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Identifying techno-economic improvements for a steam generating heat pump with exergy-based costs minimization

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Abstract:

Steam generating heat pumps show great potential for reducing carbon emissions in the industrial sector. However, predicting their performance is challenging as the exergy destruction of e.g., compressors and expansion valves increases with the temperature lift and condenser temperature. With over seventy design improvements mentioned in the literature, selecting the most effective design improvements is crucial. In this study, energy and exergy-based methods were compared in their ability to identify design improvements for a single stage subcritical heat pump to produce steam from hot condensate. The energy-based method suggested the addition of a sequential compressor with an intermediate cooler; however, this design did not improve the heat pump's techno-economic performance. The suggestion of adding either an internal heat exchanger or a flash vessel by exergy-based methods did lead in both cases to improved techno-economic performance. The internal heat exchanger performed best and increased the coefficient of performance from 2.3 to 2.8 and reduced operational costs by 0.8 M€ after 5 years of operation. Additionally, the initial investment decreased by 135 k€, and the total costs of operation decreased from 10.3 M€ to 8.7 M€. These findings show that exergy-based methods are the way forward in identifying effective design improvements for steam generating heat pumps.

Keywords:

High-temperature heat pump, Steam generation, Exergy-based costs minimization, Techno-economic analysis, advanced heat pump configuration

1. Introduction

1.1. Steam generating heat pumps

Heat pumps are being widely considered as a crucial technology for reducing carbon emissions in the industrial sector by 2030, and steam generation heat pumps (SGHPs) are a notable example. By converting renewable electricity into heat, SGHPs can effectively contribute to the sector's CO_2-reduction goals. These heat pumps have the potential to decrease energy consumption in the European industrial sector by almost 30%, as they can supply heat at a temperature of up to 300 °C [1]. SGHPs have three major advantages over other high temperature heat pumps: 1. they have a low installation costs factor because they can be connected to existing steam networks, 2. they have a good heat transfer coefficient compared to other working media (organic vapours), and 3. they are easier to control compared to hot water systems [2].

A large temperature lift is required to produce steam from waste heat. The typically used subcritical single stage heat pump, also known as simple cycle heat pump, becomes uneconomical at temperature lifts exceeding 50°C [3] due to increased inefficiencies in the compressor and expansion valve [4]. Advanced heat pump configurations, on the other hand, are an economically viable alternative at these large temperature lifts [3]. The coefficient of performance (COP), which is the key technical performance indicator of heat pumps, reflects this increase in techno-economic viability. By advancing the simple cycle through intermediate cooling in the compressor, for example, the heat pump's COP can be increased. Advanced configurations can have a COP up to three times that of the simple cycle at the same temperature difference [5]. Many researchers have explored the performance and techno-economic potential of advanced heat pump configurations, including Arpagaus [6, 7], Schlosser [5], and Adamson [8]. These studies have identified over seventy different configurations that can help improve upon the simple cycle's performance.

However, it is important to note that selecting an advanced heat pump configuration based solely on its listed performance in a different technical context can be ill-advised, as a heat pump's performance is highly sensitive to its technical surroundings. An example of the sensitivity of a heat pump's performance to its technical surroundings is demonstrated by Bless et al. [2], who found that the COP of high-temperature heat pump configurations for steam generation appeared comparable at first, but differed by almost two times when applied to the same case. This highlights the need for caution when selecting advanced heat pump configurations based solely on listed performance in a different technical context.

While there are over seventy possibilities for optimizing the simple heat pump configuration, currently, there is no systematic method for determining the optimal improvements. In other words, there is no established approach for selecting the most effective configuration for a specific application. This presents a challenge for researchers and engineers who seek to develop more efficient heat pump systems, as trial-and-error is currently the only means of determining which advanced heat pump configurations will perform optimally in a given context. It is, therefore, needed to develop a systematic method for selecting the most efficient and effective heat pump configurations for specific applications.

1.2. Economics and exergy analysis of high temperature heat pumps

Arpagaus [6, 7], Schlosser [5], and Adamson [8] have identified various advanced heat pump configurations that aim to minimize irreversibilities resulting from non-ideal pressure changes and heat exchange. These modifications involve adding expanders, compressors, flash vessels, open connections, heat exchangers, or switching the working fluid. The reduction of irreversibilities results in less work being required by the compressor and therefore a higher COP. Simple cycle heat pumps typically operate at about 50% of their ideal COP, due to these irreversibilities. Hence, they require twice the amount of shaft work to drive the compressor compared to the ideal operation [9]. Exergy analysis is a technique that helps to identify where irreversibilities occur. Opposite to energy, exergy is not a conserved measure. Hence, an exergy balance needs to account for exergy loss to the environment or loss to irreversibilities. The latter is known as exergy destruction.

