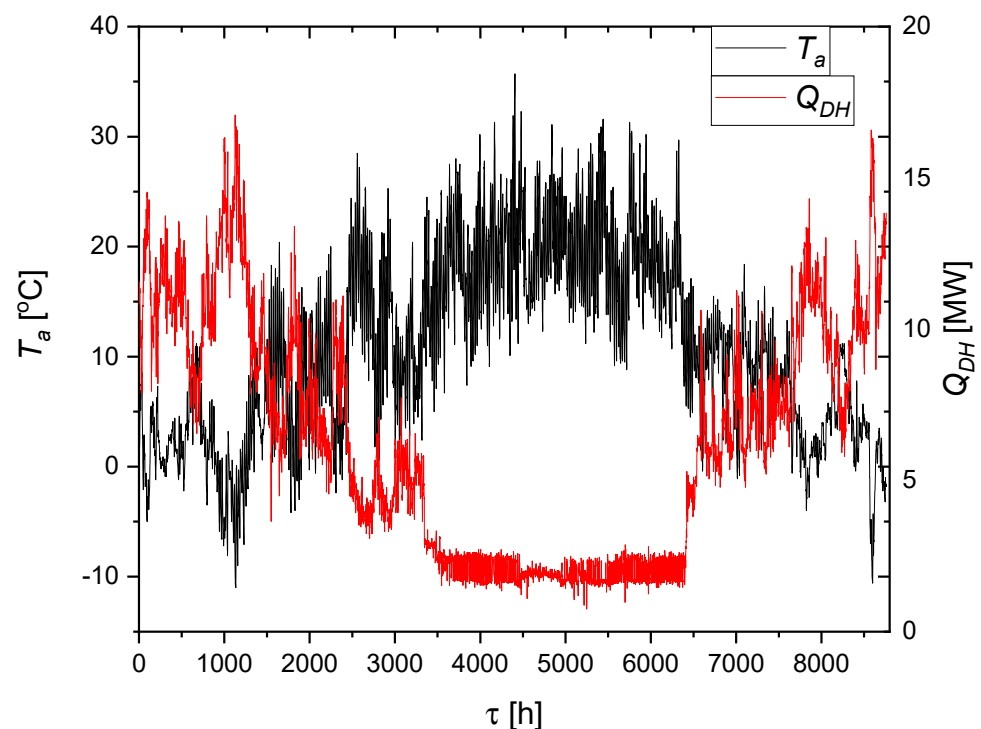


Figure 2. Annual profiles of district-heating heat demand and ambient-air temperature (own elaboration based on data from district heating, 2025).

Based on the annual load profile, the analyzed network can be classified as a small local district-heating system with pronounced seasonal variability. The annual mean load is 6.28 MW, the peak load reaches 17.10 MW, and the annual heat delivered is 54.81 GWh (197.3 TJ). This corresponds to 3227 peak-load hours per year and a load factor of 0.367. A non-negligible load is also maintained outside the main heating season. The average load in May–September is 2.53 MW, while in the transitional months April and October, it reaches 6.32 MW. This indicates a persistent contribution of domestic hot water preparation.

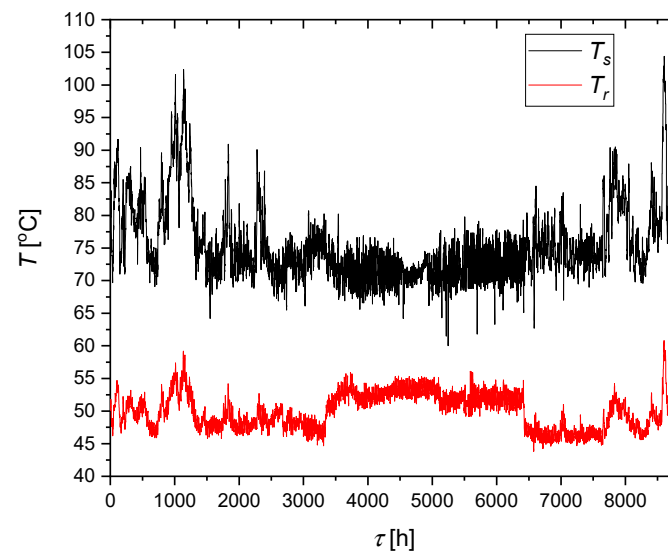


Figure 3. Annual characteristic of the district-heating system supply (T_s) and return (T_r) temperatures (own elaboration based on data from district heating, 2025).

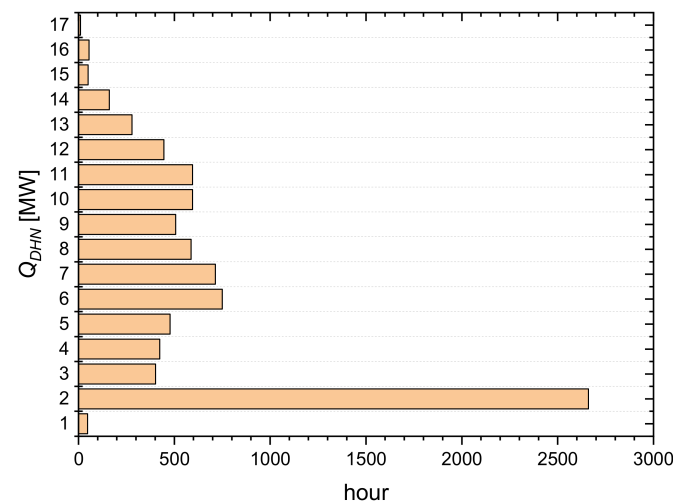


Figure 4. Frequency distribution of hourly thermal power demand in the district-heating network (own elaboration based on data from district heating, 2025).

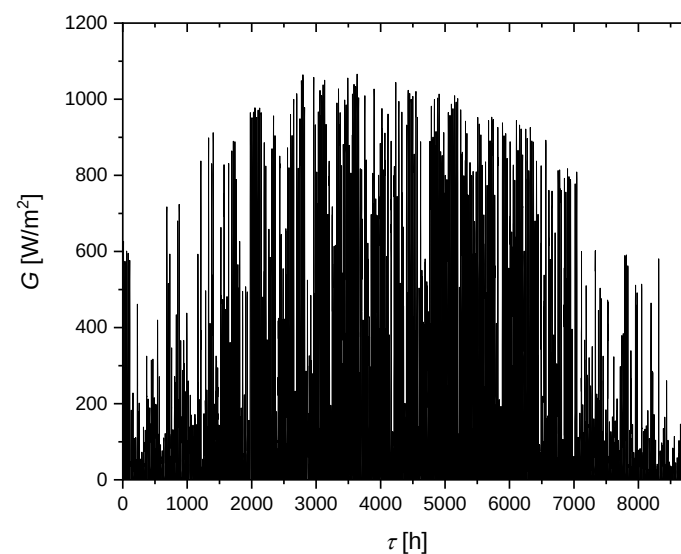


Figure 5. Annual profile of horizontal irradiance and irradiance at the plane of system inclined at 30° .

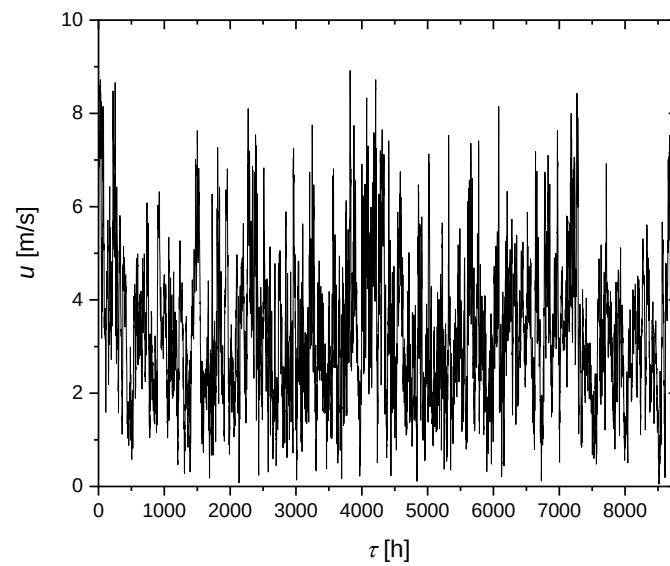


Figure 6. Annual profile of wind speed.

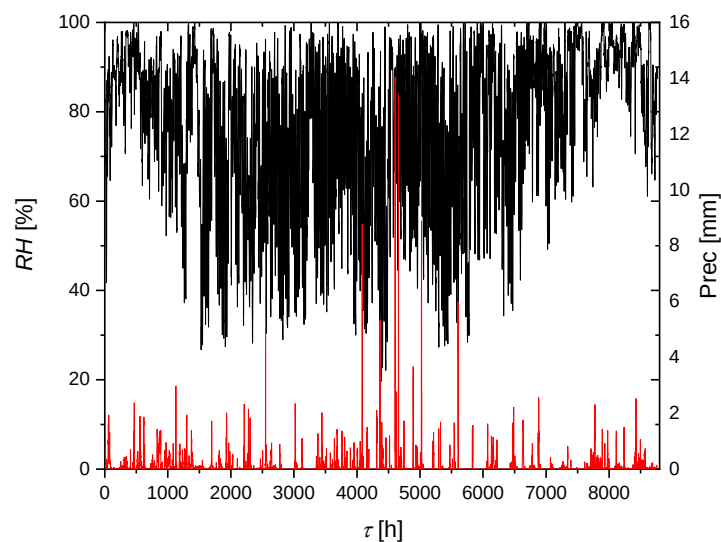


Figure 7. Annual profiles of relative humidity (black line) and precipitation (red line) used in the PVT model.

