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Published in:
Proceedings of ECOS 2022 - The 35th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems 2022

Publication date:
2022

Document Version
Peer reviewed version

Economic potential of lowering supply temperatures when using industrial heat pumps

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Abstract:

Industrial processes are highly dependent on large amounts of energy in the form of heat. This is traditionally delivered by steam or hot water heated in boilers using natural gas, oil, or biomass. High-temperature heat pumps (HTHPs) are a key technology in the transition towards sustainable energy systems. They may provide highly efficient electricity-based process heat contributing to the de-carbonization of the process industry. The implementation of HTHPs in industrial applications is limited by various operational, technological, and thermodynamic constraints. In contrast to boilers, the efficiency of HTHPs greatly increases when lowering the supply temperature. This yields potential economic savings by lowering the utility temperature of industrial processes. This paper aimed at quantifying the decrease in the levelized cost of heat when lowering the supply temperature of HTHPs. The main boiler was replaced by a HTHP, and the steam temperature was lowered. The changes in Coefficient of Performance (COP), Total Capital Investment (TCI) were observed for several HP cascade cycles, with natural working fluids, including water, hydrocarbons, and ammonia. The results showed that the price ratio between natural gas and electricity and the COP of the HTHP were the main factors for determining financial viability when comparing to a natural gas boiler. The operational cost of a HTHP used for hot utility decreases when lowering the utility temperature. Improved performance was observed for all HTHP configurations, however, it was highly sensitive to the COP and electricity price. A cascade HTHP using R-717 and R-718 as refrigerants delivered 160 °C steam at a levelized cost of heat (LCOH) of 46 EUR/MWh with 30 °C as a source temperature. Varying the delivery temperature with 10 °C resulted in a change of 1 EUR/MWh to 4 EUR/MWh in LCOH depending on the technology and reference point. This indicated the significance of correct design temperatures for process heat supply.

Keywords:

Heat pumps, electrification, high temperature, industry, boiler.

1. Introduction

To minimize CO_2 emissions from heat and power productions, the future energy system will to a large extent be based on electrical power from renewable sources such as hydro, wind, and solar power. To have an efficient production of heat based on electrical power there is a need to utilize low-temperature excess heat. Heat pumps (HP) are seen as major contributors to this transformation as they make it possible to produce more heat than the electrical power uptake from the grid. In addition, high temperature heat pumps (HTHP) enable the efficient supply of heat even for industrial processes. The main barriers for more HTHP integration are the installation cost and complexity compared to conventional boilers. Thereby, equipment must compensate for the higher investment cost by higher efficiency, the Coefficient of Performance (COP). Several studies have analysed the potential of utilising HTHPs for the supply of both cooling and heating in industrial processes. This is documented by the work of Annex 35 and Annex 48 (with Annex 58 following) of the IEA Technology Collaboration Programme on Heat Pumping Technologies in [1,2,3]. The projects exhibited the prospect of excellent economic performance and thermodynamic efficiency in factory-scale facilities. The electrification of industrial processes specifically in Denmark is analysed in [4] showing potential for the food, wood, chemical and oil & gas sectors.

Heat pump equipment is readily available and is a proven technology at various scales. However, this is classically for supply temperatures below 100 °C. The current state of the art of HTHPs is reviewed by [5] and concludes that only a few heat pump suppliers have already managed to exceed 120 °C. Utilising existing equipment, [6] experimentally show stable operation of hydrocarbon HPs at 150 °C without altering the compressor except for safety pressure valves.

Various studies like [7,8] concluded that there is a considerable need for process heat at temperatures above 100 °C. While [9] show thermodynamic competitiveness of HTHPs as a boiler substitution for heat production

for temperatures up to 180 °C within reasonable technical constraints utilising R-717 and R-718. The COP of HTHPs is highly dependent on the temperature difference between the source and the sink as well as the level of temperature glides in the source and the sink as seen in [10] determining the operational expenditure (OPEX) of the boiler substitute. Additionally, the total capital investment of HPs is shown by [11,12,13] to be dependent on operating temperature, component selection, configuration and refrigerant. The work of [14] analysed two promising concepts for higher supply temperatures and found technically and economically feasible solutions for process heat supply of up to 280 °C.
This article examines the economic potential of HTHPs as a direct boiler substitution. The HTHP works by exploiting a readily available heat source at 30 °C for generating steam at temperatures between 110 °C and 200 °C. The influence of parameters such as equipment price, fuel cost, operating hours, system efficiency, and supply temperature is studied. The novelty of this work lies in the examination of the economic boundaries for feasibility and the comparison to heat pumps both on a theoretical level and to modelled heat pump cycles to determine their influence on the levelized cost of heat (LCOH) and economic viability.
Thus, a basis for the economic potential of HTHPs as an electrification strategy in the industry requiring steam production can be provided, enhancing the working domain of HPs to high-temperature applications, which are currently not covered by state-of-the-art technologies.

2. Methods

This section presents the method and concepts used for the modelling, analysis and evaluation of the heat pump introduced as a potential boiler substitution. First, the modelling of the heat pump cycles are introduced, followed by the simple, heuristic economic framework used to investigate this electrification strategy.

2.1. Heat pumps

The heat pumps considered in the different scenarios were electrically driven vapour compression heat pumps in a cascade configuration. The model consisted of an evaporator and superheater at the heat source and a desuperheater, condenser and subcooler at the heat sink as visualised in Fig. 1. In between these, there are two closed-loop 1-stage heat pumps sharing a set of heat exchangers (HEX). The first one is ensuring superheat/desuperheat from point 12→6 and 1→2 respectively. The latter HEX exchanges latent heat fully condensing the refrigerant, 2→3, and evaporating the second refrigerant from 11→12.

