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Abstract:
Energy planning tools often consider a constant coefficient of performance (COP) for large-scale heat pumps (HPs) over the entire year, even though the optimization is usually performed on hourly basis. Because the HP performance is highly dependent on ambient conditions, the constant COP may not represent real implementation well. Ambient temperature levels can be included by assuming simple representations of COP based on a Carnot or a Lorenz cycle. These simple representations of COPs are used, because implementing a thermodynamic HP model in the energy planning software may be complicated or would result in values limited to a specific HP application. Other options of estimating the COP are the use of more advanced COP estimation methods.

This paper compares COP estimations based on four different methods to the COP obtained using a thermodynamic HP model. Four different heat sources and varying district heating supply temperatures were considered. Furthermore, the results of implementing the different COP estimations in an energy planning tool were analysed. As a case, the planning tool was used to assess HP implementation in a new development district in Copenhagen, Denmark. The change in seasonal COP of the HPs as well as the economic results and optimal HP capacities were compared.

The results show that the Jensen COP estimation method for design conditions provide good approximations compared to a thermodynamic HP model. The results indicate that this method is suitable for the use in energy planning tools. Assuming a constant Lorenz efficiency, constant exergy efficiency or constant COP over the year resulted in very large deviations in COP, especially when the operating conditions differ considerably from the design conditions. Consequently, different heat sources were found for the most economical optimum. The accuracy of the three methods was highly dependent on the initial guesses of constant COP and efficiencies.

Keywords:
COP estimation, District heating, Energy planning, Large-scale heat pumps, Low-temperature heat sources.
1. Introduction

Large-scale heat pumps (HPs) are proposed as a technology for future energy systems to help integrate a high share of renewables, such as wind power, into the energy system by coupling the power and district heating (DH) sectors [1,2].

Energy planning tools are often used to investigate the most economic and/or sustainable supply of energy, being heat or electricity, for a country, region, city or new development district, using some kind of optimization routine. In such tools, HPs are often represented in a simplified way, which makes it faster to solve the optimization problem than if a thermodynamic HP model was included. Lund et al. [3] used the energy planning tools EnergyPlan [4] and MODEST [5] for their analysis of integrating large-scale HPs in Denmark. As in many other studies [6–10], they assumed a constant coefficient of performance (COP) of the HPs. Lund et al. [11] and Østergaard and Andersen [12] used the energy planning tools EnergyPlan [4] and energyPRO [13], respectively. In both studies, the COP was related to an ideal Lorenz cycle and multiplied by a constant Lorenz efficiency. In this way, seasonal changes of inlet and outlet temperatures of the heat source and heat sink were taken into account.

Several studies show how the COP may be calculated or estimated based on different approaches. Jensen et al. [14] derived an equation for COP based on exergy efficiency, considering the heat source and sink stream temperatures above the dead state. Oluleye et al. [15] and Oluleye et al. [16] derived linear correlations of the Carnot efficiency of a HP for six different refrigerants. Coefficients were provided for condensing temperatures above 50 °C and evaporation temperatures above 10 °C. Jensen et al. [17] derived a generic equation to estimate the COP of a one-stage HP cycle in design conditions. The COP estimation depends on the heat source and heat sink temperatures as well as characteristics of the compressor, heat exchangers and refrigerant. For most of the parameters, reasonable values may be easily assumed. Simple linear approximations are given for those factors that are more difficult to estimate without a thermodynamic HP model. Furthermore, they showed that the Lorenz efficiency of a HP with fixed component characteristics decreases for a smaller temperature lift. The temperature lift was defined as the difference between the absolute mean temperatures of the heat source and the heat sink. Therefore, assuming a constant Lorenz efficiency for annual calculations may lead to an overestimation of COP for some periods.

For this study, the four mentioned COP estimations were compared to a thermodynamic HP model by calculating the hourly COP of HPs based on four different heat sources over one year. Furthermore, the different COP estimations were implemented in an energy planning model, which was used for investigating possibilities of installing large-scale HPs to supply DH to a development district in Copenhagen, Denmark. The model is based on mixed-integer linear programming and identified the most economical HP capacities depending on the heat source for supplying DH to the new development district. This new residential and commercial development area, Nordhavn, will be gradually expanded until 2060 to accommodate 40,000 inhabitants and 40,000 jobs in a floor area of 3.5 million m² [18]. Based on a preliminary study for this area, four different heat sources were relevant to integrate. The heat sources were ambient air, groundwater, sewage water and seawater. The results of using different COP estimations in the planning model were analysed in detail.

2. Method

First, the development of a thermodynamic HP model is presented for design and off-design conditions. Then, the four different COP estimation methods are described, followed by a description of the energy planning tool, the case study and considered heat sources.

2.1. Thermodynamic HP model

A thermodynamic HP model was developed in the software Engineering Equation Solver (EES), version 10.478 [19] considering the different state points of the heat pumping process, as indicated in Figure 1 (b). A two-stage HP with open intercooler was modelled, which ensures lower pressure ratios and discharge temperature out of the compressor than a single-stage HP. Ammonia was chosen
as refrigerant, because it has neither global warming nor ozone depletion potential [20]. Two-stage HPs with ammonia as refrigerant have been widely used in DH [21]. First, a HP model was developed for dimensioning the HP in design conditions. Afterwards, the dimensioning parameters were used as inputs to a HP model for off-design conditions, in which the temperatures of heat source and heat sink varied.