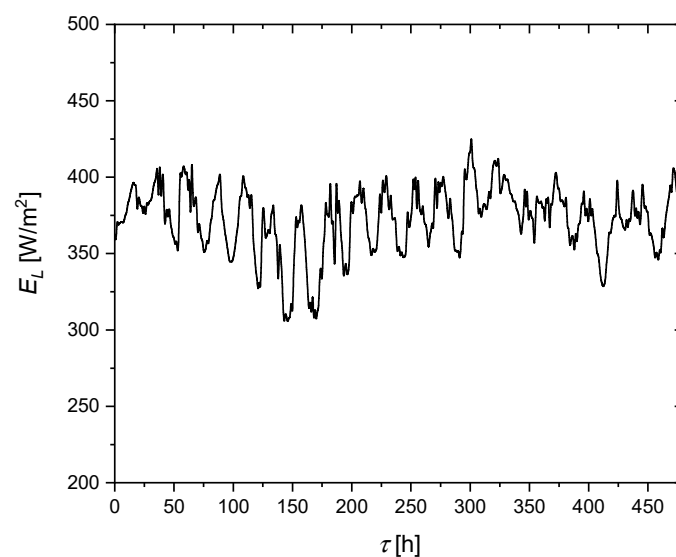


Figure 8. Example downwelling long-wave radiation calculated for days 160–180 of 2025.

The analyzed network operates with return temperatures mostly within 45–55 °C and supply temperatures mostly within 70–85 °C, increasing to approximately 100–105 °C during winter. According to the IEA DHC temperature-based classification, supply temperatures below 70 °C correspond to fourth-generation district heating, whereas 70–100 °C is typical of third-generation systems [20]. Recent analyses of implemented low-temperature networks also show that low-temperature operation encompasses various technical configurations and should not be reduced to a single fixed temperature pair [21]. The analyzed network should therefore be classified primarily as a conventional third-generation district-heating system, with short winter periods near high-temperature operation. Such levels are consistent with Polish systems serving older building stock, for which design supply temperatures of approximately 90–95 °C have been reported [21]. Its annual heat supply, calculated from hourly mean source power, is 54.81 GWh (197.3 TJ), which is approximately 0.10% of Polish heat sales through district-heating networks in 2024 and about 36% of the mean sales per reporting enterprise [22]. With a peak hourly load of 17.08 MW, the system falls within the 5–20 MW URE class and can be regarded as a small local network or the lower end of a medium-scale district-heating system [22].

The hourly operating profile of the district-heating network is characterized in Figure 4 using the frequency distribution of thermal power demand. The data were grouped into 1 MW power classes centered on integer values of Q_{DHN} ; therefore, each bar represents the number of hours for which the hourly mean thermal power falls within the corresponding ± 0.5 MW interval. This representation enables identification of the dominant operating range of the DHN and distinguishes between frequently occurring part-load conditions and short-duration peak-load operation. The distribution shows that the largest number of operating hours occurs in the 1.5–2.5 MW power class, assigned to the 2 MW bin. This range can therefore be interpreted as the dominant low-load operating regime of the analyzed district-heating network. For this reason, the system considered in the present study was sized to cover this most frequent thermal demand range.

Reliable simulation of PVT collector performance requires more than the ambient-air temperature profile. It also requires parameters that affect the amount of solar energy absorbed by the collector field. These parameters include the orientation and tilt angle of the PVT array, shading effects, the incident-angle modifier (IAM), and optical losses caused by surface soiling. Therefore, the solar irradiance on the horizontal surface was converted into the effective irradiance in the plane of the collector field using the PVsyst version 8.0.20 software (PVsyst SA, Satigny, Switzerland) [23]. In the analyzed case, the collector surface was oriented south and inclined at 30°. The inter-row distance was set to 10 m, measured from the front of one row to the front of the next row, and was used to determine self-shading losses. Incident-angle modifier losses were calculated using the default PVsyst profile, and the average annual soiling loss was assumed to be 0.7%. The resulting irradiance profile shown in Figure 5 was used as an input to the thermal and electrical PVT models.

The weather boundary conditions used in the case study indicate moderate Central European climatic conditions with pronounced seasonal variability. The annual mean ambient temperature was 10.75 °C, with hourly values ranging from −11.0 °C to 35.7 °C. The dataset included 721 h below 0 °C and 6557 h at or above 4 °C, which was adopted as the minimum PVT–HP operating threshold. The annual irradiation sum calculated from the irradiance series used in the model was approximately 1307 kWh/m² in the plane of array, with a maximum hourly irradiance of 1066 W/m². The mean wind speed and relative humidity were 3.3 m/s and 76.3%, respectively, while the annual precipitation sum was 1048 mm. These conditions directly affect PVT output, condensation potential, heat-pump source temperature, and achievable heat-demand coverage.

The precipitation profile shown in Figure 7 includes rain and snow. Since the PVT-HP system is intended mainly for summer and shoulder-season operation, only the rainfall effect was included in the PVT model, while snowfall was neglected. The rainfall mass flow rate was calculated from the horizontal projection of the PVT collector area, the hourly precipitation amount, and the 3600 s time step. Since measured long-wave irradiance and cloud-cover data were not available, the downwelling long-wave radiation shown in Figure 8 was estimated using [24,25].

3. Computational Model

The thermal dynamics of the system consisting of the PVT, heat pump, thermal energy storage (TES), electric energy storage (EES), and control valves were analyzed using a 0D lumped-parameter mathematical model based on heat balance equations. The solution of the model equations can be used to validate the selection of the system configuration. It also allows for assessing and tracking temperature levels that are likely to appear during the system operation. Furthermore, it can be used as a tool in the design of the system supervisory control loop.

In the model, the heat capacities of the liquid loop and all the aforementioned system elements were accounted for. The equations of the model are based on a balance between the rate of heat accumulation in the material of mass M and specific heat c and the difference between the supplied heat flow rate Q_{in} and the dissipated heat flow rate Q_{out} . This balance can be written in the following general form [26]:

$$\dot{Q} = Mc \frac{dT}{dt} = \sum Q_{in}(\tau) - \sum Q_{out}(\tau), \quad (1)$$

where τ denotes time.

3.1. PVT Modeling

The developed model, in the section describing a single PVT collector, accounts for the accumulation of thermal energy in both the working fluid and the collector structure. These assumptions, arising from the intent to simplify the model rationally, do not significantly affect the computation results. Due to its relatively low mass compared to the heat exchange area, the PVT collector exhibits negligible thermal inertia relative to other system components. Consequently, based on balance Equation (1), the following equation can be formulated:

$$M_p c_p \frac{dT_p}{d\tau} = \dot{m}_{b,w} c_b (T_{p,out} - T_{p,in}) + q_p, \quad (2)$$

where

M_p —equivalent mass of the PVT collector [kg];

c_p —specific heat capacity of the PVT collector [kJ/(kgK)];

c_b —specific heat capacity of the working fluid (aqueous solution of 30% propylene glycol) [kJ/(kgK)];

q_p —instantaneous thermal power of the collector [W].

The computational parameter T_p is defined as the mean PVT collector temperature and is taken as the mean working-fluid temperature:

$$T_p = (T_{p,out} + T_{p,in}) / 2. \quad (3)$$

Thermal Power Modeling

At the level of a single PVT collector, the “Status Quo of PVT Characterization” [12] report presents several models for determining the instantaneous thermal power of the

PVT collector. According to the authors, from a practical point of view, models based on standard test data are most applicable, such as the quasi-dynamic test (QDT) model specified in ISO 9806 [27]. This model describes the instantaneous thermal power as a function of solar irradiance (direct and diffuse components), working-fluid temperature, ambient temperature, wind speed, long-wave radiation flux, and temperature change rate, with model parameters ($\eta_{0,b}$, a_1 , $\dots$, a_8) determined experimentally. The model can be extended with additional terms. Particularly relevant for PVT operation below the dew point is the term involving the coefficient a_{10} , which accounts for extra thermal gains from water vapor condensation on the collector surface. It also enables the consideration of losses due to condensate evaporation [12]. However, in the simulation, it was assumed that water drained efficiently from the collector surfaces, so this effect was neglected [28]. Frosting issues were also omitted due to the assumed operating periods of the system. Finally, the heat flow rate from a PVT collector can be described as follows:

$$Q_p = Q_{sol} - Q_{loss} + Q_{lw} + Q_{cond} + Q_{rain} \quad (4)$$

where the term Q_{sol} represents short-wave heat exchange:

$$Q_{sol} = A_G \eta_0 G \quad (5)$$

Q_{loss} represents thermal losses related to conductive and convective heat exchange:

$$Q_{loss} = A_G [a_1(T_p - T_a) + a_2(T_p - T_a)|T_p - T_a| + a_3u(T_p - T_a) + a_5\left(\frac{dT_p}{d\tau}\right) + a_6uG] \quad (6)$$

Q_{lw} represents long-wave heat exchange of the front and rear side of a PVT:

$$Q_{lw} = A_G [a_4(E_L - \sigma(T_p + 273.15))^4 + a_9(\sigma((T_{bg} + 273.15)^4 - (T_p + 273.15)^4))] \quad (7)$$

Q_{cond} represents heat exchange due to condensation process:

$$Q_{cond} = A_G a_{10}(2.8 + 3.0u)(\mu_{sat}(T_a) - \mu_{sat}(T_p)) \quad (8)$$

Q_{rain} represents heat exchange with rainwater:

$$Q_{rain} = a_{11}\dot{m}_{wat}c_{wat}(T_{wat} - T_p) \quad (9)$$

where

A_G —gross collector area according to ISO 9488 [m²];

a_1 —heat loss coefficient [W/(m²K)];

a_2 —temperature dependence of coefficient a_1 [W/(m²K²)];

a_3 —wind dependence of coefficient a_1 [J/(m³K)];

a_4 —sky temperature dependence of coefficient a_1 [-];

a_5 —heat capacity [J/(m²K)];

a_6 —wind dependence of optical efficiency [s/m];

a_9 —rear collector radiation loss dependence [-];

a_{10} —condensation and evaporation dependence [J/kg];

a_{11} —precipitation-related heat gains/losses dependence [-];

E_L —long-wave radiation (>3 μm) [W/m²];

G —total irradiance in the collector plane [W/m²];

u —wind speed [m/s];

η_0 —optical efficiency [-];

T_a —ambient temperature [$^{\circ}\text{C}$];
 T_p —mean PVT collector temperature, taken as the mean working-fluid temperature [$^{\circ}\text{C}$];
 σ —Stefan–Boltzmann constant [$\text{W}/(\text{m}^2\text{K}^4)$];
 T_{bg} —temperature of the collector background [$^{\circ}\text{C}$];
 $\mu_{sat}(T)$ —saturation absolute humidity at temperature T [kg/m^3];
 $\dot{m}_{wat}$ —rainfall mass flow rate [kg/s];
 c_{wat} —specific heat capacity of rainwater [J/kgK];
 T_{wat} —rainwater temperature [$^{\circ}\text{C}$].