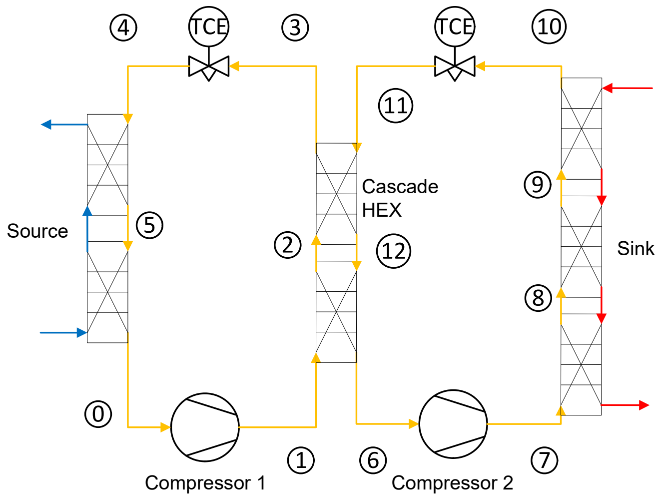


Figure 1: Layout of the cascade configuration used for modelling.

Compressor 1 (0→1) and compressor 2 (6→7) represent a compression process with an isentropic efficiency of 75 % which is common for theoretical calculations [15]. The expansion valves (3→4 and 10→11) are isenthalpic expansion processes leading into the evaporation process of both the top and bottom cycle.
All heat transfer was assumed to be isobaric processes. A minimum temperature difference of 5 K was chosen for all the heat exchangers. The subcooling process in the top cycle (9→10) was maximised so the outlet of the subcooler was set to 5 K above the sink inlet temperature. A minimum of 5 K superheating was considered and increased if required to ensure a dry compression at the outlet and midway through the two compression processes. This was necessary for the configurations utilising hydrocarbons as a refrigerant as the isentropic lines are steeper than the inclination of the saturated vapour line [10]. The intermediate temperature in the cascade HEX is optimised to yield the highest COP. The considered refrigerants were R-600a, R-601a, R-602a, R-717 and R-718 in the configurations presented in Table 1

Table 1: Considered heat pump configurations in for the economic evaluation.

	cascade, R600 + R601a	cascade, R600 + R602a	cascade, R717 + R602a	cascade, R717 + R718
Bottom cycle	R-600	R-600	R-717	R-717
Top cycle	R-601a	R-602a	R-602a	R-718

The heating COP (COP_h) of the heat pump is defined as the ratio between the heat flow to the sink ($\dot{Q}_{sink}$) and the total compressor power $\dot{W}$ as shown in Eq. (1).

$$\mathrm{COP}_h = \frac{\dot{Q}_{\mathrm{sink}}}{\sum \dot{W}} \tag{1}$$

As a broad range of operating temperatures are the focus of this paper, a more appropriate choice is the second law efficiency. In the case of HPs, the Lorenz efficiency η_{Lor} is a suitable measure for comparing the obtained COP to the COP of a Lorenz cycle (COP_{Lor}) presented in Eq. (2).

$$\eta_{\mathrm{Lor}} = \frac{\mathrm{COP}_h}{\mathrm{COP}_{\mathrm{Lor}}} \tag{2}$$

COP_{Lor} was found by dividing the logarithmic mean temperature on the hot side of the heat pump ($T_{\mathrm{sink,av}}$), by the difference between the logarithmic mean temperatures of the hot and the cold side ($T_{\mathrm{sink,av}}$).

$$\mathrm{COP}_{\mathrm{Lor}} = \frac{T_{\mathrm{sink,av}}}{T_{\mathrm{sink,av}} - T_{\mathrm{source,av}}} \tag{3}$$

The inlet and outlet temperatures at the source side were kept constant through all simulations with 30 °C and 20 °C respectively. The sink side is modelled as evaporation of water at temperatures ranging between 110 °C and 200 °C with no temperature glide. Pressure drops and possible required pump or fan power was neglected.

The individual overall heat transfer coefficients, U were assumed constant at varying temperature levels only depending on the fluids on each side of the HEX i.e. steam, water or refrigerant following the recommendations of [12,16]. This corresponds to 50 W/m^2K for the evaporator, 1250 W/m^2K for the cascade HEX. U was40 W/m^2K, 1000 W/m^2K and 750 W/m^2K for the gascooler, condenser and subcooler respectively.
U were used for sizing of the heat transfer equipment by Eq. (4) where A is the total heat transfer area and LMTD is the logarithmic mean temperature difference for a known heat transfer rate. This yields an area used to estimate the equipment cost as described later in Section 2.2..

$$\dot{Q} = U \cdot A \cdot \mathrm{LMTD} \tag{4}$$

The performance of the cycles was examined by numerical modelling based on energy and mass balances in Python [17] utilising CoolProp [18] for their thermophysical properties. The minimize function in [19] was used for the optimisation of the LCOH by changing the evaporation-, and condensing pressure of the bottom cycle and condensing pressure of the top cycle. All data was collected, analysed, and visualised using the Pandas library [20].

2.2. Component cost estimation

The estimation of the bare Purchased equipment costs (PEC) was performed based on cost correlations found in the literature based on supplier data [12]. The cost function correlations used are summarised in Table 2, for which the standard PEC were found with the parameters k and size parameter x, using Eq. (5). For heat transfer equipment x is the total heat transfer area while it is the volume flow rate at the inlet conditions for the compressors.

$$\log(\mathrm{PEC}) = k_1 + k_2 \log(x) + k_3 \cdot (\log(x))^2 \tag{5}$$

For the heat pumps these standard purchased equipment costs were further corrected by a factor of 4.16 as suggested by [11] to determine the total capital investment (TCI) at the start of the project. This factor adjusts for additional costs such as engineering hours, smaller components, and axillary items for the j^{th} component.

$$\mathrm{TCI}_{\mathrm{HP}} = 4.16 \cdot \sum_j \mathrm{PEC}_j \cdot f_j \tag{6}$$

The TCI of all solutions are corrected by the factor f_{CEPCI}, which was based on the Chemical Engineering Plant Cost Index (CEPCI) to compensate for the varying age of the price data corresponding to the guidelines in

[21]. The two HEX types are corrected with the same material factor, f, as they belong in the same process equipment category [22]. All data for the cost functions and further estimates for equipment cost are presented in Table 2 with their respective sources.

