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Simulation-Based Design of a Two-Stage Heat Pump System Utilising Treated Wastewater for Urban Heating Applications

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Abstract

In light of current energy policies, integrating renewable energy and waste heat recovery into district heating systems has become increasingly crucial. There is a particular need to identify locally available, low-temperature heat sources that can be effectively upgraded to meet the high supply temperatures required by existing networks. This study examines the technical and economic viability of high-temperature heat pumps utilising treated sewage as a renewable heat source in district heating systems. A comprehensive mathematical model of a high-capacity, two-stage centrifugal heat pump was developed and implemented in Aspen HYSYS to simulate its performance under varying operational conditions typical of Central-Eastern Europe. The model accounts for seasonal variations in sewage flow and temperature, as well as regulatory constraints of district heating networks and technical requirements such as temperature lift (ΔT_{lift}) and part-load operation. Two operating scenarios are evaluated: a conventional coal-based system and a hybrid system incorporating a 3 MW sewage heat pump. Results indicate that the heat pump can meet domestic hot water demand in the summer and reduce annual CO_2 equivalent emissions by approximately 9500 tonnes, assuming the use of renewable electricity. Despite minor degradation in the coefficient of performance at part-load and low-flow conditions, the system demonstrated stable, efficient performance. An economic assessment using the levelised cost of heat method confirmed cost competitiveness in the Polish market context. The findings underscore the strategic role of sewage-source heat pumps in decarbonising urban heating.

Keywords: High-temperature heat pumps; Waste heat recovery; District heating; Off-design conditions; Mathematical modelling

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1. Introduction

Currently, energy policy and global trends are promoting increased shares of renewables and the use of waste heat. The green energy transformation is driving changes not only in electricity generation but also in district heating [1]. The European Union (EU) has introduced an ambitious climate and energy package to reduce greenhouse gas emissions, promote renewable energy sources, and enhance energy efficiency. Initially outlined in the 2020 framework and later extended to 2030, these measures aim to transition the EU economy to climate neutrality by 2050. To achieve this transformation, the European Commission introduced the „Fit for 55” initiative, a comprehensive legislative package designed to update existing regulations and create new policies. This effort aligns EU actions with the climate goals set by the European Parliament and the Council.

In the realm of district heating (DH), significant legislative updates affect directives such as the Emission Trading System (ETS) and the Energy Efficiency Directive (EED). A key change under the EED redefines what constitutes an efficient heating

Nomenclature

a, *b* – constants
CF – correction factor
COP_i – value of COP at time range *i*
C_V – coefficient
h – enthalpy of the flow, kJ/kg
h_i, h_j – specific enthalpies, kJ/kg
k – adiabatic exponent
LMDT– logarithmic mean temperature difference, K
$\dot{m}$ – mass flow rate, kg/s
M – number of outlet streams
n – volume exponent
$\dot{n}$ – molar flow rate, kmol/s
N – number of inlet streams
p – pressure, Pa
Δp – pressure drop, Pa
P – power, kW
q – heat provided to the shell side, kW
q_{loss} – heat losses, kW
Q – heat exchanger power, MW
r – compression ratio
R – individual gas constant, J/(mol·K)
t – time, s
T – temperature, K
ΔT_{lift} – temperature lift, K
UA – overall heat exchanger heat load, W/K
v – specific volume, m^3/kg
V – volume, m^3
$\dot{V}$ – volumetric flow,

Greek symbols

ρ – density, kg/m^3

Subscripts and Superscripts

design– design point parameters
i, *j* – streams
in – inlet
lift – lift
loss – losses
new – new value
out – outlet
shell – shell side of heat exchanger
tot – total
tube – tube side of heat exchanger

Abbreviations and Acronyms

CAPEX– capital expenditures
CFD – computational fluid dynamics
COP – coefficient of performance
DH – district heating
DHC – district heating and cooling
DHCN – district heating and cooling networks
DHN – district heating networks
DHS – district heating system
EED – Energy Efficiency Directive
ETS – Emission Trading System
EU – European Union
GDH – generation district heating
GDHC – generation district heating and cooling
GEA – GEA Group, Germany
GWP – global warming potential
HFO – hydrofluoroolefin
HP – heat pump
HTHP – high-temperature heat pump
IEG – Research Institution for Energy Infrastructures and Geothermal Systems IEG
IGV – inlet guide vanes
LCOH – levelised cost of heat
ODP – ozone depletion potential
OPEX – operating expenditures
PCM – phase change material
sCOP – seasonal COP
VHC – volumetric heating capacity

system. Starting in 2028, only systems that incorporate heat generated from renewable sources or waste heat will qualify as efficient. The required share of such heat will gradually increase to 100% by 2050 [2]. These new requirements pose considerable challenges for heat suppliers and producers, compelling them to adapt their operations to meet the updated efficiency standards. It is worth noting that in 2021, in Poland, 69.5% of heat used for heating was generated from coal, while slightly more than 20% was generated from renewables [3]. When assessing potential heat sources for centralised heating systems, compliance with these evolving regulations is critical.

Currently, only a few technologies possess the technical potential to serve as significant heat sources for centralised heating systems in medium- and large-sized cities. These include solid biomass boilers, particularly those with cogeneration capabilities, and large-scale heat pumps [4]. Additionally, using treated wastewater as a low-temperature heat source for heat pumps offers numerous benefits, including predictable, stable flow, high medium temperatures and high energy density.

In many countries, such as those in Scandinavia, high-temperature sewage heat pumps are commonly used for DH. For instance, studies on the potential of heat pumps in Denmark can be found in Johansen et al. [5] and Østergaard et al. [6], while case studies on Stockholm (Sweden) are presented by Levihn [4].

High-temperature heat pumps have been studied in conjunction with district heating networks (DHN), yielding promising results. Alarnaot et al. [7] carried out a multi-perspective assessment of an experimental moderately-high-temperature heat pump (HTHP) for district heating and cooling networks (DHCN), obtaining a 1.96-year payback period when connected to 4G DHCN. Sadeghi et al. [8] used heat from a DHN at 85°C to produce steam at 160°C using HTHP and considered several configurations, including combinations of vapour compression and steam cycles. Passamonti et al. [9] presented a HTHP pilot plant at the Fraunhofer Research Institution for Energy Infrastructures and Geothermal Systems IEG (IEG) location in Bochum for 120°C heating production, which can save up to 4800 tonnes of CO_2 equivalent emissions annually. In [10], Degelin et al. presented a techno-economic analysis of a centralised heat pump dedicated to district heating, including the influence of supply temperature (10−75°C) on seasonal coefficient of performance (sCOP), which varies from 3.43 to 4.6. The levelised cost of heat (LCOH) for analysed cases remained in the range of 213−297 EUR/MWh$_{th}$. Zhu et al. [11] experimentally investigated the cooperation of booster heat pumps with ultra-low-temperature DHN, achieving a coefficient of performance (COP) of 4.95. Barco-Burgos et al. [12] presented a review of high-temperature heat pumps in DHCN. They have identified factors influencing the feasibility of heat pumps (HP) in DH, including electricity and fuel prices, supply temperature, availability of external waste heat and other local conditions. The 5th generation of DHCN was analysed by Calise et al. [13]. The integration of renewable energy sources, a ground-source heat pump, and a low-temperature network resulted in 64% primary energy savings and a 76% reduction in CO_2-equivalent emissions. The same authors in [14] analysed seawater HP cooperation with district heating and cooling (DHC). The simple payback period of the system was relatively long – 14.7 years. However, coupling the system with renewable energy plants allows for achieving zero primary energy consumption and CO_2 emissions. Another work related to seawater HP implementation in DH was presented by Ali et al. [15]. Authors report 88% electricity savings, 65% reduction in CO_2 emissions and a 76% decrease in LCOH in the Norwegian case, with slightly worse coefficients in Tallinn due to sea water freezing in winter. In the northern part of Central-Eastern Europe, the heating season extends from October to April, spanning more than half the year. Additionally, heating systems provide domestic hot water year-round, including during the summer months. Given the extended operational period, implementing environmentally friendly innovations in this sector is crucial.