The resulting expression corresponds to the ISO 9806 quasi-dynamic formulation, extended by the terms with coefficients a_9 , a_{10} , a_{11} , and without the thermal-inertia term described by coefficient a_5 . The Q_{cond} term is physically applicable only when the collector temperature is below the dew-point temperature. In the numerical implementation, Q_{cond} was set to zero when the collector temperature was equal to or higher than the dew-point temperature.

Coefficient Determination

The determination of most coefficients for the above model follows the standard procedures described in the ISO 9806. The value of the coefficient a_{10} can be derived from the equation presented in [29,30]. Its theoretical maximum depends on air parameters and typically ranges from 1700 to 1900 J/kg [31]. Experimental values vary between 1211 and 2000 (for uninsulated solar collectors). The study [32] highlighted the dependence of the coefficient on absorber area and uniformity of water coverage. Notably, the term involving this coefficient is physically meaningful only if the collector temperature drops below the dew-point temperature. Hence, in simulations, the dew-point temperature was calculated based on ambient temperature and relative humidity using CoolProp version 8.0.0 software (Ian Bell, University of Liège, Liège, Belgium) [33].

Another approach is presented in the work of Stegmann et al. [34]. The WISC-type PVT model, including condensation gains, was based on the steady-state collector model from EN 12975-2 and supplemented with an empirical formula for condensation effects. Validation against annual measurements on a pilot installation showed a 1.7% discrepancy between simulation results and measurements.

Coefficients a_1 , a_3 , and a_6 for steady-state (SST) PVT modeling according to ISO 9806:2017 were determined experimentally under conditions including condensation, snow, frost, and rain in [35]. This approach allows for the inclusion of the influence of these phenomena in coefficient values but does not isolate contributions from condensation or rain alone in the overall heat balance. Hence, it is useful mostly for comparative purposes across PVT types. Otherwise, it would require adjusting coefficient values in simulations based on ambient conditions.

Regarding coefficient a_{11} , its experimental determination is challenging because rainwater temperature is difficult to measure. In [28], this temperature was taken as equal to the wet-bulb temperature and coefficient a_{11} was set to 0.5 based on a short-term experiment.

It is noteworthy that during simulations, the term associated with the coefficient a_5 , corresponding to the PVT collector's thermal inertia, was omitted from calculations, as the heat capacity effects of the collector and the working fluid were already incorporated in Equation (2) via coefficients M_p and C_p .

Electrical Power Modeling

The performance of the electrical part of a PVT collector can be modeled using the following formula, assuming the electrical part temperature equals the overall collector temperature, i.e., the mean working-fluid temperature:

$$P_p = P_{STC} \left(\frac{G}{G_{STC}} \right) (1 + \gamma(T_p - 25)), \quad (10)$$

where

P_p —electrical power [W];

P_{STC} —electrical power at the STC conditions [W];

G —total irradiance in the collector plane [W/m^2];

γ —temperature coefficient of power [$1/\text{K}$];

T_p —mean module temperature [$^{\circ}\text{C}$].

3.2. Heat-Pump Model

Industrial heat pumps coupled with district-heating systems are usually represented by several types of models. These include thermodynamic cycle models, reduced-order COP correlations, performance map models, and simplified dispatch representations used in energy system optimization. In thermodynamic cycle models, the heat pump is modeled using the standard vapor-compression cycle. The evaporation and condensation pressures are determined from the corresponding saturation temperatures. Compressor work is calculated from enthalpy differences and assumed isentropic efficiency [36]. In reduced-order models, the COP is often estimated using a constant value, the Carnot efficiency, the Lorenz efficiency, the exergy efficiency, or linear off-design correlations [37]. These methods reduce the computational cost of annual simulations. Their accuracy decreases when the source and sink temperatures differ from the design conditions.

Simplified dispatch representations have a different purpose. They do not describe the heat pump as a refrigerant cycle. Instead, the heat pump is treated as a power-to-heat technology with capacity limits, operating-cost constraints, and a fixed or temperature-dependent COP [38]. This approach is useful for dispatch optimization, investment planning, and market-operation studies. It is less suitable for analyzing the thermal response of a specific heat-pump unit. More detailed system-level models use heat-pump performance maps. In such models, heating capacity, COP, and part-load behavior are expressed as functions of the evaporation temperature, the district-heating outlet temperature, and the load ratio [39]. These maps can be generated from product-selection software or from detailed component models. Another approach is to generate operating points from a detailed physical model and use them to identify a black-box model for techno-economic simulations [40].

In the present study, a different modeling approach was required. The lower-temperature source is a large PVT field. The heat pump must operate under continuously variable source and sink temperatures. Direct use of commercial industrial heat-pump data was not possible. Manufacturers usually do not provide performance characteristics with the resolution required for the model developed in this study. Available data are often limited to nominal points, selected operating conditions, or general product ranges. They are insufficient for direct implementation in a dynamic system-level simulation.

For this reason, the heat-pump model was parameterized using the performance characteristics of a BITZER 6FE-50Y reciprocating compressor operating with R1234yf. An additional justification for selecting BITZER compressor data as a reference is the documented use of BITZER 8-cylinder reciprocating compressors in Fenagy large-scale air-to-water heat pumps for district-heating applications [41]. The compressor was assumed to be speed-controlled by a frequency converter. Heating capacity Q_e , Q_c and electric power $P_{el,HP}$ were calculated from manufacturer selection-polynomial data as functions of the evaporation temperature T_e and the condensation temperature T_c [42]. The complete heat-pump plant was represented as a modular unit composed of 50 identical compressors.

The evaporation and condensation temperatures were linked to the corresponding HTF temperatures by fixed temperature differences. The evaporator outlet temperature on the HTF side was defined relative to the refrigerant evaporation temperature. The

condenser outlet temperature on the HTF side was defined relative to the refrigerant condensation temperature. This engineering assumption enabled the determination of the heat pump's performance over the full operating range used in the system model. In the authors' opinion, this representation is closer to real operation than simplified models based on a constant COP or a fixed efficiency ratio. It also retains low computational complexity.

The refrigerant-side description was limited to T_e and T_c . The thermal inertia of the heat pump on the refrigerant side was neglected. This simplification was accepted because the article aimed to verify the complete hybrid system model and analyze the interactions among the PVT field, heat pump, TES, EES, and district-heating demand. A detailed representation of a specific industrial heat pump was therefore less critical at this stage. The dominant thermal inertia are associated with the PVT field, storage units, and HTF loops. The heat capacities of the evaporator and condenser were retained only on the HTF side. This improves numerical stability and damps fast temperature transients. In further work, this aspect should be refined for a specific implementation case. The generic heat-pump representation should then be replaced with a model based on performance data for a selected industrial unit.

For the cooling water in the condenser, the average temperature change is related to the inlet and outlet temperatures and convective heat transfer.

$$M_{c,w}c_w \frac{dT_{cw}}{d\tau} = \dot{m}_w c_w (T_{c,out} - T_{c,in}) + Q_c(T_e, T_c), \quad (11)$$

where Q_c —condenser capacity [W].

For the HTF in the evaporator, the average temperature change is related to the inlet and outlet temperatures and convective heat transfer.

$$M_{e,b}c_b \frac{dT_{eb}}{d\tau} = \dot{m}_b c_b (T_{e,in} - T_{e,out}) + Q_e(T_e, T_c), \quad (12)$$

where Q_e —evaporator capacity [W].

3.3. Thermal Energy Storage Model

Thermal energy storage is one of the most effective methods for increasing the operational flexibility of district-heating systems. Previous studies have shown that TES can be used in operational optimization to shift heat production in time and to improve the dispatch of heat sources in district-heating networks [43]. This is particularly relevant when the heat source is variable or when its operation must be coordinated with technical and economic constraints.

In PVT-HP systems, this issue is especially important because the low-temperature heat source is weather-dependent. The thermal output of the PVT field and the heat pump's operating conditions depend on solar irradiance, ambient temperature, wind speed, and collector operating temperature. Short-term variations in solar irradiance may therefore cause rapid changes in the temperature level available at the heat-pump evaporator. These changes affect the evaporation temperature and, consequently, the COP and instantaneous heating capacity of the heat pump.

For this reason, TES may also be treated as a stabilizing component. Its thermal inertia can reduce rapid variations in the heat-transfer-fluid temperature and attenuate transient disturbances on the source side of the heat pump. In this role, the storage is used mainly to smooth fluctuations caused by weather variability and to improve the dynamic stability of the PVT-HP system.

At the conceptual stage of the present study, two possible TES locations were considered. The first option was heat storage on the sink side of the heat pump, i.e., between

the heat pump and the district-heating network. The second option was storage on the low-temperature source side, connected with the PVT field and the evaporator loop. In both cases, TES could increase the flexibility of the PVT-HP system. However, installing an additional storage unit usually involves significant capital expenditure. This is especially important for large-capacity storage required in district-heating applications.

The sink-side TES was not implemented in the model. Such a storage unit would have to be charged at temperatures higher than the district-heating supply temperature delivered by the heat pump. This would increase the refrigerant condensation temperature. As a result, the heat-pump COP would decrease. The energy performance of the whole PVT-HP system would also be reduced. For this reason, this storage option was excluded at the present stage of the analysis.