Table 2: Parameters for cost functions to be used to evaluate the equipment cost of the heat pump configurations.

Component	TCI [EUR/kW]	k_1	k_2	k_3	Year	f	source
Centrifugal compressor, R-718		2.290	1.360	-0.103	2011	2.8	[23]
Piston compressor, HC		2.245	0.790	0.0	2013	1.0	[16]
Screw compressor, R-717		3.476	0.681	0.0	2004	2.2	[22]
Flat plate HEX, sink		3.132	0.345	0.082	2004	3.3	[22]
Shell & tube HEX, source		3.247	0.226	0.095	2004	3.3	[22]
Electric boiler	200				2016		[24]
Natural gas boiler	100				2016		[24]
Biogas boiler	120				2016		[24]
Technology Catalogue HP estimate 2020	1050				2020		[25]
Technology Catalogue HP estimate 2050	700				2050		[25]

2.3. Economic evaluation

The costs for natural gas and electricity were determined for Denmark for different years based on data from [26]. The price forecasts were coupled with the expected Danish taxes for energy use in industrial processes and for the case of natural gas with CO_2-emission costs as recommended in [21]. Prices were assumed to be applicable to large industrial sites for process heating purposes. A detailed analysis of the prices and price forecasts is not in the scope of this study; however, the influence of different energy prices will be visible from the performed sensitivity analysis. Operation and maintenance costs (OMC) were further included as a continuous payment of ϕ = 1 % of the total capital investment costs each year. The operational cost and maintenance cost were varied by the inflation and energy prices forecasted in [26] while the initial economic investment was assumed to be financed by a loan with an annual interest rate of 4 %.

The annual lifetime cost (C) of implementing the solution were calculated using Eq. (7)

$$\mathrm{C} = \frac{\mathrm{TCI} \cdot \mathrm{CRF}}{\mathrm{OH}} + \mathrm{FC} \cdot \mathrm{OH} + \mathrm{OMC} \tag{7}$$

With TCI as the total capital investment, FC as the fuel cost, OH as the yearly operating hours, and the capital recovery factor (CRF) being defined by the inflation rate (d) and the interest rate (i) as in Eq. (8).

$$\mathrm{CRF} = \frac{i_{\mathrm{eff}} \cdot (1 + i_{eff})^N}{(1 + i_{\mathrm{eff}})^N - 1} \text{ with } i_{\mathrm{eff}} = \frac{1 + d}{1 + i} - 1 \tag{8}$$

A crucial factor is the operational cost (OPEX) of the individual solutions. The OPEX consists of primarily the cost of electricity or gas. As the calculations were all done at a fixed $\dot{Q}_{\mathrm{sink}}$ = 1 MW, the fuel cost was defined from the efficiency of the heat-producing equipment as in Eq. (9). Here P_{ng} is the natural gas price, and $P_{\mathrm{r,j}}$ is the price ratio between the fuel of the heat-producing equipment, j, and the natural gas price per MWh presented later in Fig. 2. η_j is the fuel to heat efficiency of j, i.e., the COP for the heat pump. For the electric boiler, the natural gas boiler, and the biogas boiler the η_j was set to 0.95, 0.9, and 0.9, respectively.

$$\mathrm{FC} = \frac{P_{\mathrm{r,j}}}{P_{\mathrm{ng}}} \cdot \frac{\dot{Q}_{\mathrm{sink}}}{\eta_j} \tag{9}$$

From this, it was possible to calculate the levelized cost of heat (LCOH) which is a measure of the average net present cost of heat generation for a system over its lifetime. It is used for investment planning and to compare different methods of heat generation on a consistent basis. The LCOH was calculated as presented in Eq. (10) using the annual lifetime cost (C), the yearly operating hours (OH) and the delivered heat to the sink ($\dot{Q}_{\mathrm{sink}}$

$$\mathrm{LCOH} = \frac{\mathrm{C}}{\dot{Q}_{\mathrm{sink}} \cdot \mathrm{OH}} \tag{10}$$

A key parameter of this analysis is the current-, future electricity prices, and gas prices. These are presented to the left in Fig. 2 in terms of EUR/MWh. The projection indicates stable biogas and electricity prices while the natural gas price is estimated to linearly increase. The price ratios between the various fuel types are shown to the right in Fig. 2, and will be relevant later for illustrating when a heat pump can be an economically attractive solution.

