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# Thermodynamic and economic performance of novel organic cycle designs powered by low-temperature waste heat

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**Abstract:**
Waste heat recovery is one of the alternative energy sources that have been investigated by scientists in recent decades to address ongoing environmental problems, such as global warming. The organic Rankine cycle is a promising waste heat recovery technology to exploit industrial waste heat, even at low-temperature levels (<100$^{o}$C), for electricity production. The proposed study presents and compares two innovative organic cycle designs feeding with the same available heat source. The first cycle includes a nearly isothermal expander, where a small fraction of the supplied waste heat is continuously provided to the expander, aiming to approach a quasi-isothermal process instead of an adiabatic one, avoiding the temperature decrease due to the expansion process. The result is an increase in the cycle's thermal efficiency and greater power output production compared to the adiabatic expansion process. The second configuration is called the trilateral flash cycle, where the working fluid does not reach the saturated vapor state during heating at the heat recovery system, while it expands into the two-phase region of the fluid. The aforementioned cycles are investigated parametrically in terms of thermodynamics with a low-temperature heat source (80-100$^{o}$C) for different organic working fluids, such as the R1234ze(E), R1234yf, R1233zd(E), and R134a. Parametric studies are carried out through Aspen Plus software, while a techno-economic comparison of the organic cycle designs is conducted based on Aspen Process Economic Analyzer and literature data. According to the final results, R1233zd(E) seems to be the most proper working fluid thermodynamically, while the organic Rankine cycle with nearly isothermal expansion achieves higher values of both electrical and exergy efficiencies, reaching the maximum values of 10.51%, and 52.27%, respectively. In terms of finance, both cycles achieve similar payback period values, reaching the value of 1.56 years in the case of the trilateral flash cycle and assuming 8,000 operating hours per year. Finally, for the trilateral flash cycle, lower net present value levels of about 30% compared to the corresponding values for the other cycle, are determined despite its lower installation cost.

**Keywords:**
Trilateral Flash Cycle, ORC, isothermal expander, thermodynamic analysis, cost analysis, low-grade WHR.

## 1. Introduction

In recent decades, the increasing energy demand, economic, and population growth, as well as the high penetration of fossil fuels into the energy sector have led to significant environmental problems, such as air pollution, and global warming. Thus, the international community and scientists worldwide have promoted the utilization of alternative energy sources, such as the recovery of waste heat. In the industrial sector, considerable amounts of heat in various temperature levels depending on the processes, are rejected to the ambient, which can be recovered and further exploited to produce electricity or heating [1]. One of the most commercially available low-grade waste heat recovery technologies is the Organic Rankine Cycle (ORC). This cycle has a similar structure to the water-steam Rankine cycle but employs organic fluids with low saturation temperature levels. Thus, ORC can exploit low-temperature heat sources, even below 100$^{o}$C to produce electricity, while this kind of installation is highly reliable and easy to maintain [2].


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Many researchers around the globe have focused on the performance of ORC modules. Indicatively, Eyerer et al. [3] analyzed experimentally the utilization of substances with low Global Warming Potential (GWP), such as R1233zd(E) and R1224yd(Z), instead of conventional fluids, such as R245fa. According to the final results, the maximum power output was achieved, when R245fa was used, which was 9%, and 12% higher than that of R1233zd(E), and R1224yd(Z), respectively. However, in the case of using the eco-friendly medium R1233zd(E), the thermal efficiency was enhanced by 2% compared to the other two fluid options. In the past few decades, the integration of the nearly isothermal expansion process in the ORC has been investigated. During the expansion process, heat is transferred to the organic medium maintaining the temperature at a higher level, compared to the conventional adiabatic process. Thus, the power output and thermal efficiency are expected to be enhanced [4]. Indicatively, Ziviani et al. [5] studied an ORC unit with liquid-flooded expansion and internal regeneration for different working media. In that case, a secondary fluid was ejected at the expander inlet being in thermal equilibrium with the organic fluid, to limit the temperature reduction due to the expansion procedure. So, the net electricity production and cycle efficiency could be increased by 20%, reaching up to 50% of the Carnot efficiency. Moreover, Kosowski and Piwowarski [6] performed the thermodynamic analysis of both a conventional ORC and an ORC with the ideal isothermal process. The incorporation of the isothermal process led to an increased cycle efficiency of up to 12% when the organic fluid reached the saturated vapor state, and up to 7% when the organic fluid reached the superheated vapor state.