It was also assumed that the district-heating network itself may provide some buffering capacity. Transient modeling studies of district-heating networks indicate that heat can be stored in pipelines and used to optimize operations by controlling the network supply temperature [44]. This supports the assumption that the network's thermal inertia can partially attenuate short-term fluctuations in heat-source operation. In the present case, variations in the PVT-HP system's thermal output may affect network temperature. However, these changes were assumed to remain limited under the adopted control strategy and operating conditions.

On the low-temperature source side, a large TES unit was also considered. Such a unit could reduce the short-term mismatch between heat collected by the PVT field and heat extracted by the heat-pump evaporator. However, this solution would require not only the construction of the storage vessel but also a large inventory of heat-transfer fluid. Since the source-side loop may also operate under shoulder-season and winter conditions, this inventory would have to consist of a brine, for example, an aqueous glycol solution. This would increase the investment cost and could also increase pumping power due to higher hydraulic losses.

Therefore, only a small buffer tank was included in the final model. Its volume was set to 10 m³. The tank was introduced to represent limited thermal inertia on the low-temperature source side, rather than to provide long-term heat storage. Stabilization of system operation was assumed to be achieved mainly by controlling the heat-pump capacity and the PVT-field inlet temperature.

The buffer tank was described using a lumped-parameter heat balance. This formulation is consistent with the modeling approach used for the other thermal components of the system. It is also consistent with previous TES modeling work, where the heat capacity of system components and heat losses to the ambient environment were included in a 0D dynamic balance [45].

Following the general energy-balance approach, the balance equation for the buffer tank was derived in the following form:

$$M_{T,b}c_b \frac{dT_{t,b}}{d\tau} = \dot{m}_b c_b (T_{t,in} - T_{t,out}) + k_t A_t (T_a - T_{t,b}), \quad (13)$$

where, in particular, $M_{T,b}$ is the effective mass of the tank and HTF [kg], k_t is the overall heat-transfer coefficient [W/m²K], and A_t is the external surface area of the tank [m²].

3.4. Electrical Energy Storage Model

In system-level simulations, electrical energy storage is usually represented by an energy-reservoir balance rather than by a detailed electrochemical cell model. In this approach, the storage state is updated based on the instantaneous charging and discharging powers. At the same time, the dominant operational constraints are imposed by conversion

efficiency, available energy capacity, and admissible power limits. Such a formulation is commonly used in multi-period energy-system models, where the main purpose of storage is to shift locally generated electricity over time and reduce grid imports. In contrast, cell voltage dynamics, internal resistance, and thermal behavior are outside the scope of the analysis [46].

In the present model, the EES was introduced to buffer the mismatch between the electricity generated by the PVT field and the electrical demand of the PVT-HP system. The electrical demand included the heat pump and circulation pumps. Electricity generated by the PVT collectors was used directly by these consumers in the first instance. If the instantaneous PVT electricity generation exceeded the system demand, the surplus was charged to the EES. If instantaneous generation was lower than demand, the deficit was first covered by discharging the EES and, if the stored energy was insufficient, by importing electricity from the grid. Therefore, the EES was not treated as a source enabling full electrical autonomy of the system, but rather as an element that reduced grid electricity imports and increased the use of locally generated PVT electricity. This interpretation is consistent with system-level analyses of PVT-HP configurations with TES and EES. Electrical storage increases the local use of PVT electricity, but the degree of electrical autonomy must be determined from the annual energy balance [47].

The following discrete energy balance describes the storage state:

$$E_{EES}(\tau + \Delta\tau) = E_{EES}(\tau) + \eta_{RT}P_{ch}(\tau)\Delta\tau - P_{dis}(\tau)\Delta\tau, \quad (14)$$

where E_{EES} is the electrical energy stored in the EES [Wh], P_{ch} is the charging power supplied to the storage [W], P_{dis} is the electrical power delivered from the storage to the system [W], $\Delta\tau$ is the simulation time step [h], and η_{RT} is the aggregate round-trip efficiency [-].

To simplify the simulation procedure, a single value representing the round-trip efficiency was adopted and set to $\eta_{RT} = 0.90$; thus, the charging and discharging efficiencies were not parameterized separately. In this simplified balance, storage-cycle losses were assigned to the aggregate charging–discharging process rather than divided between the two conversion stages.

The EES lifetime was assumed to be 8000 equivalent full cycles, and the end-of-life condition was defined as SoH (State of Health) equal 80%. Calendar aging was neglected. The cyclic degradation of the storage was represented by a linear reduction in usable capacity per equivalent full cycle, calculated from the difference between the initial SoH and the assumed end-of-life value. This gives $\Delta\text{SoH}_{\text{cyc}} = (100\% - 80\%)/8000 = 0.0025$ percentage points per cycle. This simplified formulation provides a first-order representation of EES degradation suitable for system-level simulations, while intentionally omitting the effects of depth of discharge, average state of charge, C-rate, and cell temperature.

No maximum EES energy capacity or maximum charging/discharging power was imposed a priori. The EES was therefore treated as a sizing variable rather than as a component with predefined rated parameters. The required energy capacity can be determined ex post from the maximum stored-energy excursion obtained in the simulation. In contrast, the required power rating can be inferred from the maximum charging or discharging power observed over the analyzed period. Consequently, the model results provide preliminary values of $E_{EES,max}$, and $P_{EES,max}$, which can be used in subsequent techno-economic sizing of the PVT-HP-EES system.

3.5. Circulation-Pump Model

Circulation pumps were included in the model solely to estimate auxiliary electricity consumption for the hydraulic loops. The hydraulic power required by a circulation pump

was calculated as: $P_h = \Delta p \cdot \dot{V}$, where P_h was the hydraulic power [W], Δp was the pressure drop in the hydraulic circuit [Pa], and $\dot{V}$ was the volume flow rate [m^3/s]. The corresponding electrical power was then estimated from $P_e = P_h / \eta_p$, where P_e was the electrical power consumption of the pump [W] and η_p was the pump efficiency [-]. The efficiency was calculated using the empirical relation, $\eta = 0.0721 \cdot P_h^{0.3183}$. This formulation serves as a simplified representation of circulation-pump performance and follows the approach used in standardized heat-pump performance calculations, including the ISO 14511-3 [48].

For the lower-temperature source loop, the pressure drop was estimated from the PVT-field hydraulic layout. The pressure drop of a single PVT collector at the nominal flow rate was assumed to be 1 kPa. The collectors were connected in parallel in groups of five units. In addition, 2 m of header pipe per collector was assumed for the supply and return lines. The maximum pressure drop in these pipes was set to 200 Pa/m. The resulting pressure loss was used to estimate the electrical power required for circulation through the PVT field and the heat pump's evaporator side.

The flow rate on the lower-temperature source side was kept constant in the simulation. It was determined by the assumed number of installed PVT collectors. Therefore, the electrical power consumption of the corresponding circulation pump was also treated as constant. The model did not account for changes in hydraulic resistance due to temperature-dependent variations in fluid properties. This simplification was considered acceptable because the pump model was introduced only to estimate auxiliary electricity consumption.

On the sink side of the heat pump, the water flow rate was adjusted to match the district-heating supply temperature requirement. Thus, the hydraulic power would formally vary with the instantaneous flow rate. However, the estimated pressure losses on this side were small relative to the heat-pump system's total electrical demand. For this reason, the variability of sink-side pump power was neglected. A constant total circulation-pump power of 60 kW was adopted for the simulations.

The electricity consumed by the circulation pumps was included in the system electrical balance only during periods of heat-pump operation. This assumption is consistent with the adopted supervisory strategy, in which the hydraulic loops are active only when the PVT-HP system supplies heat to the district-heating network. The pump's electricity consumption was therefore included with the compressor's electricity demand when calculating the total electrical demand of the PVT-HP system and its interactions with the PVT field, EES, and grid.

3.6. Numerical Implementation and Computational Assumptions

The equations presented in the preceding subsections form a closed set of ordinary differential and algebraic equations. The thermal part of the model was formulated using 0D lumped-parameter heat balances. Algebraic relations were used to describe the heat-pump performance, the electrical output of the PVT field, the state of the EES, and the auxiliary electricity demand of circulation pumps. The model includes the PVT collector field, heat pump, source-side buffer tank, EES, circulation pumps, and hydraulic coupling with the district-heating network.

The set of differential equations was transformed into normal form, which is suitable for numerical integration. The equations were solved in the LabVIEW Control Design and Simulation Module using the adaptive Runge–Kutta 23 method. The principal thermal state variables were the water temperature on the condenser side $T_{c,w}$, the brine temperature on the evaporator side $T_{e,b}$, the brine temperature in the PVT $T_{p,b}$, and the brine temperature in the buffer tank $T_{t,b}$. The district-heating demand, district-heating supply, return temperatures, ambient temperature, wind speed, relative humidity, precipitation, and irradiance in the collector plane were used as time-dependent boundary conditions.

