
David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)


# Proceedings of the 7th International Seminar on ORC Power Systems

Editorial Universidad de Sevilla

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# EXERGY-BASED METHODS FOR HEAT PUMPS

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## ABSTRACT

Heat pumps are undoubtedly a relevant technology option for decarbonizing the residential sector's heating supply and providing process steam in the industry. Nevertheless, thermodynamic analysis and evaluation of heat pumps needs to be more accurate.

Exergy analysis provides a method that detects the real thermodynamic inefficiencies of the components and the overall process. Determining the exergetic efficiency also provides a statement about the process quality and the maximum possible potential for process optimization. In addition, heat pump concepts with different inputs (compression heat pumps, absorption heat pumps) can be consistently compared with each other. The exergoeconomic analysis enables the identification of the cost-generation process and the determination of the heat production costs.

Thermal Engineering Systems in Python (TESPy) is a free and open-source software for the simulation and analysis of energy conversion systems like gas turbines, steam power plants, heat pumps, or refrigerators. Exergy balances are solved in postprocessing for all predefined components, and exergetic efficiencies are determined.

Simulation results and comparative exergy analyses are presented for different heat pump concepts. The paper is intended to establish the exergy-based methods for heat pump analysis. The results show which concepts and components are suitable for providing indoor heat and process steam from renewable energy sources.



## 1 INTRODUCTION

Why are we dealing with heat pumps in general? Reversing the heat engine process makes it possible to bring the thermal energy of a low-temperature level to a higher temperature level by using work. If we use renewable energy sources to provide the necessary power, it is possible to decarbonize a significant part of the final energy use (space heating, industrial process heat, district heating).

At present, technical developments can be observed, especially for high-temperature heat pumps. Within the framework of IEA Annex 58, an overview of manufacturers and product configurations can be viewed online [1]. Currently, heat pumps with a maximum thermal output in the upper tens of MW range can be purchased. According to the manufacturers, temperatures above 200 °C can already be achieved. However, ambitious projects with large temperature spreads or high temperatures often do not reach TRL 9, are demonstrators or custom-made products, and are frequently financed with public funding.

There are numerous contributions on heat pumps. At this point, only a selection of current publications, which deal with comparative thermodynamic analyses of heat pumps, is given. Adamsen et al. [2] and Jiang et al. [3] present comprehensive overviews for heat pump configuration for high-temperature applications and comparatively benchmark them. Arpagaus et al. [4] give a market overview for high-temperature heat pumps, present potential analyses, and coefficient of performance (COP) correlations depending on different process parameters. Schlosser et al. [5] focus on industrial applications, i.e., large-scale heat pumps, and present COP correlations. For a survey of exergy analyses for heat pumps, see Peng et al. [6] and Liu et al. [7].

To date, there is no consistent exergy-based consideration for heat pumps in an open-source framework with complete data availability. The paper aims to present and establish the conventional exergy analysis as an evaluation method for heat pumps. Analyses of different heat pump concepts show the influence of different process parameters and designs on the exergetic efficiency of individual components and the overall process.

The next section presents the Thermal Engineering Systems in Python (TESPy) library, a free and open source software for simulation and analysis of energy conversion systems. The basics of the exergy analysis for heat pumps are presented in Section 3 followed by an example for a simple heat pump. Section 5 is dedicated to the comparative exergy-based analysis of different heat pump concepts. The findings and further steps are summarized in Section 6.