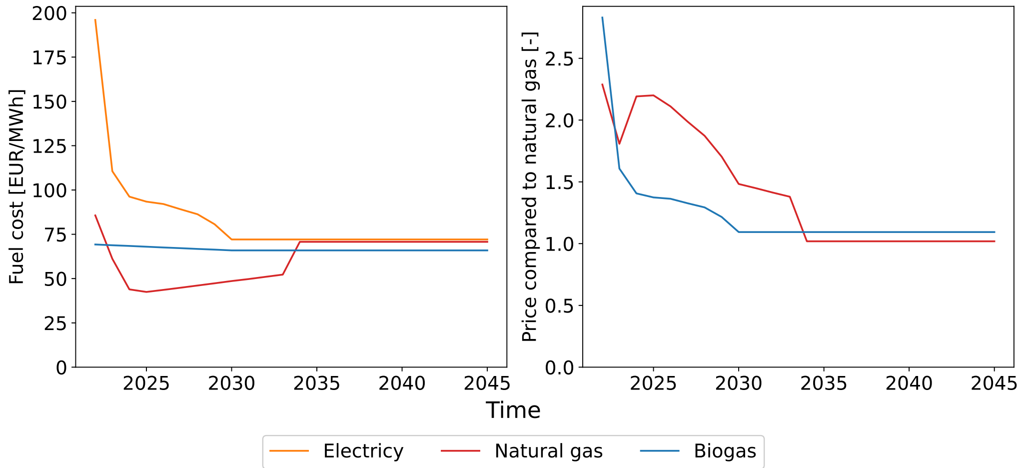


Figure 2: Prices for electricity, biogas and natural gas [EUR/MWh] in the Danish market based on the forcast from the Danish Energy Agency in February 2022 [26].

In all calculations, yearly operating hours of OH = 7000 h/a and an equipment lifetime of N = 20 a was used. The equipment was assumed to be installed in year 2030 correcting TCI by f_{CEPCI}.

3. Results

The estimated specific TCI of the simulated systems are represented in Fig. 3 as a function of the delivery temperature. The boiler and heat pump price estimates from Table 2 are presented as horizontal lines. The resulting prices of the heat pumps vary between 0.7 MEUR/MW and 1.1 MEUR/MW being close to the Technology Catalogue HTHP estimates for a generic vapour compression heat pump. Similar TCIs are presented in [27] for installed, large district heating heat pumps ranging from 0.6 MEUR/MW to 1.1 MEUR/MW depending on the type of heat source and capacity.

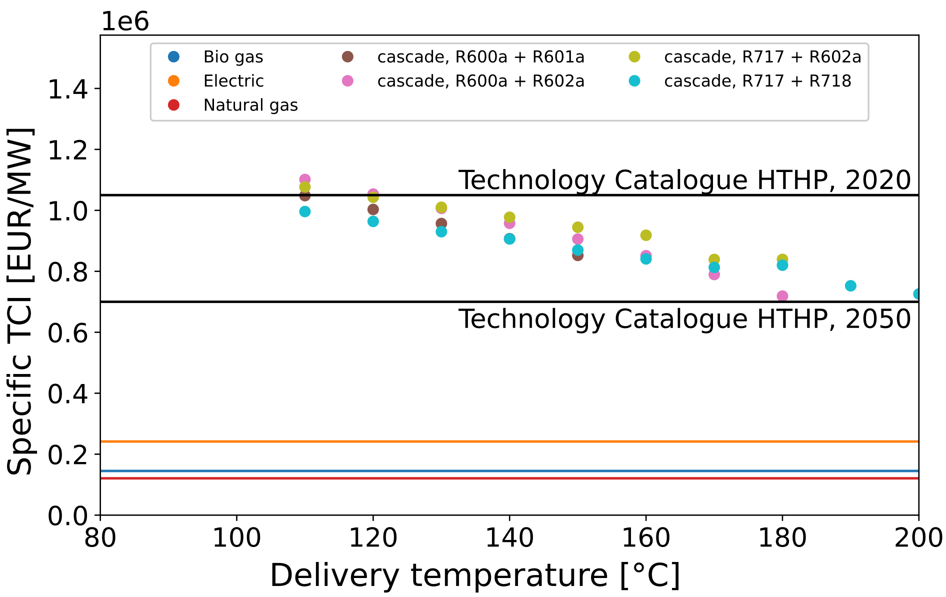


Figure 3: Estimated specific total capital investment for the simulated HTHPs for varying supply temperature. The boiler price (adjusted to 2030 prices) and heat pump price estimates from Table 2 are presented as horizontal lines.

The average price of the simulated heat pumps was 0.92 MEUR/MW with maximum a difference between the configurations of 0.14 MEUR/MW at any given temperature. The price shows a decreasing trend for increasing temperatures. The primary factor for this is a decrease in the volume flow of the top cycle due to higher densities. The HTHP with the lowest TCI is primarily the configuration utilising R-717 and R-718. However, the cheapest heat pump is 3.5 times more expensive in initial investment than the most expensive boiler. Therefore, the heat pump must compensate by a lower operational cost through the equipment lifetime to achieve better overall economic performance.

In Fig. 4 the COPs of the heat pumps are presented as a function of the steam delivery temperature. The coloured dots represent individual model simulations while the black contour line represent the COP of a heat pump with Lorenz efficiencies between 0.4 and 0.6. The coloured horizontal lines show the ratio of electricity, natural gas and bio gas price compared to the natural gas price multiplied by the boiler efficiency.

It is observed that the COP of all HP models including the Lorenz cycle decrease at higher delivery temperatures. This is expected as the temperature lift increase yielding a lower COP_{Lor} from Eq. (3). The HPs using R-717 in the bottom cycle (green and light blue) have the highest COP, while the pure hydrocarbon cycles have a lower COP. Notably, all simulations stay within Lorenz efficiencies of 0.4 and 0.6. The COP of the cascade, R-717 + R-718 is 2.28 at 110 °C and 1.60 at 200 °C corresponding to a η_{Lor} of 49.5 % and 59.2 %, respectively. The horizontal boiler lines represent the constant thermal efficiency of the boilers. As the price of natural gas is the lowest, the corresponding line is at the highest level for the boilers.

The y-axis can be seen as a simplified measure of the operating cost of the individual solutions. Therefore, the crossing points between the heat pumps and the boilers represent at which temperature a heat pump solution for a given efficiency cannot make up for its higher TCI seen in Fig. 3 no matter the lifetime of the equipment. It should be noted that the temperature of this crossing point will be highly dependent on the price ratio of natural gas. Both electric- and bio gas boilers are much lower in terms of fuel to heat price ratio.