2.1.1. HP design model

The design model was used to calculate the thermal conductance, i.e., UA-values, of the evaporator and condenser as well as the displacement rate of the compressors for design conditions, as specified in Table 1. The heat exchangers were modelled as counter-flow configuration. The heat transfer process on the condenser was modelled as three parts consisting of de-superheating, condensing and sub-cooling, each satisfying the energy balances between the refrigerant of the HP and the DH water. A heating capacity of 80% of the hourly peak heat demand for the case study was assumed. Further, the compressor heat loss was neglected.

**Table 1. Design conditions for HP dimensioning**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating capacity</td>
<td>16</td>
<td>MW</td>
<td>Air inlet temperature</td>
<td>−12</td>
<td>°C</td>
</tr>
<tr>
<td>DH Supply/return temperature</td>
<td>85/35</td>
<td>°C</td>
<td>Seawater inlet temperature</td>
<td>4</td>
<td>°C</td>
</tr>
<tr>
<td>Superheat</td>
<td>0</td>
<td>K</td>
<td>Groundwater inlet temperature</td>
<td>10</td>
<td>°C</td>
</tr>
<tr>
<td>Compressor heat loss</td>
<td>0</td>
<td>-</td>
<td>Sewage water inlet temperature</td>
<td>11</td>
<td>°C</td>
</tr>
<tr>
<td>Built-in volume ratio (all/air)</td>
<td>2.2/3.0</td>
<td>-</td>
<td>Heat source temperature difference</td>
<td>6</td>
<td>K</td>
</tr>
<tr>
<td>Isentropic efficiency, ( \eta_{is,max} )</td>
<td>0.80</td>
<td>-</td>
<td>Seawater temperature difference</td>
<td>3</td>
<td>K</td>
</tr>
<tr>
<td>Volumetric efficiency</td>
<td>0.90</td>
<td>-</td>
<td>Pinch point temperature differences</td>
<td>5</td>
<td>K</td>
</tr>
</tbody>
</table>

The built-in volume ratio, \( v \), and the intermediate pressure were optimized for design conditions to maximize COP. Properties of the different state points, such as temperature, enthalpy or entropy were determined in EES. An overview of the modelled HP cycle for design conditions is shown in Figure 1 as a temperature-heat load \((T\-\dot{Q})\) diagram \((a)\) and a pressure-enthalpy \((p\-h)\) diagram \((b)\) for sewage water as the heat source. The de-superheating (DSH), condensing (Cond) and sub-cooling (SC) part as well as the pinch point temperature difference (PP), DH supply (DH\(_s\)) and return (DH\(_r\)) temperature and state points are indicated. The refrigerant was sub-cooled by the DH return temperature down to a temperature difference of 5 K.

**Figure 1.** \( T\-\dot{Q} \) diagram \((a)\) and \( p\-h \) diagram \((b)\) for HP cycle using sewage water as heat source

2.1.2. HP off-design model

The HP model was modified to allow calculations in off-design conditions, meaning when the heat source and sink temperatures varied. The isentropic efficiency of the compressors was modelled by a relation developed for screw compressors [20], which takes into account the pressure ratio.
(\(p_{\text{HP}}/p_{\text{LP}}\)), the built-in pressure ratio (\(\pi = \nu^\kappa\)) and the polytrophic exponent \(\kappa\), as shown in (1).
Thereby, losses due to the mismatch between built-in pressure ratio and the actual pressure ratio were considered.

\[
\eta_{\text{is}} = \eta_{\text{is,max}} \frac{(p_{\text{HP}}/p_{\text{LP}})^{\frac{(\kappa-1)}{\kappa}} - 1}{\frac{(\kappa-1)}{\kappa} \pi^{1-1/\kappa} (\pi - p_{\text{HP}}/p_{\text{LP}}) - 1}
\]  

Part-load behaviour was not modelled. In off-design, the UA-values of the evaporator, the sum of the values of the condenser unit (DSH, Cond, SC) and the displacement rates of the compressors were kept constant. Instead, the intermediate pressure, the condensing temperature and the evaporation temperature were free variables. Furthermore, the low-pressure and high-pressure compressor speed (rpm) were synchronized, but could vary in off-design conditions, due to variations in heat source and heat sink temperature.

2.1.3. Considerations for air-source HP

The pressure ratios changed considerably over the year when ambient air was used as the heat source, because of the large temperature change from winter to summer. This would result in very low isentropic efficiencies of the compressor during summer periods. A different control or design may be chosen for this case and therefore, the relation from (1) was not used for the air-source HP at all. Instead, a constant isentropic efficiency of 0.80 was assumed representing a best-case scenario.

2.2. COP estimation methods

The four COP estimation methods are described in the following. Design parameters for two of the applied COP estimation methods were based on the results of the thermodynamic model. This was done to allow a fair comparison of the different estimation methods and to the thermodynamic HP model. If such model was not available, guesses must be made for the COP estimations based on reference values found in literature or from suppliers. Those values could differ considerably from the actual ones.

2.2.1. Constant COP

The COPs calculated based on the thermodynamic HP model for design conditions were used as constant COPs for the HPs of the different heat sources.

