15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

Optimization of the utilization of different industrial waste heat streams via high temperature heat pumps

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Abstract

The utilization of industrial waste heat for process or district heat supply is key for a sustainable energy supply. The potential of industrial waste heat is high, and the temperature levels are beneficial to achieve high Coefficient Of Performances (COP). However, those waste heat streams are mostly sensible and not latent. To use their full potential of usable heat, those heat fluxes have to be cooled down to ambient temperature levels within the heat pump.
This paper shows a new methodology for reference COP evaluation to assess the full potential of usable sensible waste heat sources cooling it down to ambient temperature level, because neither the Carnot or the Lorenz or the Adler/Zotter/Längauer cycle are suitable for this. In order to achieve highest heat output at the maximum of sensible waste heat streams utilization, two new reference cycles and COP assessments are introduced. The LZ reference cycle must be used for sensible waste heat utilization supplying sensible heat sinks (e.g. hot pressurized water) and the CZ for sensible waste heat utilization supplying latent heat sinks (e.g. steam). For a heat pump supplying steam at 150 °C from a sensible heat source at 90 °C, different reference COPs lead to significantly different performance indicators (COP_Carnot = 4.7, COP_Lorenz = 4.3, COP_AZL = 5.6). However, when sensible waste heat is to be fully utilized, the entire available heat flow must be cooled down to near ambient temperature. In this case, cooling the source from 90 °C to 0 °C instead of 60 °C allows approximately three times more waste heat to be recovered, but limits the achievable efficiency to a maximum COP of about 4.4 based on the CZ efficiency using the full waste heat potential. The effects of different theoretical heat pump cycles on possible COPs and their potential for industrial high-temperature heat pump applications are described in detail.

Keywords: Carnot, Lorenz, Adler/Zotter/Längauer, Entropy, COP, Exergy, cyle

1. Introduction

Industry is one of the world's largest energy-intensive sectors, and the majority of this energy is used for process heat. A significant part is discharged unused into the environment in the form of waste heat via flue gases, cooling systems, exhaust air or hot product streams. It was found that "processes with an input temperature above 200 °C are to be considered as possible sources of waste heat" [1]. Industrial waste heat is therefore one of the largest underused energy sources in the energy system and is increasingly recognized as an important lever for industrial decarbonization.

This resource needs to be quantified in order to establish that "the total waste heat potential for example in the EU, which according to [2] is about 300 TWh/year" and that "a third of this corresponds to a temperature below 200 °C" [2]. According to [2], the temperature level of the waste heat is the key parameter for the possible path of use. The reason for this is that high-temperature waste heat can sometimes be reused directly, while low- and medium-temperature waste heat is much more difficult to recover and therefore often goes unused.

In this context, High Temperature Heat Pumps (HTHPs) are an effective way for heat recovery of waste heat at low and medium temperature levels [3, 4, 5, 6] and HTHPs with higher temperature lifts offer significant potential to improve thermally demanding industrial processes due to their high temperature lifting ability [7]. According to [8], steam generation (at 200 °C) by recovering industrial waste heat at 60 °C is already demonstrated, which underlines that even very large temperature increases are thermodynamically possible with multi-stage or cascading heat pumps. This is particularly relevant as steam remains a dominant heat transfer medium in many industrial sectors at around 150 to 200 °C, while a large part of the available waste heat is well below these temperature values [9].

Despite this great technical potential, several hurdles continue to hinder large-scale distribution. It was emphasized that "the economic profitability of waste heat recovery depends strongly on site-specific boundary conditions" [1], and that central obstacles are "high investment costs, technical complexity and lack of standardization" [8] as well as low seasonal performance factors (SPF) of heat pumps [5]. Nevertheless, the central message is clear: industrial waste heat is not a niche resource, but a structurally important pillar of a future low-carbon industrial energy system.

Despite this, the performance assessment and comparison of heat pump systems is still very often based on idealized assumptions. In particular, Carnot-based reference coefficients of performance (COP) or exergetic efficiency factors are frequently computed using a single characteristic source and sink temperature. For sensible heat sources and sinks, this approach is fundamentally problematic, because it neglects the continuous temperature change during heat transfer.

Depending on how the reference temperatures are chosen, two systematic errors occur in practice. If the source inlet temperature is used as reference, the theoretical potential is often overestimated, because a large part of the heat is actually available at much lower temperatures. If, on the other hand, the source outlet temperature is used, the theoretical potential is often strongly underestimated, because the high-exergetic fraction of the heat available at higher temperatures is not properly accounted for. In addition, in many approaches the exergy content of sensible heat is still evaluated using a simple Carnot factor, which again does not properly reflect the temperature glide of the heat source.

A further practical consequence of these inconsistencies is that sensible waste heat sources are often not cooled down sufficiently, so that a significant share of high-exergetic heat is still rejected to the ambient and remains unused. This means that even if a heat pump is applied, the actual utilization of the waste heat source can remain thermodynamically inefficient due to inappropriate system boundaries and evaluation metrics.

From a thermodynamic point of view, this situation is analogous to the historical development of ideal reference processes. The Carnot cycle provides a theoretical upper limit for isothermal heat sources and sinks, while the Lorenz cycle extends this concept to sensible heat sources and sinks with temperature glide. However, for many modern industrial heat pump applications — in particular for steam-generating heat pumps utilizing sensible waste heat — no consistent and application-specific theoretical reference cycles are available so far.