Bergamini et al. [10] used exergy analysis to study the exergy destruction in a high temperature single-stage ammonia heat pump that produced heat at 140°C from an isothermal 30°C source. They found that the exergy destruction of the heat pump's components increased at different rates and that exergy destruction in the expansion valves was the most sensitive to the temperature lift, whereas its initial contribution to the total exergy destruction was limited. Hence, the dominant source of exergy destruction in the heat pump varied among components as the temperature lift was increased. Hu et al. [11] found similar results when assessing a heat pump that produced pressurized water at 120 °C from a heat source ranging from 50 to 90 °C. They concluded that there is a strong correlation between exergy loss in the heat pump and the temperature lift, while exergy destruction in the evaporator and the condenser remained relatively constant.

Moran et al. [12] combined exergy with economic analysis to help decision makers in identifying how much an additional component would increase both the technical and economic performance. Wang et al. [13], for instance, applied the exergoeconomic principle of exergy loss per total capital investment to compare the performance of mechanically and thermally driven heat pumps. Their analysis shows that exergy loss per capital investment as a function of temperature lift differs between mechanically and thermally driven heat pumps. Based on this distinction, they formulated a guide map to aid in the selection of the technologies. In a follow-up study, Wang et al. [14] used the same principles when evaluating the performance of a transcritical heat pump cycle for hot water production and increased the heat pump's cost rate by 17%.

Hence, exergy-based economic analysis has emerged as a valuable tool for enhancing the techno-economic performance of heat pumps. However, its application to high temperature heat pumps has been limited. Moreso, to the author's knowledge, no known cases exist where exergy-based economic analysis has been applied to a SGHP despite the challenges in selecting design changes and their significant potential for improvements.

1.3. Objective

Assessing the performance of advanced heat pump configurations is challenging due to their substantial dependence on the temperature of the heat source and sink. This is especially true for steam generating heat pumps, where the exergy destruction of pressure equipment increases with the temperature lift and condenser temperature. To address this issue, this paper employs exergy-based economic analysis to identify techno-economic improvements to a steam generating heat pump and compares the results with an energy-based method. The objective of this paper is to illustrate how exergy-based economic analysis can be used to identify techno-economic improvements to a steam generating heat pump and show how the results differ from an energy-based method.

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2. Method

The method section is built up out of three sections. The first section goes into the general description of the followed approach. The underlying thermodynamic analysis is explained in section 2.2 and the costs calculations in 2.3.

2.1. Identification of techno-economic improvements

Advancements to a simple cycle heat pump configuration to produce steam were explored by using a four-step method: 1. setting costs targets, 2. performing energy, economic, and exergy analysis (3E-analysis), 3. assigning costs to exergy losses, and 4. assessment of design changes.

Step No. 1: To set cost targets, the performance of an ideal (i.e., lossless) heat pump was compared to an electric boiler. This comparison is common in industry because both are a way to industrial electrification. The heat pump was defined to be economically viable when the total costs of ownership (TCO) after five years of full-time operation (8000 h) were lower than that of an e-boiler. The cost of electricity was taken to be 0.041 €/kWh [15] based on the expected average electricity costs between 2022 and 2030 in the Netherlands for large consumers. The required capital investment for an e-boiler of the required size was based on a capital costs price of 165 €/kW [15]. For both the e-boiler and the heat pump an installation costs factor of 3 was used to convert bare unit costs to installed costs. Based on the TCO of an e-boiler and the operational costs of an ideal heat pump, the maximal capital investment for a heat pump was calculated.

The next step was initiated when the capital costs price of the heat pump was at the high end of typical heat pump cost price ranges (100-1000 €/kW) [16].

Step No. 2: An energy, economic, and exergy (3-E) analysis was applied to a heat pump. The thermodynamic states of the heat pump were fixed by the outlet conditions of both the evaporator and the condenser. For both heat exchangers the pinch point temperature difference was set to 5 K with respect to the heat source and sink and conditions are assumed to be saturated [17]. When subcooling and super heating are considered using an internal heat exchanger, the amount of heat transferred was limited by the temperature at the outlet of the compressor to below 175 °C to limit the degradation of compressor lubricants and seals [3]. In the assessment, the exergy value of heat was omitted when assessing heat exchangers, as an exergy balance was made over both sides. The compressor's isentropic efficiency was stated to be 70% and its mechanical efficiency was 85%. The costs of the bare units, e.g., the heat exchangers, were based on their duty and cost functions. The bare unit costs were indexed to December 2022 with the Chemical Engineering Price Index (CEPI) [18] and converted into total costs of installation (TCI) using an installation factor to account for the cost of integrating the unit, contingencies, and other fees.

Step No. 3: Exergy-aided cost minimization was applied and the exergy losses were translated into additional operational costs [19]. Since the operational costs of a heat pump consist mainly of electricity consumption by the compressor, the amount of work required to compensate for the exergy destruction of a component can be translated to additional operational costs with the help of electricity costs and the number of operational hours.