## 2 THERMAL ENGINEERING SYSTEMS IN PYTHON

The open-source software Thermal Engineering Systems in Python (TESPy) [8] is used for the component-based simulation of the considered energy conversion processes. The software is developed as part of the open energy modeling framework (oemof) in the scientific field and by industrial users.[1] The library is written in Python and its structure is depicted in Fig. 1. The user can build up arbitrary thermodynamic processes from single predefined components like heat exchangers, compressors, valves, or turbines. In 554
addition to heat pumps, geothermal or solar thermal power plants, heat networks, or chillers can also be mapped. The software accesses the likewise open-source substance properties database CoolProp [9]. Detailed documentation with more examples and tutorials is available for the software.[2]

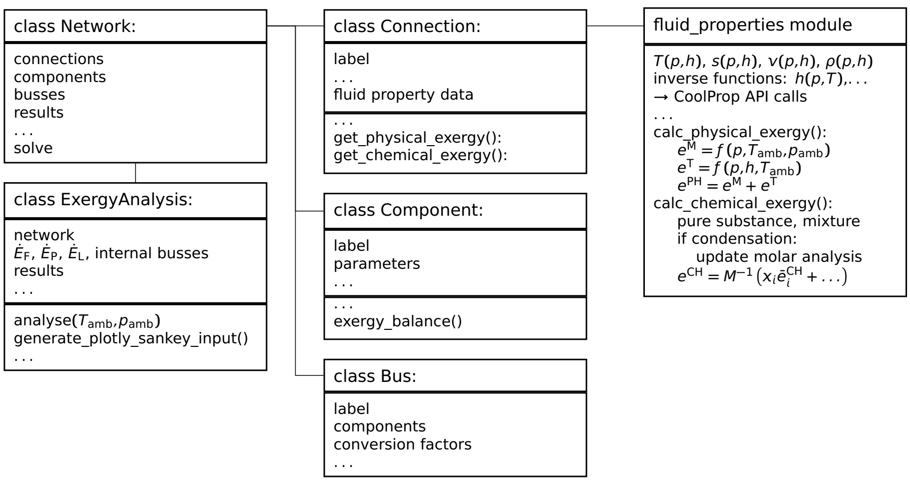


**Figure 1:** Simplified structure of TESPy

Following process simulation and design, exergy analysis can be used to identify sources of thermodynamic losses. The exergy analysis is performed automatically. Only the exergetic fuel, product, and loss of the overall process have to be passed to the tool [10, 11]. Furthermore, the simulation models can be combined with optimization methods [12].

1 https://oemof.org
2 https://tespy.rtfd.io

## 3 EXERGY ANALYSIS FOR HEAT PUMPS

A conventional thermodynamic analysis based on the evaluation of mass and energy balances fails in determining the real thermodynamic losses of a system. Consequently, the efficient use of energy sources is not guaranteed.

The misjudgment based on the coefficient of performance (COP) is not only that the impression is given that more energy is supplied than required. Moreover, the theoretical improvement potential of a thermodynamic optimization can not be calculated due to the lack of an inherent upper limit. With the help of the COP itself, no statement can be made about how far the chosen process configuration is from the thermodynamic optimum. An additional performance indicator, e.g. Lorentz COP [13], has to be calculated. In the case of the exergetic efficiency with the limits 0 and 1, this already contains the information sought.

In addition, exergy-based analysis enables the correct evaluation and comparison of processes with more than one product or fuel streams. With reference to heat pumps, Franzke [14] states that only on the basis of exergy a "flawless comparison" of compression heat pumps and absorption heat pumps is possible. In the same way, the correct consideration of the products in a cogeneration process for the simultaneous provision of electricity and heat succeeds only based on exergy [15, 16].

