TECHNO-ECONOMIC ANALYSIS OF HIGH TEMPERATURE HEAT PUMPS: A CASE STUDY

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NOMENCLATURE

COP	[-]	Coefficient of performance
c_p	[J/kg·K]	Specific heat capacity
h	[J/kg]	Specific enthalphy
$\dot{Q}$	[W]	Heat transfer rate
T	[K]	Temperature
$\dot{W}$	[W]	Power

Special characters

Δ	[-]	Difference

Subscripts

$Comp$	Compressor
$Cond$	Condenser
El	Electric
$Evap$	Evaporator
Th	Thermal

ABSTRACT

Adoption of industrial high temperature heat pumps is often hold back because of the low attainable heat sink temperature and temperature lifts. In order to increase its application potential, operation at higher temperatures is targeted. Whereas the application potential is great, high temperature heat pumps typically have high investment costs and operate at low COPs. Hence, its economic feasibility in terms of Net Present Value (NPV) and Discounted Pay-back Periods (DPP) is evaluated based on a case study. The cooling loop of a walking beam furnace acts as heat source, this heat is transferred and upgraded by a high temperature heat pump to a pressurized water based heating system. Optimization and simulation of the heat network design indicates a NPV of 730 076 € or a DPP of 13.95 years. Furthermore a comprehensive sensitivity analysis is performed, exposing the future potential of high temperature industrial heat pumps. In this analysis the sensitivity of the heat pump investment cost, distance between heat source and sink, electricity cost to fuel price ratio and the used discount rate on the NPV and DPP is assessed.

INTRODUCTION

Large amounts of industrial waste heat are released unexploited to the environment [1]. This gives the opportunity to industrial heat pumps to provide electrification of heating in various industrial branches, by converting industrial waste heat to usable process heat. The global results are enhanced energy efficiency and a reduction in CO_2 and harmful emissions. The current commercially available industrial heat pumps are constricted in terms of operating temperatures ($<$ 130 °C) and temperature lifts (50 °C) [2]. In order to increase the heat recovery and application potentials of the heat pump technology, operation at temperatures of 200 °C and temperature lifts of 100 °C are targeted [2; 3]

In this study the economic feasibility of a high operation temperature and high temperature lift heat pump will be assessed. Whereas the application potential is great, high pay-back periods (PBPs) are expected as high temperature heat pumps typically have high investment cost, while operating at low COPs. The application of interest (i.e. heat sink) is a pressurized water based heating system of a storage hall. As the heating system is mainly based on thermal radiation, high operation temperatures are desired. The supply temperature of this heating system varies between 60-170 °C, while the return temperature varies between 40-140 °C. This system operates with a quasi-constant mass flow rate of 75 kg/s. The hall heating network is currently heated by use of an internal steam network. However, the residual heat available from the cooling loop of a walking beam furnace (WBF) acts as a promising substitute heat source. This cooling loop operates at 80±12 °C with a quasi-constant mass flow rate of 390 kg/s. After heat transfer the temperature of the outlet water fluctuates between 87±12 °C, consequently the average heat transfer rate is about 11.5 MW_{th}. The heat source and sink are located at a distance of 525 m from each other. For both the heat source and heat sink, the mass flow rate and supply and return temperature are monitored with a sample interval of 1h. The yearly temperature distribution of the return temperature of the WBF cooling loop and the supply temperature of the hall heating are plotted in Fig. 1. These temperatures will be further referred to as the "hot" temperatures of respectively the heat source and the heat sink. The supply temperature of the WBF cooling loop and the return temperature of the hall heating shows a similar trend. Likewise, these temperatures will be further referred to as ”cold” temperatures of the heat source and sink.