Step No. 4: advancements to the evaluated heat pump cycles were made. The component whose performance caused the largest increase in operational costs was selected to be changed. This change was realized by adding one of the standard cycle's components; i.e., a compressor, an expansion valve, an internal heat exchanger, an ejector, a closed economiser, a flash tank, a desuperheater, a cascade condenser, or an expander. The selection among these components was based on the origin of the exergy destruction and the estimated costs of the design change. This was also done based on an energy analysis to benchmark the results of the exergy analysis. Once selected, a 3-E evaluation was done on the advanced configuration. The resulting component, together with the changes in exergy destruction, were used to evaluate the techno-economic validity of the change in the design. The performance of the advanced heat pump should have exceeded that of the e-boiler and the previously evaluated heat pump configurations. This performance is defined based on four performance indicators: 1. the total costs of ownership, 2. the initial investment, 3. the coefficient of performance, and 4. the total exergy destruction. Of these, the first two indicate the economic viability, whereas the third and fourth give insight into the technical and environmental performance of the proposed configuration, respectively.

To illustrate the application of the method, it was applied to a case study where 10 tonnes per hour (t/h) of 2 bar(a) pressure steam is produced from a water condensate steam of 50 kg/s and an initial temperature of 80 °C. This case study was selected for its common appearance in different industries like chemical, paper, and food production. All configurations were modelled using refrigerant R-1234ze(Z) due to its high critical point, low GDP and ODP [11].

2.2. Thermodynamic analysis

2.2.1. Energy balance

The basis of the energy balance was a consistent mass balance. For the simple cycle, the mass flow rate of the refrigerant (m_r) was defined by the heat transferred required in the condenser (Q_{cd}), the enthalpy after compression, and after condensation, as shown in Eq. (1):

$$Q_{cd} = m_r(h_{cd,in} - h_{cd,out}) \tag{1}$$

The refrigerant exited the condenser as a saturated liquid. All open systems were assumed to operate in a steady state and did not accumulate mass. This also holds in the case of a (flash) vessel, where an enthalpy balance defined the quality of the vapour and hence the mass ratio of its outgoing streams, and thereby the mass ratio between the top and bottom cycle.

Work added to the system (W_c) by the compressor based on the isentropic enthalpy difference (Δh_{is}) between the pressure stages and an isentropic efficiency (η_{is}) of 70%, as indicated in Eq. (2):

$$W_c = \frac{m_r \Delta h_{is}}{\eta_{is}} \tag{2}$$

The required amount of work by the compressor's drive (W_D) was based on a correction for electrical, volumetric, and mechanical losses based on overall motor efficiency (η_m) of 85%, as shown in Eq. (3):

$$W_D = \frac{W_c}{\eta_m} \tag{3}$$

The intermediate pressure (p_i) is corrected by 0.35 bar when multiple pressure stages are considered based on the work by Mateu-Royo et al. (2018), see Eq. (4):

$$p_i = \sqrt{p_1 p_2} + 0.35 \tag{4}$$

where p_1 and p_2 are the pressures before and after the compressor, respectively. Pressure relief in expansion valves is considered isenthalpic. Other forms of pressure loss are neglected, as well as heat losses. The coefficient of performance (COP) of the heat pump was based on the heat delivered at the condenser and the work required by the compressor's drive, as shown in Eq. (5):

$$COP_{hp} = Q_{cd}/W_D \tag{5}$$

The energy balance of the heat pump was closed by defining the required thermal duty of the evaporator as the difference between the duties of the condenser and the compressor.

2.2.2. Exergy balance

The influx of exergy (Ex_{in}) equals the outflux of exergy (Ex_{out}) plus exergy losses. The loss of exergy is the sum of internal exergy destruction (Ex_{des}) and transfer of exergy to external sources [20]. Since heat loss to the environment was neglected and all heat transferred from the heat pump to the environment is valuable, the exergy balance simplifies to Eq. (6):

$$Ex_{des} = \Sigma Ex_{in} - \Sigma Ex_{out} \tag{6}$$

Exergy destruction will be zero in the case of an ideal operation. In that case, the exergy flowing into the system in the form of heat at the evaporator and work by the compressor is equivalent to the exergy of the outflow of heat at the condenser. The exergy value of the streams is defined by the enthalpy (H) and entropy (S) of the stream shown in Eq. (7) [21]:

$$Ex = H - H_0 - T_0(S - S_0) \tag{7}$$

Where subscript "0" denoted the reference state at T_0 = 298,15 K and p_0 = 101325 Pa. Substituting Eq. (7) in Eq. (6) and accounting for the exergy value of heat: $Q(1 - T_0/T)$ at a thermodynamic mean temperature and that of work: W, results in Eq. (8):

$$Ex_{des} = \left(1 - \frac{T_0}{T}\right)Q + W - [H_2 - H_1 - T_0(S_2 - S_1)] \tag{8}$$

Exergy destruction due to mechanical losses in the drive was taken as equivalent to the loss of work during transfer.