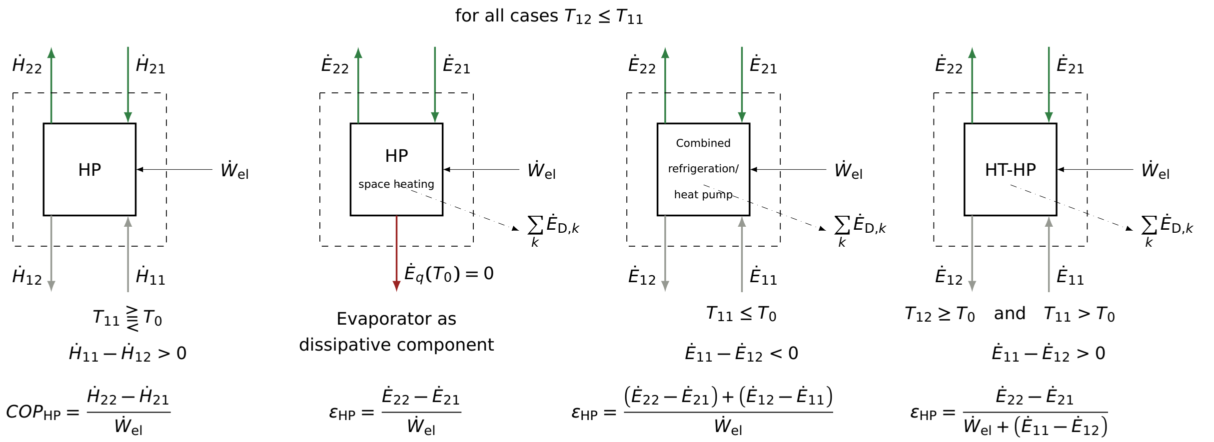


**Figure 2:** Steady-state energy balance and exergy balances of the heat pump. The different use cases must be distinguished to define the exergetic efficiency consistently. For space heating, the heat pump usually emits the exergy flow rate associated with the supplied heat at $T_0$.

Figure 2 contrasts the steady-state energy balance with the exergy balance. Following the definition of exergy (zero point at ambient conditions), evaluating the balance must consider whether the process takes place entirely above or partially below the ambient temperature. A heat rate is supplied to the heat pump at the lower temperature level. The enthalpy difference of the streams 11 and 12 is therefore always greater than 0, independent of $T_{11}$. The exergy difference of streams 11 and 12 is negative for the case $T_{11}$ is lower than or equal to the ambient temperature. This means that a heat rate is energetically supplied to the overall process, but exergetically the exergy rate associated with the heat rate is transferred to the environment or is used as additional exergetic product; the heat pump thus acts like a refrigerating machine at the same time.

In addition to the electrical power required for the motor, the exergetic efficiency of the heat pump also includes the thermal power supplied to the process at the lower temperature level. It must be addressed and is essential, for example, when integrating industrial waste heat for high-temperature heat pumps. It also results in the advantage that the exergetic efficiency has an inherent upper limit compared to the coefficient of performance. It has a maximum value of 1.

# 4 SIMPLE HEAT PUMP EXERGY ANALYSIS

## 4.1 Thermodynamic Model

A simple thermodynamic model of a heat pump is used to demonstrate the exergy-based analysis. The flow diagram of the process can be found in Fig. 3. The heat pump is used to supply a heating circuit and is characterized by the parameters listed in Table 1. In addition, we assume that the process is steady-state, all components are adiabatic, there are no pressure losses in the heat exchangers, and changes in kinetic and potential energies can be neglected.