The following computational assumptions were adopted:

- The system was represented as a 0D lumped-parameter model. Spatial distributions of temperature and other state variables in the heat-transfer-fluid loops, heat exchangers, and storage volumes were not explicitly represented. Each component was described by a representative bulk temperature or by inlet and outlet temperatures defined in the corresponding balance equation.
- Water was used as the heat-transfer fluid on the sink side of the heat pump. Its specific heat was set to 4.18 kJ/(kgK). The condenser-side mass flow rate was varied to achieve the required increase in water temperature and match the district-heating supply temperature.
- A 30% aqueous propylene glycol solution was used as brine on the low-temperature source side. Its specific heat was set to 3.80 kJ/(kgK). The source-side flow rate was kept constant and was determined from the number of installed PVT collectors and a unit flow rate of 40 L/(m²h).
- The brine temperature at the heat-pump evaporator was assumed to be equal to the source-side brine temperature supplied from the PVT and buffer loop. Additional temperature losses in the connecting pipework were not introduced.
- The constant parameters of the PVT thermal model are presented in Appendix A. The variable inputs of the PVT model included irradiance in the collector plane, ambient temperature, wind speed, relative humidity, precipitation, and estimated long-wave radiation. The mean collector temperature was taken to be the mean heat-transfer-fluid temperature. Condensation gains were considered only when the collector temperature was below the dew-point temperature. The wet-bulb temperature approximated the precipitation temperature.
- The heat-pump plant was represented by 50 identical BITZER 6FE-50Y compressors operating with R1234yf. Heating capacity and electric power were calculated from selection-polynomial data as functions of evaporation and condensation temperature. Refrigerant-side thermal inertia was neglected, while the HTF-side heat capacities of the evaporator and condenser were retained.
- Only a 10 m³ buffer tank on the low-temperature source side was included as TES. No sink-side TES was implemented. The district-heating network was treated as a component with inherent thermal buffering capacity.
- The EES was described by a discrete energy balance with an aggregate round-trip efficiency of 0.90. Its lifetime was set to 8000 equivalent full cycles, with end of life defined as SoH = 80%. Maximum energy capacity and maximum charge/discharge power were not imposed a priori.
- Circulation pumps were included only through their auxiliary electricity consumption. The source-side pump power was treated as constant. The variability of sink-side pump power was neglected, and a total circulation-pump power of 60 kW was adopted during heat-pump operation.
- The operating range of the PVT-HP system was limited by the adopted supervisory constraints. The minimum ambient temperature for operation was set to 4 °C. The PVT-field inlet temperature was constrained between 4 °C and 30 °C.
- In the implemented feedback strategy, the TES and EES states were not used as direct controller inputs. The TES was represented only as a source-side buffer tank and no separate charging or discharging rule was applied. The EES was operated according to the electrical balance described in Section 3.4. PVT electricity was used directly by the PVT-HP subsystem first. Surplus electricity was stored in the EES, while deficits were covered first from the EES and then from the grid.

4. Supervisory Control Strategy

As stated above, the proposed system must cover the entire summer heat demand of the district-heating network. A secondary objective is to maximize its utilization during the remainder of the year and thereby reduce the load on the conventional heat source. During summer, i.e., outside the space-heating season, the heat demand remains nearly constant at approximately 2.2 MW (Figure 2), which corresponds to an average daily heat supply of about 190 GJ. Owing to day-to-day variability in solar irradiance and ambient conditions, the system's ability to satisfy this demand depends not only on component sizing but also on the adopted supervisory control strategy.

The configuration shown in Figure 1 is intended to supply heat to the district-heating network and, depending on the operating mode, may also generate electricity. Its thermal performance and the instantaneous electricity balance, including grid import and export, are governed by both the system design and the control logic. The most influential design parameters include:

1. the type, number, layout, and field arrangement of the PVT collectors;
2. the heat-pump configuration, including compressor type and number, as well as refrigerant selection;
3. the capacity and design of the thermal energy storage (TES);
4. the type and capacity of the electrical energy storage (EES);
5. the thermal capacity and hydraulic configuration of the district-heating network (DHN).

In addition to the physical architecture associated with system sizing and configuration, the adopted control strategy is critical. In this context, it is necessary to define the operating objectives and permissible deviations in the supply parameters of the district-heating network. Exemplary control objectives that may be considered within the control process include:

1. full or partial coverage of the DHN heat demand over the longest possible period of the year;
2. operation based exclusively on electricity generated by the PVT collectors;
3. operation at the highest achievable coefficient of performance (COP);
4. operation aimed at minimizing operating costs.

In the present study, the control criterion was defined as the ability of the proposed system to cover the DHN heat demand during summer and, to a lesser extent, during the shoulder seasons. Preliminary model calculations indicated that at least 3400 PVT collectors would be required to achieve this objective. This preliminary sizing was based on summer operation, when the district-heating demand was dominated by domestic hot water preparation. The PVT field size was increased until the PVT-HP system was able to cover the representative summer heat-demand level under the adopted control assumptions. The value of approximately 3400 collectors represented the minimum field size satisfying this criterion, whereas 3600 collectors were selected as a rounded configuration with a small design margin. Therefore, a total of 3600 collectors was assumed in the calculations. The estimated land area required for installing a PVT field of this size is approximately 2.1 ha, which may be available in the case of some heat supply companies. This estimate was based on the practical experience one of the authors gained during the implementation of the "Ciepłownia Przyszłości" project [10]. In addition, the following simplifying assumptions were adopted for the supervisory control strategy:

1. the heat-pump inlet temperature was set equal to the DHN return temperature;
2. the heat-pump outlet setpoint temperature was set equal to the DHN supply temperature;
3. the required outlet temperature of the heat pump was controlled by adjusting the circulation-pump flow rate;

4. the minimum outdoor temperature at which the system was allowed to operate was set to 4 °C;
5. the minimum outlet brine temperature from the heat-pump evaporator, corresponding to the inlet temperature to the PVT field, was also set to 4 °C.

All of these parameters were adopted a priori, based on the authors' knowledge and engineering experience. They should therefore be regarded as preliminary assumptions that need to be verified and, if necessary, refined in subsequent control-strategy development. It should also be emphasized that the adopted minimum inlet temperature to the PVT field was intended to prevent frost formation on the collector surface. This threshold was selected conservatively and was most likely higher than strictly necessary. The 4 °C ambient-temperature limit resulted from preliminary calculations showing that, with this constraint, the PVT-field inlet temperature did not fall below 0 °C. This assumption is conservative and has a significant effect on the annual energy performance. Further model development and experimental validation will therefore focus on refining these setpoints while maintaining the compressor operating envelope and avoiding frost formation on the PVT collectors.

4.1. Effect of PVT-Field Inlet Temperature on System Thermal Output and COP

The thermal output and the overall coefficient of performance of the proposed system depend strongly on the assumed inlet temperature of the heat-transfer fluid to the PVT field. This parameter was therefore treated as a control variable in the implemented strategy. Its appropriate operating value depends mainly on the instantaneous heat demand of the DHN, weather conditions, and the admissible operating range of the heat pump. In a more advanced predictive strategy, the states of TES and EES could also be included. On the one hand, decreasing the inlet temperature to the PVT field increases the useful heat gains from the collectors. On the other hand, such a reduction requires a lower evaporation temperature in the heat pump, thereby reducing its energy efficiency. The resulting trade-off between PVT heat recovery and heat-pump performance must therefore be explicitly considered in the supervisory control strategy.

To illustrate this effect, additional calculations were performed on the 160th day of the year. This day was selected because, under the adopted control strategy, the heat pump operated continuously throughout the entire day, without interruption. It was one of the non-heating-season days for which the available PVT-HP output could exceed the district-heating demand. Therefore, it was suitable for illustrating how the proposed controller changed the PVT-field inlet temperature to reduce oversupply and improve operating conditions. This case should be interpreted as an illustrative controller-response example, not as a full robustness test. For each simulation case, the setpoint temperature of the heat-transfer fluid supplied to the PVT field was defined as the ambient temperature reduced by a constant temperature difference assigned to a given case. The analyzed temperature difference ranged from 2 to 10 K. The DHN heat demand on that day amounted to 54.42 MWh, and this value was used as the reference for calculating the dimensionless demand coverage factor Φ . The results obtained for system operation at different assumed inlet temperatures to the PVT field are presented in Table 1.

The relationships shown in Figure 9 indicate that the inlet temperature to the PVT field can be used as an effective control variable for shaping the operating profile of the proposed system. By decreasing $T_{p,in}$, a higher amount of heat can be delivered to the DHN under the same ambient conditions, which means that the degree of heat-demand coverage Φ can be directly influenced through control of this parameter. At the same time, this effect is achieved at the expense of increased system electricity consumption. Increasing $T_{p,in}$ raises the evaporation temperature of the refrigerant in the heat pump and therefore improves its

coefficient of performance, as also shown in Figure 9. Consequently, the control of $T_{p,in}$ leads to a clear trade-off between DHN heat-demand coverage and system COP. Values of Φ above 100% in Table 1 indicate that, for low PVT-field inlet-temperature setpoints, the available PVT-HP thermal output exceeds the daily district-heating demand. This does not imply the use of sink-side TES. In practical demand-following operation, the controller can increase the PVT-field inlet-temperature setpoint and reduce the heat-pump output, which lowers heat production and improves COP. Small remaining imbalances were assumed to be absorbed by the thermal inertia of the district-heating network, causing only limited supply-temperature fluctuations. For the 160th day of 2025 analyzed, Table 1 shows that parameter Θ , defined as the ratio of electricity generated by the PVT collectors to electricity consumed by the thermal system, does not reach 50% in any of the considered cases. This means that even at the highest assumed PVT supply temperature, more than 50% of the required electricity must still be imported from the grid. It should be noted, however, that for other days of the year or for a different system configuration, a higher degree of electrical self-sufficiency may be achieved. This level is also strongly affected by the type of heat pump and its operating range. Finally, the departure from the nearly linear trends observed for $\Delta T_a = 8\text{--}10\text{ K}$ results from the adopted lower limit for the PVT inlet temperature, set to $4\text{ }^{\circ}\text{C}$ to prevent frost formation on the collector surface. This constraint causes both the heat-demand coverage factor and the COP to approach a plateau in this range.

Table 1. Summary of the total electricity and heat supplied by the PVT-HP system and the average daily COP value for a variable temperature difference between ambient-air temperature and PVT-field inlet temperature (day 160).

