Integration of heat pumps into thermal plants for creation of large-scale electricity storage capacities

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HIGHLIGHTS
- Realistic round-trip-efficiencies are in range of 50–60% for different power plants.
- Efficient heat pumps are realizable at 300–600 °C and moderate internal recuperation.
- Trans-critical heat pumps are not generally superior to super-critical processes.
- Operation of heat pumps and electric heaters in series connection is promising.
- New thermodynamic assessment approach for PHES systems.

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ABSTRACT
Within Thermo-Electric Energy Storage (TEES) concepts, thermal plants are conceivable for reconversion of stored heat into electricity. By this means, new areas of application for existing thermal plants are established and the costs of the storage system are reduced. A promising TEES approach is Pumped-Heat-Electricity-Storage (PHES). In the present study, the thermodynamic potentials of the new concept of integrating PHES systems into different types of thermal plants for the creation of large-scale electricity storage units are assessed – based on exergetic quantities – including the discussion of technical aspects.

Using the environment as the heat source, recuperated heat pump designs are investigated with regards to the achievable efficiencies for different working fluids (CO₂, air and Argon) and the related processes (trans-critical/super-critical). The investigated maximum heat pump temperature range is between 50 °C and 700 °C. The heat pump designs are individually optimized concerning their remaining degrees of freedom. Finally, a combined characteristic diagram is provided, which allows to identify the most reasonable heat pump working fluid and process configuration referring to the boundaries of a specific storage concept. Electric heaters as a simpler method for power-to-heat conversion are assessed as well.

The results show that exergetic heat pump efficiencies of above 70% can be achieved if the maximum temperature of the provided heat is in the range of 300–600 °C while the minimum temperature is elevated. It is also shown that trans-critical cycle designs are not generally superior to super-critical cycle designs at these boundaries.

Based on the results of the heat pump analysis, the round-trip-efficiencies of different heat integration options into different types of thermal plants are estimated – the reachable efficiencies are roughly in the range of 50–60%. Finally, the application of heat pumps and electric heaters in series connection is assessed. Then, the round-trip-efficiencies of the storage concepts drop by a few percent points (2–5%) but the technical challenges of designing high temperature heat pumps are reduced.

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1. Introduction

1.1. Literature review

Worldwide, the share of renewable energy sources increases and induces a higher degree of volatility at the electricity supply side [1,2]. Adapting the more volatile supply side to the demand side needs the contribution of all market participants.

Conventional power plants are required to act more flexible to harmonize the supply side output [1,2]. Therefore, research is ongoing to extend the power plant capacities to variably respond on market demands. This refers to fossil-fired plants [3], but also to nuclear plants [4]. At the same time, fossil-fired plants are still improved with regards to energy efficiency and thereby contributing along with renewable sources to the environmental-friendly electricity generation. This can either be realized by improved plant designs but also by converting these plants into poly-generation systems. One example is the utilization of low-temperature waste heat of fossil-fired plants for heating purposes [5,6]. However, in some regions, as in Germany, fossil-fired power plants currently face the situation of being under-utilized because of regulations preferring renewable input [7,8]. The identification of additional areas of application could make these plants operate more economically. Also for renewable plants, as for solar thermal plants, intensive research and development is carried out referring to operational features. The integration of augmented thermal storage capacities [9–11], enabling the plant’s power generation to be less dependent on the changing solar heat input is one focus. The target is to design solar thermal plants, which are capable for base-load operation [12].

Different authors point out that economic electricity storage systems are required as a further key component for proceeding to electricity grids with even higher degrees of integration of renewable sources [1,13–16]. Electricity is usually not stored directly but used to create a potential of another form of energy. This potential is preserved and reconverted into electricity when required (e.g. batteries → chemical potential/pumped hydro storage (PHS) → mechanical potential). An overview of the various concepts of electricity storage is provided in several studies, see e.g. [15,16]. The most relevant technological rating criteria for electricity storage concepts is the round-trip-efficiency. It is defined as the ratio between the electricity output during a discharging sequence and the input during a charging sequence. A comparison of the different storage technologies shows significant differences of achievable round-trip-efficiencies [15,16]. Nevertheless, further aspects of the storage technologies have to be considered. Systems with high efficiencies usually suffer from other disadvantages. For example, the application of pumped hydro storage (PHS) and compressed air energy storage (CAES) is bound to the availability of appropriate geological sites and materials for batteries are usually expensive. Therefore, storage technology of low efficiency is still considerable for economic application [16]. Apart from efficiency considerations, storage systems of different scale and response times are necessary to meet future demands [16].

One approach to create electricity storage capacities is the utilization of a power-to-heat cycle (heat pump) during a charging sequence in order to create a thermal potential. This potential is stored and later reconverted by a heat-to-power cycle (H2P) within a discharging sequence. In general, the minimum and maximum temperatures of the thermal potential can be above or below ambient conditions. This technology is usually known as Pumped-Heat-Energy-Storage (PHES) belonging to the category of Thermal Energy Storage (TES); if the temperatures are below ambient conditions, it is also referred to as Pumped-Cryogenic-Energy-Storage (PCES) [17,18]. Theoretical approaches are developed to estimate the potentials of this technology by means of the round-trip-efficiency; based on simple but very unspecific models, it can be shown that high round-trip-efficiencies require high differences in temperatures between the heat source and the heat sink [17,18]. Within recent years, different PHES concepts were developed. Four of them ([19–43]) are briefly introduced in Table 1, including characterizing values and features. These concepts have in common that they are investigated and developed beyond basic thermodynamic assessments. Significant differences between the


Table 1
Existing approaches on pumped-heat-electricity-storage.

<table>
<thead>
<tr>
<th>PHES concept</th>
<th>System</th>
<th>Working fluid</th>
<th>Target average temperatures of the thermal potential</th>
<th>Process temperature</th>
<th>Process pressure</th>
<th>Estimated round-trip efficiency</th>
<th>References</th>
<th>Related patents</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lit-1</td>
<td>Stand-alone</td>
<td>Argon</td>
<td>(T^+ &gt; T_{\text{max}}, \ T^- &lt; T_{\text{min}}) (\approx) 500 (°C) (\approx) -170 (°C) (\approx) 12 (\text{bar}) (\approx) 1 (\approx) 0.70 (-0.72) [19,20] [23–25]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lit-2</td>
<td>CO(_2)</td>
<td></td>
<td>(\approx) 1000 (°C) (\approx) -70 (°C) (\approx) 7 (\text{bar}) (\approx) 1 (\approx) 0.67 [26] [28–30]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lit-3</td>
<td>Integrated</td>
<td>ORC-fluid</td>
<td>(\approx) 125 (°C) (\approx) 0 (°C) (\approx) 180 (\text{bar}) (\approx) 40 (\approx) 0.51 (-0.65) [31,33] [35–41]</td>
<td></td>
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</tr>
<tr>
<td>Present study</td>
<td>Argon/air/C(_2)</td>
<td>(\approx) 200 (°C) (\approx) 80 (°C) (\approx) 7 (\text{bar}) (\approx) 7 (\approx) 0.60 [42] [43]</td>
<td></td>
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</table>