2.2.2. Constant Lorenz efficiency

This estimation was based on the COP of a Lorenz cycle which was multiplied by a constant Lorenz efficiency, as shown in (2). A Lorenz cycle takes the inlet (\(i\)) and outlet (\(o\)) temperatures of heat source (\(C\)) and heat sink (\(H\)) into account, as shown in (3). The constant value of the Lorenz efficiency was determined based on the design conditions of the thermodynamic model.

\[
\text{COP}_{\eta} = \eta_{L,m} \text{COP}_{L} = \eta_{L,m} \frac{T_H}{T_H - T_C}
\]

\[
T_H = \frac{T_{H,o} - T_{H,i}}{ \ln \left( \frac{T_{H,o}}{T_{H,i}} \right) }, \quad T_C = \frac{T_{C,i} - T_{C,o}}{ \ln \left( \frac{T_{C,i}}{T_{C,o}} \right) }
\]  

2.2.3. Constant exergy efficiency

Thermal exergy was the only exergy part considered, which can be expressed for a stream of matter \(j\), as shown in (4). The exergy efficiency of the HP model, \(\varepsilon_m\), was calculated as the ratio of exergy product and exergy fuel, as shown in (5). Both the exergy product and the exergy fuel depends on the purpose of the system, its operating conditions and integration into the environment (dead state) [22].
\[ \dot{E}_j = \dot{m}_j (h_j - h_0) - \dot{m}_j T_0 (s_j - s_0) \]  

\[ \varepsilon_m = \frac{\dot{E}_p}{\dot{E}_F} \]  

For the HP, the exergy product was considered as the exergy of the supplied heat to the DH network. The exergy fuel was considered as the electrical work for operating the compressor. The dead state temperature was set as the heat source inlet temperature during operation. Therefore, the dead state temperature may vary for every hour during the year and each heat source. The exergy part that occurs below the dead state was neglected, as the installation was assumed to return this stream to the ambient. The exergy efficiency for a HP was then calculated as shown in (6) [14].

\[ \varepsilon_m = \frac{\dot{m}_H (h_{H,o} - h_{H,i}) - \dot{m}_H T_0 (s_{H,o} - s_{H,i})}{W} = \frac{\dot{m}_H c_p (T_{H,o} - T_{H,i}) - T_0 \dot{m}_H c_p \ln \left( \frac{T_{H,o}}{T_{H,i}} \right)}{W} \]  

The exergy efficiency can be expressed as a function of the Lorenz efficiency, when substituting (2) and (3) into (6), as shown in (7).

\[ \varepsilon_m = \frac{\dot{Q}_H - T_0 \dot{m}_H c_p \ln \left( \frac{T_{H,o}}{T_{H,i}} \right) (T_{H,o} - T_{H,i})}{W} = \frac{\dot{Q}_H \left( 1 - \frac{T_0}{T_H} \right)}{W} = \eta_{L,m} \frac{T_H - T_0}{T_H - T_C} \]  

The COP can be calculated rearranging (7), as shown in (8). Using this estimation of COP based on constant exergy efficiency resulted in very similar values compared to the case of constant Lorenz efficiency. The constant value of the exergy efficiency was determined based on the design conditions of the thermodynamic model.

\[ \text{COP}_e = \frac{\varepsilon_m}{1 - \frac{T_0}{T_H}} = \varepsilon_m \frac{T_H}{T_H - T_0} \]  

### 2.2.4. Jensen COP estimation for design conditions

Another COP estimation method was presented by Jensen et al. [17]. They derived a generic equation for the COP analytically for design conditions of a single-stage HP cycle, as shown in (9).

\[ \text{COP}_j = \left( \frac{\text{COP}_L \left( 1 + \frac{\Delta T_{r,H} + \Delta T_{pp}}{T_H} \right)}{1 + \frac{\Delta T_{r,H} + 2 \Delta T_{pp}}{T_H} \eta_{is,c} \left( 1 - \frac{w_{is,c}}{w_{is,c}} \right) + 1 - \eta_{is,c} - f_Q} \right) \]  

Equation (9) depends only on temperatures of the heat source and heat sink (\( \text{COP}_L, T_H \) and \( T_C \)) as well as characteristics of the compressor (isentropic efficiency \( \eta_{is,c} \) and heat loss factor \( f_Q \)), the heat exchangers (entropic pinch point temperature difference \( \Delta T_{pp} \approx \Delta T_{pp} \)) and certain characteristics of the refrigerant \( (w_{is,c}/w_{is,c}, \Delta T_{r,H} \) and \( \Delta T_{r,C} \)). Approximations were proposed by Jensen et al. [17] for \( w_{is,c}/w_{is,c}, \Delta T_{r,H} \) and \( \Delta T_{r,C} \) using ammonia as the refrigerant. Other input parameters may be found in Table 1 based on design conditions.
2.3. Energy planning model

A model was developed in GAMS, version 24.8.3, [23] with the aim of minimizing total costs including annualized investment costs ($C_{\text{Inv},p}$) and annual operational costs ($C_{\text{el}} + C_{\text{O&M}}$) of each production unit $p$. The optimization was based on mixed-integer linear programming using the CPLEX solver, version 12.7.0.0 [24]. Annual calculations were performed with one hour time step. A more detailed description of the model can be found in Pieper et al. [25]. The objective function is shown in (10).

$$\min Z = \sum_p Z_p = \sum_p C_{\text{el},p} + C_{\text{O&M},p} + C_{\text{Inv},a,p}$$ (10)

Short-term storage with a heat loss of 5% for each hour, $n$, and an electric peak load boiler with COP of 1 were also implemented [26]. The hourly heat demand had to be supplied by the production units or the storage, as shown in (11). The model determined the required HP capacity of each heat source, electric boiler capacity and storage capacity by (12) and (13), ensuring that the capacities were above the hourly production and storage level, respectively.

$$\sum_p Q_{H,p,n} = Q_{\text{Heat},n} + Q_{\text{st, char},n} - Q_{\text{st, dis},n}$$ (11)
$$\dot{Q}_{H,p} \geq Q_{H,p,n}$$ (12)
$$\dot{Q}_{\text{st, c}} \geq Q_{\text{st, level},n}$$ (13)