ΔT_a	$E_{sys.th}$	$E_{p.th}$	$E_{sys.el}$	$E_{HP.el}$	$E_{p.el}$	Θ	Φ	COP
K	MWh	MWh	MWh	MWh	MWh	[%]	[%]	-
2	19.34	11.86	8.84	7.40	4.19	47.4	35.5	2.61
3	25.78	15.66	11.39	9.95	4.20	36.9	47.4	2.59
4	32.37	19.59	14.11	12.67	4.21	29.8	59.5	2.55
5	39.41	23.72	17.02	15.58	4.22	24.8	72.4	2.53
6	46.57	27.89	20.01	18.57	4.23	21.1	85.6	2.51
7	53.24	31.78	22.81	21.37	4.24	18.6	97.8	2.49
8	58.04	34.54	24.83	23.39	4.25	17.1	106.7	2.48
9	59.93	35.63	25.63	24.19	4.25	16.6	110.1	2.48
10	60.66	36.09	25.92	24.48	4.26	16.4	111.5	2.48

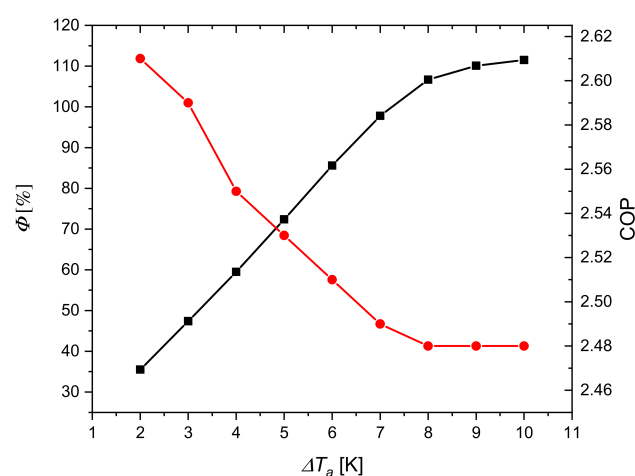


Figure 9. Dependence of the heating network demand coverage factor (black line) and the heat-pump efficiency factor (red line) as a function of the set temperature difference between the ambient temperature T_a and the inlet temperature $T_{p,in}$ (day 160).

4.2. Applied Control Strategy

The detailed control algorithm was developed from the parametric results presented in Section 4.1. These results identified the $T_{p,in}$ inlet temperature as the main operating variable. This variable affects both heat-demand coverage and system COP. The feedback structure used in the simulation is shown in Figure 10. The control objective is to match the heat-pump output to the district-heating heat demand. The algorithm also keeps the PVT-field inlet temperature within the limits required for frost prevention and compressor operation.

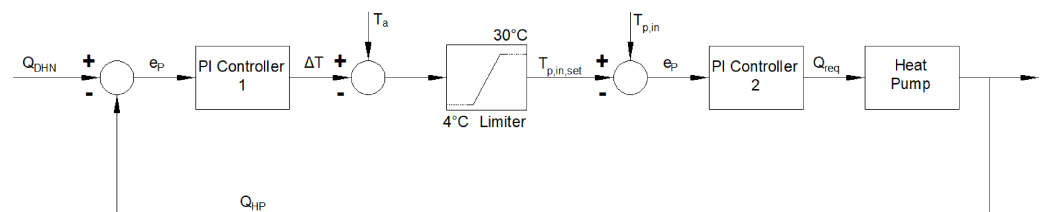


Figure 10. Control block diagram of the proposed supervisory strategy for the system.

The controller response was illustrated for the 160th day of the year. The main assumption is that system operation was allowed only when the ambient temperature exceeded 4 °C. The lower admissible temperature of the heat-transfer fluid supplied to the PVT field was also set to 4 °C. This limit was introduced to reduce the risk of frost forming on the surfaces of unglazed PVT collectors. The upper limit was set to 30 °C. This value is determined by the heat-pump compressor's permissible operating range. Thus, the PVT-field inlet-temperature setpoint was constrained to 4–30 °C.

The algorithm consists of two PI controllers (Figure 10). Both controllers were implemented using the PID Advanced VI from the LabVIEW Control Design and Simulation Module version 21.0 software (National Instruments, Austin, TX, USA). The model was solved using the variable-step Runge–Kutta 23 ODE solver, and the actual integration step was used by the controller algorithm. The proportional gains and integral times of the two supervisory PI controllers and the condenser-side process-level PI controller are listed in Appendix A, where PI1 denotes the outer heat-load controller, PI2 the PVT-field inlet-temperature controller, and PI3 the condenser-side circulation-pump controller. The first controller forms the outer heat-load control loop. The district-heating heat demand, Q_{DHN} , is compared with the actual thermal output delivered by the heat pump, Q_{HP} . The resulting error is supplied to PI Controller 1. The controller output is the temperature correction ΔT . This signal represents the required depression of the PVT-field inlet temperature relative to ambient temperature. The preliminary temperature setpoint is therefore calculated as $T_a - \Delta T$. This value is then passed through the limiter. The limiter generates the final setpoint $T_{p,in,set}$.

This control action follows from the trade-off discussed in Section 4.1. If the actual heat output is lower than the district-heating demand, PI Controller 1 increases ΔT . The PVT-field inlet-temperature setpoint then decreases. This increases heat extraction from the collector field. At the same time, it lowers the heat-pump evaporation temperature and reduces the COP. If the required heat output can be achieved with a smaller temperature depression, ΔT decreases. The PVT-field inlet-temperature setpoint then increases. This improves the heat-pump COP.

The second PI controller forms the inner temperature-control loop. Its setpoint is $T_{p,in,set}$ and the process variable is the actual PVT-field inlet temperature, $T_{p,in}$. The controller output is the required heat-pump capacity, Q_{req} . This signal is applied to the heat-pump model. It modifies the unit's instantaneous operation. The heat-pump response affects both $T_{p,in}$ and the heat delivered to the district-heating network Q_{DHN} .

The operation of the algorithm is illustrated by the daily profiles shown in Figures 11–13. Figure 11 shows the incident solar irradiance on the PVT field. The highest value occurs in the morning, when the irradiance exceeds 400 W/m^2 . The profile then decreases, with several local reductions. These fluctuations indicate variable sky conditions during the analyzed day and also influence the system efficiency.

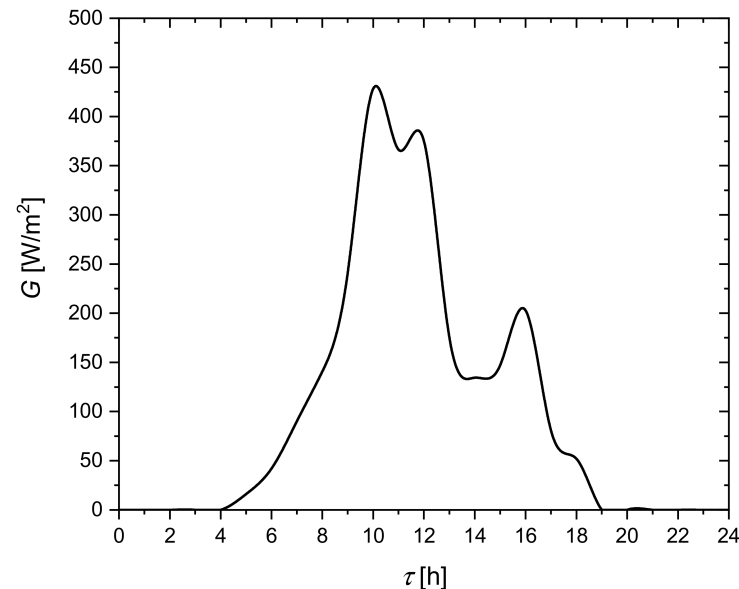


Figure 11. Daily profile of irradiance in the plane of the PVT field.

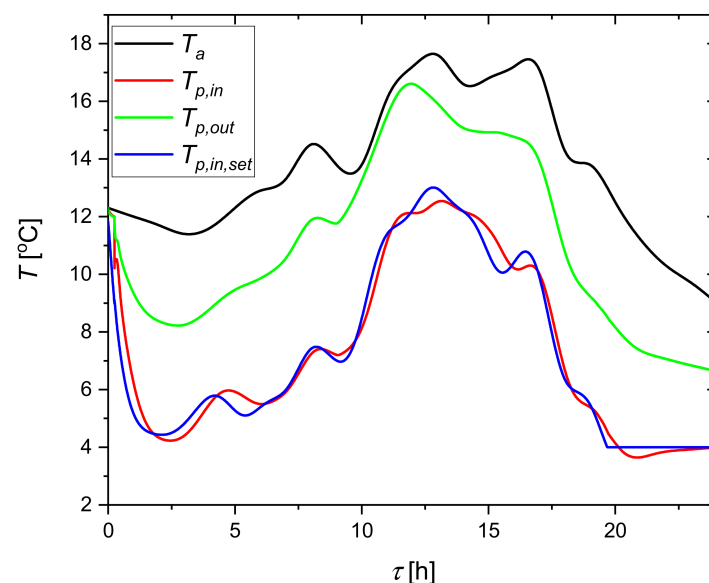


Figure 12. Daily profile of temperatures at the PVT collectors inlet and outlet (red and green) in the relation with ambient temperature (black) and the temperature setpoint (blue).

Figure 12 shows the ambient temperature, the PVT-field inlet and outlet temperatures, and the calculated temperature setpoint. The PVT-field inlet temperature follows the setpoint over most of the day. This confirms the efficient operation of the inner temperature-control loop. During the night and late evening, the inlet temperature approaches the lower admissible limit of 4°C . This behavior is driven by the heat-load controller, which lowers the PVT-field inlet temperature to increase heat extraction from the collectors. During periods of higher solar irradiance and higher ambient temperature, the setpoint increases. The actual inlet temperature increases accordingly. This reduces the temperature depression

relative to the ambient air and improves the heat-pump operating conditions. The PVT outlet temperature remains higher than the inlet temperature throughout the analyzed period. This confirms that the PVT collector field acts as an effective low-temperature heat source.