Heat pumps can be operated with various refrigerants, including hydrofluoroolefines (HFO) and other increasingly popular natural refrigerants. The selection of a refrigerant should be based on thermodynamic properties, safety considerations and environmental impact. A comparison of the main parameters for HP operation with typical refrigerants is presented in Table 1.

Table 1. Comparison of the main parameters for HP operation with typical refrigerants.

Refrigerant	R1234ze(E)	NH_3 (R717)	CO_2 (R744)
Chemical formula	$C_3H_2F_4$	NH_3	CO_2
Type	HFO (Synthetic)	Natural	Natural
GWP / ODP	<1 / 0 [63]	0 / 0 [38]	1 / 0 [64]
Safety class	A2L [65]	B2L (toxic) [65]	A1 (non-toxic) [65]
Operating pressure	Standard (~25–30 bar) [38]	Medium (10–35 bar) [66]	Very high (30–90 bar) [67]
COP range	2.5–6.2 [16]	3.0–5.9 [16,21]	1.8–4.2 [20,24]
Typical max. outlet temperature	80–95°C [37]	95–120°C [25,26]	90–120°C [38]

R1234ze(E) has emerged as a promising hydrofluoroolefin refrigerant for heat pump applications due to its favourable environmental characteristics and thermodynamic properties. Experimental studies have demonstrated that R1234ze(E) exhibits superior performance compared to other HFO refrigerants in specific applications, achieving a COP of up to 6.2 [16]. Among the disadvantages of this refrigerant, carcinogenic effects and a high medium cost could be enumerated.

Ammonia (R717) demonstrates superior thermodynamic performance in large-scale district heating applications. Industrial installations by GEA (GEA Group, Germany) show ammonia heat pumps achieving COPs above 3.5, with systems delivering 40 MW of heating capacity for district networks [17]. However, ammonia's primary limitation is its toxicity and associated safety risks. Ammonia is classified as toxic and requires stringent safety measures, extensive leak detection systems, and specialised operator training. Modern low-charge ammonia systems have reduced refrigerant inventory, but even these installations require comprehensive safety protocols due to leakage risks. In applications near populated areas, ammonia presents significant safety management challenges that can complicate implementation and increase liability concerns.

Carbon dioxide (R744) transcritical heat pumps offer unique advantages for ultra-high-temperature applications, with systems capable of producing heated water up to 120°C. Performance data show COPs ranging from 1.8 to 4.2, depending on operating conditions, which is lower than for other considered refrigerants. The critical limitation of CO_2 systems is their extremely high operating pressure. CO_2 transcritical cycles operate at pressures significantly higher than those of conventional refrigerant systems, with maximum design pressures exceeding 90 bar. These elevated pressures require specialised compressors, robust heat exchangers and heavy-duty piping components, substantially increasing capital costs. Additionally, CO_2 systems show higher sensitivity to operating condition variations than alternative refrigerants, resulting in reduced performance flexibility [18].

Urban areas often harbour significant sources of waste heat that remain underutilised or inefficiently exploited. Optimising the use of these resources represents a promising avenue for improving heating systems. One notable example of such waste heat is treated sewage, which holds substantial potential as an energy source [4]. However, the low temperatures of treated sewage make it unsuitable for direct use in heating applications.

To address this limitation, treated sewage can serve as a low-temperature heat source for HP, which raises the medium's temperature to a usable level [19]. This method aligns with decarbonisation strategies [20], which project that heat pumps will meet nearly 25% of heating needs by 2050. Consequently, heat pumps are widely regarded as a promising and forward-thinking solution for future heating systems.

Sewage water has been considered as a potential heat recovery source that can be upgraded with heat pumps. Wang et al. [18] addressed the challenges of unequal sewage water flows, scale formation, corrosion, and a rapid decline in heat transfer capacity over time by considering a non-metallic immersed heat exchanger as an evaporator in HP. Supply temperatures varied from 40.4°C to 60.6°C, with a sCOP of 5.65 achieved. Ma et al. [21] considered sewage HP integrated with phase change material (PCM) heat storage. Although integrating a phase change heat storage unit substantially improves both economic viability and energy efficiency, the simple payback period remains long – over 15 years. A techno-economic analysis of sewage HP for domestic hot water purposes was shown by Xu et al. [22]. Authors emphasise that individual requirements and conditions affect installation performance, although in the analysed case, a 64.8% reduction in CO_2-equivalent emissions was achieved. A small sewage HP dedicated to trigeneration was analysed by Gao et al. [23]. The study shows a strong dependence of COP on ambient temperature, with values of 8.97 in the summer and 2.44 in the winter. In [24], Xu et al. developed a CO_2-equivalent emission-reduction model for sewage HP implementation in an example city in northern China. Typical reductions are approximately 60%, depending on the building type. Brands and Fung [25] analysed large-scale wastewater HP performance in cold climates, achieving significant reductions in CO_2-equivalent emissions (45−60% in fossil-fuel-dependent regions and up to 95% in areas with renewable electricity production). Ozcan et al. [26] studied the energy, exergy, economic, environmental and sustainability impacts of a wastewater-source heat pump system for DH applications.

A heat pump in such an application is constrained by the existing network and its parameters, as well as by the sewage system, safety requirements and legal conditions, including power connection requirements and heat output specifications. Furthermore, it must adapt to the current demand for heat. These complex technical and operational conditions make the integration and optimisation of the heat pump unit within the existing heating system highly challenging. Heat pumps often operate under off-design conditions, making their analysis essential.

Assessing the operating conditions of large heat pumps can significantly benefit from the use of mathematical models. However, there is limited literature dedicated to this topic. Typically, heat pump-related models are implemented in Modelica, MATLAB, TRNSYS and IDEA ICE [27]. Additionally, some scientists employed analytical or semi-analytical methods to calculate HP components, as well as computational fluid dynamics (CFD) tools such as Open FOAM [28]. Tan et al. [29] analysed a low-temperature waste recovery heat pump. The system was modelled in Aspen Plus without off-design analysis. Considered cases covered waste temperatures in the range 100−120°C, which is significantly higher than the temperature levels considered in this study.

In this research, the authors developed a mathematical model of the main processes occurring in large, high-temperature HPs and implemented it within the Aspen HYSYS numerical environment. This software is typically used in the chemical industry [30], for petrochemical engineering calculations [31], in hydrogen-related installations [32], for carbon capture and storage [33], in modelling compressed air energy storage systems [34], for liquid air energy storage [35], and in other applications.

Aspen HYSYS is not widely used for HP modelling, despite being equipped with all the required functionalities. This software was chosen for its ability to perform calculations in off-design mode, a feature not widely described in the available literature. The developed model was subsequently used to analyse the HP performance.

The objective of this work was to provide a comprehensive mathematical description of the operating conditions and phenomena within the heat pump unit, facilitating its optimal utilisation, even under off-design conditions commonly encountered in urban heating systems. Poland has been considered a representative case for this work in Central-Eastern Europe. Additionally, the developed tool can assist in selecting appropriate HP units and accurately predicting their performance. The developed model was then used to predict the performance of the heating system under the green energy transformation. Another goal of this study was to evaluate the technical, environmental and economic impacts of integrating a sewage-source heat pump into an existing coal-based DH system. Two operational scenarios were analysed: the current configuration with two coal-fired boilers and a proposed hybrid system that includes a sewage heat pump. The analysis compared heat generation, fuel consumption and equivalent CO_2 emissions for both cases, demonstrating that introducing a heat pump could reduce annual coal use and lower equivalent CO_2 emissions. Furthermore, an economic assessment based on the levelised cost of heat (LCOH) was conducted to determine the cost-effectiveness and competitiveness of the proposed solution within the Polish DH sector.