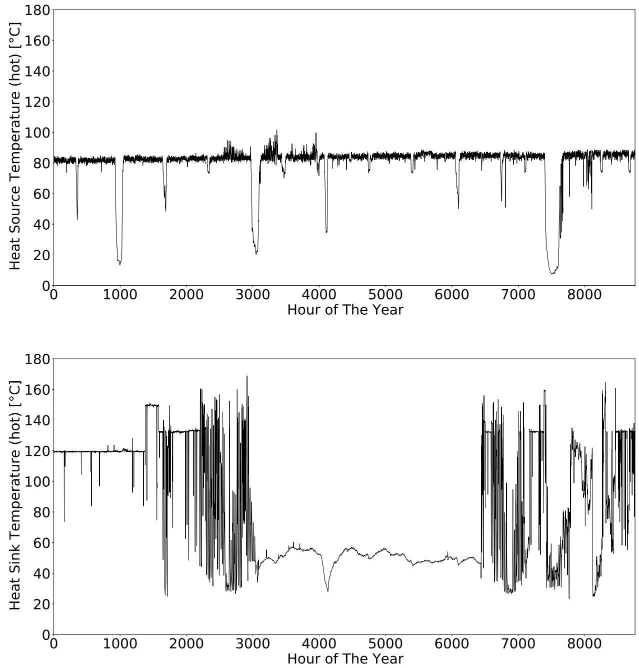

Figure 1. Temperature profiles of the cooling loop return temperature and hall heating supply temperature.

The yearly distribution of the delivered heat by the cooling loop and the requested heat of the hall heating are represented in Fig. 2.

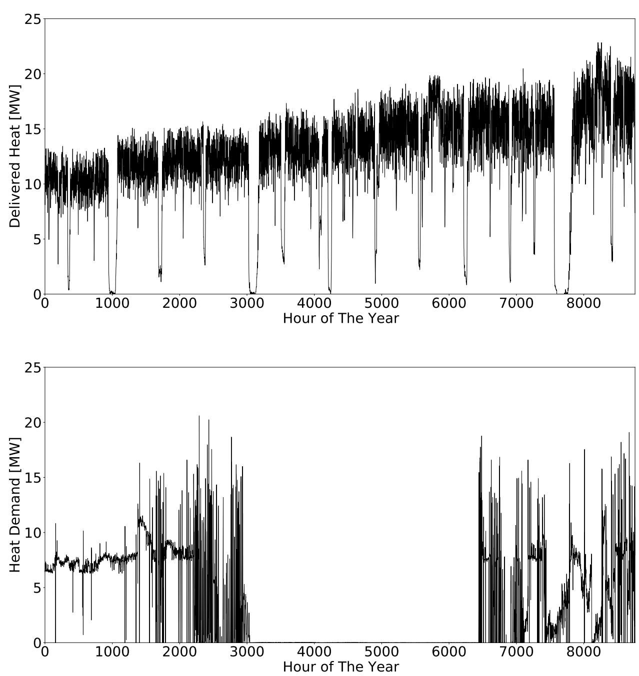

Figure 2. Heat load profiles of the cooling loop and the hall heating.

As observed in Fig. 2 the hall heating is turned off during summer, and shows strong fluctuations in heat demand. Whereas the cooling loop shows several moments with a low amount or zero value of extracted heat, on account of maintenance or shutdown of the WBF.

DESIGN AND MODELLING

Design of The Heat Network

The heat sink and heat source will be connected by use of racked piping in the open air, yet to be built. A separate heat loop from the heat source to the evaporator of the heat pump will be made. This allows some degree of freedom in the mass flow rate sent from heat source to heat sink. A high mass flow rate will increase the piping investment cost and also the operational and investment cost of the centrifugal pump. However, if the mass flow rate is high, the fluid stream will experience a low temperature glide, allowing for higher evaporation temperatures of the refrigerant and therefore higher COPs. Hence, different temperature glides of the heat source are examined by varying the mass flow rate send from heat source to heat sink, respecting that the mass flow rate cannot exceed 390 kg/s. After the loops come back together additional heat may be extracted by the conventional cooling system, when needed.