1.2. Present research

1.2.1. System concept

The challenge of creating economic large-scale electricity storage capacities is addressed in a new way by the idea of integrating heat pumps and thermal storage devices into existing thermal power plants. A recuperated heat pump using environmental heat provides high temperature heat at appropriate parameters for the later integration of this heat into a power cycle. By this means, a Thermo-Electric Energy Storage (TEES) system in analogy to Pumped-Heat-Energy-Storage systems (PHES), as for those in Table 1, is established. In the following, the present concept is referred to as Integrated Pumped-Heat-Energy-Storage (I-PHES). I-PHES further addresses the under-utilization of fossil-fired power plants by extending their field of application. Thermal power plants could then either act as electricity providers or consumers. Potentially, the number of complete power cycle shut-downs can also be decreased because the overall power output of the plant can virtually be decreased below the technically constraint minimum load of the fossil-fired plant by simultaneous operation of the power cycle and the heat pump. Additionally, the utilization of existing facilities (thermal plant power cycle) most likely allows for decreasing the costs of such large-scale storage systems.

1.2.2. Study scope and setup

Scope of the present study is to identify the achievable round-trip-efficiencies for I-PHES systems with different options of integrating stored heat into different types of power plants. Additionally, appropriate heat pump designs are systematically assessed regarding thermodynamic but also technical aspects. The goal is to identify optimized heat pump designs for the investigated I-PHES systems but also to reveal reasonable parameter ranges of heat pumps for I-PHES applications from a general point of view. Some thermodynamic aspects of thermal storage are derived to be considered within the identification process. However, the determination of suitable thermal storage technology for each of the analyzed I-PHES systems is not part of this paper.

Various options for heat integration into power cycles, especially for water/steam cycles (WSC), are conceivable [62]. For example, heat can be integrated into the preheating train of a WSC or into the air flow at the compressor outlet of a gas turbine for the purpose of pre-warming and fuel saving. This study focuses on the creation of large-scale electricity storage capacities with high storage densities and high charging and discharging rates. Therefore, the study limits to a few appropriate options referring to different types of fossil-fired or solar-heated WSCs and combined cycles; it is regarded that the design parameters within one type of power plant can differ noticeably.

A consistent and general approach – referring to exergetic efficiency measures – is used for the evaluation of the round-trip-efficiency of PHES systems. This approach provides beneficial features compared to the very specific approaches used for the evaluation of the PHES systems summarized in Table 1. The approach is applicable to any PHES system independent of the chosen modeling depth of the systems. One of the special features of this approach is that it explicitly includes the efficiencies of the involved sub-processes in the manner that undesirable physical cross-dependencies are avoided.

This feature is used to systematically identify optimized heat pump designs in order to generate understanding of the relevant design criteria and parameters. An appropriate and simple heat concepts can be observed due to the chosen working fluids (CO\(_2\), air, Argon), the design temperature ranges and consequently the thermodynamic cycle designs (trans-critical/super-critical). The proposed systems are mainly dedicated to be small-scale stand-alone applications. Some demonstration projects are planned, although, none of them is realized up to now. The large number of related patent documents shows that intensive development efforts are conducted. The aforementioned indicators allow the conclusion that a feasible technical solution for PHES systems is not available yet. Recently, a concept is reported by [44], which focuses on the integration of a super-critical CO\(_2\) heat pump into solar towers. The heat source of the heat pump is dedicated to be low-temperature heat from turbine steam extractions. Prior to the present work, the authors of this article published a study [45], relating to the option of integrating electricity storage capacities into a specific modern large-scale fossil-fired power plant. Stored heat is either supplied to the steam generator or the preheating train during a discharging sequence. Both concepts [44,45] refer to first approaches to realize integrated PHES systems.

Generally, heat pumps are used for domestic heating – either for warm water supply but also for space heating – or in industrial processes. Ongoing research about heat pumps focuses on (a) their design optimization [46–48], (b) their application within advanced concepts of district heating, (c) heat supply of single buildings [49–55] and (d) the efficiency optimization of industrial processes [56,57]. Heat pumps are considered as appropriate flexible electricity consumers allowing to support the integration of renewable energy sources [58]. They are currently becoming more attractive as emission-free alternatives for low-temperature heat supply [59]. Heat source of the heat pumps is either environmental heat or waste heat from industrial processes or power plants. Environmental heat is extracted from different sources. These sources are mainly ambient air, seawater [60], ground heat [55,61] but also others [60]. Due to their stable temperature level, seawater and especially ground heat are useful and still in focus of current research activities. The typical maximum heat pump temperatures are relatively low. For domestic heating purposes, the temperatures are usually below 75 °C [56] and often significantly lower [51,53,55]. In contrast, their range within industrial processes is usually slightly higher up to about 100 °C [56]. These temperatures are significantly lower compared to the corresponding heat pump temperatures of more or less all mentioned PHES concepts.
Pump design is a recuperated heat pump belonging to the category of compression heat pumps. This design allows to provide heat in different temperature ranges depending on the degree of recuperation. The maximum temperature of the temperature range in which heat is provided by the heat pump is assessed within the limits of 50 °C and 700 °C, which is far beyond the parameter ranges of typical heat pumps. To the authors knowledge, a systematic analysis of recuperated heat pumps within these temperature limits is not yet available. This temperature range covers many conceivable heat integration options for common power cycles and partly also for high temperature power cycles [63] as combined cycles. At the same time, the required heat pump components can be considered as commercially available or not to be too far from state-of-the-art technology; e.g. air compressor technology up to maximum temperatures of several hundred degree Celsius is state-of-the-art within gas turbines. The analyzed concepts rely on the assumption that the heat pump heat source is the environment. Consequently, the concepts do not involve cold storage; this simplifies the system complexity referring to cost-effective systems. Even though, it is possible to reasonably integrate low temperature heat into power cycles, as for example to decrease the WSC condenser temperatures or the gas turbine inlet temperatures. These approaches are practically strongly limited with respect to the exploitable temperature range (e.g. problems with icing). Thereby, the constraint of disregarding cold storage is further justified. Apart from that, appropriate technology to extract heat from the environment can be considered as state-of-the-art due to the common use within classic heat pump applications.

The analysis includes the investigation of different working fluids proposed for PHES systems. In combination with the specified maximum temperature range, this leads to trans-critical or solely super-critical heat pump designs. The remaining degrees of freedom of the heat pumps – the chosen pressure levels – are subject to optimization; technical feasible constraints are regarded. The optimization is carried out to allow for a reasonable comparison of the heat pump designs with regards to the achievable exergetic efficiency and the working fluid. The study includes the assessment of electric heaters as a technically simpler alternative to heat pumps. The characteristic diagram of the efficiency and the related diagrams of other design parameters are then used to identify promising target design ranges of heat pumps for I-PHES applications.