2. Materials and methods

2.1. System considered

2.1.1. Sewage scenario

Heat pumps that use treated sewage as a heat source are dependent on both demand and supply factors. The supply side consists of waste heat from sewage, with varying temperature and mass flow over time. Typically, temperature fluctuations are seasonal [36] (weather-dependent) and generally range between 10–22°C [37]. Data on sewage and outside temperatures collected over 4 years in a specific city in Poland enabled the creation of a correlation between these variables (Fig. 1), which was used in this study.

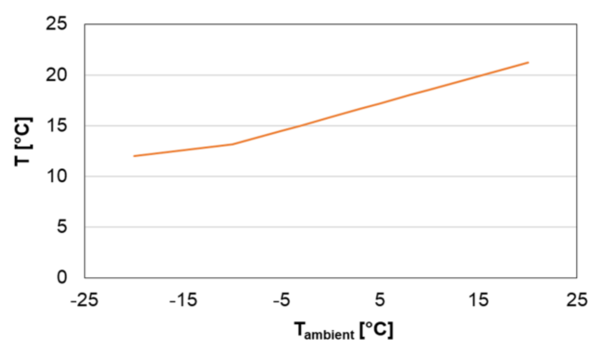

Fig. 1. Treated sewage temperature as a function of ambient temperature.

The amount of available sewage depends on the size of the agglomeration. However, it can be assumed that the daily sewage production per adult in Central-Eastern Europe is approximately 150 litres. Despite this, sewage mass flow is not constant and fluctuates throughout the day. Example relative data from one Polish city are shown in Fig. 2.

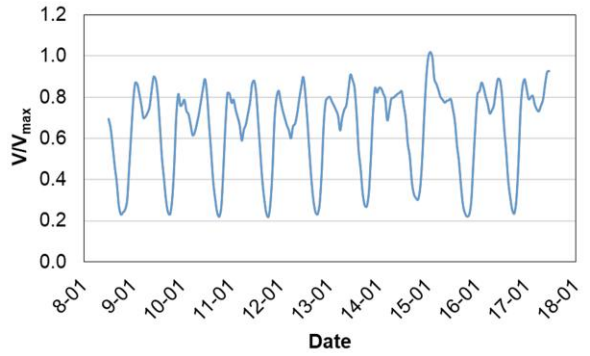

Fig. 2. Relative sewage flow fluctuations during a representative week.

On the other hand, there is the demand side. When considering heat pumps for integration with an existing DHN, they must meet network requirements. Therefore, high-temperature units should be considered [38]. Typically, the supply and return temperatures of DH water are determined by a regulatory table (Table 2). In Poland, high-temperature networks are typical; however, the transition to green energy may necessitate lowering the supply temperature, as in Scandinavian countries. Despite this, the heat pump must be capable of operating under these fixed conditions, taking into account both the available sewage parameters and the district heating system requirements.

Table 2. Selected values of the regulatory table for the district heating network in a Central-Eastern European city.

Operational point	1	2	3	4	5	6	7	8	9	10
Ambient temperature [°C]	20	10	8	5	3	0	−3	−5	−10	−20
DHN return [°C]	43.0	43.5	44.0	46.0	47.5	50.0	50.5	52.0	55.0	60.0
DHN feed [°C]	73	74	74	80	85	92	95	99	105	122

2.1.2. Heat pump

There are various configurations of the HP systems, starting from simple, relatively cheap constructions to more complex sets characterised by, i.e., higher COP. The simple system is typically a subcritical cycle with single-stage compression. The compressor can be a piston, scroll or centrifugal type, depending mainly on the unit's power [39]. This is currently the most widely used cycle in heat pumps. Although it achieves a high COP with a relatively low temperature difference, the maximum temperature rise is limited by the single-stage compressor's compression limit. Another characteristic feature is that as the temperature difference increases, expansion losses also increase, and their value depends primarily on the thermo-physical properties of the working medium, particularly its viscosity. To overcome these drawbacks, various HP configurations were developed [40]. A review of possible solutions can be found in [41]. The system, commonly found in industrial applications, is the HP cycle with two-step compression. In such a system, two compressors allow for a higher temperature lift. Furthermore, the gaseous fraction after the first reduction valve is combined with the gas from the first compressor, thereby reducing energy consumption during compression.

As the considered application requires a high-temperature output, a two-stage centrifugal compressor system was selected, similar to the one described in [42]. Such a system ensures a compromise between efficiency, the desired ΔT_{lift}, and system complexity, which is strictly related to investment costs [43]. The system used in this study is illustrated in Fig. 3.

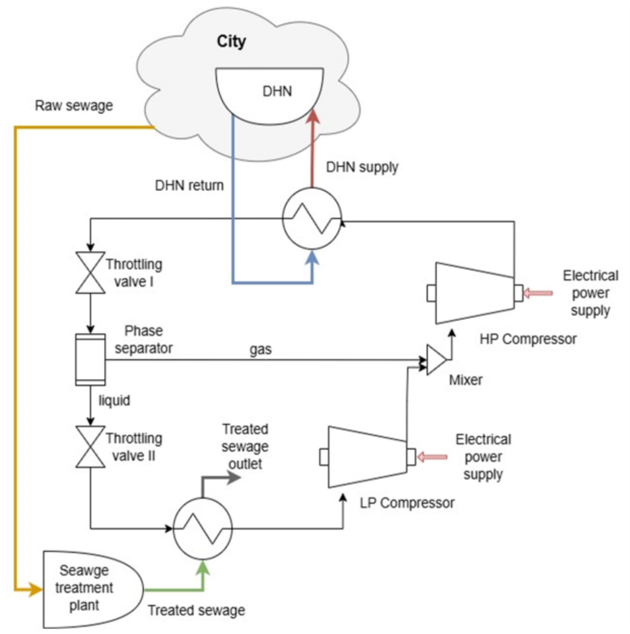

Fig. 3. Heat pump unit scheme.

2.1.3. Working fluid

For effective and safe HP operation, the proper selection of the working fluid (refrigerant) is essential. There are several factors determining its choice, which can be divided into categories [38]: thermo-physical: high critical temperature, low critical pressure; environmental: global warming potential (GWP) below 10, ozone depletion potential (ODP) = 0, future-proof regarding regulations; safety: non-toxic, non or low flammable, non-reactive with steel, copper, and aluminium; efficiency: high COP, even with high ΔT_{lift}, high volumetric heating capacity (VHC); and availability: low price, accessible in the market.

One of the popular refrigerants in European applications is R1234ze(E) [44]. This medium has a relatively low negative global warming potential [14] and simultaneously allows the desired phase-change temperatures to be achieved at a reasonable pressure. It has been classified as A2L [42] and is considered relatively safe. Its physical properties allow it to operate within the assumed temperature range. Several studies have highlighted the high energy efficiency of vapour-compression systems using this fluid, even at high condensation temperatures [7,51,52]. Taking the above into account, R1234ze(E) was selected for this research, as in [47].

2.1.4. Boundary conditions and connection with district heating networks

As boundary conditions, the sewage temperature and DHN water temperature at the return (HP inlet) were assumed to be functions of the outside temperature (Table 3). The selected values were based on experimental data and a regulatory table for a specific city in Poland. Ten operational points of the system were selected. The HP water-side outlet temperature should generally match the DHN feedwater temperature; however, due to HP constructional conditions during the winter season, several discrepancies occur. It will be discussed in the further part of this study.

Table 3. Boundary conditions for selected operational points of HP.

Operational point	1	2	3	4	5	6	7	8	9	10
Atmospheric temperature [°C]	20.0	10.0	8.0	5.0	3.0	0.0	−3.0	−5.0	−10.0	−20.0
Treated sewage temperature [°C]	21.2	18.5	18.0	17.2	16.6	15.9	15.0	14.5	13.2	12.0
DHN return [°C]	43.0	43.5	44.0	46.0	47.5	50.0	50.5	52.0	55.0	60.0
DHN feed [°C]	73.0	74.0	74.5	80.0	85.0	92.0	95.0	99.0	105.0	122.0

2.2. Equations

Numerical analysis encompassed the development of mathematical models, their implementation in numerical software, and the selection of initial and boundary conditions.