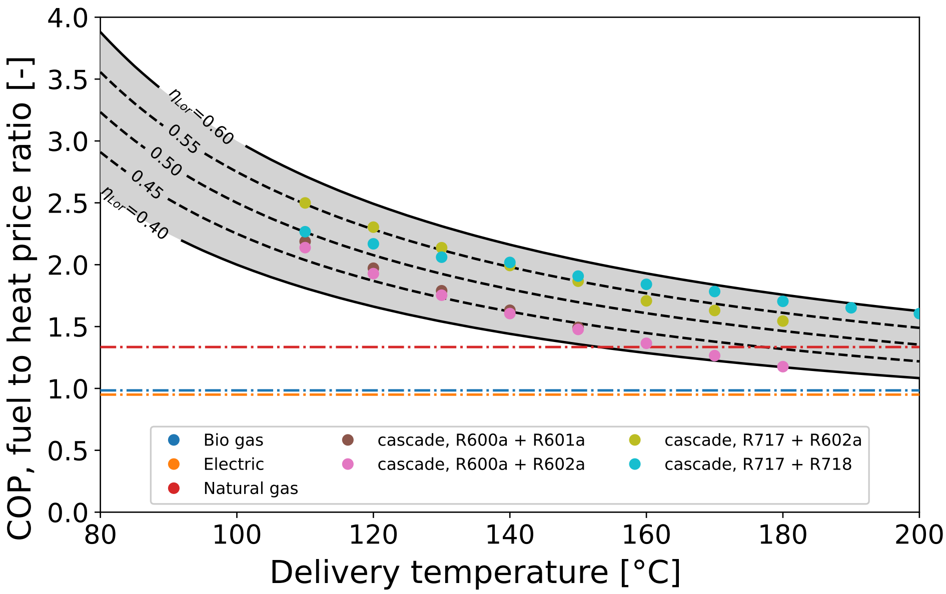


Figure 4: Fuel to heat price ratio of the heat pumps and boiler alternatives.

Combining the information from Figs. 3 and 4 and including the estimates of the maintenance cost, yields the levelised cost of heat for various temperatures as presented in Fig. 5. The boiler equipments show the trend of constant LCOH independent of the delivery temperature. The black contour lines are drawn for a specific TCI of 0.92 [MEUR/MW] again for varying Lorenz efficiencies of the heat pumps. As the COP of the heat pumps decreases with increasing temperature, it is expected that the LCOH increases with the delivery temperature which is confirmed by the results.

The best performing heat pumps deliver heat at 36 EUR/MWh at 110 °C and 48 EUR/MWh at 200 °C. Heat pumps with a Lorenz efficiency even below 40 % have a lower LCOH than all the boiler types below 140 °C, while the best performing heat pump always has a better LCOH than the boilers. Neither the electric- nor bio gas boilers are economically viable compared to the other solutions due to their higher operating cost. This shows the importance of low operational cost for systems with many yearly operating hours. Again, it is remarked that the temperature of the crossing point will be highly dependent on the price ratio electricity compared to natural gas.

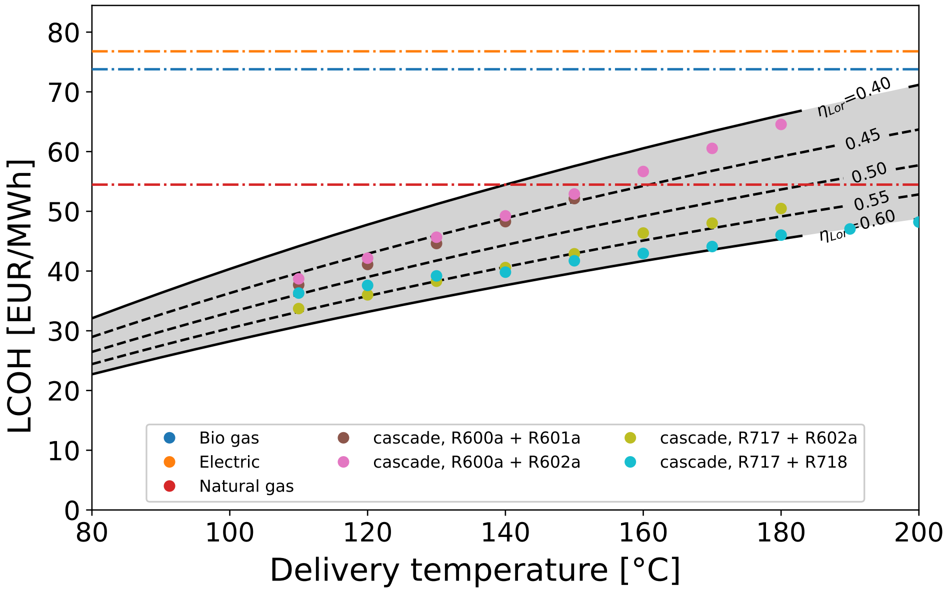


Figure 5: Levelised cost of heat of the three different boilers and the heat pump depending on the delivery temperature.

The contributions to the LCOH of the various factor for the heat pumps and boiler solution are of interest to give insight to the most contributing factors. This distribution is presented for the best heat pump solution identified from the simulations and the natural gas boiler in Fig. 6. The blue is the invested capital distributed over the lifetime of the system, orange is the estimated maintenance cost. Red and green combined equate for the fuel cost where the green is the part contributed by CO_2 quota.

The error bars represent the variance of the respective fuel price in the period from 2025 to 2045. The relative contribution to the LCOH from the equipment decreases with increasing temperatures as the TCI decreased on top of the increasing fuel consumption. It is clear that the main contributor to the LCOH is the fuel cost when looking over the entire equipment lifetime. This is also the case for the heat pumps with the higher CAPEX and maintenance cost. This underlines that an improvement in the performance of the heat pump might be financially beneficial even at high investment costs.