Based on the results of the heat pump assessment, the thermodynamic evaluation approach is used for the estimation of the practically achievable round-trip-efficiencies of the indicated different integration options into power plants; promising options are identified and discussed; the indicated heat pump working fluids and the related cycle designs are outlined. Finally, the combination of heat pumps and electric heaters in series connection is investigated with regards to the achievable I-PHES round-trip-efficiencies. Thereby, the maximum heat pump temperatures can be reduced and the heat pump design becomes technically less challenging.

2. Thermodynamic assessment approach

The round-trip-efficiency (RTE) is defined as

$$\eta_{RT} = \frac{E_{out}}{E_{in}} = \frac{\int_{t_{in}}^{t_{out}} P_{el, out} dt}{\int_{t_{in}}^{t_{out}} P_{el, in} dt}$$

whereby $E_{in}$ is the input of to-be-stored electricity and $E_{out}$ the electricity output, which is finally regained.

A reasonable split of PHES storage concepts into sub-processes and related exergetic efficiency measures can be made according to Fig. 1. The way of splitting the overall process is guided by the idea that the sub-processes with relevant irreversible energy conversions are explicitly shown. The basic assumption is that the sub-processes are adiabatic to the environment. Exceptions are: (a) electric motor and generator heat losses because of friction, (b) heat losses of the storage tanks and (c) heat rejection to the environment, which is required if the overall process is not ideally reversible, see e.g. [19]. The round-trip-efficiency can then be defined using exergetic quantities $E$ according to

$$\eta_{RT} = \frac{E_{out}}{E_{in}} = \frac{E_{GM}}{E_{GM}} = \frac{E_{P2H} - E_{P2H}}{E_{GM}} = \frac{E_{in, GM}}{E_{in, GM}} = \frac{E_{out, H2P} + E_{H2P}}{E_{GM}} = \frac{E_{out}}{E_{GM}}$$

(2)

Fig. 1. Energy flow diagram of PHES systems divided into sub-processes and related efficiency measures for the estimation of the round-trip-efficiency. Case: PHES system with hot storage (above ambient temperature) and cold storage (below ambient temperature); this is the case for the first three (Lit-1 to Lit-3) PHES systems in Table 1.
in connection with the relation

\[ \frac{E_{\text{HP}}} {E_{\text{P2H}}} = \frac{E_{\text{S,in}}}{E_{\text{S,in}}} \cdot \frac{E_{\text{S,out}}}{E_{\text{S,out}}} \cdot \frac{E_{\text{HP2H}}}{E_{\text{HP2H}}} \]  

(3)

where the superscripts + and − refer to the upper and lower characterizing (thermodynamic averaged) temperatures establishing the thermal potential. The exergies \( E \) of the heats \( Q \) are defined by

\[ E = \eta \cdot Q = \left(1 - \frac{T_{\text{f}}}{T_{\text{i}}} \right) \cdot Q \]  

(4)

The sign of \( E \) is determined by the level of the characterizing temperature \( T \) in relation to the ambient temperature \( T_{\infty} \); it denotes whether the exergy flow is directed accordingly or contrary to the heat flow. Consequently, the signs are dependent on whether the PHES system includes hot and/or cold storage.

A useful feature of splitting the overall process according to Eqs. (2) and (3) is that most of the efficiency measures clearly point to only a single loss category.

Within the concept analysis of the present study, the heat source of the heat pump is the environment; a cold storage is not part of the system. Eq. (2) then simplifies to

\[ \eta_{\text{RT}} = \frac{E_{\text{out}}}{E_{\text{in}}} = \prod_i c_i = E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \cdot E_{\text{S,in}} \]  

(5)

while Eq. (3) becomes irrelevant. Compared to approaches based on energetic quantities, the benefits of this approach are (a) equal scale for all subsystem efficiency measures, (b) no inherent physical cross-dependencies between the efficiency measures and (c) a clear and consistent relation between the sub-system efficiencies and the overall efficiency such that the maximization of each subsystem efficiency leads to a maximization of the RTE.

3. Heat pump model & constraints

A recuperated heat pump is the most simple heat pump design that allows to provide heat of a specified quality dependent on the degree of recuperation. Fig. 2 shows the heat pump configuration and the corresponding qualitative T-s-diagram introducing the used nomenclature.

With regards to the application of such a heat pump design in I-PHES applications, the only required boundary condition is the definition of the temperature range of the heat provided by the heat pump. According to Fig. 2, it is of interest to identify optimized heat pump designs for different combinations of the minimum temperature \( T_{\text{HP4}} \) and the maximum temperature \( T_{\text{HP3}} \), which characterize the quality of the provided heat. Alternatively, to the explicit specification of the two temperatures, only the maximum temperature in combination with the following parameter defines the HP design.

\[ \gamma_{\text{HP}} = \frac{T_{\text{HP4}} - T_{\infty}}{T_{\text{HP3}} - T_{\infty}} \]  

(6)

This parameter can be conceived as a recuperation rate with the following properties

\[ \gamma \to 0 \equiv \text{simple HP cycle (no recuperation)} \]

\[ \gamma \to 1 \equiv \text{HP cycle with high degree of recuperation} \]

The combination of the parameters \( T_{\text{HP4}} \) and \( \gamma_{\text{HP}} \) allows to present the results of the optimization in compact form by 2D characteristic diagrams. They are therefore chosen as design defining parameters.

However, the two parameters do not completely define the cycle. There are further degrees of freedom, which require optimization for the identification of the best process design for a specific application. This refers to the pressure levels. Refer to Fig. A.14 in Appendix A for the setup of the optimization problem.

Certain parameters of the HP devices need to be predefined. In terms of this, it is avoided to overestimate the potentials of the technology by assuming too optimistic device properties.

The only exception refers to the availability of high temperature compressors. Air compressors are available up to maximum temperatures of about 600 °C, e.g. implemented in modern gas turbines. Compressors for the working fluid CO2 are only available up to allowable temperatures of about 450 °C; significant engineering effort is required to design CO2 compressors at equal maximum temperatures as for air compressors, although it is theoretically possible [64]. Nevertheless, the maximum temperature for the purpose of this study is chosen at 700 °C to cover the
operational temperature ranges of many different types of thermal plants.

The main source for irreversible energy conversion inside the heat pump process is the compressor, especially in trans-critical HP processes. Therefore, the isentropic compressor efficiency is assumed to be a conservative value of 80%. The turbine efficiency is therefore the isentropic compressor efficiency is operational temperature ranges of many different types of thermal plants.