The COP of the production units determined the electricity usage and consequently the costs related to that, as shown in (14).

$$\text{COP}_{p,n} = \frac{Q_{H,p,n}}{P_{p,n}}$$ (14)

2.4. Case study

The relative load duration curve based on hourly measurements in 2018 of the heat demand of the already existing building area of Nordhavn was used as input for this study. These measurements include heat losses of the existing part of the DH network, which include both commercial and residential buildings, and are shown in Figure 2. The maximum hourly peak demand was assumed to be 20 MW resulting in an annual heat production of 51 GWh. If all buildings in Nordhavn until 2060 would be residential and comply with the Danish building standard 2020 (30 kWh/m²/a) [27], the annual heat demand would be 105 GWh.

![Figure 2. Relative load duration curve for area of Nordhavn (based on [28])](image-url)
A dependency of DH supply temperatures on outdoor temperature $T_{\text{amb}}$ was determined in Pieper et al. [29] based on measurements for the area of Nordhavn [30], as shown in (15). The temperature-dependency was assumed to increase to a maximum of 85 °C during cold periods. This maximum temperature corresponds to the pressure and temperature limitations of state-of-the-art ammonia compressors for HPs [31].

$$T_{H,o,n} = \begin{cases} 
70 \degree C, & \text{for } T_{\text{amb},n} > 10 \degree C \\
85 \degree C, & \text{for } T_{\text{amb},n} < 2.5 \degree C \\
-2 T_{\text{amb},n} + 90 \degree C, & \text{otherwise}
\end{cases}$$ (15)

This dependency was further simplified for the sake of clarity of the results, as shown in Figure 3. Constant temperatures were assumed in winter and summer. A linear trend was assumed for April and November. The return temperature was assumed to be constant at 35 °C.

2.4.1. Heat sources

The heat sources considered for this study were ambient air, groundwater, seawater and sewage water. Hourly values of the temperature and the relative humidity of ambient air for the area of Nordhavn for 2018 were provided [32]. The groundwater temperature was assumed to be 10 °C and to be accessed at 100 m depth. Measurements showed a groundwater temperature between 10 °C to 11 °C at this depth and location [33]. This is at the upper end of what is typically found in Denmark (8 °C to 10 °C), which also depends on the depth of the groundwater reservoir [21].

It was assumed that the groundwater HP capacity was limited due to area constraints. This complies with the largest groundwater HP installations found in Denmark [21]. Such HP capacities require a large amount of groundwater, which is difficult to extract and reinject without compromising the long-term stability. Therefore, the practical limit might be at around 5 MW [34]. An analysis for Nordhavn has shown that pumping 50 m$^3$/h of water, which may correspond to approximately 0.7 MW HP capacity, influences the groundwater level temporarily by 0.5 m at a distance to the pumping location of approximately 900 m [35]. The width of the total area of Nordhavn is around 3 km and it is mainly surrounded by the sea.

Seawater was assumed to be pumped from the maximum possible depth near the coastline of Nordhavn at 10 m depth (distance approximately 200 m). In this way, freezing problems during cooler periods can be reduced or avoided, as the temperature in winter is higher at greater depth than at the water surface. This is also done for large-scale HPs in Oslo, Norway [36], and in Stockholm, Sweden, [37], however at greater depth compared to Danish conditions. An hourly seawater temperature profile for a depth of 10 m was used [29] based on measurements from 2015 and 2017. The data was provided by the National database (ODA) by the Danish Environmental Protection Agency [38].

Measurements of daily sewage water temperature and daily volume flow rate after the cleaning process were provided for several years for the sewage water treatment plant, Lynetten, located 2 km away from Nordhavn [39]. Cleaned sewage water was considered as the heat source, because the
biological treatment of cleaning the sewage water is sensitive to changes in temperature and thus not to be disturbed [40]. In addition, using untreated water may require additional attention when sending it through the evaporator of the HP in terms of cleaning equipment and heat exchanger design. Mean values for each day were created based on the different years of available data. The same temperature was assumed for all hours of one day and the daily volume flow rate was evenly distributed for each hour of the day. The hourly volume flow rate varied between 2100 m$^3$/h and 4350 m$^3$/h with the mean value being 3035 m$^3$/h and corresponding to 45 MW HP capacity. An overview of the hourly temperatures of all considered heat sources may be found in Figure 4.

![Figure 4. Hourly heat source inlet temperatures of ambient air (air), groundwater (gw), sewage water (sew) and seawater (sea)](image)

### 2.4.2. Economic input parameters

Year 2018 was used as the reference year. Hourly electricity prices for the region DK2 were taken from Nord Pool [41]. Electricity taxes, system and transmission fees were applied. Distribution fees were applied for the network of Radius, using a weighted mean value of the triple tariff for simplicity [42]. The public service obligation (PSO) was neglected, as it will be phased out by 2022 [43]. Further, the electricity tax will be reduced by 100 DKK/MWh from 2019 if electricity is used to produce heat for DH [44]. Investment costs and operating and maintenance (O&M) costs were obtained from different references. The investment costs were annualised assuming a discount rate of 4% and a lifetime of piping, the HPs, the storage and electric boilers of 30 years, 25 years, 20 years and 15 years, respectively [45]. Only part of the investment was taken into account for equipment that had a longer lifetime than the HPs. On the contrary, additional investments were required for the electrical boiler due to their shorter lifetime. In case additional DH piping was required to allow the use of certain heat sources, a maximum velocity of 2 m/s was assumed. For the case of sewage water a heat loss of 5% was assumed for the additional 2 km DH pipe. An overview of the input parameters can be found in Table 2 to Table 4.