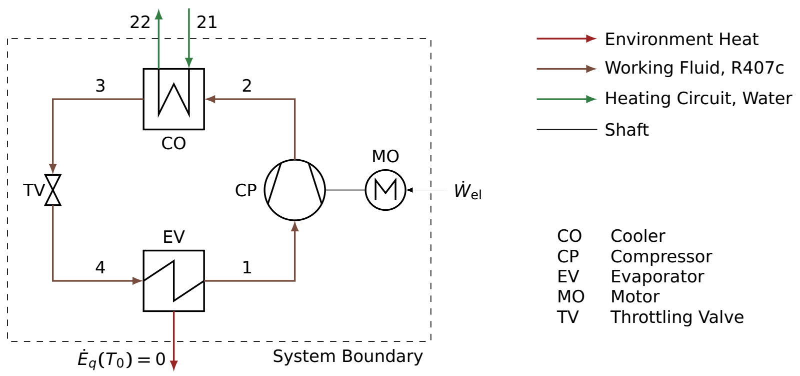


**Figure 3:** Thermodynamic model of a heat pump for the supply of a heating circuit

**Table 1:** Process parameters of the heat pump and ambient conditions

| Parameter | Symbol | Unit | Value |
|---|---|---|---|
| Heat rate | $\dot{H}_{22} - \dot{H}_{21}$ | kW | 20 |
| Lower pressure level | $p_1$ | bar | 5 |
| Upper pressure level | $p_2$ | bar | 20 |
| Superheating at the evaporator outlet | $T_1$ | °C | $T_{sat}(p_1) + 2\,K$ |
| Cooler outlet temperature | $T_3$ | °C | 40 |
| Isentropic efficiency of compressor | $\eta_s$ | % | 75 |
| Electric-mechanical efficiency of motor | $\eta_{el,mech}$ | % | 95 |
| Ambient temperature | $T_0$ | °C | 15 |
| Ambient pressure | $p_0$ | bar | 1.013 |

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The mass and energy balances are solved for the selected process parameters using TESPy, see Section 2. The model, input data, and all results can be found in a git repository.[3] The exergy values can be calculated for all material and energy flows. Using the exergy balance, the exergy destruction of the components is determined. The definitions of the exergetic efficiencies of the components are noted in Eqs. (1) to (5).

## 4.2 Exergy Analysis of Components

Electricity is supplied to the motor via the system boundary to provide mechanical power. The latter consequently represents the exergetic product of the component.

$$\varepsilon_{MO} = \frac{\dot{W}_{mech}}{\dot{W}_{el}} \tag{1}$$

Mechanical power is used to compress from state 1 to 2. However, stream 1 is supplied at a temperature below the ambient temperature. Thus, to define the exergetic efficiency, it is necessary to divide the exergy rates into the thermal and mechanical parts [17]. The thermal part of the physical exergy of stream 1 must

3 https://github.com/tub-hofmann/heat-pumps-tespy

be considered as an exergetic fuel in addition to the necessary mechanical power. The increase of the mechanical part of the physical exergy and the thermal part of the physical exergy of stream 2 represents the exergetic product of the compressor.

$$\varepsilon_{\mathrm{CP}} = \frac{\dot{E}_2^{\mathrm{T}} + \left(\dot{E}_2^{\mathrm{M}} - \dot{E}_1^{\mathrm{M}}\right)}{\dot{W}_{\mathrm{mech}} + \dot{E}_1^{\mathrm{T}}} \tag{2}$$

The cooler connects the process with the heating circuit. If the heat capacity flows of the hot and cold sides are not exactly superimposed (infinitely large heat exchanger), any heat transfer is associated with exergy destruction. The exergetic product is the temperature rise in the heating circuit. The exergetic fuel is the exergy difference on the hot side of the heat exchanger.

$$\varepsilon_{\mathrm{CO}} = \frac{\dot{E}_{22} - \dot{E}_{21}}{\dot{E}_2 - \dot{E}_3} \tag{3}$$

If the throttling valve operates entirely above the ambient temperature $T_0$, it can be considered as a dissipative component. In the present case, $T_0$ is surpassed during expansion. It is necessary to divide the exergy flows into the thermal and mechanical parts. The thermal part of the physical exergy below ambient temperature represents the exergetic product of the throttling valve [10]. The pressure loss, i.e., the change of the mechanical part of the physical exergy and the thermal part of the physical exergy above the ambient temperature, is considered as exergetic fuel.

$$\varepsilon_{\mathrm{TV}} = \frac{\dot{E}_4^{\mathrm{T}}}{\dot{E}_3^{\mathrm{T}} + \left(\dot{E}_3^{\mathrm{M}} - \dot{E}_4^{\mathrm{M}}\right)} \tag{4}$$

In the example presented here, the evaporator operates completely below the ambient temperature $T_0$. If we assume that the exergy flow rate $\dot{E}_{\mathrm{q}}$ associated with the heat flow supplied from the environment surpasses the system boundary at $T_0$, $\dot{E}_{\mathrm{q}}$ results in zero. Consequently, no exergetic product can be defined for the evaporator. The evaporator must be considered as a dissipative component. 557

$$\varepsilon_{\mathrm{EV}} = \mathrm{n/a} \tag{5}$$

### 4.3 Exergy Analysis of Overall Process

For the exergy analysis of the overall process, the steady-state exergy balance can be used. Simultaneously, the resulting information of the exergy destruction of the overall process can be compared with the sum of the exergy destruction within the components. Thus, it is easy to ensure that the analysis has been performed correctly.