2.3. Costs equations

The total cost of ownership and the total capital investment are taken as the key performance indicators for the economic evaluations. These costs are based on the indexed bare unit costs of the components, the cost

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of installation, and operation. The bare unit costs ($C_{0,p}$) required for the heat pump's components were based on the costs function provided by Zuhlsdorf et al. (2019), which is presented in Eq (9):

$$\log(C_{0,p}) = k_1 + k_2 \log x + k_3 (\log x)^2 \quad (9)$$

where "x" is the scaling parameter of a certain technology and "k_i" is a calibrated value. Table 1. shows the used values adapted from Zuhlsdorf et al., [17].

Table 1. Parameters for estimation of component capital costs according to Zuhlsdorf et al., [17]

Component	Scaling Parameter X	Range	k_1	k_2	k_3	f_{cepi}	f_{bm}
Centrifugal compressor	Fluid power	450 - 3000 kW	2.2897	1.13604	-0.1027	2.3749	2.8
Drive	Shaft power	75 - 2600 kW	1.9560	1.7142	-0.2282	2.3749	1.5
Plain vessel	Volume	1 – 800 m^3	3.5970	0.2163	0.0934	2.0793	3.0
Shell & tube heat exch.	Area	10 – 900 m^2	3.2476	0.2264	0.0953	2.0793	3.2
Radial turbine	Fluid power	100 – 1500 kW	2.2476	1.4965	-0.1618	2.3749	3.5

These values were homogenized into the equivalent costs of the components for December 2022 with f_{cepi} based on the Chemical Engineering Price Index (CEPI) [18]. Bare module costs were converted into total capital investment (TCI) using the installation costs factor (f_{IF}), as shown in Eq. (10):

$$TCI_p = C_{0,p} f_{cepi} f_{IF} \quad (10)$$

For the centrifugal compressor and its drive, Eq. (4) was used as input for the scaling parameter by either including or excluding η_m, respectively. The resulting costs were benchmarked to the costs data provided in the DACE-booklet [22] and found to be plausible. In case multiple compressors were used, their scaling parameters were combined to account for the economics of scale. Their respective costs were based on the ratio between the scaling factor of the individual component and the scaling factor of the combined components. The volume of a vessel was based on being able to supply the outlet streams for 10 minutes without an influx of new refrigerant. The heat exchanging area (A) of the heat exchangers was calculated using Eq. (11):

$$Q_{hx} = U \cdot A \cdot \Delta T_{lm} \quad (11)$$

where "U" is the heat transfer coefficient and ΔT_{lm} is the logarithmic mean temperature difference between the hot and cold streams. A heat transfer coefficient of 1000 W/m^{2}K is used for heat transfer between a liquid and an evaporating liquid and 1250 W/m^{2}K was used when both sides were changing phases [17]. The total costs of the installation were equal to the sum of all the TCI of the components. This value is expressed as a factor of the condenser duty for benchmarking purposes.

The operational costs of the heat pump were defined to be equivalent to the electricity consumption by the compressor. Hence, the amount of work required to compensate for the exergy destruction of a component was translated to costs (c_i) by using electricity costs (c_{el}), and the number of operational hours in a certain period (t), as indicated in Eq. (12):

$$C_i = Ex_{des,i} \cdot c_{el} \cdot t \quad (12)$$

herein, the payback time of the process change should be less than five years of full-time operation or 40,000 hours. The costs resulting from non-ideal operation of the components were listed in descending order.

3. Results

Exergy-based cost minimization is used to advance the design of the heat pump. The performance of an ideal heat pump is compared to that of an e-boiler to indicate whether a heat pump would be an economically viable alternative. Thereafter, the performance of a single stage subcritical heat pump, a simple cycle, is assessed. Alterations to this base design are suggested based on its energetic and exergetic performance. The techno-economic performance of these advanced designs is thereafter compared.

3.1. Costs targets for a steam generating heat pump

To identify the techno-economic best heat pump configuration, an exploration of its technical surroundings and technical competitor, of the e-boiler was carried out. Producing 10 t/h of 2.0 bar(a) steam with an e-boiler required 6.5 MW of electricity. Based on the assumed bare unit costs of 165 €/kW and an installation factor of 3 this resulted in the costs as listed in Table 2. The total costs of ownership of the heat pump must be below that of the e-boiler to be competitive. In an ideal operation, this heat pump required 1.0 MW to operate, or 1.7 M€. Hence, the total installed costs must be below 12.3 M€, or 625 €/kW$_{th}$, which seemed plausible based on the benchmarked 100-1000 €/kW$_{th}$ [22]. The heat pump would then operate with a COP of 6.38 and reduce electricity consumption by 85%.