The applied pressure constraints are depicted in Fig. 3. In particular, the efficiency of trans-critical CO2 HP cycles is dependent on the pressure level; as the results will show, optimized designs at low and medium pressure constraints are high maximum pressure levels if the HP is dedicated to provide high-temperature heat. In this parameter range, maximum pressures of several hundred bar are indicated. This is not realistic for practical applications. Therefore, the maximum pressure is limited to $p_{lim\text{-max}} = 200 \text{ bar}$. According to [65], this is a reasonable technoeconomic limit for the design of CO2 compressors. The limit of the minimum pressure in direction of low values is set to $p_{lim\text{-min}} = 1 \text{ bar}$. Potential problems with the intrusion of environmental air into the HP cycles (sealing) is thereby avoided. Additionally, high volume flow rates and related large-size component designs are restricted. In case of trans-critical cycles, the minimum pressure is also limited in direction of high pressures. This refers to the minimum evaporator temperature difference. Although, it is technically possible and thermodynamically desirable to realize lower values, the minimum evaporator temperature difference is restricted to $\Delta T_{lim\text{-eva}} = 15 \text{ K}$. As a result, the maximum evaporation temperature is $T_{eva} = 15 \text{ °C}$ and related maximum evaporation pressure is $p_{lim\text{-max}} = p_c(T_{eva}) = 34.84 \text{ bar}$. The practical reason is that large-sized heat exchanger designs are avoided with the goal to create a compact system.

Total pressure losses of all components are assumed negligible with regards to this conceptual study.

Also the minimum temperature of the heat pump cycle $T_{HP}$ is subject to limitations. In case of trans-critical CO2 HPs, a difference of 20 K to the triple point is kept; for super-critical cycles using air or Argon, the same difference is applied to the critical point. Both limits refer to a save design in proximity to the sublimation region or the wet steam region.

4. Optimized heat pump designs

4.1. Optimized CO2 heat pump designs

Fig. 4 shows the characteristic diagrams of CO2 heat pumps for different parameters. The maximum efficiencies of COEP ≈ 80% are reachable at recuperation rates of $\gamma_H = 0.8$. At moderate to high maximum temperatures of $T_{max,H} > 300 \text{ °C}$ and medium recuperation rates of $\gamma_H = 0.3 - 0.7$, an extended region of achievable efficiencies of COEP > 70% is present. A narrow part of this area extends down to $T_{max,H} = 200 \text{ °C}$ and $\gamma_H = 0.2$.

The characteristic diagrams are limited by the blue curve in direction of high maximum temperatures and low recuperation rates. The limit is reasoned by the expander outlet temperature.

For four selected HP designs in different regions of the characteristic diagram, the impact of a change of the efficiency of the compression and expansion devices (reference: $\eta_{C,\text{ref}} = 0.8/\eta_{\text{ref}} = 0.9$) is shown. The sensitivity of the COEP on the device efficiencies is less significant at high temperatures and recuperation rates. Therefore, related designs are more suitable for off-design operation.

The COPs take small values of ≈1.2–1.5 in the range where the maximum COEPs are located. Generally, the coefficient of performance is lower than COP ≈ 2 in the area of promising exergetic efficiencies of COEP > 70%.

The pressure ratios are below $\Pi \approx 15$. In the area where the maximum COEPs are established, they range between $\Pi = 2 - 6$.

The efficiencies of the HP designs are strongly dependent on the chosen pressure levels. The maximum pressures are high even at low maximum temperatures. In direction of low recuperation rates and high maximum temperatures, the designs are constrained by the maximum pressure limit of $p_{lim\text{-max}} = 200 \text{ bar}$. Here, the optimized designs are defined by lower minimum pressures, which at the same time correspond to lower evaporation temperatures. This explains the corresponding drop of the COEP: The heat pump receives heat at lower temperatures. As long as the designs are not constraint by the allowed maximum pressure, efficient heat pump designs are characterized by the highest possible minimum pressure ($p_m(T_{HP}) = 34.84 \text{ bar}$) resulting from the constraint evaporator temperature difference. Focusing on the area of $\gamma_H > 0.5 - 0.6$, optimized HP process designs are less influenced by the pressure constraints. Here, the required maximum pressures are below 200 bar even at high maximum temperatures. In the range of $\gamma_H = 0.6 - 0.8$, the designs pass over to purely supercritical designs.

4.2. Optimized air & Argon heat pump designs

The air and Argon heat pumps are characterized by pure supercritical designs; at all locations of the HP cycle, the states are gaseous within the analyzed parameter range. Fig. 5 visualizes that high COEPs can be reached as for CO2 designs. However, the COEPs are significantly lower (up to about 20%) at low temperatures and recuperation rates compared to CO2 heat pumps. This is caused by the shape of the super-critical cycle. Environmental heat, significantly below ambient temperature, is received by the heat pump. In general, the same effect of the device efficiencies (compressor & expander) on the overall process efficiency as for CO2 heat pumps can be observed. Only at low temperatures and low recuperation rates, noticeable differences can be recognized. The expander efficiency is naturally more important for the supercritical HP designs compared to trans-critical processes using CO2.

The working fluid Argon leads to an almost identical characteristic diagram and equal achievable efficiencies. Both fluids – air and Argon – show ideal gas properties in the investigated parameter range. The only difference between air and Argon is the limiting line in direction of high maximum temperatures and low recuperation rates. The possible heat pump design area is less

For interpretation of color in Figs. 4 and 9, the reader is referred to the web version of this article.
Fig. 4. CO₂ heat pumps – characteristic diagrams of different parameters.
extended for Argon (dotted blue line). This is caused by the fact that the wet steam region (critical point) of Argon extends to higher temperatures. Apart of this, the diagram for air is valid for Argon as well.

Referring to the COP in Fig. 6, the trends are comparable to the COP of the CO₂ designs. Only at low temperatures and recuperation rates, the differences are more pronounced.

Fig. 7 shows the required pressure ratios of the heat pumps for air and Argon. For air, these ratios are below \( P/C_2 \) and hence slightly lower compared to the related CO₂ designs. For the Argon designs, the pressure ratios are significantly lower. Even at high temperatures and low recuperation rates, values of \( P = 5 \) are not exceeded. This feature of Argon heat pumps is already well addressed in literature related to PHES applications, see e.g. [20,26].

In contrast to the findings for CO₂ heat pumps, the pressure levels of super-critical air or Argon heat pumps have almost no influence on the efficiency. The COEPs tend to be highest at low pressure levels. Here, the differences in the COEP are lower than 0.5% and therefore in range of the modeling uncertainty. As a result, the pressure levels can be chosen referring to other design aspects.

4.3. Electric heaters

Optimization of the application of electric heaters for power to heat conversion is not required; there are no degrees of freedom in the design. A temperature increase of a liquid storage material can directly be realized by transferring heat from the hot ohmic resistor to the storage medium. The COP is always at a value of one, whereas, the exergetic efficiency depends strongly on the target temperature. Fig. 8 shows the characteristic diagram of the exergetic efficiency of electric heaters.

The achievable efficiencies are significantly lower than for all assessed heat pump designs. The difference is lowest where the heat pumps achieve the highest efficiencies – at high maximum temperatures and recuperation rates. Here, the electric heater efficiencies are 15–20% lower.