This paper presents a new methodology to quantify the maximum achievable Coefficient Of Performance (COP) of high-temperature heat pumps (HTHPs) for the recovery of different kinds of industrial waste heat sources (see Section 3) and for various heat supply applications (see Section 4). The proposed approach is described in detail in Section 5.

2. Overview of electrical driven HTHPs Cycle for suitable waste heat recovery

According to [9] electrically driven high-temperature heat pumps (HTHPs) show a high potential for industrial applications and a rising market. However, different electrically driven heat pump cycles and working fluids are used. Fig. 1 provides an overview of the considered heat pump cycle concepts. From left to right, the figure shows the (subcritical) vapor compression (Perkins/Evans) cycle [10], the (transcritical) Lorentzen cycle [11], the Osenbrück cycle [13], the (supercritical) reverse (counterclockwise) Joule/Brayton cycle (rJC) [9], the(supercritical) reverse Joule/Brayton cycle in a Rotation Heat Pump (RHP) [15] or the (supercritical) Adler/Zotter/Längauer-Cycle [9], with isothermal heat rejection and a heat absorption with temperature glide. All cycle layouts are illustrated in terms of their principal component structure as well as by their corresponding representation in the temperature–entropy (T–s) diagram. In the central part of the figure, these cycles are highlighted consisting of an isentropic compression process in the compressor and an isentropic expansion process in a turbine or expander. In contrast, the state change in throttling devices is isenthalpic. For the idealized representations, pressure losses in heat exchangers and piping are neglected. In the lower part of the figure, the same cycle concepts are shown including non-isentropic compression and expansion processes, thereby illustrating the effect of real component inefficiencies.

Among these concepts, vapor compression heat pumps based on the Perkins/Evans cycle (PEC) currently represent the dominant technological solution on the market for high-temperature applications, as they achieve reasonably high COP.

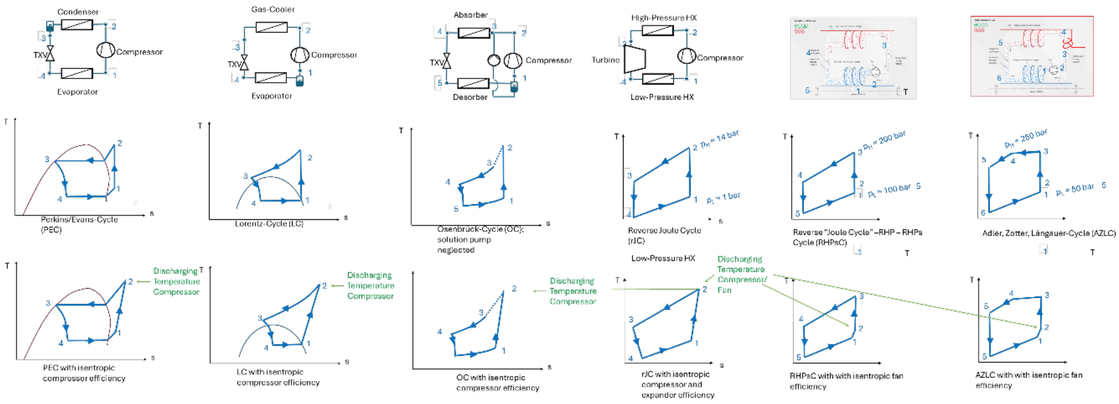


Figure 1: Overview of the cycles – from left to right – (subcritical) Perkins/Evans cycle [10], the Lorentzen (transcritical) cycle [11], the Osenbrück cycle [13], the reverse (counterclockwise and supercritical) Joule/Brayton cycle (rJC) [10], the reversible Joule Cycle in a Rotation Heat Pump (RHP) [14] and the Adler/Zotter/Längauer-Cycle [9]

In practice, the selection of a suitable heat pump cycle is primarily determined by the characteristics of the heat sink, like if hot pressurized water is required (sensible heat) or steam is required (latent heat, which means at constant temperature (isothermal state of change)). A vapor compression-based cycle like Perkins/Evans cycle fits for latent heat source and unlimited heat source, like ambient air, where a high mass flow could be provided, which also leads to very low temperature changes during the heat absorption.

A major disadvantage of vapor compression-based cycles, besides their limited suitability for sensible heat sources (see chapter 3), is the very high compressor discharge temperature. Even advanced variants such as the Lorentzen cycle, the Osenbrück cycle, and Perkins/Evans cycle -based systems using zeotropic mixtures are not well suited for applications with large temperature glides at the heat source.

Hower, all those different cycles are mostly evaluated in comparison to Carnot cycles [15], which is an inappropriate methodology when it comes to waste heat utilization with heat pumps. The reason is that the kind of heat fluxes is mostly not latent and the full potential of waste heat streams is not considered in this methodology.