Table 2. Costs comparison of an ideal e-boiler with an ideal heat pump based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh – iterated values in italic

Unit	Supplied power [MW$_{th}$]	Required power [MW$_e$]	Specific costs [€/kW$_x$]	Bare unit costs [M€]	Total installed costs [M€]	Operational costs [M€]	Total costs of ownership [M€]
e-boiler	6.5	6.5	165	1.1	3.2	10.8	14.1
Heat pump	6.5	1.0	<625	<4.1	<12.3	<1.7	<14.1

3.2. Energy, Exergy, and Economic (3-E) performance assessment of a simple heat pump

The layout of a simple, single stage subcritical, heat pump thermodynamically consists of an evaporator, compressor, condenser, and expansion valve. The results of the 3E analysis of this cycle are presented in Table 3. These results show that the compressor's drive is the largest energy consumer. The electric drive of the compressor required 2.9 MW, and the COP was therefore 2.3. Hence, 1.9 MW is required to compensate for exergy destruction compared to the ideal heat pump. No costs were assigned to the expansion valve as the required capital investment was two orders of magnitude less than the other components. The compressor and its electric drive cost 3.2 M€ and made up for more than 70% of the total installed costs. Total costs of ownership (TCO) of 9.2 M€ were based on the electricity consumption of the drive and the total installed costs. The economic performance of the simple heat pump is a cost competitive option to the e-boiler, with a bare module cost of 234 €/kW.

The energy analysis presented in Table 3 highlights that work is solely required by the compressor and its driver. Consequently, enhancing the compressor's efficiency emerges as a viable solution based on the energy analysis. However, when subjected to exergy analysis, it is revealed that the expansion valve is responsible for the majority of exergy destruction, accounting for 689 kW (or 37% of total exergy destroyed) and incurring operational losses of over 1.1 M€. In addition, both the compressor and its drive account for 53% of the total 1848 kW destroyed. As a result, modifying the process to utilize the work potential before the expansion valve, and thereby reduce work requirement, is viewed as a promising avenue for improving the cycle based on the exergy analysis.

Table 3. 3E-evaluation of a simple cycle heat pump based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.1	0.0	95	156	278	161	514	
Compressor	0.0	2.4	545	894	2,448	724	2,028	
Drive	0.0	2.9	432	708	2,880	412	1,153	
Condenser	6.6	0.0	87	142	1,054	248	792	
Exp. Valve	0.0	0.0	689	1,131				
Total			1,848	3,031		1,544	4,488	9,2

3.3. Improving the heat pump configuration

Increasing the efficiency of the compressor, as suggested by the energy-analysis, was pursued in advanced configuration No. 1 (Figure 1.A.), which depicts the common [4, 8] configuration of sequential compression with intermediate cooling to improve upon the compressor's efficiency. Utilization of the exergy before the expansion valve, as suggested by the exergy analysis, was pursued with advanced configurations No. 2 (Figure 1.B.) and No. 3 (Figure 1.C.). The aim of advanced configuration No. 2 is to use the exergy available before the expansion valve to reduce work requirements by the compressor with the help of an internal heat exchanger (IHX), also known as a closed economizer. Various authors cite this option as an interesting configuration due to its low equipment costs [23]. Advanced configuration No. 3 was designed to utilize the exergy before the expansion valve and to increase the compressor's efficiency by reducing the temperature of the pressure gas after the first compressor and utilizing the exergy available after the condenser [24]. All three configurations are well-established in current practices and have a relatively high performance [8].

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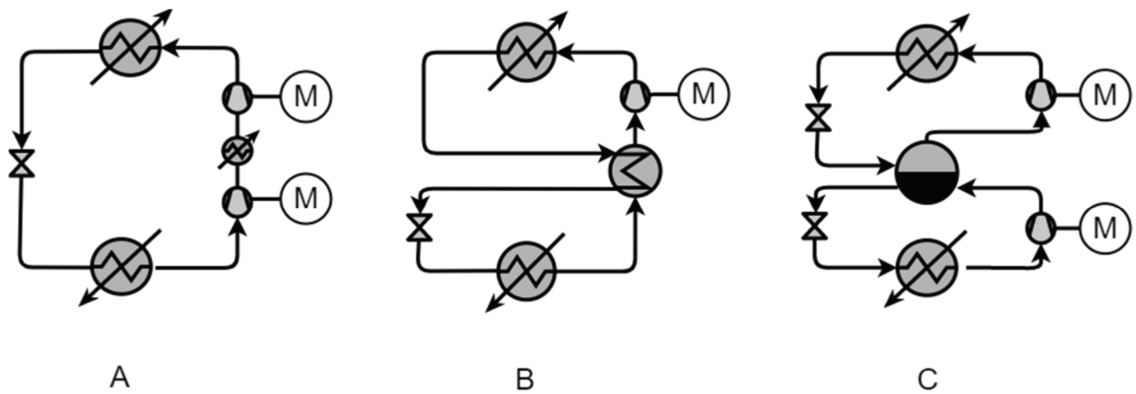

Figure. 1. Overview of advanced configurations. A) Advanced configuration No. 1: the reference simple cycle updated with an additional compressor and intermediate intercooler, B) Advanced configuration No. 2: the reference simple cycle updated with an internal heat exchanger (IHX), and C) Advanced configuration No. 3: the reference simple cycle updated with an open economizer.