5. Heat pump designs – discussion

The COEP diagrams of the heat pumps of different working fluids are super-positioned with the goal to identify the parameter regions in which a certain working fluid and the related heat pump design provides the highest efficiency. Fig. 9 shows the resulting combined diagram, which is cut-off for efficiencies lower than 50%.

The most important observation is the limited range in which trans-critical CO₂ heat pumps allow to achieve the highest efficien-
cies; trans-critical designs are not in any case the best choice, particularly not in range of high maximum temperatures and recuperation rates. The reason is that the advantage of a trans-critical design, which allows to pick up heat very close the temperature of the heat source, is not dominant here. However, the differences between CO2 cycles and air or Argon cycles are low; they are in range of less than 3% points. Consequently, it could also be reasonable to design CO2 heat pumps in this region.

The minimum evaporator temperature is assumed to be $\Delta T_{\text{lim, ev}} = 15 \, \text{K}$. A reduction of this temperature difference is desirable in order to increase the average temperature of the heat input of the CO2 HP; by this means the efficiency is increased. The blue colored area within the CO2 region shows HP designs, which are constraint by this temperature difference. Only for these designs a reduction of the evaporator temperature difference has a beneficial impact. A decrease of the evaporator temperature difference has almost no impact on the size of the CO2 region and consequently on a preferred application of CO2 heat pumps; only at the part of the border between the CO2 and the air & Argon region – where the blue colored region touches the border – the area would extend in direction of higher recuperation rates.

The CO2 designs in the area above the blue colored region (at higher temperatures in range of $\gamma_{\text{HP}} \approx 0.25 - 0.5$) would not profit from a modification of the evaporator temperature difference, as long as the maximum pressure constraint of $P_{\text{HP,max}} = 200 \, \text{bar}$ is not shifted to higher values as well. Referring to practical realizations, the constraint designs in this region provide advantages. The evaporator can be designed at higher evaporation temperature differences; hence, the evaporator designs are of lower size (smaller heat transferring surfaces) and contribute to a compact overall design of the HP.

Especially, at low maximum temperatures and recuperation rates, the CO2 HP designs are characterized by expansion from the liquid state region into the wet steam regions at high steam fractions. It might be technically challenging to design appropriate turbines. The area highlighted in red color shows the corresponding design region, which is affected. Especially in this part of the characteristic diagram, the CO2 designs show significantly higher COEPs than the other options. However, if it is not possible to realize the required expansion devices, throttle valves have to replace the turbines. If the turbine is replaced by a throttle valve, a significant drop of the COEP can be observed in range of 6–12%, dependent on the specific design. The COEPs are then in order of about 60%. Consequently, these designs are less promising for I-PHES applications.

Out of the red colored area, in direction of higher maximum temperatures and recuperation rates, the designs are characterized by expansion from the gaseous state region into the wet steam region. Consequently, the steam fractions at the outlet of the expander are higher but still at values where droplet erosion of the blades can occur. Usually, the limit of the steam fraction for steam turbines in direction of low values is in range of 85–90% [66]. However, the fluid properties of CO2 are differing from those of water; it might be possible to realize turbines operating at lower steam fractions. In any case, designs located close to the red colored area are not preferable.

In [33] it is referred to a study, which reveals that turbo devices using CO2 can potentially be designed at higher efficiencies compared to air devices at the same boundary conditions. The analysis carried out in the present study does not take this aspect into account – the assumed compressor and turbine efficiencies are independent of the fluid type. Consequently, it might be possible to extend the region in which it is preferable to apply CO2 heat pumps. Further investigation is required to clarify this point.

The assessment shows that the efficiency of CO2 heat pumps is strongly dependent on the chosen pressure levels, especially, if low or medium recuperation rates are established. Consequently, operation in off-design due to changed boundary conditions is accompanied by significant changes of the efficiency. With regards to
applications within I-PHES systems this is a disadvantage. The operational boundary conditions are influenced by natural fluctuations of the ambient temperature, for example. Additionally, it has to be considered that a storage system might not operate at nominal conditions for long times; it is dedicated to respond on the current residual load of the electricity grid. Consequently, it is likely that the operation is characterized by frequent start-ups and/or off-design operation. In particular, CO₂ heat pumps are affected negatively thereby due to their sensitivity on changes of the boundary conditions. However, this affects HP designs using other working fluids too. In any case, further investigation with regards to off-design operation of the heat pump is indicated. This requires more detailed heat pump models covering the part-load characteristics. Also, the dynamic properties of the heat pump should be analyzed.

An I-PHES system, which is fast responding to an electricity surplus of the grid, requires fast start-up features. Comparing the CO₂ and air or Argon heat pumps, the main difference in the practical design is the evaporator, which is required in the first case. Assuming typical start-up times of gas turbines in order of 10–20 min [67] as a rough estimate for the start-up times of super-critical heat pumps, trans-critical CO₂ HPs will require even more extended start-up times. This aspect might change the choice of the HP design with regards to the working fluid to a preferred application of super-critical designs using air or Argon, especially, in range of the border area in the characteristic diagram. In

\[\gamma_{HP} [-]\]

\(\approx\) Region of maximum P2H efficiency using an electric heater

\(\approx\) Region of maximum P2H efficiency using a CO₂ heat pump

\(\approx\) Region of maximum P2H efficiency using an Air / Argon heat pump

\(\approx\) Region characterized by \(\Delta COEP_{CO₂\rightarrow Air} \leq 3\%\)

\(\approx\) Region characterized by liquid to wet steam phase change inside the turbine (CO₂)

\(\approx\) Region characterized by optimized heat pump designs restricted by \(T_{eva} = 0{\degree}C\) (CO₂)

Fig. 9. Combined characteristic diagram – exergetic heat pump and electric heater efficiencies COEP.
Fig. 9, the area highlighted in yellow color is defined by a difference of the HP efficiency between CO₂ and air or Argon designs of less than 3% points. As can be seen, this area is wide and covers large parts of the very promising design region of CO₂ HPs. Here, air or Argon heat pumps are a promising alternative if the low efficiency drop is accepted.

A comparison of the size of the heat pump designs for different working fluids is carried out. The volume flow rate at the compressor outlet is taken as a measure for the relative comparison of the heat pump sizes. Air or Argon heat pumps allow for a free choice of the pressure level with regards to the COEP, while CO₂ heat pumps do not. Accordingly, it is assumed that the air heat pumps are designed for maximum pressures of $p_{\text{max}} = p_{\text{HP}} = 200$ bar, while the CO₂ heat pumps are designed according to their optimized pressure levels; Fig. 10 shows the ratio of the volume flow rates $\nu$ at the compressor outlet.

It can be seen that $\nu$ is always higher than a value of one; air heat pumps are of larger size compared to related CO₂ designs. However, the differences in the sizes are moderate. Especially in ranges of COEP > 70%, the volume flow ratio is $\nu < 2$. If the pressure level of the air HP is chosen lower, the differences in the sizes increase significantly. In case of HP designs using Argon, lower volume flow ratios can be realized; the reason is the higher density of Argon compared to air. Finally, it is subject to an economic optimization if large-sized components or increased mechanical requirements on the components at high pressure loads are less problematic. In any case, air heat pumps have to be designed at elevated pressure levels if their facility size is intended to be comparable to CO₂ HPs.