$$\dot{E}_{\mathrm{D}}^{\mathrm{TOT}} = \sum_k \dot{E}_{\mathrm{D},k} = \dot{W}_{\mathrm{el}} + \dot{E}_{21} - \dot{E}_{22} \qquad \text{with} \quad \dot{E}_{\mathrm{q}} = 0 \tag{6}$$

### 4.4 Results

Table 2 summarizes the results of the exergy analysis for the components and the overall process. For the components the exergy destruction ratio can be calculated.

$$y_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F,tot}}} \tag{7}$$

The highest exergy destruction occurs in the compressor. It is followed by the evaporator, the cooler, and the throttling valve. The exergetic efficiency of the overall process illustrates that the predominant part of the electrical power is required to overcome the thermodynamic inefficiencies of the components.

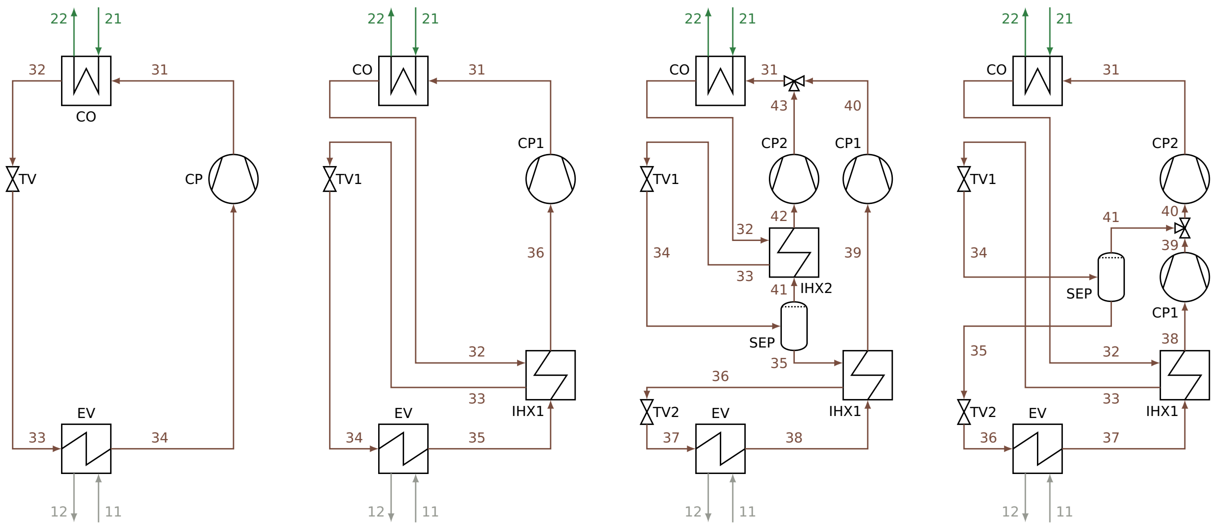
**Table 2:** Exergy-based analysis, overall process and component results

| | R407c, $\dot{m}_1$ = 0.101 kg/s<br>$\dot{E}_{\mathrm{D}}^{\mathrm{TOT}}$ = 3.1 kW | |
|---|---|---|
| Component $k$ | $\varepsilon_k$ [%] | $y_{\mathrm{D},k}$ [%] |
| Compressor, CP | 79.1 | 19.9 |
| Cooler, CO | 80.6 | 12.1 |
| Throttling valve, TV | 60.9 | 8.6 |
| Evaporator, EV | diss. | 18.4 |
| Heat pump, HP | 35.9 | |