First, the energy balance equation and mass balance equation need to be fulfilled:

$$\sum_{i=1}^{i=N} \dot{m}_i h_i = \sum_{j=1}^{j=M} \dot{m}_j h_j, \quad (1)$$

$$\sum_{i=1}^{i=N} \dot{m}_i = \sum_{j=1}^{j=M} \dot{m}_j, \quad (2)$$

where N is the number of inlet streams, M is the number of outlet streams, $\dot{m}_i$ represents the mass flow rate at the inlet side [kg/s], $\dot{m}_j$ means the mass flow rate at the outlet side [kg/s], h_i and h_j are the specific enthalpies of each stream [kJ/kg].

In the system, two fluids are used: water (treated sewage and heating water) and refrigerant (R1234ze(E)) in the heat pump circuit. Due to the various properties of the fluids in the proposed case, two mathematical models of the media were selected. Water circuits were described using the ASME steam model [54], while the refrigerant cycle was modelled using the Peng-Robinson model of real gas in the form (3) [48]:

$$p = \frac{RT}{v-b} = \frac{a(T)}{v(v+b)+b(v-b)}, \quad (3)$$

where p is the pressure [Pa], R represents the individual gas constant [J/(mol·K)], T is temperature [K], v means specific volume [m^3/kg], and a and b are constants.

The energy balance equation for the shell side is expressed in Eq. (4):

$$\dot{m}_{shell} \cdot (h_{in} - h_{out})_{shell} - q_{loss} + q = \rho \frac{d(V \cdot h_{out})_{shell}}{dt}, \quad (4)$$

where $\dot{m}_{shell}$ is the mass flow rate at the shell side [kg/s], h_{in} and h_{out} are the inlet and outlet enthalpy of the flow [kJ/kg], q_{loss} represents heat losses [kW], q means heat provided to the shell side [kW], ρ is the density [kg/m^3], t represents time [s] and V is volume [m^3].

The energy balance equation for the tube side is given by (5):

$$\dot{m}_{tube} \cdot (h_{in} - h_{out})_{tube} - q_{loss} + q = \rho \frac{d(V \cdot h_{out})_{tube}}{dt}, \quad (5)$$

where $\dot{m}_{tube}$ is the mass flow rate at the tube side [kg/s], h_{in}, h_{out} are inlet and outlet enthalpies of the flow [kJ/kg], q_{loss} represents heat losses [kW], q is heat provided to the tube side [kW], ρ means density [kg/m^3], t is time [s], V means volume [m^3].

Heat exchanger power was described by Eq. (6):

$$Q = |m_{in} h_{in} - m_{out} h_{out}|, \quad (6)$$

where: Q means heat exchanged, m_{in} and m_{out} are inlet and outlet mass flow rates [kg/s], h_{in} and h_{out} are specific enthalpies of inlet and outlet streams, respectively.

The overall heat exchanger heat load (UA, W/K) can be expressed as [36]:

$$UA = \frac{Q}{LMTD}, \quad (7)$$

where Q represents the heat exchanger power [MW], $LMTD$ is the logarithmic mean temperature difference [K], which can be expressed as:

$$LMTD = \frac{\Delta T_{side\,1} - \Delta T_{side\,2}}{\ln\left(\frac{\Delta T_{side\,1}}{\Delta T_{side\,2}}\right)}. \quad (8)$$

In off-design mode, changes in the medium's volume flow rate affect the exchangers' heat load – Eq.(9), where $\dot{V}$ is the volumetric flow of the media entering the heat exchanger and subscripts denote design point conditions.

$$\frac{UA}{UA_{design}} = \left(\frac{\dot{V}}{\dot{V}_{design}}\right)^{0.8}. \quad (9)$$

The flow through the valves was calculated based on [49]:

$$\dot{m} = k\sqrt{\Delta p}, \quad (10)$$

where $\dot{m}$ is the mass flow rate [kg/s], k is the coefficient representing the inverse of flow resistance (conductivity) [$\sqrt{\mathrm{kg} \cdot \mathrm{m}}$], Δp is the pressure drop [Pa].

The mass flow rate was calculated as a function of inlet (p_{in}) and outlet pressure (p_{out}) and the coefficient C_v, which corresponds to the inverse of flow resistance (11):

$$\dot{m} = f(p_{in}, p_{out}, C_V). \quad (11)$$

The compressor assumed in this study is a centrifugal one [50]. Its electrical consumption power was calculated as shown in Eq. (12):

$$P = \dot{n}_{in} \cdot M \cdot \left(\frac{n}{n-1}\right) \cdot CF \cdot \left(\frac{p_{in}}{\rho_{in}}\right) \cdot \left[\left(\frac{p_{out}}{p_{in}}\right)^{\left(\frac{n-1}{n}\right)} - 1\right], \quad (12)$$

where P is the device's electrical consumption power [kW], $\dot{n}$ means the molar flow rate [kmol/s], M represents the molar mass of the working medium [kg/kmol], n is the volume exponent, p_{in} and p_{out} are the inlet and outlet pressures and ρ_{in} is the inlet density [kg/m^3].

The correction factor (CF) was Eq. (13):

$$CF = \frac{h'_{out} - h_{in}}{\left(\frac{n-1}{n}\right)\left(\frac{p_{out}}{\rho'_{out}} - \frac{p_{in}}{\rho_{in}}\right)}. \quad (13)$$

The power imparted to the working medium during the compression process is described in Eq. (14):

$$P = \dot{n}_{in} \cdot M \cdot (h_{out} - h_{in}). \quad (14)$$

In off-design mode, the maximum compressor pressure depends on the design pressure and the medium inlet temperature, as shown in Eq. (15), where r is the compression ratio, T is the temperature [K], and k is the adiabatic exponent:

$$r_{new} = \left[\frac{T_{inlet,design}}{T_{inlet,new}}\left(r_{design}^{\frac{k-1}{k}} - 1\right) + 1\right]^{\frac{k}{k-1}}. \quad (15)$$

The compressor was assumed to be a constant-speed machine, as is typical for high-power units. In some operating states, the heat pump is controlled by throttling. This is achieved in practice by changing the angle of the inlet guide vanes (IGV). Assumed characteristics of the efficiency change depending on the blade closure angle, based on data presented in [50], are shown in Fig. 4.

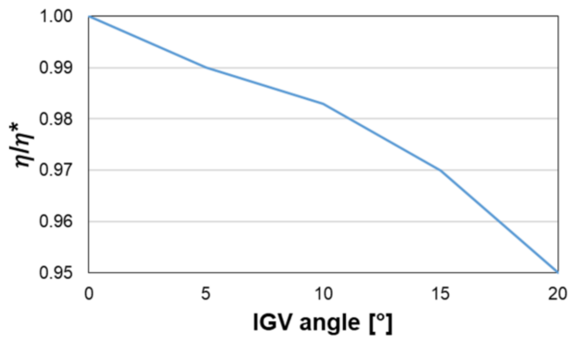

Fig. 4. Relative efficiency as a function of the compressor IGV angle (own work based on [50]).

As the medium's parameters change, its specific volume changes. Compressors, as displacement machines, operate at a constant volume flow. Reducing it requires choking. An example compressor throttling characteristic, used in this study, is shown in the chart (Fig. 5). The mathematical model presented above was implemented in the Aspen HYSYS numerical environment, which is designed explicitly for heat-flow calculations, including complex systems.

An economic assessment of the investment in water heat pumps was conducted based on the levelised cost of heat (LCOH) parameter [51].