With regards to the basic assumptions of the HP model, three of them require further discussion. First, total pressure losses are disregarded at all. However, pressure losses are introduced by the recuperator; the larger its size, the more significant are the pressure losses. This means that the designs with a high recuperation rate are overestimated in their efficiency. The results show that the COEP increases sharply in direction of high temperatures and recuperation rates. Therefore, an analysis including pressure losses will not change the basic findings. Furthermore, disregarding the pressure losses has no impact on the identified regions in which different working fluids allow for the most promising heat pump designs.

The second assumption which requires discussion refers to the isentropic device efficiencies. Caused by its definition, the isentropic efficiency is dependent on the pressure ratio. Therefore, a direct comparison of devices operating at different pressure ratios with the same isentropic efficiency is complicated. A compressor working at a higher pressure ratio effectively has a higher efficiency; for an expander the situation is reverse. Therefore, the effects tend to compensate each other. For the assessment of a specific design (equal pressure ratios of compressor and expander), this effect does not introduce significant problems. However, a comparison of different designs in different regions of the characteristic diagram is affected with regards to the absolute values of the heat pump efficiency. The pressure ratios are rising in direction of high temperatures and low recuperation rates; accordingly, the efficiencies are generally overestimated in this direction. At the same locations in the characteristic diagram, the HPs using CO₂ or air have similar pressure ratios. Therefore, a direct comparison is possible. HP designs using Argon have significantly lower pressure ratios and are consequently underestimated. More detailed studies are required to quantify the differences.

The third point relates to the assumption of the terminal temperature differences of the heat exchangers of 5 K. Considering economic system designs, this assumption might be too optimistic. At higher terminal temperature differences, the heat pump efficiency tends to decrease.

Taking the discussed aspects into account, the reasonable design range of heat pumps for I-PHES applications using environmental heat as heat source is characterized by medium to high maximum temperatures of the provided heat of $T_{\text{max,HP}} \approx 300 – 600$ °C and moderate recuperation rates of $\gamma_{\text{rec}} = 0.25 – 0.6$. High efficiencies of COEP > 70% can be achieved. At the same time, the dependence of the heat pump efficiency on changed boundary conditions is comparatively low. Especially, super-critical designs using air or Argon are promising in wide parts of this design range; trans-critical CO₂ designs are superior in the lower part of the above specified parameter range. In the identified design region, also a good utilization of the heat storage material is possible. Considering the preferred application ranges, the required compressors are available. If it is possible to design appropriate expander, high efficient CO₂ heat pumps can also be created in range of maximum temperatures of $T_{\text{max,HP}} = 150 – 300$ °C and recuperation rates of $\gamma_{\text{rec}} = 0.1 – 0.25$. The design of heat pumps for I-PHES applications in range of high maximum temperatures and recuperation rates of $\gamma_{\text{rec}} > 0.6$ is not indicated, although high efficiencies are theoretically possible; the utilization of the heat storage material is low and very large recuperators are required. Electric heaters are generally not competitive to heat pumps referring to the efficiency.

6. Estimation of the round-trip-efficiencies of different I-PHES concepts

6.1. Assessed I-PHES concepts

The investigated options for the integration of large amounts of heat into thermal plants are summarized in Table 2.

Several types of WSCs are assessed. The reason is that they are used for different applications with regards to the energy source. Consequently, they differ in their design parameters and their applied level of technical sophistication and as a result in their efficiencies. Additionally, as was shown in the previous sections, the achievable heat pump efficiencies are strongly dependent on the specific design parameters. Hence, the integration of PHES storage capacities into these cycles results in different round-trip-efficiencies. Even for one type of WSC, the design parameters distin-
guish. Therefore, typical temperature ranges for the different types of WSCs are considered, according to Table 2.

In case of heat integration into the steam generator of the WSC, the plant is solely supplied by heat from storage during a discharging sequence; no fossil or solar supply is required. Practically, a second steam generator would have to be integrated in parallel to the original one, which is suitable for the chosen storage media or heat transfer fluid. In case of fossil-fired WSCs, the integration of a molten salt storage and an appropriate steam generator could be suitable for a practical realization of an I-PHES concept on this plant type.

Another option is the integration of heat into the preheating train of the WSC. It is distinguished between heat integration into the low-pressure and the high-pressure preheating train. The reason is that the corresponding heat pump design are very different referring to the maximum temperature as well as the degree of recuperation. In this case, the WSC operates with dual heat supply; the steam generator operates as usual and additional heat from storage is used for the supply of preheaters from outside; the corresponding steam extractions for the supply of the preheating train are closed and additional power is generated by the turbine; a series of preheaters in parallel connection to the original preheaters is required. Although, many heat integration options are theoretically available – referring to the possible combinations of supplied preheaters – the two options regarded here can serve for the estimation of the order of the achievable round-trip-efficiencies.

Two options are assessed with regards to combined cycles. Similar to the heat integration into steam generators of stand-alone WSCs, heat could also be utilized to supply the heat recovery steam generator. In this scenario, the gas turbine would not be operated within a discharging sequence. The second option is to use stored heat for the pre-warming of compressor outlet air; this reduces the required natural gas input for the gas turbine.

6.2. Assumptions

Referring to the assumptions about the temperature ranges in Table 2, the most promising heat pump fluid type (and process) as well as the corresponding efficiency can be evaluated based in Fig. 9. The assumed efficiencies of the heat-to-power cycles are summarized in Table 2. They are result of an analysis based on detailed simulation models of these cycles. The efficiency ranges differ between the types of water/steam cycles. Because of the fact that WSCs reject heat to the environment via the condenser at temperatures almost identical to ambient temperature, the exergetic efficiency is in good approximation a pure measure for internal losses – irreversible energy conversion within the cycle. Therefore, it is a measure for the technological sophistication of the power cycle configuration and its devices. For example, WSCs of coal-fired plants are usually more sophisticated in their designs compared to waste-to-power WSCs. This refers to e.g. the length of the preheating trains.

For similar reasons, differences in the efficiencies of the heat transfer from storage to the H2P cycles are present; the utilized values are shown in Table 2. It is always presumed that the storage medium or the heat transferring fluid does not undergo a transcritical phase change during the heat transfer to the H2P cycle. The situation is different for the working fluid of the H2P cycle. In case of steam generators of WSCs, heat is received while the working fluid is subject to a multi-phase state change. In case of preheating trains, heat is obtained while the fluid stays liquid. The heat transfer efficiencies in Table 2 take these aspects into account.

Table 3 summarizes the presumed efficiencies of the other sub-processes.