The pipe diameter is sized according to a typical maximum fluid velocity of 2 m/s as often applied in refrigeration units [4]. The heat pump will be placed at the location of the heat sink, which is beneficial compared to a heat pump at the location of the heat source in terms of ambient heat losses. In this particular case study residual heat, which has no associated fuel cost and is generally abundantly available, is used. Hence from a purely financial viewpoint insulation of the waste heat intermediate thermal loop could be omitted. However, in order to avoid scalding and limit the temperature drops, proper insulation is necessary. Therefore the insulation is designed according to the recommended minimum thickness of the manufacturer.

The hall heating induces a strongly fluctuating heat demand profile, with peaks in heat demand up to 20 MW_{th}. Sizing the heat pump based on these short-term peaks would cause a strong increase in heat pump, centrifugal pump and piping investment cost and in operational cost of both the heat pump and centrifugal pump. In addition these short term-peaks only contributes to small amounts of heat compared to the annual heat demand. Therefore, it is often more interesting to use the heat pump as base load and to handle the peaks with the conventional heating system. However, if the size of the heat pump would be reduced, the associated fuel costs of the conventional heating system could become dominant. The conventional heating system is also used as back-up during moments of heat shortages, introduced by the peaks in heat demands or maintenance of the WBF. As there is generally an oversupply of heat, the introduction of a thermal energy storage (TES) could be another option to further increase energy efficiency. However, due the lack of distributed heat shortages and the high investment cost of the TES, high or undefined PBPs are to be expected. Hence, its implementation is not assessed.

Thermal Modelling of The Heat Network

The heat network is modelled by a quasi-steady-state simulation. In this method steady-state operation is assumed during each individual time step. A period of 1 hour is chosen as time step, as this time interval corresponds with the measurement interval of both heat source and sink. Hence, 8760 time steps are calculated for each set of input parameters, allowing for the calculation of the operational cost on an annual basis. These input parameters differs in terms of temperature glide (i.e. mass flow rate) of the heat source and size of the heat pump.

During each time step the COP is calculated by simulating a sub-critical heat pump cycle. According to Frate et al. [5] benzene is the most suited working fluid for the case of high source (> 65 °C) and high sink temperatures (> 130 °C). The critical temperature of benzene is 288.87 °C [5], assuring sub-critical operation. Other interesting fluids allowing for sub-critical operation are 1336mzz(Z), dichloroethane and pentane [5; 6]. The heat pump cycle is modeled with use of the CoolProp library [7], assuming isobaric condensation, isenthalpic expansion, isobaric evaporation and compression with a fixed isentropic efficiency of 0.7 [8]. The subcooling and the superheat temperature, are both set to 5 K, as done in the work of Kosmadakis et al. [4]. Within all heat exchangers a temperature difference of 5 K at the pinch point is imposed, being a typical value for large-scale heat pumps [4]. After modelling the heat pump cycle, the COP is calculated by dividing the enthalpy change over the condenser (Δh_{Cond}) by the enthalpy increase over the compressor (Δh_{Comp}):

$$COP = \frac{h_{Cond,in} - h_{Cond,out}}{h_{Comp,out} - h_{Comp,in}} = \frac{\Delta h_{Cond}}{\Delta h_{Comp}} \tag{1}$$

During some periods the "hot" temperature of the heat source may exceed the "hot" temperature of the heat sink, eliminating the need of a heat pump. Instead a heat exchanger is sufficient. In this particular case the heat pump is converted to a heat exchanger by virtually implementing an infinitely large COP.