3.3.1. Advanced configuration No. 1: Two stage compression with intermediate intercooling

The introduction of a second compression stage with an intermediate cooling step reduced the COP from 2.3 to 2.2, as work requirements by the driver increased to a total of 2.6 MW (Table 4.). Exergy losses are dominated by the expansion valve at 641 kW, or 32%. However, combined losses in both compressors and drives account for 1,042 of the total 2,016 kW exergy destroyed, or 53%. Though the specific work requirements by the compressor were slightly reduced by the intermediate cooling step, these gains are negated by the required increase in refrigerant mass flow to meet the energy demand in the condenser. This is partially a result of not being able to utilize the apparent heat in the intercooler due to its relatively low temperature of 92 – 89 °C and an initial sink temperature of 80 °C with an advised minimal temperature difference of at least 5 K [17]. Due to the higher work requirements and the additional investment, the total cost of ownership of this configuration is higher than that of the simple cycle. The total installed costs of the configuration are 5.3 M€, with a TCO of 10.3 M€, or 255 €/kW as a bare module.

Table 4. 3E-evaluation of advanced heat pump configuration No. 1 with two compression stages and intermediate cooling based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.3	0.0	174	286	282	161	516	
Compressor 1	0.0	1.4	343	563	1,392	397	1,112	
Drive 1	0.0	1.6	246	403	1,637	223	625	
intercooler	0.3	0.0	55	90	102	121	388	
Compressor 2	0.0	1.2	264	433	1,183	338	946	
Drive 2	0.0	1.4	209	342	1,392	190	946	
Condenser	6.6	0.0	84	138	1,054	248	792	
Exp. Valve	0.0	0.0	641	1,052				
Total			2,016	3,307		1,678	5,325	10.3

3.3.2. Advanced configuration No. 2: Internal heat exchanger

The addition of an internal heat exchanger (IHX) is a viable option to significantly increase the overall performance of the heat pump, leading to a reduction in the total costs of ownership (TCO) by 0.5 M€ in comparison to the simple cycle. Although the installation of the IHX increases the initial installed costs by 0.5 M€, the reduction in the size of the compressor and electric drive results in savings of 135 k€ in installed costs and a reduction of 0.8 M€ in operational costs after 5 years (Table 4.). The total cost of ownership for the heat pump with the IHX is 8.7 M€ or 249 €/kW as a bare module.

The lower operational cost is primarily attributed to the increase in the COP to 2.8 from 2.3, resulting from the addition of the IHX. This decrease in the work required by the drive from 0.6 MW to 2.3 MW is the main reason behind the increase in the COP. The total exergy destruction was reduced by 490 kW to 1,358 kW, with the compressor contributing the most to the total exergy destruction at 379 kW (28%), followed by its driver at 351 kW (25%), and the condenser at 476 kW (22%). The exergy destruction became evenly distributed among the components with the introduction of the IHX. The increase in exergy in the evaporator is due to the higher COP, requiring more energy from the sink and a temperature drop from 55 to 54 °C of the source. The exergy destruction in the condenser significantly increases due to its high inlet temperature, which is increased by superheating the suction gas before the compressor with the IHX. The introduction of the IHX led to an exergy destruction of 64 kW.

Table 5. 3E-evaluation of advanced heat pump configuration No. 2 with an internal heat exchanger based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Required power [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.6	0.0	195	319	288	162	520	
Compressor	0.0	2.0	379	621	1,991	682	1,909	
Drive	0.0	2.3	351	576	2,342	406	1,137	
IHX	2.3	0.0	64	105	185	143	456	
Condenser	6.6	0.0	290	476	1,054	248	792	
Exp. Valve	0.0	0.0	79	130				
Total			1,358	2,227		1,640	4,814	8.7

3.3.3. Advanced configuration No. 3: Two stage heat pump with open economizer

The addition of an open economizer splits work requirements over two compressors with a combined duty of 2 MW, or 2.4 MW at the electric drive. Hence, the required 6.6 MW at the condenser can be delivered with a COP of 2.8. Exergy destruction is evenly distributed among the components. The second stage compressor was the main source of exergy destruction at 269 kW (19% of total exergy destroyed), followed by its drive at 212 kW (15%) and the expansion valve directly after the condenser at 207 kW (15%). Together with the first stage and their drives, the compressors accounted for 60% of total exergy destruction, compared to 21% of both expansion valves, with a total of 1,4 MW destroyed. The exergy destruction in the evaporated increased by 100 kW as more heat is transferred. The intermediate cooling in the vessel slightly reduced exergy destruction in the condenser compared to the simple cycle. The vessel itself also has a negligible amount of exergy destruction, when considering heat transfer between the top and bottom cycle. Exergy destruction in the expansion valves went from 689 kW in the simple cycle to 294 kW for both valves. As a result, total exergy destruction was reduced by 445 kW. The reduced size of the compressors and electric drive reduced investment costs by 43 k€. However, the installation of the vessel requires an additional 0.55 M€. The TCO is 8.8 M€, or 255 €/kW as a bare module. Hence, the initial investment increases compared to the simple cycle, but the increased efficiency mitigates the impact of operational costs and reduced the TCO by 0.4 M€ during the five years.