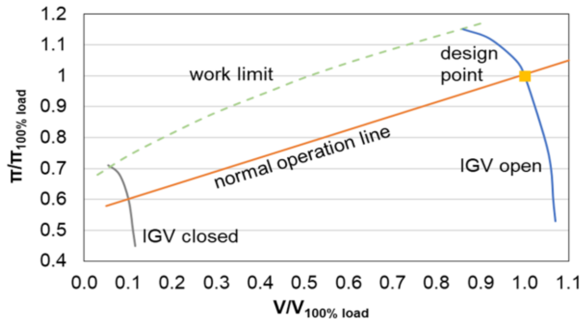

Fig. 5. Pressure ratio as a function of volumetric flow at the compressor inlet.

LCOH provides a comprehensive measure of the unit cost of heat production throughout the installation's lifecycle. This parameter accounts for investment expenses, operational costs, and system efficiency, enabling objective, consistent comparisons of technologies. Consequently, it serves as a key tool for the economic optimisation of heat source portfolios and informed investment decision-making, which is why it was selected as an assessment parameter in this study. LCOH was calculated using Eq. (16):

$$LCOH = \frac{\sum_{i=0}^{N}\left(\frac{I_i + M_i}{(1+r)^i}\right)}{\sum_{i=0}^{N}\frac{Q_i}{(1+r)^i}}, \quad (16)$$

where I_i represents investment expenditures in year i [EUR/a] (CAPEX and OPEX), M_i are operating expenditures and operational costs [EUR/a], Q_i is heat production in year i [EUR/a], r means discount rate, and takes a value of 5% in this study as in [52], and N represents the lifetime of the installation (economic operational period), 20 years.

2.3. System balance model

Small heating plants in Poland are still predominantly based on water-coal boilers, such as WR-10. This study analysed the integration of systems utilising such sources with a heat pump, where the lower heat source is treated wastewater. Based on the typical heating load of a system for a medium-sized city under Polish weather conditions (Fig. 6), a balance model with an hourly resolution was developed.

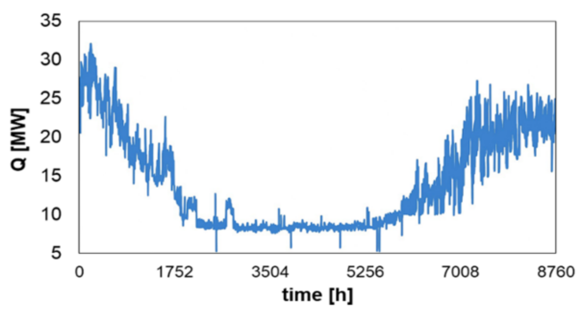

Fig. 6. Heat demand in the system over the year.

The system scheme is shown in Fig. 7. In each hour of system operation, the amount of heat produced by each source was determined. Priority was given to the heat pump installation as a renewable source, provided treated wastewater was available as the lower heat source, and its temperature allowed achieving the desired supply water parameters in the network. Two WR-10 boilers operated as supplementary sources, with constraints such as the power range (3−15 MW). The size of the heat pump installation was selected based on the potential for utilising sludge waste heat (3 MW).

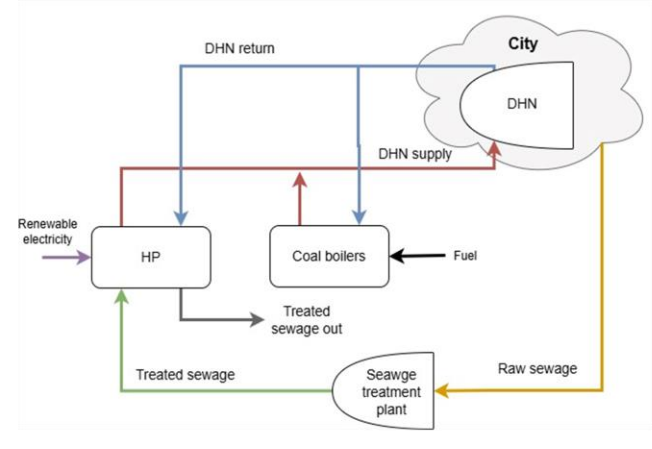

Fig. 7. Scheme of the analysed system.

3. Results and discussion

Determining the HP design point is a critical yet complex task. It is influenced by various factors, including the maximum temperature lift, mass flows of sewage and district heating water, and their variations, as well as the anticipated temperature ranges of the operating mediums. The selection of design-point parameters has a significant impact on the HP unit's operating conditions, as these parameters are interdependent. This section analyses how the variability of selected parameters affects the operation of the HP unit under the given conditions, using a developed mathematical model. The data obtained from the HP modelling were then used in the system balance model to predict the annual heat production by source and the environmental impact of the implemented changes (equivalent CO_2 emission reduction).

3.1. Designed temperature lift

First design condition, which should be determined, is temperature lift, defined as the maximum temperature difference between the heating water outlet and treated sewage inlet temperatures:

$$\Delta T_{lift} = T_{H,out} - T_{L,in}. \tag{17}$$

The maximum values of ΔT_{lift} occur during the winter season, when sewage temperatures are at their lowest. The required heating water temperatures are at their highest (Fig. 8). Conversely, considering the thermodynamic cycle in heat pumps, the highest value of ΔT_{lift} is achieved with the most significant difference between evaporator and condenser pressures, which corresponds to the highest compression rate. Based on this outcome, the system should be designed to operate in winter conditions. During the summer season, the pressure ratio can be reduced, for instance, by throttling; however, if the system is designed with a pressure ratio that is too low, there will be no way to increase it when needed. In this analysis, the design point was assumed to correspond to winter conditions.

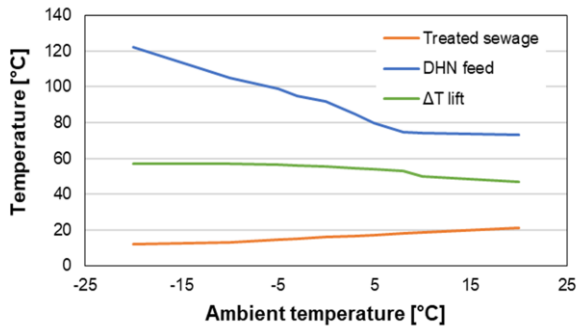

Fig. 8. Treated sewage, DHN feed and ΔT_{lift} temperatures as a function of ambient temperature.

3.2. Outflow temperature

The second key parameter of the design point is the outflow temperature (district heating supply water at the outlet of the HP module), which significantly influences the system's operating parameters, such as the coefficient of performance (COP). It should also be adjusted to reflect the evolution of district heating systems, which have progressed from the first generation based on steam to today's fifth-generation low-temperature networks, illustrating a consistent trajectory toward higher energy efficiency, reduced distribution losses, and greater integration of renewable and waste heat sources. While the first and second generations relied on high-temperature steam and pressurised hot water systems fuelled by fossil energy [52,53], subsequent generations progressively lowered operating temperatures (to around 70–100°C in the third generation) [54]. They introduced technological innovations such as pre-insulated pipes and automatic control systems [53]. The fourth generation district heating (4 GDH) marked the shift to low-temperature networks (30−70°C) based on renewable energy and smart control technologies [12], whereas the fifth generation (5 GDHC) expanded this concept to include both heating and cooling within decentralised, bidirectional networks operating near ambient temperatures (10–40°C), where buildings can act as prosumers of heat [54]. In the light of increasing demands for energy efficiency, decarbonisation and system flexibility, existing district heating networks now face the urgent need to transform toward 4 GDH and 5 GDHC solutions. This transformation requires integrating heat pumps, thermal storage, waste and low-temperature heat sources, along with smart management systems that enable dynamic balancing of supply and demand while minimising the use of solid fuels, which are key steps toward achieving a sustainable future for urban heating.

Given the dynamic development of district heating networks and current local network requirements (3 GHD), two operating scenarios for the heat pump module were selected and analysed under specific weather conditions. The analysis was divided into two scenarios (see also Table 4):

- **Scenario 1**: A higher upper heat source temperature (maximum 95°C), adjusted for high-temperature networks (2 GDH).
- **Scenario 2**: A lower upper heat source temperature (maximum 75°C), suitable for low-temperature district heating networks (3–5 GDH).