For the I-PHES concepts, which include dual heat supply by stored heat and fossil or solar sources, it is not obvious how the efficiency of the H2P power cycle can be estimated. It is required to determine the individual conversion efficiencies of the heat sources with regards to their contribution to the overall power output. Both energy conversions take place at the same time inside the same system. Because of the numerous interactions inside the cycle and the fact that the system properties are subject to nonlinearities, a clear distinction cannot be made. For this reason, it is assumed that both heats are converted with equal efficiency; this corresponds to the assumption that the individual conversion efficiencies are identical to the overall efficiency of the heat to power cycle

\[
\eta_{\text{H2P}} = \frac{\sum F_{G1}}{\sum F_{\text{H2P}1}} = \frac{\sum F_{G1}}{\sum F_{\text{H2P}1}} \Rightarrow \eta_{\text{H2P}} = \eta_{\text{i}}
\]

According to the definition of the exergetic efficiencies, this assumption is reasonable because the conversion efficiency is almost a pure measure of internal irreversible energy conversions and the applied technology respectively.

6.3. Results and discussion

Fig. 11 shows the estimated round-trip efficiencies (RTE) for the different I-PHES concepts. Additionally, the most promising heat pump working fluid is indicated according to the findings of the characteristic diagram in Fig. 9. Related to the process designs, all CO2 heat pump cycles are trans-critical; the air/Argon cycles are super-critical. If both, CO2 and air/Argon are mentioned for

| Table 2
<table>
<thead>
<tr>
<th>Thermal plants - heat integration options and assumed parameters.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal plant type</td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>Coal-fired WSC</td>
</tr>
<tr>
<td>High-pressure preheating train</td>
</tr>
<tr>
<td>Low-pressure preheating train</td>
</tr>
<tr>
<td>Waste-to-Power WSC</td>
</tr>
<tr>
<td>Solar tower WSC</td>
</tr>
<tr>
<td>High-pressure preheating train</td>
</tr>
<tr>
<td>Low-pressure preheating train</td>
</tr>
<tr>
<td>Combined cycle</td>
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<td></td>
</tr>
</tbody>
</table>

7 A confidential handling of the information does not allow to present further details. However, the measures can be considered as good estimates of the order of magnitude of the efficiencies, also relatively to each other. The efficiency ranges cannot be considered as hard boundaries; they are also not the result of a strict stochastic analysis.
one concept, this means that CO₂ heat pumps are preferably indicated by the results of the thermodynamic assessment. However, a RTE of a similar value with a difference of less than −2% can be achieved using the working fluid air or Argon.

It can be seen that the achievable RTEs for most concepts are in the range of 50–60%; the uncertainty introduced by the assumed efficiency ranges of the sub-processes is about 10–15%.

Most promising with regards to the RTE are the I-PHES concepts with a high maximum temperature and a medium to high minimum temperature of the to-be-integrated heat. Especially, this refers to the concepts where heat is integrated into the steam generator of the WSC (CF-SG & CSP-SG) and into the compressor outlet air of a combined cycle (CC-GT).

In the case CC-GT, the highest RTEs of above 60% are reachable. However, the required maximum temperatures are in range of the current limit of the allowed maximum temperatures of state-of-the-art air compressors required for the heat pump cycle; for a practical application, the enhancement of the permissible operational range of the compressors would be necessary. Additionally, the utilization of the storage material is low; consequently, large storage units would be required. Also, the necessary modifications on the gas turbine are probably difficult. Therefore, a realization of this I-PHES concept is technically challenging.

The concepts with heat integration into the steam generator of WSCs are within the reasonable design range of heat pumps, which was identified in the previous sections. Especially, if the parameters of the WSCs are elevated, the achievable RTEs are high because of two reasons. On the one hand, the corresponding heat pump cycles can be designed highly efficient. On the second hand, the efficiencies of the H₂P cycles are of high level because of their sophisticated technical design. For the concepts CF-SG & CSP-SG, heat pumps using the working fluids CO₂ or air/Argon are suitable. However, state-of-the-art CO₂ compressors for the heat pumps are not available because the required maximum temperatures are too high. Consequently, this I-PHES concept is preferably realized using super-critical heat pumps. In case of heat integration into WSCs of lower parameters, like waste to power cycles, the application of CO₂ compressors is indicated; appropriate compressors are available, although they would have to be designed close to their current temperature limit. According to the heat pump assessment, the required turbines operate at high steam fractions but out of the region of liquid to wet steam expansion; suitable turbine designs are required and may introduce technical challenges for the heat pump realization. The option of heat integration into steam generators of WSCs is promising from the thermodynamic point of view; also the technical challenges are moderate. In particular, this I-PHES concept is interesting in connection with WSCs of solar tower plants (CSP-WSC); heat storage in form of molten salt tanks is already available; only the installation of the heat pump is required.

The integration of storage capacities into combined cycles by means of heat supply to the heat recovery steam generator (HRSG-CC) also leads to RTE values above 50%. Appropriate heat pump designs use the working fluid air. The design is defined by a low degree of recuperation, which leads to an efficient utilization of the storage material. The identification of suitable heat storage fluids might be difficult, because they are used in a wide temperature range. An alternative could be the utilization of solid storage materials. In this case, the applied assumptions with regards to the heat transfer efficiencies and the storage efficiency are optimistic; the achievable RTEs are most likely lower than estimated here.

The I-PHES concepts on preheating trains of WSCs are not within the preferred design range of heat pumps. The achievable round-trip-efficiencies differ strongly for the assessed application cases. Only if extended preheating trains with high pre-warming temperatures are present (CF-HPT & CF-LPT), RTEs up to 55% are possible. In case of less extended preheating trains (CSP-HPT & CSP-LPT), the RTEs are significantly lower in range of 45%. For these I-PHES concepts, the application of CO₂ heat pumps is indicated.

<table>
<thead>
<tr>
<th>Motor</th>
<th>Heat pump</th>
<th>Heat transfer to storage</th>
<th>Storage</th>
<th>Heat transfer to H₂P cycle</th>
<th>H₂P cycle</th>
<th>Generator</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.99</td>
<td>Characteristic diagram (Fig. 9)</td>
<td>0.95–0.99</td>
<td>0.99</td>
<td>According to Table 2</td>
<td>According to Table 2</td>
<td>0.99</td>
</tr>
</tbody>
</table>

Table 3: Exergetic efficiencies of the sub-processes.

Fig. 11. Estimated round-trip-efficiencies of I-PHES concepts on different thermal plants.
However, in case of I-PHES concepts on the HPTs, also super-critical designs using air/Argon can be used without significant impact on the achievable RTE. The technical risks are comparatively low for I-PHES concepts on the low-pressure preheating train; only low-pressure steam extractions are closed and consequently the modifications of the operational conditions of the WSC are low. In contrast to that are applications on the high-pressure preheating train; the closing of the high-pressure steam extractions changes the operational conditions of the steam generator; the mass flows in the reheat section are increased. Additionally, large parts of the steam turbine are operated in off-design at higher load. Therefore, it is questionable if this I-PHES concept is practically realizable.