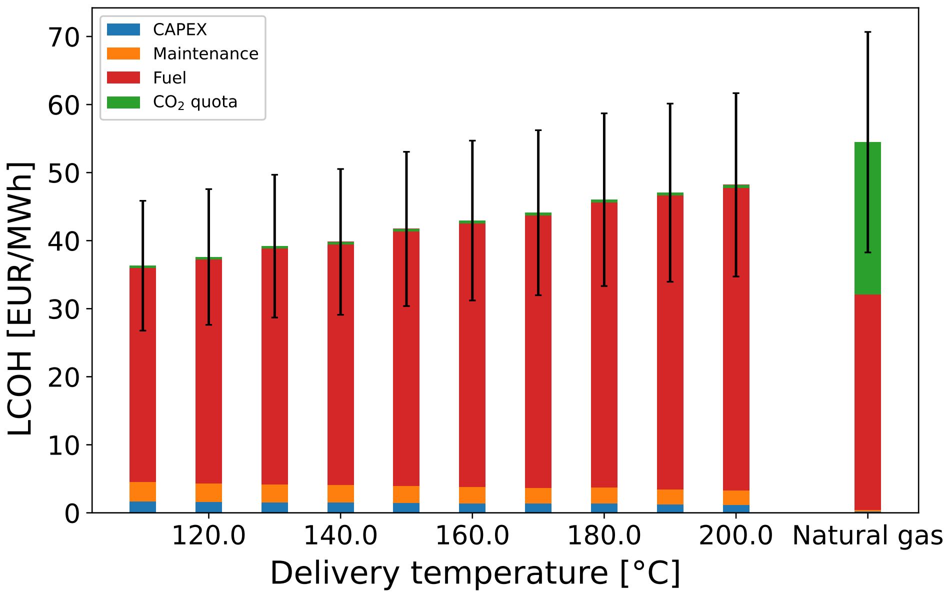
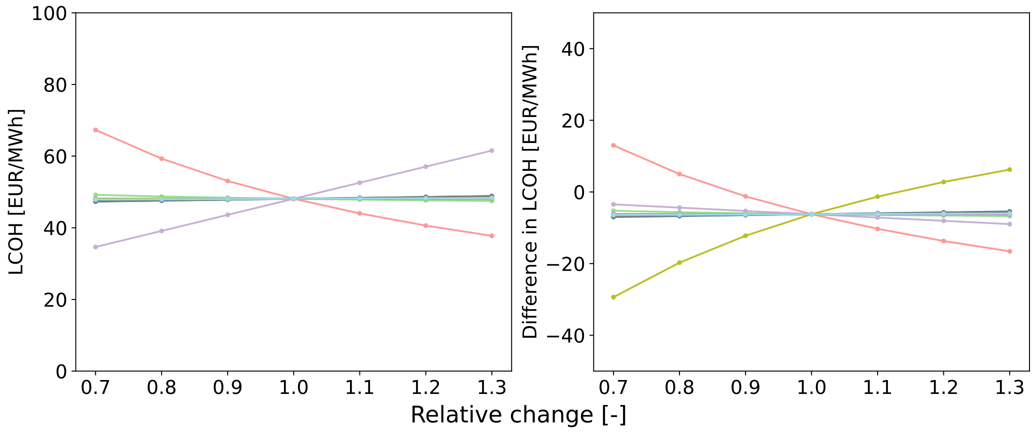


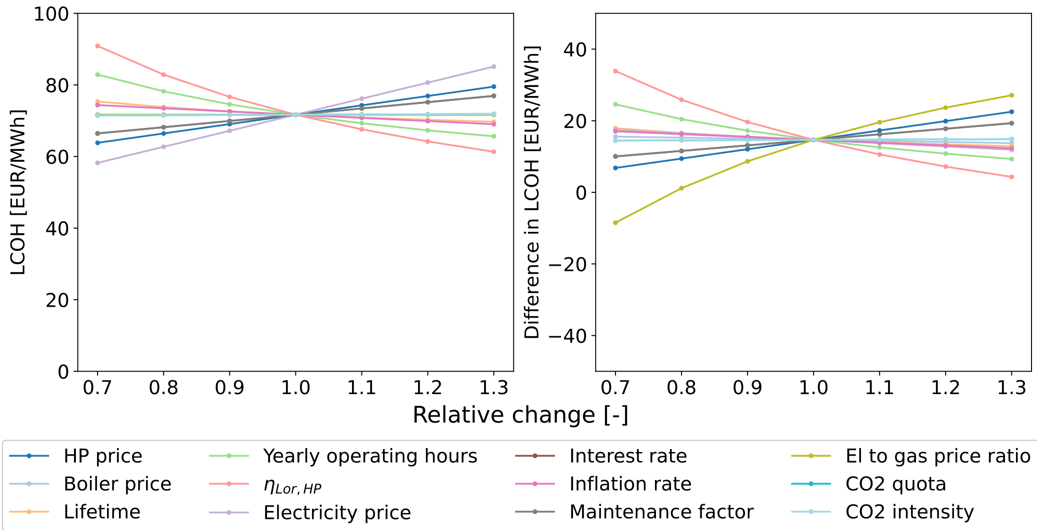
Figure 6: Distribution of the levelised cost of heat for the cascade, R-717 + R-718 solution and the natural gas boiler.