As the COP and the condenser power ($\dot{Q}_{Cond}$) are known, the compressor power ($\dot{W}_{Comp}$) and the evaporator power ($\dot{Q}_{Evap}$) can be calculated according to:

$$\dot{W}_{Comp} = \frac{\dot{Q}_{Cond}}{COP} \tag{2}$$

$$\dot{Q}_{Evap} = \dot{Q}_{Cond} \cdot \left(1 - \frac{1}{COP}\right) \tag{3}$$

Based on the evaporator power and the predefined temperature glide (ΔT_{Glide}), the mass flow rate of the heat distributing network ($\dot{m}$) can be calculated as:

$$\dot{m} = \frac{\dot{Q}_{Evap}}{c_p \cdot \Delta T_{Glide}} \tag{4}$$

The mass flow rate allows for pressure drop calculations. The pressure drop over the pipes are calculated by the Darcy-Weisbach equation, the pressure drop over the bends are calculated by use of a total loss coefficient represented by Coker [9]. On recommendation of the industrial plant an absolute pipe roughness of 0.1 mm is used. Based on the relative roughness the fanning friction factor for flows with Re $\geq$ 2300 is calculated by correlations of Fang et al. [10], implemented within a Python library. Ultimately, generalized pressure drop correlations of the evaporator are retrieved based on the toolbox SSP G8 [16] of manufacturer SWEP [11]. The pressure drop over the condenser is considered equal to the pressure drop over the heat exchanger connecting the internal steam network and the hall heating network. Therefore the pressure drop over the condenser is not considered, since an equal cost is associated compared to the conventional heating system.

Cost Correlations and Optimization Strategy

Different scenarios are compared in terms of Net Present Value (NPV) and Discounted Payback Period (DPP). For both capital budgeting methods the Weighted Average Cost of Capital (WACC) of the industrial plant (8.4%) is used as discount rate. Furthermore, all future costs are adapted according to an inflation rate of 1.9%, which is the 25 year average (1995-2020) in Belgium according to recent trading economics [12].

For the racked piping and insulation investments costs, internal costs functions of the industrial plant are used. The centrifugal pump investment cost, including the electric motor, is calculated with use of a power law described by Smith [13]. The heat pump investment cost is determined by a cost function of Hervas et al. [14], based on the condenser power. In practise however, the temperature glide of the heat source will determine the evaporator size, influencing the heat pump investment cost. This dependency is disregarded in this analysis. For the specific costs of electricity and steam, internal cost functions are used. As few data is available in literature on O&M costs, valves, control systems, etc., these are indirectly considered by implementing equal maintenance and replacement cost for the conventional heating system.

The NPV is calculated over 45 years, being a typical lifetime for a heat network [15; 16]. Other components typically have shorter lifetimes, for both the centrifugal pump [17] and the heat pump [18] a lifetime of 15 years is used.

TRENDS AND RESULTS

The NPV in function of the heat source temperature glide and the heat pump size is illustrated in Fig. 3. A maximum NPV of 730 076 € with a corresponding DPP of 14.24 year is found for a heat pump size of 7 MW_{Th} and a temperature glide of 6 K. This plot exposes the clear trade-off in heat pump size and temperature glide of the heat source. Furthermore the plot exposes that the optimum heat pump size is dependant on the temperature glide of the heat source and vice versa. For increased heat source temperature glides smaller heat pumps are favoured, as the COP decreases, decreasing the profitability of the heat pump.

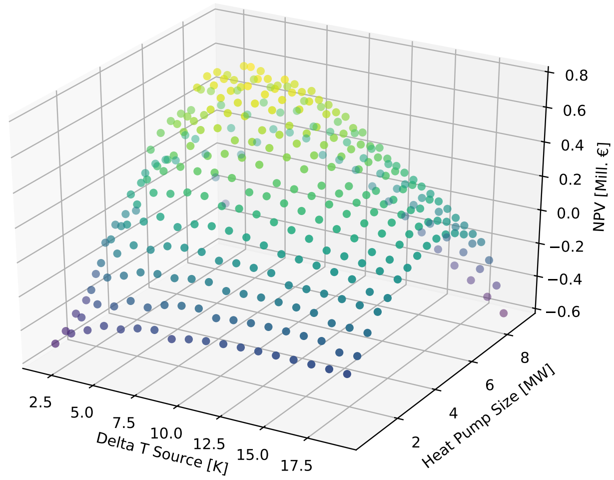

Figure 3. NPV in function of heat source temperature glide and heat pump size.