## 5 COMPARATIVE EXERGY-BASED ASSESSMENT OF HEAT PUMPS

This section analyzes different heat pump process designs, particularly for district heating and industrial process steam use cases. Figure 4 shows the various schematics: (a) simple cycle, (b) cycle with internal heat exchanger (IHX), (c) cycle with IHX and parallel compression (PC), and (d) cycle with IHX and compressor with intercooler (IC). For all heat pump designs the parameters are taken from Table 3. In addition, we assume that the processes are steady-state, all components are adiabatic, there are no pressure losses in the heat exchangers, and changes in kinetic and potential energies can be neglected. The different topologies are analyzed with the application of two natural and two synthetic working fluids: R600, R601, R1233zd(E), and R1336mzz(Z). Unfortunately, the CoolProp implementation of R1336mzz(Z) is lacking and bugs occur in the usage of R601 and R1233zd(E)[4], which makes the utilization of the closed source fluid property database REFPROP [18] necessary. The model, input data, and all results can be found in a git repository[5]. The results of the process simulations are analyzed according to the method shown above.






**(a)** Simple cycle **(b)** With IHX **(c)** With IHX and PC **(d)** With IHX and IC

**Figure 4:** Heat pump process designs

According to the contributions of Sulzgruber et al. [19, 20], the optimal high pressure of the different processes is identified. With the application of the exergy-based method, we extend their evaluation. The results of the energy analysis and the exergy analysis are compared.

The results are shown in Fig. 5a for the topology with IHX for all working fluids. The shapes of the R600, R1233zd(E), and R1336mz(Z) curves are similar to those of Sulzgruber et al., with a slight deviation of

4 https://github.com/CoolProp/CoolProp/issues/2070
5 https://github.com/tub-hofmann/heat-pumps-tespy

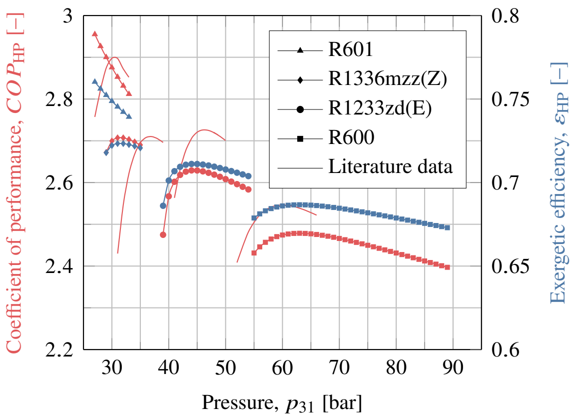
**Table 3:** Process parameters of the heat pumps and ambient conditions

| Parameter | Symbol | Unit | Value |
|---|---|---|---|
| Heat rate | $\dot{H}_{22} - \dot{H}_{21}$ | MW | 1 |
| Heat sink temperatures | $T_{21}; T_{22}$ | °C | 160; 200 |
| Heat sink pressure | $p_{21}$ | bar | 20 |
| Heat source temperatures | $T_{11}; T_{12}$ | °C | 95; 90 |
| Heat source pressure | $p_{11}$ | bar | 5 |
| Lower terminal temperature difference of cooler | $\Delta T_{\mathrm{l,CO}}$ | K | 10 |
| Lower terminal temperature difference of evaporator | $\Delta T_{\mathrm{l,EV}}$ | K | 5 |
| Upper terminal temperature difference of IHX | $\Delta T_{\mathrm{u,IHX}}$ | K | 15 |
| Isentropic efficiency of compressor | $\eta_s$ | % | 76 |
| Electric-mechanical efficiency of motor | $\eta_{\mathrm{el,mech}}$ | % | 95 |
| Ambient temperature | $T_0$ | °C | 15 |
| Ambient pressure | $p_0$ | bar | 1.013 |