#### Table 2. Electricity costs

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nord Pool average price 2018</td>
<td>46.20</td>
<td>€/MWh$_{el}$</td>
<td>[41]</td>
</tr>
<tr>
<td>Electricity tax</td>
<td>41.34</td>
<td>€/MWh$_{el}$</td>
<td>[44]</td>
</tr>
<tr>
<td>Transmission and system tariff</td>
<td>10.74</td>
<td>€/MWh$_{el}$</td>
<td>[46]</td>
</tr>
<tr>
<td>Distribution tariff</td>
<td>13.10</td>
<td>€/MWh$_{el}$</td>
<td>[47]</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>111.38</strong></td>
<td>€/MWh$_{el}$</td>
<td></td>
</tr>
</tbody>
</table>

#### Table 3. Investment costs of DH pipes and storage

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Investment costs</th>
<th>Unit</th>
<th>Pipe length</th>
<th>Unit</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>DH pipe (sew)</td>
<td>$(505+3.1 \dot{V}_H)$</td>
<td>€/m</td>
<td>2000</td>
<td>m</td>
<td>[48]</td>
</tr>
<tr>
<td>DH pipe (sea)</td>
<td>$(505+3.1 \dot{V}_C)$</td>
<td>€/m</td>
<td>200</td>
<td>m</td>
<td>[48]</td>
</tr>
<tr>
<td>Storage</td>
<td>$205+0.087 V_{st}$</td>
<td>T€</td>
<td></td>
<td></td>
<td>[49]</td>
</tr>
</tbody>
</table>
Table 4. Investment costs and O&M costs

<table>
<thead>
<tr>
<th>Technology</th>
<th>Investments, M€</th>
<th>Ref.</th>
<th>O&amp;M costs, €/MWh</th>
<th>O&amp;M costs, €/MW/a</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>HP (air)</td>
<td>0.188+0.677 $\dot{Q}_H$</td>
<td>[50]</td>
<td>1.0</td>
<td>2000</td>
<td>[21,45]</td>
</tr>
<tr>
<td>HP (gw)</td>
<td>0.505+0.640 $\dot{Q}_H$</td>
<td>[50]</td>
<td>2.0</td>
<td>2000</td>
<td>[21,45]</td>
</tr>
<tr>
<td>HP (sew)</td>
<td>0.484+0.550 $\dot{Q}_H$</td>
<td>[50]</td>
<td>1.3</td>
<td>2000</td>
<td>[21,45]</td>
</tr>
<tr>
<td>HP (sea)</td>
<td>0.484+0.550 $\dot{Q}_H$</td>
<td>[50]</td>
<td>1.3</td>
<td>2000</td>
<td>[21,45]</td>
</tr>
<tr>
<td>Electric boiler</td>
<td>0.110 $\dot{Q}_H$</td>
<td>[48]</td>
<td>0.54</td>
<td>1177</td>
<td>[48]</td>
</tr>
</tbody>
</table>

2.5. Performance indicators

The different COP estimation methods were implemented in the energy planning model and the results of the optimizations were compared with each other. Evaluated parameters were the HP capacity for each heat source, the seasonal COP (SCOP) considering storage losses and the levelized costs of heat (LCOH).

The SCOP takes hourly variations of heat supply $Q_n$ into account by calculating the ratio of the annual supplied heat and the annual consumed electricity, as shown in (16).

$$\text{SCOP} = \frac{\dot{Q}_{\text{tot}}}{P_{\text{tot}}} = \frac{\sum_{n=1}^{N} Q_n}{\sum_{n=1}^{N} P_n}$$

(16)

The LCOH were calculated as the sum of the heat production costs, including costs for electricity $C_{\text{el}}$, operating and maintenance costs ($C_{\text{O&M}}$) and investment costs ($C_{\text{Inv}}$), divided by the annual heat supply, as shown in (17).

$$\text{LCOH} = \frac{C_{\text{el}} + C_{\text{O&M}} + C_{\text{Inv}}}{\dot{Q}_{\text{Heat}}}$$

(17)

2.6. Sensitivity analysis

The COP of the thermodynamic HP model was changed by ±20 % in order to investigate the influence of varying COP on the LCOH of the optimization results.

The constant COP$_{\text{con}}$ was changed by ±20 % to show what the impact on the optimization would be, if a different value than the COP for design conditions was chosen. In addition, the SCOP based on heat demand variations was used as constant COP$_{\text{con}}$, which would represent the best possible guess one could make when choosing a constant COP$_{\text{con}}$.

For the same reason, the Lorenz efficiency was decreased by 20 %, since it already had high values. Furthermore, the Lorenz efficiency was assumed to be 0.5 for all heat sources, which was a value used in literature [11,12].

For the Jensen COP estimation method, the coefficients for the two approximations proposed by Jensen et al. [17] for $w_{\text{is,}e}/w_{\text{is,}c}$ and $\Delta T_{r,H}$ were varied by ±20 %.

3. Results

First, the dimensioning values of the thermodynamic HP model for design conditions are presented. The comparison of COP with the other COP estimation methods follows. Then, the impact of using different COP estimation methods for planning the installation of new HPs and the results of the economic analysis of the energy planning model are presented. Finally, the results of the sensitivity analysis are shown.