The influence of assumed parameters such as equipment price, fuel cost, operating hours, system efficiency, and supply temperature was studied to determine their influence on the LCOH. This is shown in Fig. 7a for a

heat pump case with η_{Lor} = 50 % and a specific TCI of 0.92 MEUR/MW delivering steam at 160 °C. The left graph shows the influence on the LCOH of the heat pump. The right graph shows the difference in LCOH of the HP compared to a natural gas boiler where a negative difference favours the HTHP. All input parameters were varied with ±30 % for both the heat pump and the natural gas boiler.
There are two parameters having a large effect on the simulated LCOH of the HP namely the electricity price and the efficiency of the HP (η_{Lor}). These directly influence the operational cost of the heat pump which is the main contributing factor as seen from Fig. 6. However, when comparing the HP to the boiler solution the importance of the electricity price is substituted by the electricity to gas price ratio that dictates the difference in operational cost. Varying the the electricity to gas price ratio by 30 % results in a difference of 20 EUR/MWh in LCOH which is 40 % of the base price at 160 °C. It is noted that the effect of the equipment price is negligible as earlier seen from Fig. 6. Also, varying the yearly operating hours and the lifetime of the equipment have little effect when the total operating hours are high.



(a) 7000 yearly operating hours.



(b) 2000 yearly operating hours.

Figure 7: Parameter variation of the assumed parameters to investigate the influence on the LCOH. The delivery temperature is assumed to be 160 °C corresponding 6 bar steam. The case is a heat pump with η_{Lor} = 0.5 % and a specific TCI of 0.92 MEUR/MW. The right figures are comparing to a natural gas boiler.

The same analysis as in Fig. 7a was performed for 2000 yearly operating hours and shown in Fig. 7b. Generally, the LCOH is higher when decreasing thee operating hours, going from 52 EUR/MWh to 64 EUR/MWh. The influence of the other parameters is more pronounced now that the initial investment is a larger part of the LCOH. The influence of the Lorenz efficiency and the gas to electricity price ratio has decreased but are still the biggest contributors.
A tool to quantify the change in LCOH when changing the delivery temperature of the steam is the sensitivity index of the LCOH. Calculating these from Fig. 5 yields the economic penalty of increasing the supply temperature of HTHPS presented in Table 3. Inversely, decreasing the supply temperature of the heat pump will improve

the LCOH by the reported amount. The changes in LCOH pr. 10 °C vary between 1.1 EUR/MWh and 4.1 EUR/MWh but maximally by 0.5 EUR/MWh within a specific heat pump configuration. For the base case in Table 3 lowering the temperature from 6 bar to 4 bar yields an improvement in LCOH of 2.4 EUR/MWh. This equates to 336 000 EUR over the equipment lifetime of 20 years.

Table 3: Comparison of sensitivity index of LCOH for heat supplying equipment above 110 °C.

Technology	Change in LCOH pr. 10 °C EUR/MWh	Relative change pr. 10 °C %
Cascade HP, R-600a + R-601a	3.6 to 4.1	7 % to 9 %
Cascade HP, R-600a + R-602a	3.6 to 4.0	6 % to 9 %
Cascade HP, R-717 + R-602a	2.4 to 2.7	4 % to 7 %
Cascade HP, R-717 + R-718	1.1 to 1.6	2 % to 4 %
Base case, η_{Lor} = 50 %, TCI = 0.92 MEUR/MW	1.1 to 1.8	3 % to 8 %

4. Discussion

This study focused on the evaluation of HTHPS as a boiler substitute on an economic basis. A series of assumptions have been made to be able to evaluate the heat pump models and boilers at a broad range of operating conditions. The assumed values were varied in Fig. 7a to study their individual influence on the levelized cost of heat. Here the cost of the heating equipment and the maintenance cost show a negligible effect on the LCOH due to the high amount of operating hours. An industrial system will typically have a high time of activity to keep an ongoing operation. However, for cases with much lower usage, the influence of operating hours is very different as seen in Fig. 7b. Therefore, a recommendation regarding attractive temperature levels for HTHP are hard to generalize. Nonetheless, for cases with a high number of operating hours, comparing the COP of a heat pump with a Lorenz efficiency of 50 % to the electricity to gas price ratio yields a good initial guess of economic competitiveness.

The examined cascade, R-717 + R-718 HTHP delivered heat at 36 EUR/MWh at 110 °C and 48 EUR/MWh at 200 °C. LCOHs of 29 EUR/MWh to 49 EUR/MWh which presented by [27] for large scale heat pumps delivering district heating. This is in a similar range of LCOH, however, large variations in the initial assumptions of electricity price, yearly operating hours and source temperature make it hard to compare on even ground.

This study purely examined the potential of lowering the delivery temperature of steam systems. It was assumed that the condensate returned in saturated conditions, but this is not always the case. Subcooling of the returning condensate will result in a potentially higher Lorenz efficiency but it will be hard to fulfil. Additionally, the hot utility of industrial processes can also use hot water as a heat carrying medium significantly decreasing overall distribution losses as described by [28]. This yields a linear temperature glide of the sink which might lower the COP of the HP if it does not match the profile of the refrigerant in the condenser by changing refrigerant or HP configuration. This case will require additional work to determine the economic viability. However, as mentioned by [28], hot water systems traditionally operate at lower temperatures than steam systems favouring HTHPs as the provider of hot utility.

4.1. Accessible heat sources of higher quality

The work of this paper analysed heat available at 30 °C as a source. By analysing the equation for the Lorenz COP, Eq. (3), the COP_{Lor} of the HTHP will increase with smaller temperature lifts. This is assuming that the Lorenz efficiency is not lower when looking at alternative temperature conditions. According to [10] it is possible to obtain similar Lorenz efficiencies at a broad range of sink and source temperatures with a concise portfolio of heat pumps. This is supported by [9] presenting Lorenz efficiencies between 45 % and 67 %. Therefore, it is expected that a source available at a higher temperature level will increase the COP of the HP, improving the LCOH of the HP system.