Table 4. Design parameter points for the analysed scenarios.

Parameter	Scenario 1	Scenario 2
DHN supply temperature [°C]	95.0	75.0
Sewage inlet temperature [°C]	12.0	
ΔT_{lift} max	89.0	69.0
Total pressure ratio	10.7	7.6
LP compressor pressure ratio	3.3	2.8
HP compressor pressure ratio	3.3	2.8
Evaporator pinch point temperature [°C]	3.3	
Condenser pinch point temperature [°C]	5.0	
Adiabatic efficiency of the compressors [%]	75	

Figure 9 compares the district heating water temperatures provided by the HP unit with the requirements in the regulatory table. The heat pump system is primarily designed as a low-temperature source due to its specifications. Heating the network water to meet regulatory requirements for temperatures below −3°C would significantly decrease efficiency. This is partly due to the need to design the system for conditions that occur for only a few hours annually. Therefore, the maximum source temperature at the network water outlet was set to 95°C in scenario 1. While this configuration accommodates most conditions, particularly during very low outside temperatures, achieving higher desired system supply temperatures may still require additional measures.

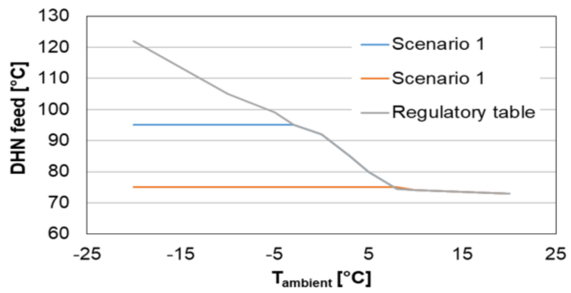

Fig. 9. District heating water supply temperature as a function of ambient temperature.

In scenario 2, the maximum network water temperature at the HP outlet is limited to 75°C. This configuration is sufficient to meet high-temperature network requirements for over half the year and serves as an example of a system suitable for low-temperature networks. Such networks are expected to become increasingly common in Poland in the future and are already successfully implemented in several European countries, including those in Scandinavia.

During periods when higher temperatures are needed, district heating water at the supply side is heated using an auxiliary heat source, such as an electric boiler or other supplementary systems, to cover the heat load. To compare these scenarios, the systems' operations in both configurations were simulated.

Figure 10 illustrates COP for each analysed scenario as a function of the outside temperature. While scenario 2 achieves a higher COP, it requires additional heating over a more extended period throughout the year or necessitates investments in network transformation, including demand-side adaptations. The results clearly indicate that lowering the design temperature enhances the overall system efficiency, reducing electricity consumption and improving the economic performance of the heat pump installation. Similar trends have been reported in the literature, where a decrease in the required supply temperature of district heating networks led to a noticeable improvement in heat pump efficiency. For instance, Degelin et al. [10] showed that reducing the supply temperature from 55°C to 45°C can increase COP by approximately 33%. Likewise, Pospíšil et al. [55] observed a correlation between the temperature difference between the water temperature at the condenser outlet and the air temperature at the evaporator inlet. When the measured temperature difference remains below 30°C, COPs above 4 were obtained.

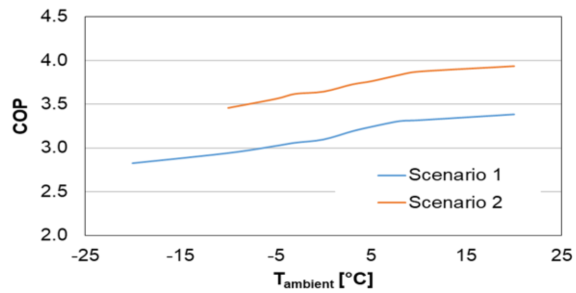

Fig. 10. COP as a function of ambient temperature.

A seasonal COP (sCOP) was used as a comparative parameter, calculated as the average annual COP, weighted by the number of hours at the selected system operating point (Eq. 18). For each scenario, the year was divided into ten periods (Table 5).

$$\text{sCOP} = \frac{\sum \text{COP}_i \cdot t_i}{t_{tot}}, \tag{18}$$

where COP_i is the value of COP at time range i, t_i is the amount of hours installation is being operated with COP_i [h], and t_{tot} is the total amount of hours when the installation is operated [h].

Table 5. Data for sCOP calculations.

Operational point	1	2	3	4	5	6	7	8	9	10
Ambient temperature [°C]	20.0	10.0	8.0	5.0	3.0	0.0	−3.0	−5.0	−10.0	−20.0
Amount of hours [h]	2560	1712	789	763	865	1031	452	284	278	26
COP_i scenario 1	3.47	3.41	3.36	3.30	3.21	3.14	3.07	3.06	2.99	2.91
COP_i scenario 2	3.87	3.85	3.74	3.65	3.66	3.50	3.52	3.46	3.38	3.25

In the analysed case, sCOP equals 3.32 in scenario 1, which complies with the literature. For scenario 2, sCOP was 3.72, which is significantly higher; however, this HP could be operated for a limited time over the year. Considering the Polish technical and weather conditions, scenario 1 was taken as the base model for further calculations. In the literature, sCOP for treated sewage HP equals 3.29 [36] and 3.3–3.5 [4]. It means that even the sCOP value obtained for scenario 1 remains at a similar level.

3.3. Treated sewage mass flow

Using the previously described mathematical model, various scenarios of the HP unit's operation were analysed, ranging from minimum to maximum sewage flow under winter conditions, which were selected as less favourable. The operational data presented in Fig. 2 indicated a relatively high daily variability of sewage flow. The variability characteristics were relatively constant throughout the year, in repeatable daily fluctuations. The minimum flow occurs for approximately 5–6 hours a day.

Designing the heat pump module for minimum flow requires throttling the compressors in operating conditions with higher output, which occurs most of the time. Throttling reduces compressor efficiency and COP, so it should be avoided whenever possible. Designing the system for maximum flow would entail the risk of failing to meet the network water parameters at lower flows (low pressure, making it impossible to achieve the desired temperature on the network water side at the HP module outlet).

To examine the impact of changes in the mass flow of treated sewage on the HP module's operation, three simulations were performed at 100% (max. mass flow), 70% and 30% (min. mass flow). The simulation results are shown in the form of graphs of the dependence of COP and thermal power on ΔT_{lift} (Fig. 11a and Fig. 11b, respectively).

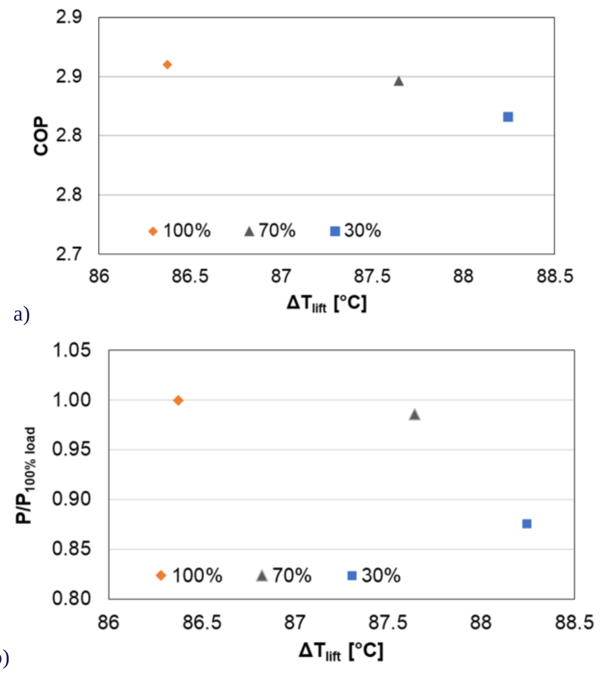

Fig. 11. COP (a) and heating power (b) as a function of ΔT_{lift}; treated sewage mass flow: 100%, 70%, 30%.