The DPP in function of the heat source temperature glide and the heat pump size is illustrated in Fig. 4. In this graph a DPP of 45 years corresponds to a NPV of zero. In some instances the operational cost is higher than the gains, resulting in a DPP which is undefined. A minimum DPP of 13.95 year with a corresponding NPV of 678 423 € is found for a heat pump size of 6 MW_{Th} and a temperature glide of 8 K. Compared to the case of maximum NPV, a smaller heat pump size and a larger temperature glide is observed. This is explained by the characteristic that the DPP optimization stronger penalizes configurations with higher investment costs. Noteworthy is the formation of a low-lying plateau at the centre (blue), just below a DPP of 15 years. A life time of 15 year is implemented for the heat pump and centrifugal pump, consequently their investment costs are reoccurring at that specific year, as a result a sudden increase in DPP is observed.

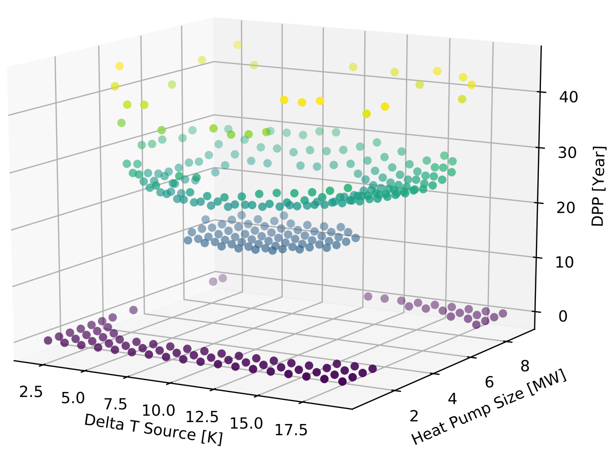

Figure 4. DPP in function of heat source temperature glide and heat pump size.

SENSITIVITY ANALYSIS

Several boundary conditions are uncertain or could vary on a project specific, regional or even temporal basis. Hence the influences of these boundary conditions on the economic feasibility and design of the heating network are considered. In this consideration some parameters might drastically decrease or eliminate the application potential of the heat pump. To counteract elimination of the heat pump by the optimization algorithm, a minimum heat pump size of 0.5 MW_{Th} is set as constraint. In case that the investment will never pay back, the DPP will not be plotted. Moreover, the base scenario is indicated by a dotted vertical axis

Sensitivity of The Heat Pump Investment Cost

Within literature, investment costs for high temperature heat pumps in industrial processes vary between 250 to 900 €/kW [6; 19]. Due to the differences in cycle configuration, compressor type, unit size, refrigerant and temperature levels of both heat sink and source strongly different specific costs are observed [4]. In this case, a large-scale single stage heat pump is used, hence the base scenario of 312.7 €/kW is acceptable, although it is a quite optimistic scenario. The NPV in function of the investment cost range is illustrated in Fig. 5.

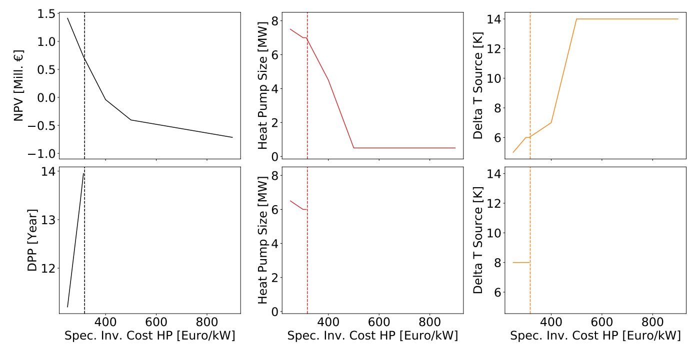

Figure 5. Sensitivity of the heat pump investment cost on the optimum parameters.