In addition, another design has also gained attention and is called Trilateral Flash Cycle (TFC), which is a cycle similar to ORC. In TFC application, the working fluid reaches the saturated liquid state without evaporation and expands into the two-phase region. More specifically, Iqbal et al. [7] investigated in terms of energy a TFC system in comparison with a conventional ORC, considering a low-temperature heat source, up to 100$^{o}$C. According to this study, TFC could produce at least 50% further useful electricity compared to the conventional ORC for the same heat source and heat sink conditions. Ajimotokan [8] analyzed energetically and exergetically four TFC configurations, i.e. the simple TFC, the recuperated TFC, the reheat TFC, and the regenerative TFC. The results indicated that the aforementioned cycles could achieve thermal efficiencies of 21.97%, 23.91%, 22.07%, and 22.9%, respectively if the highest temperature of the cycle was equal to 473 K, and n-pentane was utilized as the working medium.

In parallel, many publications are concentrated on the comparison of different cycle designs. First, Zhar et al. [9] studied thermodynamically and economically three ORC configurations, i.e. the basic ORC, the reheat ORC, and the regenerative ORC, for different working fluids. According to the final outcomes, the regenerative ORC performed better energetically and exergetically, as the energy efficiency was enhanced by 13%, and the exergy destruction was reduced by 44%, contrasted to the basic ORC installation. From an economic point of view, all the configurations achieved similar payback periods and levelized cost of energy values. Furthermore, Kanno and Shikazono [10] carried out a thermodynamic comparison between a Rankine cycle, a TFC, and a supercritical cycle, taking into consideration different working media. Two different heat source cases were examined i.e. exhaust gas at a temperature level of about 400$^{o}$C, and hot water at a temperature level of about 80$^{o}$C. Assuming the 80$^{o}$C hot water as the heat source, the maximum exergy efficiency was determined based on the sink temperature for the TFC unit, using hydro-fluorocarbon refrigerants. Finally, for the 400$^{o}$C exhaust gas case, the ethanol supercritical cycle achieved the highest sink-temperature-based exergy efficiency.

According to the previous literature review, scientists are interested in the investigation of innovative organic cycle designs, such as the ORC with nearly isothermal expansion (ORC-NIE) or the TFC. Nevertheless, most of the aforementioned studies focus on a single design or compare these designs with other ones. There is a lack of research articles that compare these two configurations in terms of energy, exergy, and economics. So, the present work investigates the thermodynamic performance of both ORC-NIE and TFC, taking into account different working fluids. The heat source of both cycles is hot water with a temperature of up to 100$^{o}$C. Additionally, the two cycles are analyzed financially. The thermodynamic simulations are carried out in Aspen Plus software, while the cost estimation analysis is based on Aspen Process Economic Analyzer, and data from the literature studies [11,12].

# 2. Organic cycle designs' fundamentals

## 2.1. Organic Rankine cycle with nearly isothermal expansion

The proposed configuration includes the basic ORC devices, i.e. the pump, the heat recovery system, and the condenser. It is assumed that the organic fluid exits the heat recovery system as superheated steam, with a superheating level of 10 K. The subcooling level at the condenser is considered at 2 K. Additionally, the innovative element of this design, which is the incorporation of the nearly isothermal expansion process, has been decided to be modeled as a two-stage expansion, with intermediate additional heat input ($Q_{heater}$). It is also assumed that the fluid exits the first stage at an intermediate pressure level, then is heated up reaching the same temperature level as the one of the superheated steam at the heat recovery system outlet, and, finally, expands up to the condenser saturation pressure in the second stage. The intermediate pressure level ($P_{med}$) is defined taking into account the high ($P_{high}$) and the low-pressure level ($P_{low}$) as [13]:

$$P_{med} = \sqrt{P_{high} \cdot P_{low}}, \tag{1}$$

The ORC high pressure ($P_{high}$) is determined by taking into consideration a temperature difference of 10 K between the heat source inlet temperature ($T_{w,in}$) and the fluid temperature at the outlet of the heat recovery system. The main outputs of the ORC-NIE, which are the net electricity production ($P_{el,net,ORC-NIE}$), the electrical efficiency ($\eta_{el,ORC-NIE}$), and the exergy efficiency ($\eta_{ex,ORC-NIE}$), are described by the following expressions:

$$P_{el,net,ORC-NIE} = P_{el,exp,ORC-NIE} - P_{el,pump,ORC-NIE}, \tag{2}$$

$$\eta_{el,ORC-NIE} = \frac{P_{el,net,ORC-NIE}}{Q_{HRS,ORC-NIE}+Q_{heater}}, \tag{3}$$

$$\eta_{ex,ORC-NIE} = \frac{P_{el,net,ORC-NIE}}{E_{HRS,ORC-NIE}+E_{heater}}, \tag{4}$$

The exergy rate ($E$) due to a heat rate ($Q$) at a temperature level of ($T$), considering the reference temperature ($T_0$) of 25$^{o}$C (298.15 K) can be defined as [14]:

$$E = Q \cdot \left(1 - \frac{T_0}{T}\right), \tag{5}$$

The model of the ORC-NIE in Aspen Plus is depicted in Figure 1.

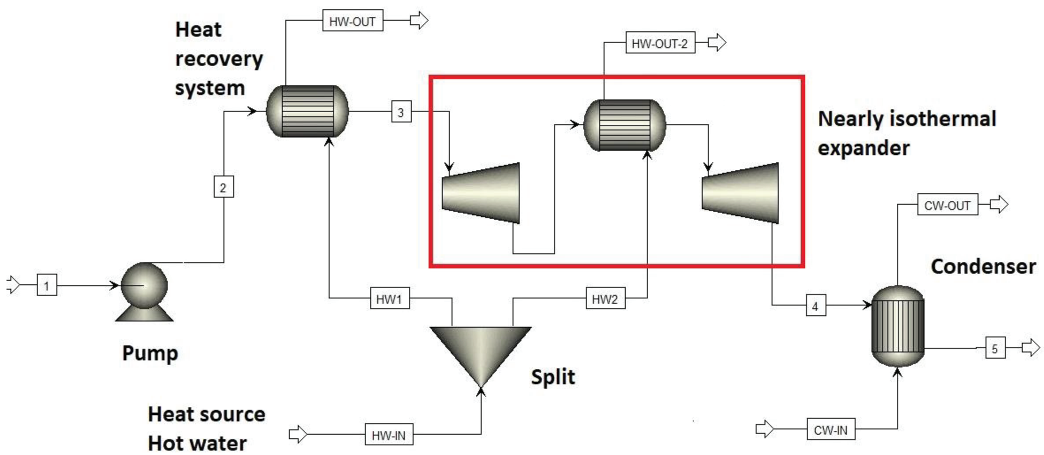


**Figure. 1.** Model of the ORC-NIE in Aspen Plus.

## 2.2. Trilateral flash cycle

The other examined design includes all the basic ORC devices, i.e. the pump, the heat recovery system, the expander, and the condenser. The main difference between the basic ORC module and the proposed one (TFC) is that the working medium exits the heat recovery system in the state of saturated liquid, without any evaporation or superheating. Subsequently, the fluid expands into the two-phase region of the substance. Additionally, the subcooling level at the condenser is considered at 2 K. The ORC high pressure ($P_{high}$) is determined by taking into consideration a temperature difference of 10 K between the heat source inlet temperature ($T_{w,in}$) and the fluid temperature at the outlet of the heat recovery system. The main outputs of the TFC, which are the net electricity production ($P_{el,net,TFC}$), the electrical efficiency ($\eta_{el,TFC}$), and the exergy efficiency ($\eta_{ex,TFC}$), are described by the following equations:

$$P_{el,net,TFC} = P_{el,exp,TFC} - P_{el,pump,TFC}, \tag{6}$$

$$\eta_{el,TFC} = \frac{P_{el,net,TFC}}{Q_{HRS,TFC}}, \tag{7}$$

$$\eta_{ex,TFC} = \frac{P_{el,net,TFC}}{E_{HRS,TFC}}, \tag{8}$$

The model of the TFC in Aspen Plus is depicted in Figure 2.