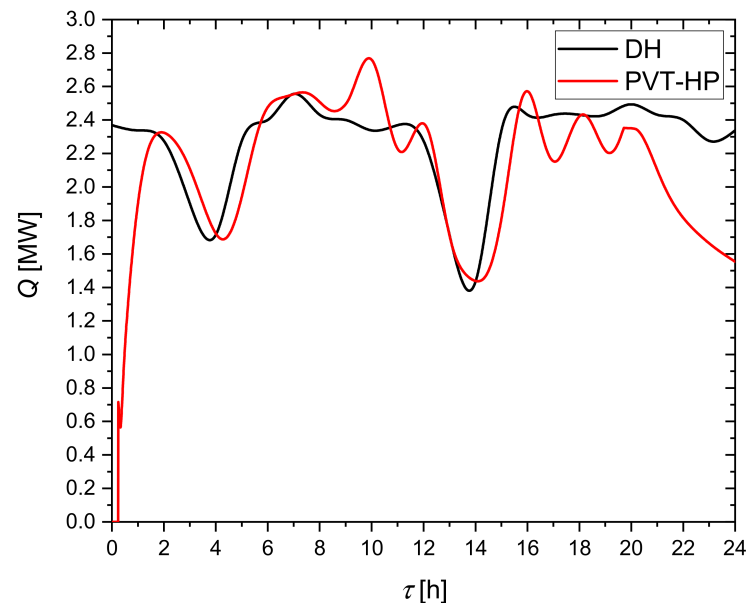


Figure 13. Relation between the district-heating heat demand and thermal output of the PVT-HP system.

Figure 13 compares the district-heating heat demand with the thermal output obtained from the PVT–heat pump system. The demand remains within a relatively narrow range during most of the day. The heat-pump output follows the demand profile with visible dynamic deviations. These deviations are caused by interactions among the heat-load loop, the temperature-control loop, and the changing thermal conditions of the PVT field. The largest variations occur during periods with rapid changes in the PVT inlet temperature and solar irradiance. Despite these transients, the system can supply heat to the district-heating network throughout the analyzed day.

A relevant result is the system’s nighttime operation. Nevertheless, the PVT field can still operate as a low-temperature source for the heat pump. Heat is then extracted mainly from the ambient air and from heat exchange with the collector surroundings. This allows the system to also cover the district-heating load during periods with no solar irradiance. The result is important for the analyzed configuration because the district-heating demand is maintained during the night, mainly due to domestic hot water preparation.

The flow rate through the PVT field and the evaporator side of the heat pump was assumed to be constant. The number of installed collectors determined it. Therefore, the low-temperature source-side flow rate was not used as an independent control variable. On the sink side, the heat-pump inlet temperature was assumed to be equal to the district-heating return temperature. The outlet temperature was set as equal to the district-heating supply temperature. The corresponding condenser-side mass flow rate was calculated from the required heat-pump heating capacity and the temperature difference between the district-heating supply and return lines. This process-level flow adjustment was not included in the supervisory-control diagram.

The adopted strategy should be treated as a preliminary feedback-control structure. It enables heat-demand tracking and keeps the PVT-field inlet temperature within the imposed safety and compressor-operation limits. Further development should include predictive variables such as solar irradiance, ambient temperature, the storage state of

charge, electricity prices, and grid-import constraints, so that heat-demand coverage, COP, electrical self-sufficiency, and operating cost can be balanced simultaneously.

5. Calculation Results and Discussion

Annual simulations of the PVT-HP system were conducted using full-year district-heating and weather data described in the previous sections. The results are presented for the full year. Although the PVT model does not include frost and snow effects, the aim was to show the system limitations during the winter period regardless of this model simplification. The calculations were performed for the system configuration and supervisory control strategy defined in Section 4.

Figure 14 shows the annual variation in the district-heating-network heat-demand coverage by the PVT-HP system. During the summer period, the proposed plant configuration and adopted control strategy achieve a high degree of demand coverage. For a considerable part of the non-heating season, the daily heat supplied by the PVT-HP system is close to, or exceeds, the district-heating network's daily heat demand. There are also days when the daily mean heat supplied by the PVT-HP system is lower than the network demand. However, these deficits are not large during the summer. Depending on the network's thermal capacity and available thermal storage volume, they would mainly result in short-term temperature fluctuations rather than a complete loss of heat supply. The annual profile also confirms that the proposed system cannot serve as the main heat source during the winter. During this part of the year, demand coverage decreases markedly due to lower ambient temperatures, lower solar irradiance, and the imposed operating limits of the PVT field and the heat pump. In general, the percentage of the total annual district-heating demand covered by the PVT-HP system is equal 44.2% and the annual seasonal performance factor (SPF) meets 2.52.

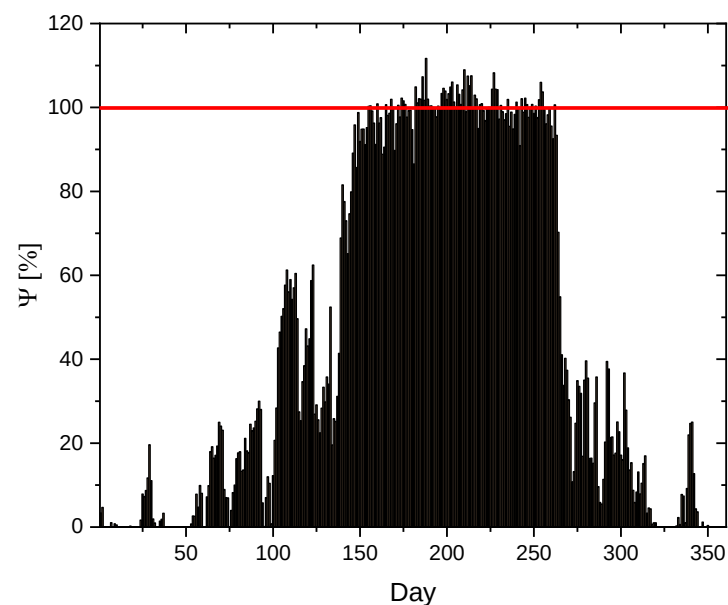


Figure 14. Annual district-heating heat-demand coverage by the PVT-HP system.

The adopted control strategy and the integration of the PVT field with the electric energy storage system reduce the grid electricity import. In the analyzed configuration, the electricity generated by the PVT collectors is used only to supply the PVT-HP system, including the heat pump and circulation pumps. When instantaneous PVT electricity generation exceeds system demand, the surplus is stored in the EES for later use. However, the electricity generated by the PVT field and recovered from the EES is insufficient to

cover the system's total annual electricity demand. Therefore, partial grid electricity supply remains necessary. The main operational objective is thus not full electrical autonomy, but minimization of grid electricity import while maintaining the required district-heating heat supply.

The simulated annual variation in the daily mean heat-pump COP is shown in Figure 15. During most operating periods, the daily mean COP remains within approximately 2.0–3.1. The obtained values depend on several parameters, including the number of PVT collectors, the nominal capacity of the heat pump, the compressor performance, the district-heating supply and return temperatures, and the adopted control strategy. The control strategy appears particularly important because the inlet temperature to the PVT field directly affects the heat pump's evaporation temperature. Therefore, further refinement of this control variable may improve the energy performance of the entire system.

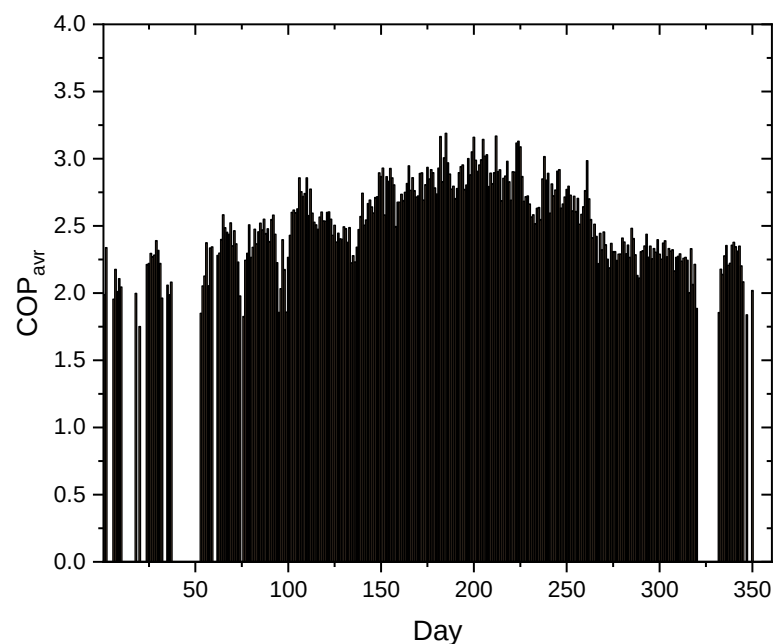


Figure 15. Simulated annual variation in the daily mean heat-pump COP.

The calculated daily mean COP values are not high compared with typical values reported for residential heat pumps operating during the summer. This comparison should be interpreted with caution. In the analyzed configuration, the heat pump actively shapes the operating conditions of the low-temperature source by controlling heat extraction from the PVT field. The system, therefore, does not operate in the same way as a residential heat pump connected to a comparatively stable source or to ambient air under standard control logic.

This distinction is also important for interpreting the COP. The COP shown in Figure 15 should be understood as the heat-pump COP based on the total electrical input to the heat pump, in accordance with the conventional definition. If the locally generated electricity from the PVT field and the EES is accounted for as grid electricity consumption, an additional system-level indicator may be defined. Such a grid-electricity-based COP would be higher than the conventional heat-pump COP because only the electricity imported from the grid would be included in the denominator. This indicator should therefore be reported separately and should not be confused with the normative COP of the heat pump.

The economic and ecological analysis of decarbonization heating systems in Poland was conducted, for example, by Kalina [40], who examined large-scale high-temperature heat pumps using wastewater treatment plant effluent as a heat source in a coal-fired

district-heating system in Gliwice. This work showed that, in the base scenarios, the system was unprofitable ($NPV < 0$). However, after modifying the operating strategy to control the heat pump based on the achievable COP and the actual energy cost ratio, the result improved to approximately $NPV = 0.7285$ million EUR and $SPB = 11.15$ years. Kalina indicated that such projects should be subsidized by approximately 30% of CAPEX.