3.1. Dimensioning values of thermodynamic HP model

The dimensioning values of the HP design model for the different heat sources are shown in Table 5. As shown, the COPs were the highest for the heat sources with the highest source temperature. The
Lorenz efficiency was higher for lower heat source temperatures. The exergy efficiencies were slightly below the Lorenz efficiency, as expected. The optimal intermediate pressure and built-in volume ratio as well as the displacement rates were very different for using ambient air compared to using the other heat sources.

Table 5. Dimensioning values of HP in design conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Air</th>
<th>GW</th>
<th>Sew</th>
<th>Sea</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP</td>
<td>2.72</td>
<td>3.42</td>
<td>3.46</td>
<td>3.29</td>
<td></td>
</tr>
<tr>
<td>Lorenz efficiency</td>
<td>0.61</td>
<td>0.54</td>
<td>0.53</td>
<td>0.56</td>
<td></td>
</tr>
<tr>
<td>Exergy efficiency</td>
<td>0.58</td>
<td>0.51</td>
<td>0.50</td>
<td>0.55</td>
<td></td>
</tr>
<tr>
<td>UAe</td>
<td>1329</td>
<td>1488</td>
<td>1496</td>
<td>1747</td>
<td>kW/K</td>
</tr>
<tr>
<td>UA_{c,\text{tot}}</td>
<td>878</td>
<td>933</td>
<td>934</td>
<td>929</td>
<td>kW/K</td>
</tr>
<tr>
<td>Displacement rates (LP/HP)</td>
<td>6.49/3.21</td>
<td>3.35/1.52</td>
<td>3.25/1.50</td>
<td>3.66/1.60</td>
<td>m³/s</td>
</tr>
<tr>
<td>Pressure ratios (LP/HP)</td>
<td>2.8/8.3</td>
<td>2.7/3.7</td>
<td>2.7/3.7</td>
<td>2.9/3.9</td>
<td></td>
</tr>
<tr>
<td>Intermediate pressure</td>
<td>4.63</td>
<td>11.24</td>
<td>11.45</td>
<td>10.62</td>
<td>bar</td>
</tr>
</tbody>
</table>

3.2. HP model COPs for different heat sources

The hourly COPs for the HPs based on the four different heat sources calculated with the thermodynamic model may be found in Figure 5.

![Figure 5. Hourly COPs for HPs calculated with thermodynamic model](image)

As shown, the COPs were generally higher from April to November, due to decreased DH supply temperatures and for most heat sources increased inlet temperatures, except for groundwater. The COP of the air-source HP fluctuated more during summer, because of the changes in ambient temperature. The COP of the sewage water HP was always higher than for the other water-based HPs. The groundwater HP resulted in higher COPs in winter than the seawater HP. On the contrary, the seawater HP COP was higher during summer, because of the warmer source inlet temperature. The SCOP considering the hourly variations in heat demand of the case study are shown in Table 6.

Table 6: Seasonal COP of HPs based on different heat sources considering the heat demand

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Air</th>
<th>GW</th>
<th>Sew</th>
<th>Sea</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seasonal COP</td>
<td>3.30</td>
<td>3.51</td>
<td>3.67</td>
<td>3.48</td>
<td></td>
</tr>
</tbody>
</table>

3.3. COP deviations based on different estimation methods

For each hour, the COP of the four different estimation methods was compared to the thermodynamic HP model. The hourly deviations of COP are shown in Figure 6.
For the water-based HPs, each of the four COP estimation methods resulted in very good estimations of COP during winter compared to the thermodynamic model, because the conditions from December until April were very similar to the design conditions. The Jensen estimation method for design conditions (Jensen) underestimated the COP slightly during this period, likely because it was developed for one-stage HP cycles, while a two-stage HP was modelled here. The deviations for the different methods increase during summer when the operating conditions differ considerably from the design conditions.

Using a constant COP (Constant) based on design conditions resulted in an underestimation of COP of up to −20 % for sewage water and seawater HPs in summer. The deviation was reduced by approximately 50 % for the groundwater HP. However, for the air-source HP, the deviations of the COPs were up to −40 % in summer and −15 % to −20 % in winter. This may be explained by the design conditions set at −12 °C for Denmark, while the lowest occurring temperature in 2018 for the area of Nordhavn was −8.6 °C.

The COP calculated with the Lorenz method (Lorenz) deviates considerably during summer from the one obtained from the thermodynamic model. For the use of ambient air, deviations in COP of up to 60 % were observed. Considering that ideal isentropic efficiencies of the compressors were assumed for the thermodynamic model, the deviation could be even larger. If sewage water and seawater HPs were used, the COP deviated more than 20 % during some periods. Using a constant Lorenz efficiency throughout the entire year would result in an overestimation of COP for heat sources with seasonal temperature variations. The chosen Lorenz efficiency, based on design conditions, may be too high, since these conditions occur only rarely during the year and because the Lorenz efficiency decreases with smaller temperature lifts, so if the ambient temperature increases and/or the DH temperature decreases.

The COP calculated with a constant exergy efficiency resulted in very similar values compared to using the Lorenz method, as expected and described earlier. Due to this similarity, the method based on constant exergy efficiency was not analysed further.

The deviations of the Jensen method are below 10 % during summer for all heat sources. The deviations are even lower when the heat source temperature does not vary, as for groundwater.
Therefore, the Jensen method gives good approximations of COP throughout the year. In particular, the deviation in summer will have a smaller impact, since the heat demand is typically lower.

3.4. Comparison of optimization results

The results of using the different COP estimation methods in the energy planning tool may be found in Table 7 as well as typical values for some of the parameters for implemented large-scale HPs in Denmark.