3. Analysis of waste heat potential

A major limitation of many existing waste heat potential assessments is that, apart from the temperature level, they rarely distinguish between the thermodynamic characteristic of heat source, i.e. whether the available waste heat is sensible or latent:

Most large-scale assessments classify waste heat streams mainly according to temperature bands [1,2], although this alone is not sufficient to describe the actual recoverability and usability of the heat. In industrial systems, most waste heat streams – such as flue gases, exhaust air, cooling water or product cooling streams – are predominantly sensible heat sources [3,4], characterized by a continuous drop in temperature during heat absorption [10]. In contrast, latent heat sources, such as the condensation of process vapors or recooling waste heat from chillers, emit heat at almost constant temperature and are therefore thermodynamically more favourable for heat recovery (neglecting the sensible heat for to cool down the superheated vapor from discharging temperature of the compressor down to condensing temperature) and. For example, if 90 °C $(t_{Sourc,in})$ hot water will be cooled down to 60 °C $(t_{Source,out})$ instead to 80 °C, 200% more waste heat capacity $(\dot{Q})$ could be recovered by the HTHP, see Fig. 2 (according to Eq. 1).

$$\dot{Q}_{Source,sensible} = \dot{m}_{Source} * c_p * (t_{Source,in} - t_{Source,out}) \quad \text{Eq. (1)}$$

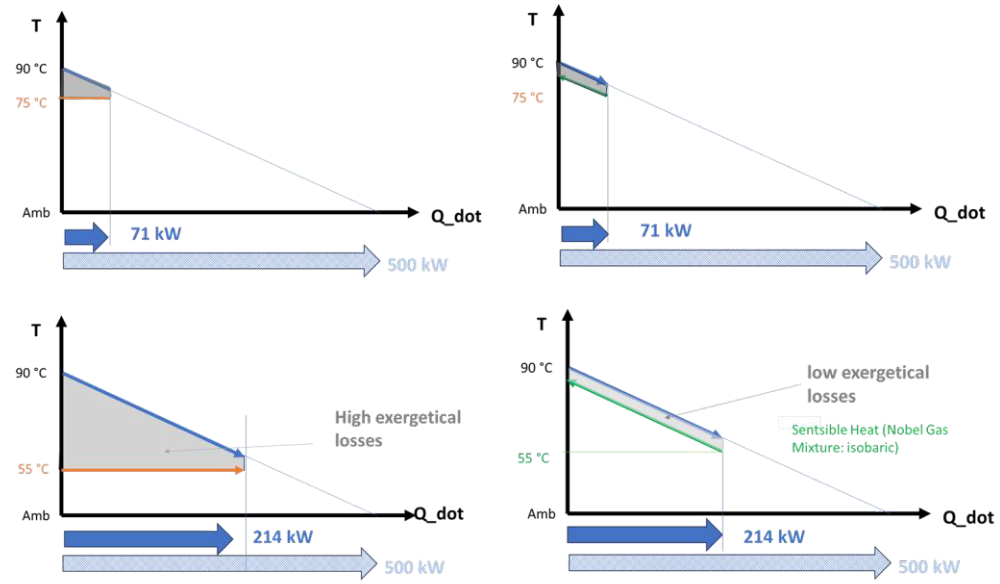


Figure 2: Comparison of the heat extraction of a sensible waste heat stream as source of a heat pump in the t vs $\dot{Q}$ – diagram in the low pressure heat exchanger (left: in an evaporator with latent heat absorption like with a single refrigerant and right: in a low pressure heat exchanger with sensible heat adsorption like in a reverse Brayton - or Osenbrück Cycle), top: heat absorption to 80 °C and bottom: heat absorption to 60 °C (setting a pinch point temperature difference of 5 K, ambient temperature 20 °C) [10]

From an energetical point of view to recover the entire potential of a waste heat stream ($\dot{Q}_{Source,max}$), the waste heat stream has to be cooled down to ambient temperature, according to Eq. 2. So the usable entropic mean temperature according to Lorenz [16] COP evaluation for temperature glides of sources (see Eq. 4).

$$\dot{Q}_{Source,max} = \dot{m}_{Source} * c_p * (t_{Source,in} - t_{Amb}) \qquad \text{Eq. (2)}$$

$$\dot{Q}_{Source,max} = \oint T * \dot{m} * ds = \bar{T}_{Source,Qmax} * \dot{m}_{Source} * \Delta s_{Source,Qmax} \qquad \text{Eq. (3)}$$

$$\bar{T}_{Source} = \frac{\Delta h_{Source}}{\Delta s_{Source,Qmax}} = \frac{c_p*(t_{Sourc,in}-t_{Amb})}{c_p*\ln\left(\frac{T_{Sourc,in}}{T_{Amb}}\right)} \qquad \text{Eq. (4)}$$

Furthermore, from a thermodynamic point of view, heat pumping cycles, like the rJC or the RHP (working in the supercritical region at an isobaric state of change) offer a temperature glide while the heat absorption, which fits best for using heat sources with high temperature glides, in comparison to heat pumping cycles operating with an evaporator. Therefore, HTHPs cycles with temperature glides in heat absorption are perfectly fitting for sensible heat sources where high temperature spreads are required. As already mentioned, for the energetical utilization of industrial waste heat streams it is important to extract as much heat as possible. Fulfilling that, irreversibility is less in the low-pressure heat exchanger (HX) of HTHPs cycle with temperature glides compared to the evaporator of the PEC for sensible heat sources (Fig. 2. The reason therefore is that the exergy value of a sensible heat can be determined according to Eq. 5 (where T is the inlet or absolute temperature level of the heat and ΔT the temperature glide of it and T_{amb} is the ambient temperature (all in K)).