In Fig. 5 a strong hyperbolic dependency on the heat pump investment costs is observed. At the lower boundary, a NPV and a DPP of respectively 1 413 228 € and 11.2 years are found. At the higher boundary however the NPV drops to -713 913 €, causing for the investment to be never paid back. As expected the optimum heat pump sizes increases with decreasing heat pump investment costs and vice versa. The optimum heat source temperature glide on the other hand shows an opposing trend. Because, for smaller heat pump sizes the profit potential by enhanced efficiencies is minor.

Sensitivity of The Distance Between Heat Source and Sink

The distance between heat source and sink should linearly increase the investment cost of the racked piping and its insulation. Furthermore it increases the operational and investment costs of the centrifugal pump. Hence, several distances between heat source and sink are evaluated. The NPV in function of the shortest distance between heat source and sink is illustrated in

Fig. 6.

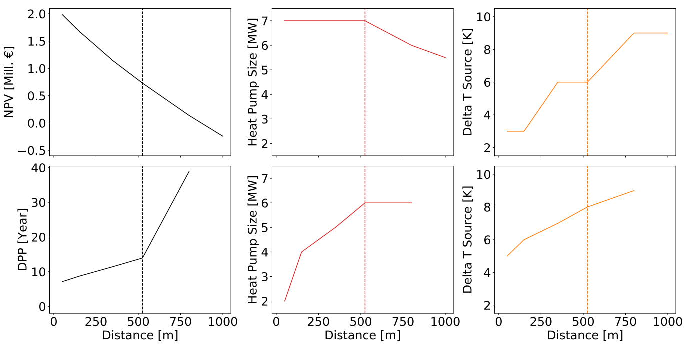

Figure 6. Sensitivity on distance between heat source and sink.

In Fig. 6 the importance of the distance between heat sink and source is clarified. If the heat source and sink would be at a distance of 50 m from each other, the NPV will rise to 1 985 243 €, while the DPP decreases to 7.12 years. If the distance would increase to 1000 m however, a negative NPV of -242 029 € would be observed, introducing undefined DPPs. With respect to the NPV the optimum heat pump size decreases with the distance and vice versa. Larger heat pumps requires bigger piping and therefore insulation in order to respect the velocity limitations. Hence its implementation is more penalized for large distances, causing the characteristic to be non-linear, as one might have expected. Contradictory for the DPP an opposite trend is observed. For small distances the total investment cost is low, hence configurations with small heat pumps can be paid back faster, even thought the NPV strongly reduces. For increasing distances the operational and investment costs of the centrifugal pump becomes more critical, therefore larger temperatures glides are observed for both capital budgeting methods.

Sensitivity of The Electricity Cost to Fuel Price Ratio

The variation in electricity cost to fuel price ratio can be a significant barrier for the commissioning of high temperature heat pumps in some countries. In this analysis a electricity to fuel price ratio of 2.2 is used. According to de Boer et al. [20] electricity to fuel price ratios ranging between 1 to 5 are reported. The oil and gas industry have shown considerable volatility in recent years. Hence there is opted to leave the electricity cost fixed and vary the fuel cost. The NPV in function of the electricity cost to fuel price ratio can be found in Fig. 7.

Fig. 7 exposes the exceptionally strong dependency of the economic profitability on the electricity cost to fuel price ratio. If the fuel cost would become identical to the electricity cost a NPV of 13 749 775 €, being almost the twenty-fold of the base case, and DPP of 2.94 years is found. For high ratios however, negative NPVs and undefined DPPs are observed. Although the decrease in NPV is limited as the optimal heat pump size decreases with increasing price ratios. In respect to the optimal temperature glide of the heat source opposing trends are observed, due to the differences in objectives and properties of the capital budgeting methods as indicated earlier on.