Table 7. Results of optimization using different COP estimation methods

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Model</th>
<th>Constant</th>
<th>Lorenz</th>
<th>Jensen</th>
<th>Reference [21]</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source</td>
<td>Sea</td>
<td>Sea/GW</td>
<td>Air/GW</td>
<td>Sea</td>
<td></td>
<td>MW</td>
</tr>
<tr>
<td>HP capacity</td>
<td>15.4</td>
<td>10.4/5.0</td>
<td>11.1/5.0</td>
<td>15.4</td>
<td></td>
<td>MW</td>
</tr>
<tr>
<td>Storage capacity</td>
<td>32.2</td>
<td>32.2</td>
<td>28.4</td>
<td>32.2</td>
<td></td>
<td>MW</td>
</tr>
<tr>
<td>Seasonal COP</td>
<td>3.4</td>
<td>3.3</td>
<td>3.7</td>
<td>3.4</td>
<td>3.5 to 4.5</td>
<td>-</td>
</tr>
<tr>
<td>LCOH</td>
<td>45.2</td>
<td>46.9</td>
<td>44.0</td>
<td>45.3</td>
<td>42 to 49</td>
<td>€/MWh_h</td>
</tr>
<tr>
<td>O&amp;M costs</td>
<td>31.7</td>
<td>32.9</td>
<td>29.7</td>
<td>31.8</td>
<td>25 to 32</td>
<td>€/MWh_h</td>
</tr>
</tbody>
</table>

As shown, using the different estimation methods resulted in a similar total HP capacity, which is 77 \% to 81 \% of the hourly peak demand. The remaining load was supplied by discharging the storage, which had sufficient heating capacity. The hot water storage tank would approximately have a volume of 564 m\(^3\), if completely filled with 85 °C warm water.

The selected heat source depended on the COP estimation method. A 10.4 MW seawater HP and a 5.0 MW groundwater HP was the optimal choice based on the constant COP method. Using the Lorenz method for determining COP resulted in the use of a 11.1 MW air-source HP and a 5.0 MW groundwater HP. The choice for air in this case originated from the large overestimation of COP. Using the Jensen method resulted in the same selection of heat source (seawater) as if the COP was calculated using the thermodynamic model.

The SCOP was very similar for most of the estimation methods, since the overestimation during summer was less significant when the low heat load was considered. The Lorenz method was an exception, which resulted in a high SCOP of 3.7.

The difference in COP is reflected in the economic parameters, e.g. the LCOH were higher when the Constant COP method with lower COPs was used and lower when the Lorenz method was used having higher COPs. A difference in LCOH of 1 €/MWh\(_h\) would be a loss or gain of 51,000 € for the 51 GWh annual heat supply. The impact on the economic parameters could be even larger, if the chosen efficiencies and COPs would have been determined without the use of the thermodynamic model.

3.5. Sensitivity analysis

The LCOH and its individual contributions are shown in Figure 7 for changes of hourly COP of the thermodynamic model. The LCOH changed by −11 \% and 17 \%, while the COP was changed by ±20 \% for each hour. As shown, the COP has an effect only on the electricity costs and associated taxes and tariffs. The piping costs to access the seawater were also affected, because a change in COP required different volume flow rates of the heat source and consequently smaller or larger pipe diameters. The investment costs of the HP and the storage as well as the maintenance costs remained unchanged. Therefore, the effect on the LCOH due to changes in COP may be limited.
The results of the optimization when varying input parameters of the different COP estimation methods are shown in Table 8.

### Table 8. Results of sensitivity analysis

<table>
<thead>
<tr>
<th>Parameter</th>
<th>0.8 $\eta_L$ $\eta_L=0.5$</th>
<th>0.8 $c_J$</th>
<th>1.2 $c_J$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HP capacity, MW</td>
<td>Sea/GW</td>
<td>10.4/5.0</td>
<td>10.4/5.0</td>
</tr>
<tr>
<td>Storage, MW</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Seasonal COP, -</td>
<td></td>
<td>2.6</td>
<td>3.4</td>
</tr>
<tr>
<td>LCOH, €/MWh$_h$</td>
<td></td>
<td>55.2</td>
<td>41.6</td>
</tr>
<tr>
<td>O&amp;M, €/MWh$_h$</td>
<td></td>
<td>41.3</td>
<td>27.5</td>
</tr>
</tbody>
</table>

As shown for the constant COP, the most economical heat source and HP capacity remained unchanged for variations of ±20 %. However, the SCOPs changed considerably and therefore also the O&M costs and consequently the LCOH. For a SCOP of 2.6 and 4.0, the LCOH changed by 18 % and −11 %, respectively. Such a difference may have a significant impact on the decision of investing into HPs compared to alternative supply options and in which heat source.

If the SCOP of the thermodynamic model was used as constant value over the year, the optimization results were very similar than the ones obtained when using the thermodynamic model. However, this would require knowledge about the expected heat demand and COPs over the year.

With a decreased Lorenz efficiency, the optimal solution was a 11.9 MW seawater HP and a 3.8 MW air-source HP, which was different than the initial found optimum using this method. If the Lorenz efficiency was assumed 0.5 for all HPs, the most economical solution would look again quite different and be based on a large sewage water HP.

If the coefficients used for the approximations were varied by ±20 %, the optimal choice of heat source would switch from seawater to a mix of seawater and groundwater. The deviations in SCOP were smaller than for the sensitivity analysis of the other estimation methods. This would result in similar economic parameters than before. Consequently, the Jensen method may be less sensitive to a wrong choice of input parameters than the use of the constant COP method or the constant Lorenz efficiency method.