The heat pumps with the lowest LCOH performed better than natural gas boilers in all of the investigated temperature domains. This is agreeing with [9] presenting heat pumps as good solutions for natural gas boiler replacements up to 180 °C with temperature lifts up to 160 K. This is also supported by [29] comparing the HTHPs to boilers on a CO_2 emission basis. However, the manufacturing of the boiler and heat pump equipment is not considered.

4.2. Instability of gas and electricity prices

The price of electricity compared to natural gas was shown to be an important parameter in determining the profitability of installing a HTHP instead of a gas boiler. The price ratio was determining the minimum COP required of the HTHP to reach economic feasibility.

However, highly unstable gas and electricity prices have been reported by [30] in 2021 and continues into

2022. The Danish Energy Agency released a similar report in 2019 regarding the forecast of energy prices in Denmark which are presented in Fig. 8. They show more stable price levels compared to the report from 2022 [26] presented in Fig. 2 for both electricity and natural gas. The electricity price is expected be on a higher level until 2030, where it converges towards 75 EUR (the same as in the old report). The natural gas price also rose in 2021 but is expected to drop below the former expected figures. The numbers in the report are, however, created before the war in Ukraine having gas and oil prices skyrocket [31], which increases the attractiveness of electrifying the industrial processes. All in all the price volatility of gas and electricity create a high degree of uncertainty on the expected LCOH for steam production thus hindering the implementation of HTHPs.

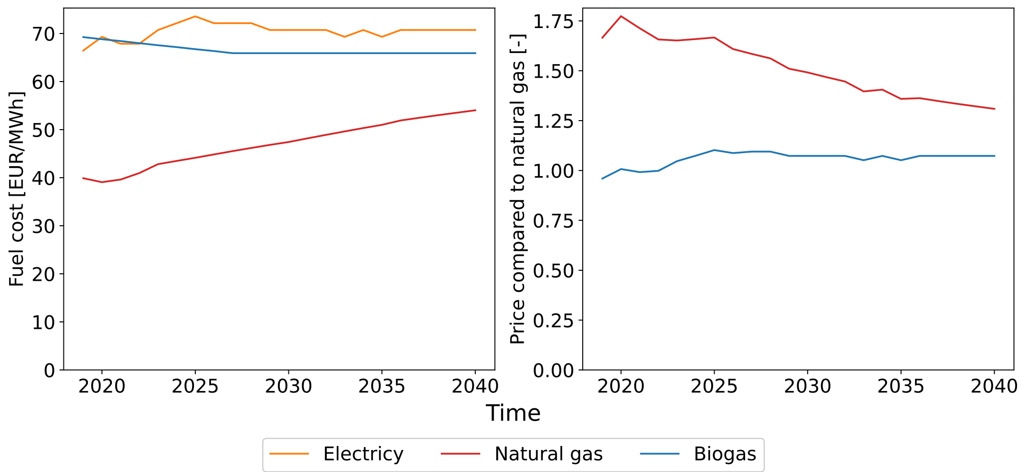


Figure 8: Prices for electricity, biogas and natural gas [EUR/MWh] in the Danish market based on the forecast from the Danish Energy Agency February 2019 [32].

5. Conclusion

An increased use of electricity in the industry will be necessary to reduce its CO2-emissions generated by burning fossil fuels. Besides this reduction, electrification can have other opportunities for industries, such as a reduction in total energy consumption.

The paper focused on evaluating HTHPs for producing steam between 110 °C and 200 °C for process industries using a readily available 30 °C source.

The results showed a decrease in the operational cost of a HTHP when lowering the delivery temperature. This was due to an improved COP which was observed for all HTHP configurations. A cascade HTHP using R-717 and R-718 as refrigerants delivered 160 °C steam at a LCOH of 46 EUR/MWh. Varying the delivery temperature by 10 °C resulted in a change of 1 EUR/MWh to 4 EUR/MWh in LCOH depending on the refrigerant pair and reference point. This indicates the significance of correct design temperatures for process heat supply. However, further work is necessary to determine the influence on the process side and when producing hot water instead of steam. Additionally, the price ratio between natural gas and electricity and the COP of the HTHP are the main factors for determining financial viability when comparing to a natural gas boiler. The LCOH is predominately determined by the COP of the HP and the electricity price at a high number of operational hours and equipment lifetime. The initial investment is uncertain but was shown to have a negligible influence.

Acknowledgments

This research project was funded by The Energy Technology Development and Demonstration Programme (EUDP), under the project title: “SuPrHeat - Sustainable process heating with high-temperature heat pumps using NatRefs” number 64020-1074. The authors are grateful of the financial support from the EUDP and the collaboration with the project partners of SuPrHeat.

Nomenclature

A Overall heat transfer area, m^2

C Annual lifetime cost, EUR/a

CRF Capital recovery factor, EUR/a

COP Coefficient of performance, –

d Inflation rate, –

f Material factor, –

f_{CEPCI} Plant cost index factor, –

FC Fuel cost, EUR/h

i Interest rate, –

k Cost function parameter, EUR, EUR/x or EUR/x^2

LCOH Levelized cost of heat, EUR/MWh

LMTD Log mean temperature difference, W/m^2K

N Equipment lifetime, a

OH Yearly operating hours, h/a

OMC Operational maintenance cost, EUR/a

P Price, EUR/MWh

P_r Price ratio, –

PEC Purchased equipment cost, EUR

$\dot{Q}$ Heat flow rate, W

T Temperature, K

TCI Total capital investment, EUR

U Overall heat transfer coefficient, W/m^2K

$\dot{W}$ Power, W

x Size parameter, A or m^3/s

Greek symbols

η Efficiency

ϕ Maintenance factor

Subscripts and superscripts

a Air

av Average

eff Effective

h Heat

HP Heat pump

Lor Lorenz

sink Sink side of heat pump

source Source side of heat pump

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