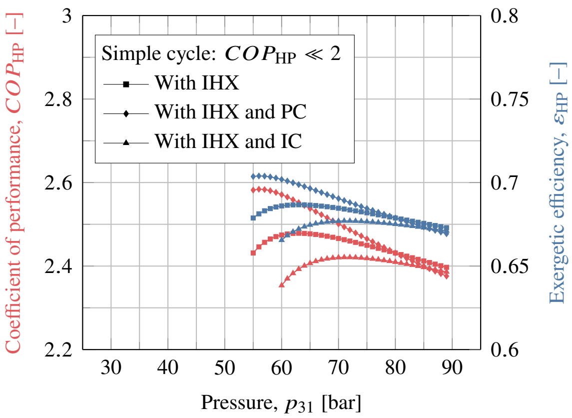
COP depending on working fluid. The models used by Sulzgruber et al. are private. Therefore, parameters, assumptions, and used property functions cannot be fully reproduced. Consequently, deviations due to different input data cannot be excluded. In contrast to Sulzgruber et al., the results plot for R601 shows no local maximum. The absent maximum at decreasing high pressure is likely due to partial subcritical cooling and heat exchanger modeling.

As can be seen from Fig. 5a, the qualitative characteristics of energetic and exergetic analysis are identical. But the exergy analysis can reveal thermodynamic improvement potential more easily. From the maximum value of the exergetic efficiency, 68.7%, of the IHX cycle using R600, it can be concluded directly that 31.3% must be used to overcome the thermodynamic inefficiencies of the components, and represent the quantitative improvement potential of the process.

Figure 5b shows the results for different topologies using the working fluid R600. The values for COP and exergetic efficiency of the simple cycle are lower than for all other process designs by a discernible margin and not depicted in the diagram. The underlying reason for the low values is the low enthalpy difference in the cooler of the simple cycle, as the isentropes shift inwards towards the critical point close to the saturated vapor line. Therefore, any topology improvement leading to superheating after the evaporator will almost double the exergetic efficiency. The highest efficiency is achieved with IHX and PC, see Fig. 4c, followed by the process with IHX, see Fig. 4b and finally, the design with IHX and IC, see Fig. 4d. Generally, lower high-pressure performance maxima correspond with a higher absolute COP and exergetic efficiency, analogous to the results in Fig. 5a. 559




**(a)** IHX with each working fluid




**(b)** R600 with each topology

**Figure 5:** Coefficient of performance and exergetic efficiency for varying high pressure. For Fig. 5a in comparison with literature data from Sulzgruber et al. [19, 20]

The exergy analysis is featured by the possibility of a component-specific assessment. The results are shown in Fig. 6 for the analysis of the process design with IHX using R600 as working fluid. The exergy destruction ratio of each component, defined in Eq. (7), is calculated and plotted for varying high pressure within the cooler. The evaporator, valve, and compressor do not show a significant change in their respective exergy destruction ratio, with only a slight decrease visible in the quantitative results. On the other hand, the exergy destruction rate of the cooler increases with rising high pressure. The reason for this is the rising exit temperature of the compressor with higher pressure which in consequence increases the mean logarithmic temperature. In contrast, the exergy destruction ratio of the internal heat exchanger decreases with higher pressure. The combination of these considerable influences results in the minimum exergy destruction ratio occurring at 63 bar.