### 4. Discussion

The aim of the study was to investigate how using different COP estimation methods in an energy planning tool would affect the optimization results. Therefore, the most economical solution found
should be rather seen as an indication. A detailed sensitivity analysis on the economic input parameters was not performed.

The optimum seemed flat, so that different combinations of HP capacities and heat sources could result in a solution close to the optimum. This may be seen by the different optimal choices of heat source depending on the used estimation method and the performed sensitivity analysis.

The piping costs may have a significant impact, as shown in Figure 7 for the seawater HP. Therefore, a sewage water HP was not selected as the most economical solution, even though the SCOP was the highest. Using sewage water required a 2 km long pipe. Furthermore, a heat loss of 5% was added. If sewage water could be accessed without additional piping, investment costs could be reduced. Furthermore, the heat loss would be avoided resulting in lower operating costs. This would decrease the LCOH by around 5 €/MWh [51]. Examples of a 10 MW and a 40 MW sewage water HP exist in Kalundborg, Denmark [21] and in Malmo, Sweden [52].

Part-load operation of the HPs was not represented in the energy planning model, the COP estimation methods and therefore also not in the thermodynamic model. Auxiliary electricity consumption for powering fans or pumps was not considered.

Other COP estimation methods were not applied due to the lack of coefficients for the required temperature range, e.g. [15]. The thermodynamic model was of a two-stage HP with open intercooler and ammonia as refrigerant. A polynomial for screw compressors was applied to represent losses due to a mismatch in pressure ratio. Therefore, the results may be limited to these kinds of applications and different for others. The deviations of COP were compared to the thermodynamic model, which may calculate different COPs than a real plant. The chosen constant COPs and efficiencies were based on the design conditions of the thermodynamic model. Deviations in COP and the economic solution would be even larger, if less optimal guesses were made, as was shown in the sensitivity analysis.

The considered dead state temperature did not differ considerably from the entropic mean temperature of the heat source, $T_C$, because it was chosen to be the heat source inlet temperature. Furthermore, the temperature difference between heat source inlet and outlet was assumed to be 5 K. Therefore, the exergy efficiency and Lorenz efficiency were very similar.

5. Conclusion

The hourly COPs based on four different estimation methods were compared to the COPs calculated by a thermodynamic HP model for four heat sources for one year. The COPs were used in an energy planning tool to investigate the impact of using different COP estimation methods on the most economical solution for a new development district.

It was found that using a constant COP based on design conditions underestimated the COP considerably if the operating conditions are very different to the design conditions. This was the case especially in summer and if ambient air was used as heat source. The economic parameters, obtained using a constant COP, were very dependent on the choice of constant COP. If the constant COP was chosen less optimal, the economic parameters changed considerably. If the SCOP was used as constant value, similar results were obtained as for the thermodynamic model.

Using a constant Lorenz efficiency resulted in very large deviations in COP of up to 60% in summer, when ambient air was used. Deviations of 20% were seen for using sewage water and seawater. Consequently, ambient air played an important role for the economic optimum. If the Lorenz efficiency was assumed to be 0.5 for all HPs, the optimum HP capacity was based on sewage water.

Using a constant exergy efficiency resulted in similar deviations as for the case of assuming a constant Lorenz efficiency.

Using the Jensen COP estimation method for design conditions resulted in good COP approximations with deviations of less than 10% in summer and, for the water-based HPs, less than 2% in winter.

Using the Jensen method in the energy planning tool resulted in solutions very close to the ones obtained using COPs from the thermodynamic model. Furthermore, this method was less sensitive to
uncertainties in input parameters than the other two estimation methods. Using a constant COP, a constant exergy or constant Lorenz efficiency can result in wrong investment decisions if wrong assumptions on COP and efficiencies are made.

**Acknowledgments**

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**Nomenclature**

- $C$: cost, €
- $c$: coefficients for approximations, -
- COP: coefficient of performance, -
- $c_p$: specific heat capacity, MJ/kg/K
- $\dot{E}$: exergy rate, MWh/h
- $f_Q$: compressor heat loss ratio, -
- LCOH: levelized cost of heat, €/MWh
- $\dot{m}$: mass flow rate, kg/s
- $P$: electricity consumption, MWh
- $p$: pressure, bar
- $Q$: heat, MWh
- $\dot{Q}$: heat rate, MWh/h
- $s$: specific entropy, MJ/kg
- SCOP: seasonal coefficient of performance, -
- $\bar{T}$: entropic mean temperature, K
- $T$: temperature, ºC or K
- UA: overall heat transfer coefficient times area, kW/K
- $v$: built-in volume ratio, -
- $\dot{W}$: power, MWh/h
- $w$: work, MJ/kg
- $Z$: objective function, €

**Greek symbols**

- $\Delta$: difference
- $\varepsilon$: exergy efficiency, -
- $\eta$: efficiency, -
- $\kappa$: polytrophic exponent, -
- $\pi$: built-in pressure ratio, -

**Subscripts and superscripts**

- $0$: dead state
- $a$: annualised
- amb: ambient
- $C$: heat source
- $c$: compression
con constant
char charging
d design conditions
dis discharging
e expansion
el electricity
F fuel
H heat sink
Heat heat supply
HP high pressure
i inlet
Inv investment
is isentropic
J Jensen
j stream
L Lorenz
level storage level
LP low pressure
m thermodynamic HP model
max maximum
N hours of a year
n current hour of the year
o outlet
O&M operating and maintenance
P product
p production unit
pp pinch point
r refrigerant
st storage
tot total

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