For the average and maximum flows, the assumed network water-side parameters (60/95°C) were met. In the case of minimum flow, due to the specific operation of heat pump units and the need to maintain a minimum temperature difference (pinch point) on the exchangers, the temperature of the network water at the outlet was 90°C. The decrease in flow was associated with a slight decrease in COP. In contrast, in the minimum flow case, the unit power drops noticeably (lower treated sewage flow is associated with lower manageable waste heat, as wastewater cannot be cooled below 6°C for design and operational reasons).

The power drop is expected only in winter conditions with low sewage flow (night conditions). Analysis of actual data on sewage flow and temperature indicates that this situation occurs for approximately 300 hours a year, mainly during winter months and at night (when temperatures are low and sewage flow is minimal). In other operating conditions (higher external temperature and/or higher sewage flow), the assumed unit power will be maintained. Such conditions occur for at least 8 000 hours per year.

3.4. Partial load states

The operation of the HP unit depends not only on the availability of treated sewage but also on the heating network's requirements. Due to demand-side constraints, the unit will have to operate at partial load (the system will not be able to absorb heat at the rate the HP could produce under the given conditions) or cooperate with the heat store [56]. In the first case (no store), the HP unit can be operated at partial load by throttling (closing the IGV guide vanes). The analysis aimed to determine the impact of HP unit operation at partial load on COP. Three simulations were performed for load levels of 100%, 75% and 50% under selected operating conditions: an external temperature of 8°C and average flow. As shown in the graph (Fig. 12), varying the HP module load from 50% to 100% has little effect on COP.

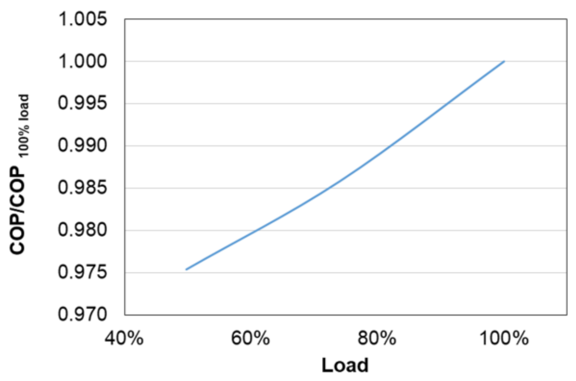

Fig. 12. Relative COP as a function of relative load for the selected operating condition and average flow, at external temperature of 8°C.

This is a consequence of the overlap of several competing characteristics. A change in the flow of the working medium in the system necessitates throttling the compressors, thereby decreasing compressor efficiency. Within the analysed range, efficiency also decreases with increasing pressure. As a result, the impact of compressor behaviour at reduced load should reduce COP. However, in the evaporator, less heat can be received from the same treated sewage stream, so the temperature of the treated sewage leaving the evaporator increases as the load decreases. As a result, ΔT_{lift} decreases while maintaining the upper source parameters at a constant level (Fig. 13). These are phenomena that increase COP. Due to the comparable impact on the system, COP remains almost constant (in the example considered, decreasing by 0.08 when the load is changed from 100% to 50%).

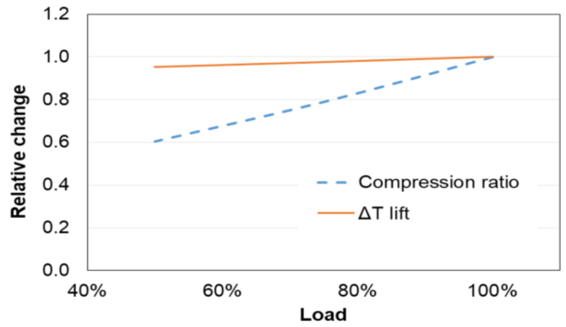

Fig. 13. Relative compression ratio and ΔT_{lift} as a function of relative load for the selected operating condition and average flow, at external temperature of 8°C.

3.5. District heating system analysis

Two district heating system (DHS) scenarios of heat supply in the analysed system were evaluated:

- **DHS Scenario 1:** Two existing coal-fired water boilers with a capacity of 15 MW, without a heat pump (current state),
- **DSH Scenario 2:** Two existing coal-fired water boilers with a capacity of 15 MW and a sewage heat pump with a capacity of 3 MW (proposed).

The heat generation by sources for each scenario is illustrated in Fig. 14. As shown, using waste heat from sewage in a small city, combined with a heat pump, can typically meet domestic hot water demand during the summer season (Fig. 13). However, when heat demand exceeds 3 MW, the heat pump must be switched off. This is because the difference between heat demand and the heat pump's capacity is insufficient to maintain the technical minimum for coal boiler operation.

To improve heat pump utilisation, a different energy source mix, such as incorporating an electric boiler, could be considered. However, this approach poses challenges, including the need for additional power supply agreements. Given the occasional use of an electric boiler, such arrangements could lead to high costs and reduce the system's economic viability. While a heat storage system could be introduced, it would result in frequent on-and-off cycling of the coal-fired boiler, which is technically discouraged.

During peak heat-demand periods (typically winter), the heat pump is not used due to its inability to achieve the required output temperatures and its relatively low COP. This period can instead be utilised for scheduled maintenance work.

The heat generated by each source is summarised in Table 6. The environmental benefits of implementing Scenario 2, measured by equivalent CO_2 emission reduction, were calculated using the CO_2 emission indicator from source [57].

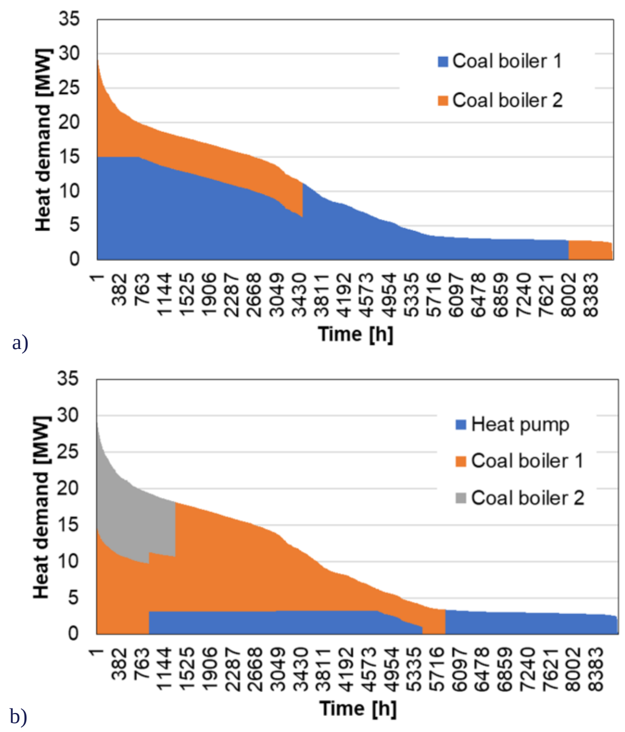

Fig. 14. Heat supplied by the source, ordered chart: a) scenario 1, b) scenario 2.

Table 6. Annual heat generation by source.

Source	DHS Scenario 1	DHS Scenario 2
Coal boiler 1	229.4 TJ	178.3 TJ
Coal boiler 2	77.2 TJ	47.5 TJ
Heat pump	0	80.7 TJ (electricity consumed 6.22 GWh)

The proposed system transformation results in a significant reduction in CO_2-equivalent emissions, estimated at approximately 9500 tonnes annually compared to the baseline scenario (Table 7). This value assumes that the electricity used to power the HP is 100% green. This reduction primarily results from decreased coal consumption and from substituting fossil-based heat generation with renewable electricity powering the heat pumps. Such a shift not only lowers the district heating system's carbon intensity but also aligns with broader decarbonisation and energy transition goals.

Table 7. Fuel consumption and CO_2-equivalent emission reduction.