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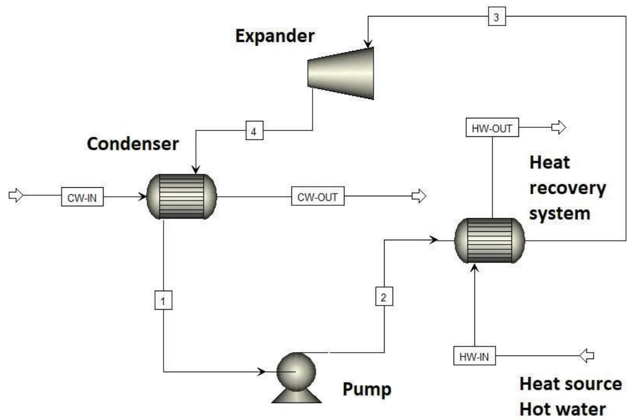


**Figure. 2.** Model of the TFC in Aspen Plus.

## 2.3. Techno-economic analysis

First, the investment cost (*IC*) of each cycle is defined taking into account the values that come from Aspen Process Economic Analyzer, and the literature studies [11,12]. In addition, the annual cash flow (*CF*) is defined considering the annual inflows and outflows. The annual inflows are consisted of the revenues from electricity selling, while the annual outflows include the operation & maintenance costs ($K_{O\&M}$). Taking into account the annual electricity production in kWh ($Y_{el}$), the operating hours per year (*hours*), and the electricity selling price in €/kWh ($K_{el}$), the annual cashflow is described by the following expression:

$$CF = Y_{el} \cdot K_{el} - K_{O\&M} = P_{el,net} \cdot hours \cdot K_{el} - K_{O\&M} \tag{9}$$

Then, the major financial indexes are defined. First, the payback period (*PBP*) is calculated as:

$$PBP = \frac{ln\left(\frac{CF}{CF - IC \cdot i}\right)}{ln\,(1+i)} \tag{10}$$

The net present value (*NPV*) is calculated as:

$$NPV = -IC + CF \cdot \frac{(1+i)^N - 1}{i \cdot (1+i)^N} \tag{11}$$

The aforementioned financial parameters are presented in Table 1.

**Table 1.** Parameters of financial analysis

| Parameters | Values |
|---|---|
| Electricity selling price ($K_{el}$) | 0.2 €/kWh |
| Project lifetime (*N*) | 20 years |
| Discount factor (*i*) | 4% |
| Operation & maintenance cost ($K_{O\&M}$) | 2% of the investment cost |