For the economic and ecological analyses in the present study, a constant heat price of 0.378 PLN/kWh was assumed, based on the GPEC (Gdańskie Przedsiębiorstwo Energetyki Ciepłej Sp. z o.o.) tariff [49]. The constant electricity price was assumed to be approximately 0.850 PLN/kWh, based on the average price offered by Energa S.A. in 2024 [50]. Two CO₂ emission factors were considered: one for electricity imported from the grid and another for heat produced in the district-heating system. According to the National Center for Emissions Management (KOBiZE) [51], the emission factor for electricity from the Polish grid in 2025 is 0.766 kg CO₂/kWh. The emission factor for heat production in a district-heating system is 0.295 kg CO₂/kWh, reflecting the structure of the national energy mix. The EU ETS CO₂ allowance price was assumed to be 318 PLN/tCO₂.

Based on the calculated hourly thermal power supplied to the district-heating network, the electrical power generated by the PVT field, and the electrical power required to supply the heat pump and circulation pumps, an illustrative operating and CO₂-emission balance of the PVT-HP system was estimated. For the analyzed variant, annual avoided CO₂ emissions amounted to approximately 1.253×10^6 kg CO₂, corresponding to an approximately 8% reduction across the whole system. The monetary value of the avoided CO₂ emissions was approximately 0.3988 million PLN per year. The annual electricity cost for operating the PVT-HP system was estimated at approximately 2.189 million PLN. The gross monetary value of heat supplied by the PVT-HP system, calculated using the assumed heat tariff, amounted to approximately 4.133 million PLN. Consequently, the illustrative annual operating benefit, including the value of avoided CO₂ emissions and excluding CAPEX and fixed O&M costs, was estimated at approximately 2.343 million PLN. This value should not be interpreted as a full economic feasibility result because investment costs, fixed operating costs, maintenance costs, financing conditions, and site-specific implementation factors were outside the scope of this study.

Figure 16 shows the monthly electricity autarky coefficient for the PVT-HP system operating with the PVT field and electric energy storage. In this context, the electricity autarky coefficient denotes the fraction of the system electricity demand covered by electricity generated in the PVT field, either directly or via the electric energy storage. The system's electricity demand includes the heat pump and circulation pumps. The results show significant seasonal variability. In spring and summer, the coefficient exceeds approximately 40% in several months and reaches approximately 59% in March. In July and August, the autarky coefficient remains close to approximately 48–49%. In autumn and winter, the coefficient decreases substantially, reaching approximately 12% in November. This confirms that the potential for autonomous operation of the PVT-HP system is strongly dependent on solar radiation and on the seasonal relation between heat demand and PVT electricity production.

The results confirm a strong seasonal variation in the role of the heat pump in the district-heating system. During the non-heating season, the PVT-HP system can operate as the main heat source and cover approximately 90–100% of the district-heating demand over extended periods. During the heating season, coverage decreases sharply, often dropping below 5% on many days. The proposed system should therefore be interpreted primarily as a seasonal heat source for summer and shoulder-season operation. In such a role, it can reduce heat production from the conventional source, decrease CO₂ emissions, and partly replace grid electricity with electricity generated by the PVT field. However, the annual

results also show that the final performance depends strongly on the control strategy, the size of the PVT field, the heat pump's operating limits, and the capacity of the electrical and thermal storage units.

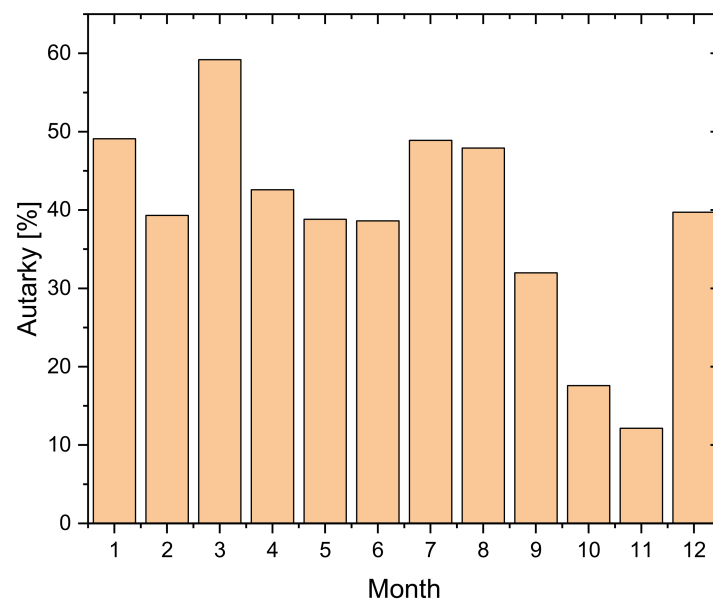


Figure 16. Monthly autarky coefficient for electricity of the PVT-HP system.

The extended PVT formulation includes wind influence, precipitation, condensation-related heat gains, and operation without solar irradiance. It therefore supports the assessment of WISC-type PVT collectors as low-temperature heat sources under nighttime and condensation-prone conditions. However, because long-wave radiation above $3\ \mu\text{m}$ was estimated rather than measured, the calculated thermal yield remains uncertain and should be verified experimentally.

6. Conclusions and Future Perspectives

The study assessed a hybrid PVT-HP system with TES and EES as a possible seasonal heat source for a small- to medium-scale district-heating network. The model was applied to hourly data from an existing network and used to determine whether the proposed configuration could replace the conventional source during periods of low heat demand. The results show that this is feasible mainly outside the heating season. Over extended non-heating-season periods, the PVT-HP system covered approximately 90–100% of the district-heating demand, while the daily mean COP remained within approximately 2.0–3.1. Electricity generated by the PVT field was either consumed directly within the system or temporarily stored in the EES. However, it was not sufficient to cover the total electricity demand of the heat pump and circulation pumps, so partial grid supply remained necessary.

The system can also contribute to heat production during shoulder periods, although its performance is then more dependent on control assumptions and operating constraints. Further refinement of the configuration, operating range, and supervisory control strategy could improve useful heat gains and overall performance. This mainly concerns the admissible PVT-field operating temperatures, heat-pump capacity modulation, storage use, and coordination of heat production with weather-dependent PVT performance. The PVT-field inlet temperature was identified as an especially important control variable. Lowering this temperature increases heat extraction from the PVT field, including periods without solar irradiance. At the same time, it lowers the heat-pump evaporation temperature and may reduce COP. This trade-off shows why static annual-average calculations are

insufficient for this type of system. A dynamic model is needed to capture the interaction between weather-dependent PVT gains, heat-pump operation, storage states, district-heating temperatures, and control constraints.

The developed model should therefore be treated as a concept-assessment tool for the analyzed PVT-HP configuration, not as a general-purpose optimization tool for district-heating systems. In its present form, it supports preliminary sizing, assessment of the selected architecture, and comparison of control assumptions under time-varying weather and load conditions. Further work should include refinement of control strategies, analysis of the influence of TES and EES sizing on performance, more detailed modeling of the industrial heat pump, and assessment of a multi-source heat pump using the PVT field together with other renewable or waste-heat sources. Overall, the presented configuration should be regarded as a seasonal renewable source for district heating rather than a stand-alone annual heat source. Its value lies in reducing conventional heat production during periods when the network load and temperature requirements are compatible with PVT-assisted heat-pump operation.

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

Abbreviations

Abbreviations

DHN	District-Heating Network
DHS	District-Heating System
EES	Electric Energy Storage
HP	Heat Pump
HTF	Heat-Transfer Fluid
PI	Proportional–Integral controller
PVT	Hybrid photovoltaic–thermal collector
TES	Thermal Energy Storage

Nomenclature

A	heat-transfer area, [m ²]
c	specific heat, [J/kgK]
E	energy, [J], [kWh]
k	overall heat-transfer coefficient, [W/m ² K]
l	length, [m]
$\dot{m}$	mass flow rate, [kg/s]
M	mass, [kg]
P	power, [W]
Q	heat flow rate, [W]
R	thermal resistance, [K/W]
T	temperature, [K], [°C]
$\dot{V}$	volume flow rate, [m ³ /h]

Greek symbols

α	heat-transfer coefficient, [W/m ² K]
δ	heat conduction thickness, [m]
λ	thermal conductivity, [W/(m·K)]
τ	time, [s], [h]

Subscripts

<i>a</i>	surroundings
<i>b</i>	heat-transfer fluid—brine
<i>bg</i>	background
<i>c</i>	condenser
<i>con</i>	condensation
<i>d</i>	district-heating network
<i>e</i>	evaporator
<i>el</i>	electric
<i>lw</i>	long-wave radiation
<i>loss</i>	thermal losses
<i>m</i>	modeled parameter
<i>out</i>	outlet
<i>p</i>	hybrid photovoltaic–thermal (PVT) collector
<i>rain</i>	rain
<i>set</i>	setpoint value
<i>sol</i>	solar
<i>sys</i>	the PVT-HP system
<i>t</i>	thermal energy storage
<i>th</i>	thermal energy
<i>w</i>	heat-transfer fluid—water
<i>wat</i>	rainwater

Appendix A**Table A1.** Values of constant parameters used in the simulation process.

Symbol	Value	Unit	Remarks
A_G	2.17	m ²	
a_1	15	W/(m ² K)	
a_2	0	W/(m ² K ²)	
a_3	2	J/(m ³ K)	
a_4	0.5	-	
a_5	0	J/(m ² K)	
a_6	0.014	s/m	
a_9	0.5	-	
a_{10}	2000	J/kg	For both sides of a collector
a_{11}	0.5	-	
M_p	31	kg	
c_p	1.94	kJ/(kgK)	
η_0	0.37	-	
$c_{p,wat}$	4.18	kJ/(kgK)	
P_{STC}	450	W	

Table A1. Cont.

Symbol	Value	Unit	Remarks
γ	−0.0036	1/K	
K_{pPI1}	−0.1	-	The outer heat-load controller
Ti_{PI1}	5	min	
K_{pPI2}	0.0002	-	The inner PVT-field inlet-temperature controller
Ti_{PI2}	10	min	
K_{pPI3}	−0.05	-	The condenser-side circulation-pump controller
Ti_{PI3}	5	min	

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