	DHS Scenario 1	DHS Scenario 2
Coal boiler 1 [TJ/year]	270.0	205.6
Coal boiler 2 [TJ/year]	90.3	54.8
Total coal consumption [TJ/year]	360.2	260.4
Difference [TJ/year]	99.8	
Emission indicator [kg/GJ]	95.05	
Equivalent CO_2 emission reduction [t/year]	9486.44	

3.5. Economic analysis

The investment expenditures were estimated based on data from the literature for similar investments. They are presented in Table 8. The estimation was made using unit economic expenditures (Fig. 15). In this study, CAPEX (capital expenditures) was assumed to be 4.8 million EUR. The OPEX (operating expenditures) was 3.5% of the CAPEX yearly. The electricity cost was 155 EUR/MWh, while the sewage heat price was 2.5 EUR/MWh. The levelised cost of heat for a heat pump is 19.35 EUR/GJ. This value can be compared with other prices available in the literature for the analysed region. In [58], the 25-year horizon is considered, and the average heat process varies from approximately 7 EUR/GJ to nearly 19 EUR/GJ. The presented systems are primarily based on fossil fuels, cement plants and municipal waste. In [59], LCOH for two heating scenarios is presented: in the centralised one (heat pump and biomass boiler, with auxiliary operation of an existing coal boiler), LCOH was 20.7 EUR/GJ. In the distributed system (modules consisting of PV panels, heat pumps, an electric boiler and a heat store), LCOH was higher at 85.6 EUR/GJ. It can be concluded that the obtained result falls within the standard price range for district heating in Poland. The obtained result can also be compared with prices in other countries. For example, in Helsinki, Finland, heat prices vary from 11.3 EUR/GJ (in summer) to 33.9 EUR/GJ (in winter) [60].

Table 8. CAPEX for water heat pumps.

Location	Year	Nominal power [MW]	CAPEX [MLN EUR]	Source
Helsinki, Finland	2025	90	100	[69]
Wroclaw, Poland	2024	12.5	19.3	[70]
Vienna, Austria	2023	110	70	[71]
Hamburg, Germany	2023	20	15	[72]
Viborg, Denmark	2021	90	41	[73]

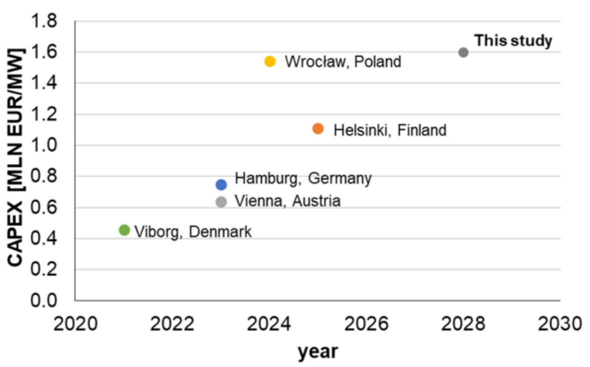

Fig. 15. Unit CAPEX for water heat pumps.

To show the impact of CAPEX and the electricity prices on LCOH, the sensitivity analysis was conducted. Changes of both within ±25% were considered. Results are presented in Table 9. The electricity price has a greater influence on the final LCOH value, although both factors have comparable impacts on the final results.

Table 9. Sensitivity analysis of LCOH.

LCOH [EUR/GJ]		Electricity price		
		−25%	**base**	**25%**
CAPEX	**−25%**	14.65	17.63	20.62
	base	16.36	19.35	22.33
	25%	18.08	21.06	24.04

4. Conclusions

The green energy transition necessitates changes not only in electricity generation but also in the sources of heat production. One potential solution for cities with existing district heating systems is the use of heat pumps powered by treated sewage. Due to the unique operating conditions of heat pumps in urban systems, each case must be analysed individually. In this paper, a mathematical model of the processes occurring in an HP with centrifugal compressors is presented. This model serves as a valuable tool for analysing HP units.

Based on a literature review, the heat pump configuration and working medium were selected. Since the unit uses treated sewage as a heat source and operates within a district heating system, a high-temperature lift is required, along with a significant power output (on the order of several megawatts). This demand exceeds the capabilities of a single-compressor, simple unit due to insufficient pressure ratios. Therefore, a two-stage configuration was chosen, as it enables higher temperature lifts and improved performance under high-demand conditions. This configuration strikes a balance between efficiency, investment costs and technical complexity. Centrifugal compressors were selected for the desired power range. The working medium R1234ze(E) was chosen for its suitable temperature ranges, reasonable pressure levels and relatively low environmental impact.

The implemented model was used to assess HP unit performance under various operating conditions typical of a city in a Central-Eastern European climate. Winter conditions were assumed as the design parameters, with an external temperature of −20°C, network water parameters within the HP module at 60°C and 95°C, and a sewage temperature of 12°C. Design point parameters for winter conditions correspond to the highest ΔT_{lift} required, resulting in the highest compression ratio in the HP circuit.

Modelling part-load states for large sewage heat pumps is critical, as these systems operate under highly dynamic conditions in which variable sludge availability significantly affects heat extraction, leading to transient performance changes that steady-state, full-load models cannot accurately capture [27]. Fluctuations in heat demand necessitate models that account for rapid shifts in operational loads and corresponding changes in the system's coefficient of performance [61]. Furthermore, network requirements impose additional constraints that demand flexible, modulating operation to support demand response and stability in power networks. Detailed part-load state modelling thus enables an accurate representation of operational parameters under varying load conditions, ensuring that system control strategies remain robust in real-world applications. This comprehensive modelling approach is essential for optimising energy management, enhancing operational flexibility, and ensuring system resilience in response to rapidly changing environmental and demand scenarios. Part-load state modelling in off-design mode is essential for large sewage heat pumps to address fluctuations in sludge supply, occupancy-driven heat demand, and network integration challenges. This approach ensures that dynamic operating conditions are represented more accurately than in simple balance models. Heat pump systems operating at partial loads exhibit minimal COP degradation. This suggests that HPs can operate efficiently without heat storage systems, although storage may enhance system flexibility and performance.

The seasonal coefficient of performance (sCOP) values demonstrate that heat pumps can achieve higher efficiency in low-temperature district heating networks. There is a relatively small difference in the sCOP between a system operating with a network water heating outlet temperature of 95°C (scenario 1) and a system operating with a reduced outlet temperature (75°C, scenario 2). Under current district heating conditions, it is recommended to design the system according to scenario 1 to maintain operational flexibility. However, when planning future modernisations or designing new networks, low-temperature networks should be considered. These networks are more effective, with reduced heat losses, higher COP values, and better compatibility with renewable heat sources, such as HPs. Modernising district heating systems to accommodate lower feed temperatures could significantly improve overall efficiency and compatibility with renewable sources.

The system should be designed to operate with treated sewage mass flow rates close to the average. Low flow rates, which occur only during a short period of the year, can be offset by higher flow rates and power output during summer. Operating the HP unit under partial loads, as determined by demand-side conditions, does not significantly worsen COP. This indicates that HP can operate effectively even in systems without heat storage, although incorporating heat storage could still enhance system performance.

An environmental analysis of heat pump operation in a typical district heating system for a small city in Poland, representative of Eastern European cities, was conducted. The amount of waste heat in sewage is suitable for a domestic hot water supply in the summer season. There are periods when the technical minimum load of other existing sources limits HP's operation. Despite that, HP implementation may help reduce CO_2-equivalent emissions by nearly 9,500 tonnes annually.

The obtained LCOH value of 19.35 EUR/GJ indicates that the proposed system remains economically competitive within the typical price range for district heating in the region. The sensitivity analysis confirmed that the electricity price has a greater impact on LCOH than investment costs, although both parameters have a comparable influence on the overall result. These findings demonstrate that integrating sewage-source heat pumps can provide cost-effective, sustainable heat generation, particularly under stable electricity prices and favourable investment frameworks [62].

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