## 2.4. Simulation methodology

First, the aforementioned cycles are studied parametrically in terms of thermodynamics in steady-state conditions through the developed models in Aspen Plus. More specifically, three main parameters are examined, the heat source inlet temperature ($T_{w,in}$), which strongly affects the cycle's high pressure ($P_{high}$), the condenser saturation temperature ($T_{cond}$), which strongly affects the cycle's low pressure ($P_{low}$), and the expander isentropic efficiency ($\eta_{is,exp}$). These values vary into a specific range to investigate their influence on the energetic and exergetic performance of the system. The heat source inlet temperature ($T_{w,in}$) ranges from 80 to 100$^o$C, with a default value of 100$^o$C, the condenser saturation temperature ($T_{cond}$), ranges from 10 to 40 $^o$C, with a default value of 30$^o$C, and the expander isentropic efficiency ($\eta_{is,exp}$) ranges from 0.4 to 0.8, with a default value of 0.7. Other parameters that remain constant during this analysis for both cycles are presented in Table 2. In parallel, the systems' operation is also examined for four different working fluids, which are depicted in Table 3. REFPROP is utilized as a proper method for refrigerants' properties [15]. Three of them (R1234ze(E), R1234yf, R1233zd(E)) are eco-friendly media with low GWP values, while R134a is a conventional refrigerant with a high level of GWP. All these fluids have zero Ozone Depletion Potential (ODP) and are selected as they have a critical temperature close to the examined heat source temperature levels. Then, the 2 designs are investigated financially. For this study, different values of operating hours per year are assumed. All the defined thermodynamic and financial indexes ($\eta_{en}$, $\eta_{ex}$, *PBP, NPV*) are used to compare the aforementioned two cycles and specify the most techno-economically viable one when the two cycles are fed with the same heat source. Finally, it is important to mention that the two cycles are decided to be compared for similar heat input rates. That's why, a constant temperature difference of 20K and a constant flow rate is considered for the hot water stream.

**Table 2.** Constant parameters of both cycles.

| Parameters | Values |
|---|---|
| Pump isentropic efficiency | 0.7 |
| Pump driver efficiency | 0.93 |
| Expander mechanical efficiency | 0.95 |
| Heat source (hot water) volume flow rate | 75.5 $m^3$/h |
| Heat source (hot water) pressure | 3 bar |
| Heat source (hot water) temperature difference | 20 K |

**Table 3.** Examined working fluids [14,16].

| Working fluid | Flammability & Toxicity | $P_{crit}$ (bar) | $T_{crit}$ ($^oC$) | ODP | GWP |
|---|---|---|---|---|---|
| R1234ze(E) | No | 36.35 | 109.26 | 0 | 7 |
| R1234yf | No | 33.82 | 94.70 | 0 | 4 |
| R1233zd(E) | No | 36.24 | 166.45 | 0 | 4.5 |
| R134a | No | 40.56 | 101.03 | 0 | 1320 |

# 3. Results and discussion

## 3.1. Thermodynamic analysis

In this section, the results of the thermodynamic analysis are presented. More specifically, the influence of the three aforementioned parameters, i.e. the heat source inlet temperature the condenser saturation temperature, and the expander isentropic efficiency, on the electrical and exergy efficiency of both cycles for different organic working media, is examined.

Figures 3 and 4 indicate that both electrical and exergy efficiencies increase with increasing heat source inlet temperature for both cycles and all working fluids. The maximum achieved electrical and exergy efficiencies of the ORC-NIE are equal to 7.7% and 38.32% respectively, while the same values for TFC are found at 5.19%, and 25.82% respectively when the heat source inlet temperature is equal to 100$^o$C and R1233zd(E) is used as the fluid.

On the other hand, the electrical and exergy efficiencies have a decreasing rate depending on the condenser saturation temperature for both cycles and all fluids, as shown in Figures 5 and 6. The lower the condenser saturation temperature, the lower the cycle's low pressure, so the cycle's useful work increases. The maximum electrical efficiency of the ORC-NIE and the TFC is calculated at 10.51% and 7.46%, respectively, for a condenser saturation temperature of 10$^o$C and R1233zd(E) as the working medium. At the same temperature and for the same fluid, the exergy performance reaches the value of 52.27%, for the case of ORC-NIE, and 37.11%, for the case of TFC.

Moreover, the electrical and exergy efficiencies increase when the expander isentropic efficiency increases for both cycles and all the fluids, as it is illustrated in Figures 7 and 8. When the isentropic efficiency increases, the process is getting closer to the ideal isentropic one, leading to higher power output. The maximum determined electrical and exergy efficiencies of the ORC-NIE are equal to 8.79% and 43.72% respectively, while for TFC these values are defined at 6.08% and 30.26% respectively when the expander isentropic efficiency is equal to 0.8 and R1233zd(E) is the fluid used.