The exergetical losses ($\dot{E}x_{loss}$) can be determined according to the Gouy-Stodola Theorem [17], see Eq. 6, where the irreversibility is calculated based on T_{amb} and the difference of total entropy change between the waste heat source and refrigerant-side. Due to the fact that the refrigerant has to absorb the same amount of heat capacity ($\dot{Q}_{Source,ref}$) as from the source gets emitted ($\dot{Q}_{Source,sensible}$), the total entropy change of the refrigerant ($\dot{m}_{ref} * \Delta s_{Source,ref}$) is higher than the total entropy change of the source ($\dot{m}_{Source} * \Delta s_{Source,qmax}$) if the average temperature of the source ($\bar{T}_{source}$) is higher than the average temperature of the refrigerant ($\bar{T}_{Source,ref}$), see Eq.7.

A HTHP-cycle offering a temperature glide in heat absorption fits for sensible heat sources as far as a Perkin/Evans-cycle fits better to latent heat sources. However, for a lot of waste heat utilization the exergetical losses are up to 6 times higher in isothermal heat absorption compared to the one with glide.

$$\dot{Ex}_{\dot{Q},sensible,Qmax} = \left(1 - \frac{T_{amb}}{T}\right) * d\dot{Q} = \oint \left(1 - \frac{T_{amb}}{T}\right) * \dot{m}_{Source} * c_p * dT = \dot{Q}_{Source,sensible,max} - T_{amb} * \dot{m}_{Source} * c_p * ln\left(\frac{T_{Sourc,in}}{T_{Amb}}\right) = \dot{Q}_{Source,sensbile,max} * \left(1 - T_{amb} * \frac{1}{t_{Sourc,in} - t_{Amb}} * ln\left(\frac{T_{Sourc,in}}{T_{Amb}}\right)\right) \quad \text{Eq. (4)}$$

$$\dot{Ex}_{Q,source,latent} = \left(1 - \frac{T_{amb}}{T}\right) * d\dot{Q} = \left(1 - \frac{T_{amb}}{T_{Source}}\right) * \dot{Q}_{Source,slatent} \quad \text{Eq. (5)}$$

$$\dot{Ex}_{loss,source} = T_{amb}\left(-\dot{m}_{Source} * \Delta s_{Source,qmax} + \dot{m}_{ref} * \Delta s_{Source,ref}\right) \quad \text{Eq. (6)}$$

$$\dot{Q}_{Source,sensible} = \oint \bar{T}_{source} * \dot{m}_{Source} * ds_{Source} = \dot{Q}_{Source,ref} = \oint \bar{T}_{Source,ref} * \dot{m}_{ref} * ds_{Source,ref} \quad \text{Eq. (7)}$$

From the perspective of the second law of thermodynamics, latent heat sources show a higher thermodynamic quality at a certain temperature level, since the heat transfer can be absorbed in the heat pump at higher entropic mean temperatures. Neglecting this distinction can lead to a systematic overestimation of the practically recoverable proportion of industrial waste heat, since the effective entropic temperature for sensible heat sources is lowered by the necessary cooling of them. Consequently, two waste heat streams with identical input temperatures into the evaporator or heat source of the heat pumps may differ significantly in their actual thermodynamic recoverability, depending on whether the heat is available in sensible or latent form. A comprehensive and technology-relevant assessment of industrial waste heat potentials should therefore not only classify sources according to temperature level, but also explicitly distinguish between sensible and latent heat fluxes and take into account their fundamentally different thermodynamic properties.

4. Analysis of heat supply

Many industrial processes require heat supply in the form of steam (latent heat) or pressurized hot water at temperatures of up to 200 °C (sensible heat). Consequently, HTHPs should be selected according if they have to provide sensible or latent heat, as different cycles are available (see Fig. 2). This requirement places stringent demands on the thermodynamic design of the chosen heat pump cycle, particularly with respect that the cycle matches the sink and the minimization of irreversibilities.

As described for the heat sources, the exergetic losses $\left(\dot{Ex}_{loss}\right)$ are determined according to the Gouy–Stodola theorem [17], see Eq. (8). In this formulation, the irreversibility is calculated based on the ambient temperature T_{amb} and the difference in total entropy change between the heat sink and the refrigerant side. This approach allows a consistent quantification of the thermodynamic quality degradation occurring during the heat transfer process.

A key factor influencing the magnitude of these exergetic losses is the difference of the temperature profiles of refrigerant and heat sink during heat rejection. A HTHP cycle offering a temperature glide during heat rejection is particularly well suited for sensible heat sinks, such as hot pressurized water or thermal oil-circuits with significant temperature spreads. In contrast, a phase-change-based heat exchanger configuration (PEC) is better suited for latent heat sources, as for steam generation where heat is released at nearly constant temperature, for example during condensation.

$$\dot{Ex}_{loss,sink} = T_{amb}\left(\dot{m}_{Sink} * \Delta s_{Sink} - \dot{m}_{ref} * \Delta s_{Sink,ref}\right) \quad \text{Eq. (8)}$$

5. Analysis of HTHPs using different kinds of heat sources for different heat sinks

The COP (see Eq. 9) can be approximated in a simplified form according to Carnot [15]. This is a common approach, and used for example for evaluation of heat pump exergetical quality for funding in Germany. However, the Carnot cycle includes two isentropic state of changes for compression and expansions, as well as an isothermal heat absorption and an isothermal heat rejection. So the evaluation regarding Carnot is fitting for a latent heat source and latent heat sink (Eq. 10).