Table 6. 3E-evaluation of advanced heat pump configuration No. 3 with an open economizer and two stage compression based on 5 years of operation, 8000 h/year, and an electricity price of 0.041 €/kWh.

Component	Heat transfer [MW]	Work transfer [MW]	Exergy destruction [kW]	Operational losses [k€]	Scaling factor [X]	Indexed bare unit costs [k€]	TCI [k€]	TCO [M€]
Evaporator	4.5	0	192	314	287	162	519	
Compressor 1.	0	0.8	204	335	826	279	781	
Driver 1.	0	1.0	146	239	972	165	463	
Vessel	5.4	0	1	2	690	183	548	
Compressor 2.	0	1.2	269	442	1,207	407	1,140	
Driver 2.	0	1.4	213	349	1,420	241	676	
Condenser	6.6	0	84	138	1,054	248	792	
Exp. Valve 1.	0	0	207	339		2		
Exp. Valve 2.	0	0	87.4	143		162		
Total			1,403	2,301		1,685	4,919	8.8

4. Discussion

The central question of this study was whether an exergy-based economic analysis provides a solid basis for identifying improvements to a heat pump's configuration. This question is answered by first evaluating the quality of the output and thereafter discussing the usability of the method.

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4.1. Quality of output

The approach described in this paper yields outputs in two phases. Firstly, the feasibility of an ideal heat pump is compared to an e-boiler based on the findings of section 3.1. The second phase involves modifying the heat pump's configuration to improve its techno-economic performance, based on the results of sections 3.2 and 3.3. It is worth noting that advanced configuration No. 1 is not included in this evaluation, as it was the outcome of design changes stemming from an energy analysis.

The first phase provides a broad indication of the feasibility of a heat pump compared to an e-boiler. The simplicity of this approach makes it inherently vulnerable to the quality of the data input. For example, changes in the installation costs factor or the costs of electricity are likely to move the solution in a certain direction. A “plug-and-play” e-boiler design is likely to reduce the installation costs factor, which will result in the capital costs being a smaller percentage of the TCO, and a lower TCO overall. Changes in the electricity price can have a similar impact on the decision between an e-boiler and a heat pump, as an e-boiler uses more electricity. The easiness of the approach makes it a valuable tool for a first assessment and conducting a sensitivity analysis will aid in exploring tipping points in the decision-making process. Moreover, additional considerations like the size of the grid connection and the dynamic behavior of both systems should be considered when comparing these techniques.

The second phase explored the advancement of the simple cycle heat pump configuration. This option is already an economically interesting alternative based on its COP of 2.3 and a bare unit cost of 234 €/kW compared to a COP >1 and a bare unit cost of 165 €/kW for the e-boiler. The TCO of the simple cycle is 9.2 M€ after 5 years of full-time operation, which is significantly less than the 14.1 M€ of the e-boiler. The expansion valve is the largest single source of exergy destruction in the simple cycle, as it accounts for 37% of the total exergy destruction. This is in line with the findings of Bergamini [10] and Hu [11]. Moreso the COP of 2.3 compared to the listed COP of 1.7 to 2.3 when transferring heat at 60 to 100 °C to a heat sink of 140 °C by Adamson et al. [8]. However, when considering the compressor (29%) and its drive (23%) as a single operational unit, they become the dominant source of exergy destruction at a combined 53%, on a total exergy destruction of 1,85 MW. Exergy destruction by the compressor’s drive is quite significant, despite its efficiency of 85%. This impact can be explained by the high exergy-value of the drive’s energy, work. The drive’s losses are entirely dependent on its duty and electric, volumetric, and mechanical efficiency. Hence, after the duty is reduced by advancing the cycle, this destruction could be reduced by more advanced mechanical equipment.

The addition of the internal heat exchanger reduced the amount of work that must be delivered to the compressor compared to the simple heat pump and increased the COP from 2.3 to 2.8. This is 22% larger than the 2.3 reported by Adamson et al. [8] at the same temperature lift with the same working media. This difference is the result of the 20 °C higher condenser temperature in the case of Adamson’s findings. This highlights the sensitivity of a heat pump's techno-economic performance to the temperature of the heat source and sink.