The present investigation does not include economic aspects. For this reason, the conclusions might be modified if they are taken into account as well. It might be found that concepts of low or medium temperatures are preferable, even if the achievable efficiencies are lower. Furthermore, a more detailed technical analysis of the I-PHES concepts is necessary, which goes beyond the brief analysis provided here. A general finding related to the design of high efficient I-PHES concepts should be considered. The partial derivatives of RTE are

\[
\frac{\partial \eta_{RT}}{\partial \omega_j} = \prod_{i \neq j} \gamma_i
\]

As a result, the improvement of the sub-process with the lowest efficiency provides the most significant benefit on the overall efficiency. Consequently, the development of sophisticated heat pump designs is indicated for all I-PHES concepts as the most effective way to improve these systems.

7. Combination of heat pumps and electric heaters

Within the identification of efficient heat pump designs for the conversion of power into heat, electric heaters were found to be less efficient than heat pumps. The resulting round-trip-efficiencies of the I-PHES concepts using electric heaters are shown in Fig. 12. It can be seen that the achievable round-trip efficiencies are significantly below 50% for most of the cases. Only, the concepts with high maximum temperatures are in range of 45%.

However, the usage of electric heaters can provide advantages. First, their response times with regards to start-ups are shorter compared to heat pumps. Second, their design with regards to high temperature applications is less challenging, because there are no
high-precision rotating components and significant thermal masses. For this reason, it can be reasonable to combine electric heats with heat pumps in order to exploit their beneficial properties and to avoid their specific disadvantages. For this purpose, it is reasonable to combine the two devices in series connection.

In this configuration, the heat pump generates heat in the lower temperature range and the electric heater within the upper temperature range. By this means, the maximum temperature of the heat pump can be decreased and the involved components can be designed for lower thermo-mechanical loads. At the same time, the electric heater works at high temperatures and high recuperation rates; this is the application in which the electric heater provides the highest efficiencies. Additionally, the start-up times during a charging sequence can be minimized, which allows for a faster response of the system to surplus electricity in the grid. During a start-up, the efficiency of the P2H unit is then significantly decreased because the electric heater has to provide heat at lower recuperation rates as long as the heat pump does not operate in its design point.

Referring to nominal operation of the power to heat unit, Fig. 13 shows the achievable RTEs. The cases with HP = 100% & EH = 0% correspond to the average RTE values already discussed in the previous section. The two other cases show the reachable RTEs if the lower 90% or 80% of the temperature range of the provided heat of the P2H unit are realized by the heat pump and the remaining 10% or 20% are generated by the electric heater.

It can be seen that the RTE values decrease only by a few percent points. Especially, for the I-PHES concepts with a high maximum temperature, the decrease is low. Exactly these cases would be preferable for a combined application of heat pumps and electric heaters for the above mentioned reasons. For I-PHES concepts with lower maximum temperature and significant recuperation rates (e.g. CF-LPT), this conclusion cannot be made: the RTEs decrease by values up to 10%.

8. Summary

For Pumped-Heat-Electricity-Storage systems, a thermodynamic assessment approach based on exergetic quantities was presented. The approach provides advantageous features compared to other approaches. One of these features is that the efficiency measures of the involved sub-processes allow for individual optimization independent of other sub-processes.

This feature was exploited in order to systematically identify optimized recuperated heat pump designs using environmental heat as the heat source for the purpose of high temperature heat supply. The pressure levels of the heat pumps were optimized for different qualities of the provided heat and consequently for different potential applications within integrated Pumped-Heat-Electricity-Storage (I-PHES) systems integrated in thermal plants. The heat pump assessment included three types of working fluids, which are proposed in literature related to PHES systems – CO2, air and Argon. A characteristic diagram was derived, which combines the results of the heat pump optimization; for defined heats provided by the heat pump, the most efficient working fluid and related process design is indicated.

It was found that the exergetic heat pump efficiency increases with rising temperature of the heat. At maximum temperatures above 300 °C, the super-critical designs using air or Argon are equally or even more efficient than trans-critical designs using CO2. The most reasonable design range of heat pumps for applications within storage concepts was identified at maximum temperatures in range between 300–600 °C and medium recuperation rates of $\gamma_{\text{lim}} \approx 0.25 - 0.6$. Here, efficiencies of above 70% can be achieved. The target design range was identified including technical aspects. Heat pumps using the working fluid CO2 are able to establish high efficiencies also at temperatures below the identified design range. In this design area, the expansion inside the turbine is characterized by phase changes from the liquid into the wet steam region at low steam fractions. For such applications, appropriate turbines are required, which might be technically challenging to realize. An alternative replacement of the turbine by a throttle valve leads to a significant drop of the heat pump efficiency by 6–12%.

Based on these results of the heat pump analysis, different options of heat integration into several types of thermal plants were assessed and the achievable round-trip-efficiencies estimated. The order of the achievable round-trip-efficiencies is roughly in range of 50–60%. Most promising are the concepts with heat integration into the steam generators of water/steam cycles. This is especially the case if the design parameters of the steam cycle are high and the preheating trains extended. Round-trip-efficiencies of about 60% are realistic. The heat pumps can either be trans-critical using CO2 or solely super-critical using air or Argon. The application of this concept on solar tower water/steam cycles is comparatively simple. This is caused by the heat storage in form of molten-salt tanks and the possibility to use the related steam generator; both are already part of these types of power plants.

It was shown that electric heaters are not competitive to heat pumps with regards to the achievable efficiencies; they are in best case 15–20% lower. For technical and operational reasons, a combination of heat pumps and electric heaters in series connection can be useful. The efficiency of such a combined power-to-heat unit decreases only by 2–5% points. This is the case if the electric heater is only used for the heating of the upper 10–20% of the temperature range of the provided heat and if the temperature level is high in general.

The present study did not include the assessment of economic aspects, which might modify some of the conclusions derived here. So, an economic assessment is required in a next step. Some technical aspects were outlined and briefly discussed, which require a more detailed analysis. This refers to (a) the technical analysis and/or development of certain devices required for the heat pumps, (b) the analysis of the heat pump off-design characteristics, (c) the investigation of the dynamics of Pumped-Heat-Electricity-Storage systems with regards to start-ups and load changes and (d) the choice of specific storage materials for different integrated PHES systems.

Appendix A. Optimization problem

The basic non-linear equation system being part of the optimization problem is shown in Fig. A.14. All variables are written in mass-specific form; they are related to the heat pump working fluid mass flow. The given equations are forming a solvable equation system with the to-be-optimized degrees of freedom – the pressure levels.

For the sake of compactness, the utilized fluid property functions for temperatures, enthalpies & entropies are only presented once in a general form; dependent on the heat pump working fluid one of the two given options for the fluid property functions is valid for each of the six heat pump state points. With regards to the constraints, the superscripts $\lim^-$ and $\lim^+$ refer to limitations of the variables in direction of high and low values, respectively. For the meaning of the single variables refer to the nomenclature.
References