$$COP = \frac{\dot{Q}_H}{P_{el}} \qquad \text{Eq. (9)}$$

$$COP_{Carnot} = \frac{T_H}{T_H - T_0} \qquad \text{Eq. (10)}$$

While the Carnot cycle is well suited for the evaluation of heat pumps operating between latent heat sources and sinks, due to the isothermal heat absorption and rejection matching such applications, the Lorenz cycle is more appropriate for systems involving sensible heat sources and sinks, as originally shown by Lorenz [16] (Eq. 11), using the mean entropy-based (logarithmic) source temperature $\bar{T}_{0M}$ (see Eq. 12) and the mean logarithmic sink temperature $\bar{T}_{HM}$ (see Eq. 13). Both temperatures depend on the respective inlet and outlet temperatures ($t_{0,in}$ & $t_{0,out}$) or ($t_{H,out}$ & $t_{H,in}$). The Carnot and Lorenz efficiency factor ν_{Lor} is defined as the ratio between the actual COP and the simplified Carnot respectively Lorenz COP (see Eq. 14 & 15). According to [16], the Lorenz efficiency is preferable to the Carnot efficiency for determining the optimal COP of water-to-water HPs, since both the heat source and sink are sensible and no latent heat is involved.

$$COP_{Lor} = \frac{\bar{T}_{HM}}{\bar{T}_{HM} - \bar{T}_{0M}} \qquad \text{Eq. (11)}$$

$$\bar{T}_{0M} = \frac{t_{0,\text{in}} - t_{0,\text{out}}}{ln\left(\frac{T_{0,\text{in}}}{T_{0,\text{out}}}\right)} \qquad \text{Eq. (12)}$$

$$\bar{T}_{HM} = \frac{t_{\text{S}} - t_{\text{R}}}{ln\left(\frac{T_{\text{S}}}{T_{\text{R}}}\right)} \qquad \text{Eq. (13)}$$

$$v_{Carnot} = \frac{COP}{COP_{Carnot}} \qquad \text{Eq. (14)}$$

$$v_{Lor} = \frac{COP}{COP_{Lor}} \qquad \text{Eq. (15)}$$

5.1. Analysis of HTHPs-Cycle utilization of a latent heat source and providing heat for a latent sink

However, if both COPs are compared to each other for specific applications, it becomes evident that different theoretical cycle concepts achieve higher COPs depending on the characteristics of the heat source and the heat sink. For example, in the case of a latent heat source at 90 °C and a latent heat sink at 150°C the theoretical COP_{Carnot} reaches 7.1, which is significantly higher than corresponding COP_{Lor} of 3.8, see Fig. 3.

The Carnot and Lorenz cycles are represented in the temperature–entropy (T–s) diagram instead of the temperature–enthalpy (T–h) diagram because the derivation of these theoretical reference cycle is fundamentally based on the requirement that the entropy change associated with heat absorption equals that associated with heat rejection (Δs_in = Δs_out). Lorenz and Carnot are defined by entropy balance not by enthalpy balance. This condition, as well as the different evolution of the product of mass flow rate and specific entropy, can be visualized directly and unambiguously only in the T–s diagram. Furthermore, the T–$\Delta\dot{S}$ diagrams is used, because this allow a direct geometric interpretation of irreversibility (exergy losses) as area differences between the reference cycle and the real heat source (from 0 K to ambient temperature), which is not possible in the T–h representation [1]

[1] For better visualization of the T–$\Delta\dot{S}$ diagram, a larger pinch-point temperature difference must be chosen, since a direct geometric interpretation of irreversibility requires the temperature scale to start at 0 K. However, for typical pinch-point temperature differences in the range of 3 to 5 K, the exact value has only a minor influence on the results.

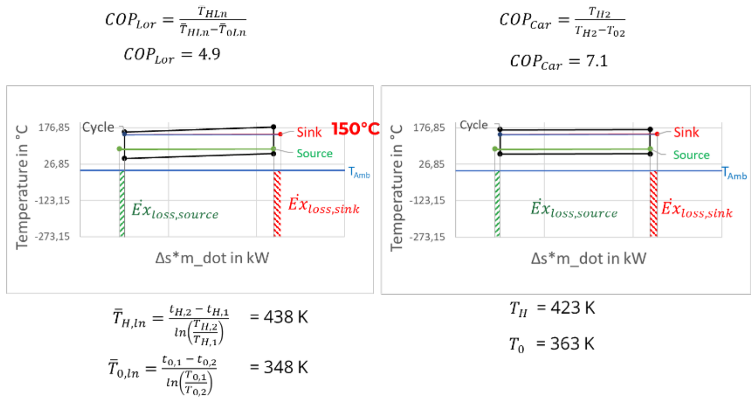


Figure 3: Comparison of the Lorenz and Carnot Cycle (and COP) for latent heat source at 90 °C and a latent heat sink at 150 °C

5.2. Analysis of HTHPs-Cycle utilization of a sensible heat source and providing heat for a sensible sink

In contrast to that, if the heat sink and source exhibit sensible temperature profiles, the situation changes fundamentally. If the inlet temperature of the heat sink is 110°C and the outlet temperature is 150°C, while the inlet temperature of the heat source is 90°C and the outlet temperature is 60°C, the theoretical COP_{Lor} reaches 6.9, while the corresponding COP_{Carnot} is only 4.9, see Fig. 4. The reason for that is that the outlet temperature of the source defines T_0 and the outlet temperature of the sink T_H, neglecting any required pinch point temperature difference between external heat transfer media and the refrigerant. So in this case, the Lorenz cycle represents the more appropriate theoretical reference, since it accounts for the temperature glide occurring during heat absorption and heat rejection. Due to the improved temperature matching between the refrigerant and the external heat streams, the average temperature difference in the heat exchangers is reduced, which leads to lower entropy generation and, consequently, to lower exergy losses and therefore to higher achievable COPs. In an ideal way the temperature glides of the heat sink and heat source cannot be identical, since this would be implying an infinite COP.