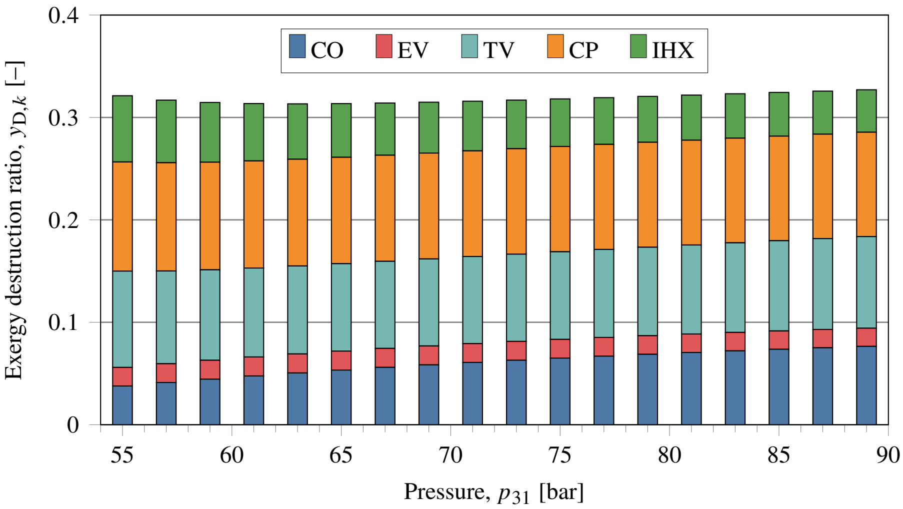



**Figure 6:** Exergy destruction ratios of the components in IHX with R600

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## 6 CONCLUSION

Exergy analysis detects the real thermodynamic inefficiencies of the components and the overall process, which makes it advantageous compared to energy-based evaluation for evaluating and optimizing thermodynamic processes such as heat pumps. Exergy-based methods, especially for heat pumps, are presented within the framework of this contribution. The analysis results have shown that different operating optima are derived from different working fluids and topologies. The influences of the individual system components are presented.

A generic heat pump model would be suitable for a broader analysis of topologies, which requires stable simulations for different working fluids with different process parameters and boundary conditions. For this reason, the co-authors Malte Fritz and Jonas Freißmann are currently developing a framework with a graphical user interface, which will be extended in cooperation with Technische Universität Berlin.

As academic research flourishes with cooperation and knowledge exchange, it is regrettable that using closed-source fluid property databases is necessary for some working fluids or cases. Future research should consider this point as a precondition.

## CREDIT AUTHOR STATEMENT

**Mathias Hofmann:** Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Resources, Data Curation, Writing - Original Draft, Writing - Review & Editing, Visualization, Supervision **Jonas Freißmann, Malte Fritz (both equally):** Software, Validation, Formal analysis, Data Curation, Writing - Original Draft, Writing - Review & Editing, Visualization **Jeremias H. Alexe:** Software, Validation, Formal analysis **Francesco Witte:** Conceptualization, Methodology, Software, Validation, Formal analysis, Data Curation, Writing - Review & Editing **George Tsatsaronis:** Review & Editing, Supervision

**NOMENCLATURE**

### Abbreviations

| | |
|---|---|
| CO | Cooler |
| CP | Compressor |
| EV | Evaporator |
| HP | Heat pump |
| IHX | Internal heat exchanger |
| MO | Motor |
| SEP | Separator |
| TV | Throttling valve |

### Latin Symbols

| | |
|---|---|
| $e$ | Mass specific exergy, J/kg |
| $\bar{e}$ | Molar specific exergy, mol/kg |
| $\dot{E}$ | Exergy rate, W |
| $h$ | Specific enthalpy, J/kg |
| $\dot{H}$ | Enthalpy rate, W |
| $\dot{m}$ | Mass flow rate, kg/s |
| $M$ | Molar mass, kg/mol |
| $p$ | Pressure, bar |
| $\dot{Q}$ | Heat rate, W |
| $s$ | Specific entropy, J/kg K |
| $T$ | Temperature, °C |
| $\dot{W}$ | Work rate, W |
| $x_i$ | Molar fraction, – |
| $y$ | Exergy ratio, – |



### Greek Symbols

| | |
|---|---|
| $\Delta$ | Difference, – |
| $\varepsilon$ | Exergetic efficiency, – |
| $\eta$ | Energetic efficiency, – |

### Superscripts and Subscripts

| | |
|---|---|
| 0 | At ambient state |
| amb | At ambient state |
| CH | Chemical |
| D | Destruction |
| el | Electric |
| F | Fuel |
| i | Stream |
| in | At inlet |
| k | Component |
| KN | Kinetic |
| L | Loss |
| mech | Mechanic |
| M | Mechanical |
| out | At outlet |
| P | Product |
| PH | Physical |
| PT | Potential |
| s | Isentropic |
| tot | Total |
| T | Thermal |

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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