Advancing the simple cycle heat pump with an internal heat exchanger seems to be the preferred route compared to the open economizer, as it has similar operational (and emission) savings at a lower cost. This is in part due to the high efficiency with which the IHX can utilize the work potential compared to the vessel and the lower unit costs. This outcome is the opposite of what is expected based on the commonly found link between higher exergy efficiency and capital costs, as mentioned by Rosen and Dincer [25]. One possible explanation for this discrepancy is that the use of a constant compressor efficiency in the analysis does not account for the volumetric efficiency gains offered by the open economizer. Overall, the analysis highlights the importance of considering a detailed unit operation when evaluating the performance and economic feasibility of heat pump configurations.

4.2. Usability of exergy-based costs minimization

The starting point of this study was that a framework is needed that aids in identifying advanced configurations that improve the techno-economic performance of a considered heat pump and help decision-makers in their quest to identify techno-economically feasible options. The proposed approach helps visualize where losses manifest together with their accompanying costs. This is different from an energy analysis, which will only be able to account for the costs of the operation of the drives and the profits from supplying heat to the condenser. Advanced configuration No. 1., where an additional compression stage and intermediate cooling are added to the simple cycle design, is a logical result of energy analysis, as it uses a common strategy to improve upon the efficiency of the compressor. However, the changes did not increase techno-economic performance in this case study. Moreso, it reduced the COP by 0.1, increased exergy destruction by 168 kW, increased the total capital investment by 837 k€, and increased the total cost of operation by 1.1 M€. Hence, reducing the overall performance of the simple cycle heat pump. Another route based on the outcomes of an energy analysis would be to buy a more efficient compressor and drive. For these upgrades to be economically competitive with the second advanced configuration with the internal heat exchanger, the overall motor efficiency must increase from 85% to 95% and the isentropic efficiency from 70% to 85%, whilst the cost increase of the compressor increases is limited to 13%.

In contrast, both design changes based on exergy analysis improved the techno-economic performance of the simple cycle heat pump. However, both advanced options still had an exergy destruction of 1.4 MW. The degree to which this can be reduced is currently unknown.

5. Conclusions and recommendations

Exergy based economic analysis can be used to identify techno-economic improvements to a steam generating heat pump. Not only was the method able to quickly assess the boundaries to its economic performance compared to other technologies (e.g., an e-boiler), but also improved on the technical performance of a single stage subcritical heat pump layout, whereas suggestions by an energy analysis were not.

The exergy-based costs minimization method identified the expansion valve as the source of most operational costs, as it accounts 37% of the total exergy destruction. However, the compressor and its drive combined contributed 53%. The work potential available before the expansion valve can be used to improve on the compressor efficiency by either a closed or an open economizer. The closed economizer, or internal heat exchanger, increased the heat pump's COP from 2.3 to 2.8 at the cost of an additional investment of 0.5 M€. This additional investment reduces the total costs of ownership by 0.6 M€ after five years of operation. The open economizer also increased the COP from 2.3 to 2.8 and required an additional investment of 0.5 M€. The total costs of ownership are reduced by 0.4 M€ after five years of operation. The approach thus succeeded in identifying techno-economically beneficial process changes and helped to produce two options that both increased the technical and economic performance of the heat pump, whereas the energy-based approach did not.

However, the exergy-based approach only pinpoints the exergy destruction that could be utilized. It is not able to identify how much the exergy destruction could be reduced. A variation of exergy analysis link *Advanced exergy analysis* as proposed by Tsatsaronis [20] may aid in this process. Moreover, the method does not suggest the technology that should be selected. Hence an extension of the framework is needed in which a strong link is made between the techno-economic performance gains of an additional component and the operational context of the heat pump. The same holds for a link to other working media, where a link between the gradient of the isentropic curves and exergy destruction is a likely candidate based on the presented results (i.e., the two stage compressor with intermediate cooling requiring more work). Lastly, it is recommended to use a more detailed thermodynamic analysis of the operational units to provide a fair basis for the comparison of added components. The assumption of a constant isentropic efficiency is likely to skew the presented results.

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Nomenclature

A	heat exchanger area (m2)
C	costs (€)
COP	Coefficient of Performance
Ex	Exergy [kJ/kg]
f	factor (-)
h	enthalpy [kJ/kg]
IHX	internal heat exchanger
K	Costs factor (-)
m	mass flow rate [kg/s]
q	heat transfer rate [kW]
S	Entropy [kJ/kgK]
T	Temperature [K]
t	time (s)
TCI	Total capital investment (€)
TCO	Total costs of ownership (€)
U	heat transfer coefficient [W/m2K]
W	Work/power [kW]

Subscripts & superscripts

0,p	bare unit value of component "p"
c	compressor
cd	condenser
CEPI	Chemical Engineering Plant Costs Index
d	drive
des	destruction
el	electricity
evap	evaporator
hp	heat pump
hx	heat exchanger
i	intermediate
if	installation factor
in	influx
is	isentropic
lm	logarithmic mean
m	mechanical

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X	scaling factor	*out*	outflux
Greek symbols		*r*	refrigerant
η	efficiency		
Δ	difference		

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