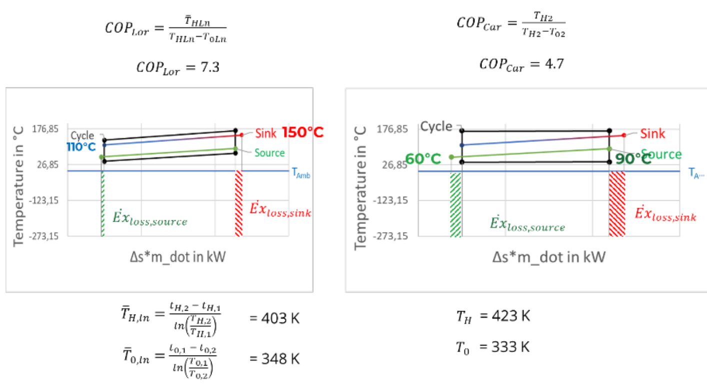


Figure 4: Comparison of the Lorenz and Carnot Cycle (and COP) for sensible heat source (90 °C to 60 °C) and a sensible heat sink (110 °C to 150 °C)

5.3. Analysis of HTHPs-Cycle utilization of sensible heat sources and providing heat for latent sinks

However, the Carnot cycle represents the ideal cycle for a latent heat source and a latent heat sink, while the Lorenz cycle typifies the ideal cycle for a sensible heat source and a sensible heat sink, for HTHP applications that convert a sensible heat source into steam (latent heat sink), neither the Lorenz nor the Carnot cycle are suitable as an ideal comparison cycle. The Adler/Zotter/Länglauer (AZL) -cycle [9] corresponds as an ideal comparison cycle for these HTHP applications. Considering a sensible heat source and latent heat sink, the ideal COP_{AZL} can be determined according to Eq. (16).

$$COP_{AZL} = \frac{T_H}{T_H - \bar{T}_{0M}} \qquad \text{Eq. (16)}$$

If the inlet temperature of the latent heat sink is 150°C, while the inlet temperature of the heat source is 90°C and the outlet temperature is 60°C, the theoretical COP_{Carnot} reaches 4.7, while the corresponding COP_{Lor} is 4.3, however, in this case having a latent sink and a sensible source the AZL-cycle represents the ideal theoretical cycle for comparison achieving the highest theoretical COP_{AZL} of 5.6, see Fig. 5.

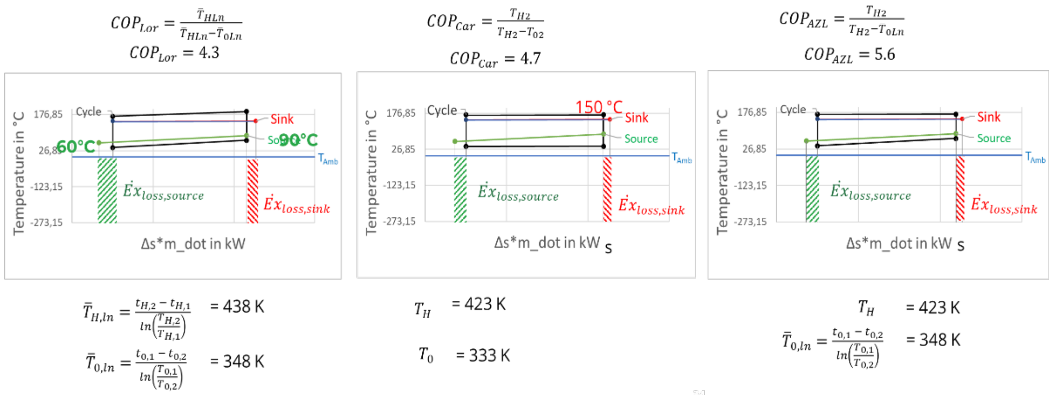


Figure 5: Comparison of the Lorenz, Carnot Cycle and Adler/Zotter/Längauer-Cycle (and COP) for sensible heat source (90 °C to 60 °C) and a latent heat sink (150 °C)

5.4. Analysis of HTHPs-Cycle utilization of sensible waste heat sources

However, considering the argument discussed in Section 3, a sensible waste heat source is energetically only fully utilized if it is cooled down as much as possible. From an exergetic point of view, this lower limit corresponds to the ambient temperature. Consequently, to assess the *full* theoretical utilization potential of sensible waste heat sources, a fundamentally new performance metric is required.

For this purpose, this work introduces the coefficient of performance $\mathrm{COP_{CZ}}$, which is defined as the maximum achievable COP under the constraint that the heat to provide is for a latent sink while the sensible heat source is cooled down to ambient temperature, i.e., that its exergy content is exploited as completely as possible, see Fig. 6. For sensible heat sink applications using sensible waste heat streams the so called $\mathrm{COP_{LZ}}$ is defined as maximum achievable COP under the premise that the heat to provide is for a sensible sink while the sensible heat source is cooled down to ambient temperature, see Fig. 7.