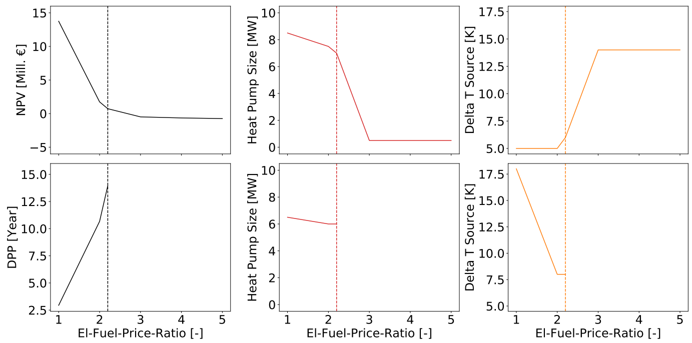

Figure 7. Sensitivity on electricity cost to fuel price ratio.

With the current EU climate targets it is expected that fossil fuels will be subjected to increasing costs as a direct result of the associated carbon taxes. Hence, low electricity cost to fuel price ratios are expected, pointing out that industrial heat pumps can become the preferred heat supply method for processes with supply temperatures between 100 °C and 200 °C.

Sensitivity of The Discount Rate

A high WACC of 8,4 % is used as discount rate during NPV and DPP calculations, entailing strong discounting of future savings. As a result long term decisions are strongly penalized while the NPV strongly decreases and the DPP strongly increases. The NPV and DPP in function of the discount rate can be found in Fig. 8.

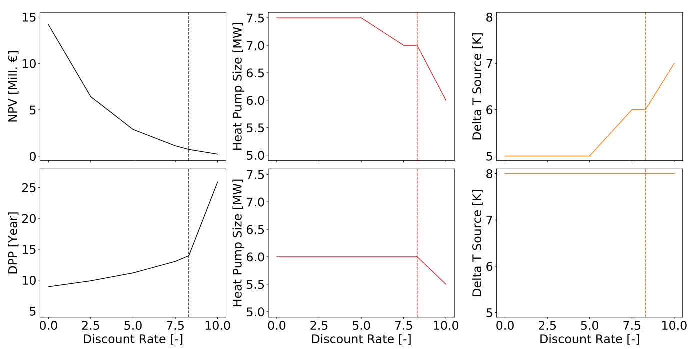

Figure 8. Sensitivity on the discount rate.

In case that no cost discounting is implemented (i.e. discount rate of 0 %) the DPP corresponds with the conventional PBP. Hence the actual PBP of the project is 8.93 years, as the optimum design parameters are identical. In the same case an optimal NPV of 14 175 442 € is achieved. For higher discount rates the NPV strongly decreases while the DPP strongly increases. Although the NPV stays positive in the considered interval. For increasing discount rates the future savings are more discounted, consequently smaller heat pumps are preferred. As for high discount rates investment costs plays a more significant role, the temperature glide at the heat source increases along with the discount rate in case of the NPV. With respect to the DPP however, the temperature glide at the heat source stays fixed.

Several authors tends to use unrealistically low discount rates. Discount rates as low as 2 % have been reported for industrial heat pump applications [21]. Compared to actual industrial business cases this could lead however to an overestimation of the NPV by several times.

CONCLUSION

A techno-economic simulation model is made, able to calculate the economic performance of high temperature heat pump projects. If applied to a case study, a positive NPV of 730 076 € is found, indicating a business case. The optimal DPP and the corresponding non discounted pay-back time of respectively 13.95 and 8.93 years however is high, amongst other due to the large distance between heat sink and source. Furthermore, depending on the capital budgeting method, different optimal design parameters are observed. Hence it is important to clearly identify if a maximum NPV or a minimum DPP is desired. In general, high temperature heat pumps will be an attractive technology in scenarios where the heat pump investment cost is low, the electricity price is low relative to the cost of the alternative energy source and the distance between heat source and sink is acceptable. In these cases industrial heat pumps can become substitutes for gas boilers, producing heat up to 200 °C.

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