The determination of $\mathrm{COP_{CZ}}$ is shown in Eq. 15 and for $\mathrm{COP_{LZ}}$ in Eq. 17. $\mathrm{COP_{CZ}}$ is similar to the determination of COP_{Carnot} (Eq. 9) and $\mathrm{COP_{LZ}}$ (Eq. 18) is comparable to $\mathrm{COP_{Lor}}$(Eq. 10), the significant difference between them is that for $\mathrm{COP_{CZ}}$ and $\mathrm{COP_{LZ}}$ as source temperature the mean entropic source ambient temperature ($\bar{T}_{0AM}$, see Eq. 19) is used.

$$COP_{CZ} = \frac{T_H}{T_H - \bar{T}_{0AM}} \qquad \text{Eq. (17)}$$

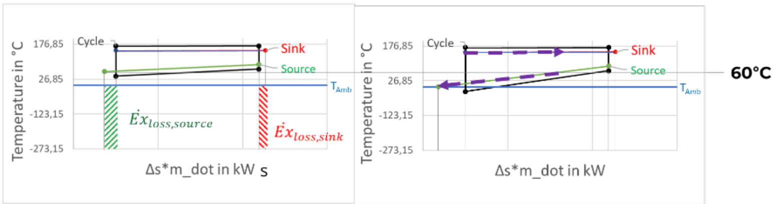
$$COP_{LZ} = \frac{\bar{T}_{HM}}{\bar{T}_{HM} - \bar{T}_{0AM}} \qquad \text{Eq. (18)}$$

$$\bar{T}_{0AM} = \frac{t_{0,\mathrm{in}} - t_{\mathrm{Amb}}}{ln\left(\frac{T_{0,\mathrm{in}}}{T_{\mathrm{Amb}}}\right)} \qquad \text{Eq. (19)}$$

The proposed $\mathrm{COP_{CZ}}$ and $\mathrm{COP_{LZ}}$ as evaluation references therefore represent a theoretical limit for heat pump performance, which is consistent with both first- and second-law considerations and explicitly accounts the total potential of sensible waste heat sources. This means, those two COP approaches consider the maximum heat exploitation and temperature slide down to the limit of the ambient temperature. In contrast to conventional COP definitions, used as an evaluation reference, which are typically based on fixed source and sink temperatures reflect the thermodynamically optimal integration of a heat pump into a sensible waste heat source while using the maximum available heat source potential. However, COP_{CZ} is lower than COP_{AZL} but reflects the maximum yield of the waste heat source and addresses the temperature slide. These new comparison methods can be used for a fair and meaningful evaluation of heat pump applications for industrial waste heat. However, COP_{LZ} is lower as COP_{Lor} and COP_{CZ} is lower as COP_{Carnot} because the entropic mean source temperature is lower when considering the full potential up to ambient temperature in each case, but the maximum heat output of each heat pump application is much higher because the source power is higher, allowing more needed heat to be recovered from the same waste heat streams. Exergetically, it still makes sense, since all heat sources above ambient temperature level allow higher COPs compared to ambient heat pumps.

$COP_{AZL} = \frac{T_{H2}}{T_{H2}-\bar{T}_{0Ln}}$ $\qquad$ $COP_{CZ} = \frac{T_{H2}}{T_{H2}-\bar{T}_{0Am}}$

$COP_{AZL} = 5.6$ $\qquad$ $COP_{CZ} = 4.4$



$T_H = 423$ K

$\bar{T}_{0,ln} = \frac{t_{0,1}-t_{0,2}}{ln\left(\frac{T_{0,1}}{T_{0,2}}\right)} = 348$ K

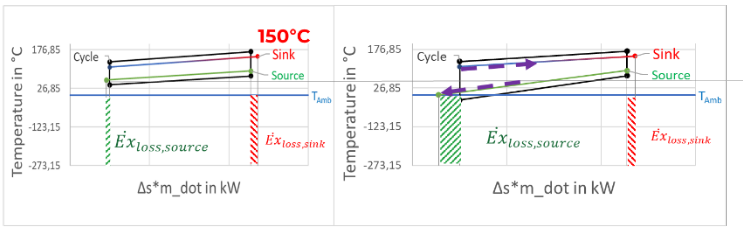
$T_H = 423$ K

$\bar{T}_{0AM} = \frac{t_{0,1}-t_{Amb}}{ln\left(\frac{T_{0,1}}{T_{AMB}}\right)} = 327$ K

Figure 6: Comparison of the Adler/Zotter/Längauer-Cycle and the LZ-Cycle (and COP) for sensible heat source (90 °C to 60 °C) and a latent heat sink (150 °C)

$COP_{Lor} = \frac{\bar{T}_{HLn}}{\bar{T}_{HLn}-\bar{T}_{0Ln}}$ $\qquad$ $COP_{LZ} = \frac{T_{H2}}{T_{H2}-\bar{T}_{0Am}}$

$COP_{Lor} = 7.3$ $\qquad$ $COP_{Car} = 4.7$



$\bar{T}_{H,ln} = \frac{t_{H,2}-t_{H,1}}{ln\left(\frac{T_{H,2}}{T_{H,1}}\right)} = 403$ K

$\bar{T}_{0,ln} = \frac{t_{0,1}-t_{0,2}}{ln\left(\frac{T_{0,1}}{T_{0,2}}\right)} = 348$ K

$T_H = 423$ K

$\bar{T}_{0AM} = \frac{t_{0,1}-t_{Amb}}{ln\left(\frac{T_{0,1}}{T_{AMB}}\right)} = 327$ K

Figure 7: Comparison of the Lorenz-Cycle and the LZ-Cycle (and COP) for sensible heat source (90 °C to 60 °C) and a sensible heat sink (110 °C to 150 °C)

6. Conclusion

From an overall energy point of view, the utilization of waste heat flows by HTHPs is an indispensable source of heat in order to reduce primary energy consumption and CO_2 emissions and to increase efficiency in energy supply. HTHPs can be used to lift the temperature level of the waste heat flows to a usable temperature level for industrial process heat or district heat supply. However, there are various industrial waste heat flows, many of which are sensible heat, such as hot air, hot water, exhaust gas streams, etc., and no latent heat flows such as condenser waste heat. Often, the potential of waste heat flows is only taken into account on the basis of their inlet temperature. The fact that there is a necessary temperature glide when using these sensible waste heat flows as source is mostly not considered. But sensible waste heat streams have to be cooled down when heat is extracted from it. For example, if a wastewater stream of 90 °C gets cooled down to 30 °C instead of 60 °C two times more heat can be extracted.

To evaluate the exergetic quality of an HTHP analysis, the COP according to Carnot is usually used as a common approach. In this work, it has been shown that the use of $\mathrm{COP_{Carnot}}$ is only an appropriate approach to evaluate applications with latent heat sources and sinks, but not for any other applications.

If the heat pump application requires a sensible heat source and a sensible heat sink, then the $\mathrm{COP_{Lor}}$ is much more accurate since the Lorenz cycle is better suited to this application as a reference cycle. For example, if hot water should provide with supply temperature of 150 °C and a return temperature of 110 °C by a heat pump using a sensible source of 90 °C and cooling it down to 60 °C $\mathrm{COP_{Carnot}}$ would be 4.7 and $\mathrm{COP_{Lor}}$ 7.3 for the same application. But, in practice the evaluation is often based on the simple Carnot approach using the sink supply temperature (sink outlet) and the source inlet temperature. This leads to an incorrect Carnot COP of 7.0, as is, for example.

However, if the heat pump application should convert a sensible heat source into a latent heat sink, which is usually the case when the waste heat flows are used for steam-generating heat pumps, then neither the Carnot nor the Lorenz cycle are suitable. In this case, only the Adler/Zotter/Längauer cycle reflects this application from a thermodynamic point of view and should be used as a reference cycle. For example, if steam should provide with at 150 °C by a heat pump using a sensible source of 90 °C and cooling it down to 60 °C, $\mathrm{COP_{Carnot}}$ would be 4.7, $\mathrm{COP_{Lor}}$ 4.3 but $\mathrm{COP_{AZL}}$ 5.6. By contrast, if the waste heat flows are to be used sensibly, the entire existing waste heat flow must be absorbed in its entirety, which means that this source must be cooled down to ambient temperature in the heat pump, which means if 90 °C gets cooled down to 0 °C, for example, instead of 60 °C three times more waste heat can be utilized within the heat pump. As a comparison cycle for these applications, new methods must be introduced, as shown in this work. For example, the $\mathrm{COP_{CZ}}$ will be introduced for latent sinks and sensible waste heat sources and the $\mathrm{COP_{LZ}}$ for sensible sinks and sensible waste heat sources, using the maximum waste heat streams (cooled down to ambient temperature). Both accurately reflect the sinks but also consider the full potential of the sources via the entropic mean temperature. This approach is important to harness the full potential of waste heat and to assess it correctly.

Nomenclature

	Description	Unit
AZL	Adler/Zotter/Längauer cycle	–
COP	coefficient of performance	–
c_p	(isobaric) specific heat capacity	$kJkg^{-1}K^{-1}$
DH	district heating	kW
$\dot{E}x$	exergy flux	kW
$\dot{E}x_{irr}$	exergy losses	kW
h	enthalpy	$kJkg^{-1}K^{-1}$
HTC	heat transfer coefficient	$Wm^{-2}K^{-1}$
HTHP	High temperature heat pump	
$\dot{m}$	mass flow rate	kgs^{-1}
$\dot{m}_{ref}$	refrigerant mass flow rate	kgs^{-1}
p	pressure	Pa
P_{el}	electrical power	kW
$\dot{Q}$	heat capacity, heat flux	kWm^{-2}
$\dot{Q}_{Sink}$	heat flux emitted from the HP to the heating circuit (sink)	kWm^{-2}
$\dot{Q}_{Source}$	heat flux absorbed to the HP (source)	kWm^{-2}
RHP	Rotational Heat Pump	
rJC	reverse Joule/Brayton cycle	
s	Specific entropy	$kJkg^{-1}K^{-1}$
SPF	seasonal performance factor	–
$\Delta\dot{S}$	Difference in entropy flux	$kJK^{-1}s^{-1}$
T, t	temperature	$K, °C$
t_{Amb}	ambient temperature	$°C$
t_{in}	inlet temperature	$°C$
t_{out}	outlet temperature	$°C$
$\overline{T}$	Entropic mean temperature	$°C$
ΔT_{PP}	pinch-point temperature difference	K
Δt	temperature spread, temperature glide	K

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