Consequently, according to Figures 3-8, in all cases, the electrical and exergy efficiencies of the ORC-NIE are higher compared to the corresponding values of TFC, as the energy content of the fluid at the expander inlet is greater for the case of the ORC-NIE, leading to higher power output. Furthermore, the most efficient fluid in terms of energy and exergy performance is R1233zd(E) for both cycles. At this point, it is important to mention that the influence of the working fluid type on the expander isentropic efficiency is not taken into account. It is assumed that the entire range of isentropic efficiency from 0.4 to 0.8 can be achieved utilizing all the examined working media.

The maximum achieved value of ORC-NIE electrical efficiency, which is equal to 10.51%, is similar to the corresponding optimized value calculated in the study [5]. For a heat source temperature of 100$^o$C, the optimum efficiency was found at 9.6%. Moreover, according to one study [10], the TFC electrical efficiency reached the value of 7.4 %, when the heat source temperature was equal to 80$^o$C. In the present study, the maximum defined level of TFC electrical efficiency is equal to 7.46%, which is close to the value in the literature.


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The main thermodynamic results for both cycles at the default scenario conditions (heat source inlet temperature, the condenser saturation temperature, and the expander isentropic efficiency equal to 100°C, 30°C, and 0.7, respectively) and R1233zd(E) as the working fluid, including the main results of the heat exchangers, are presented in Table 4. The TFC condenser requires greater surface area as the condenser load is larger in this case compared to the case of ORC-NIE. Additionally, for the ORC-NIE a greater exchange area at the heat recovery system is needed to be installed, as in this case, three processes are taking place, which are the preheating, the evaporation, and the superheating of the working medium. On the other hand, TFC requires a heat recovery system with a smaller area, because the fluid reaches the state of saturated liquid.

In addition, the Sankey diagrams for the default case of each cycle are illustrated in Figures 9 and 10. It is important to mention that the available heat source load has been defined taking into account the temperature difference between the heat source inlet temperature and the reference (ambient) temperature. According to the Sankey diagrams, the pump and the condenser loads of TFC are greater than the ones of ORC-NIE, leading to poorer exploitation of the available heat source. In addition, a temperature-specific entropy (T-s) chart for both cycles is depicted in Figure 11. According to this diagram, it is obvious that the enclosed surface of the ORC-NIE, which represents the cycle's useful output, is larger compared to the case of TFC. Taking all of the above into consideration, the ORC-NIE is the most proper choice in terms of thermodynamics for low-grade heat sources.

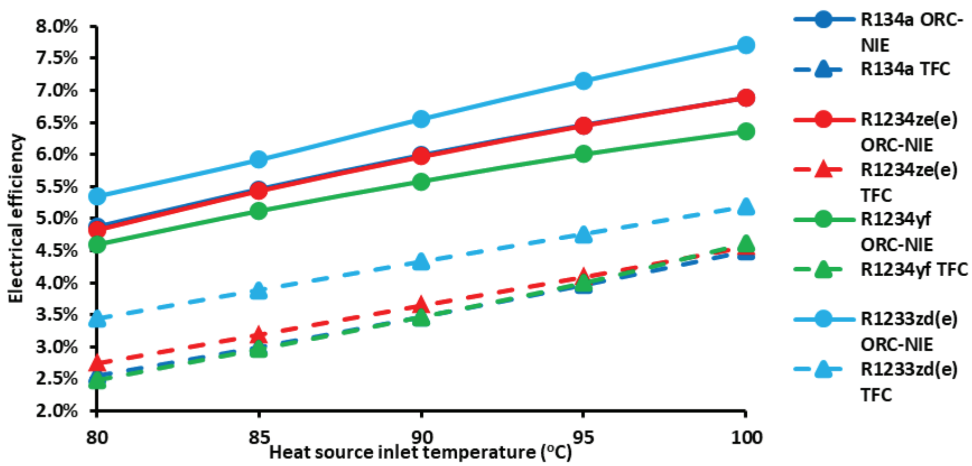


**Figure. 3.** Electrical efficiency depending on the heat source inlet temperature for both cycles and different organic working fluids.

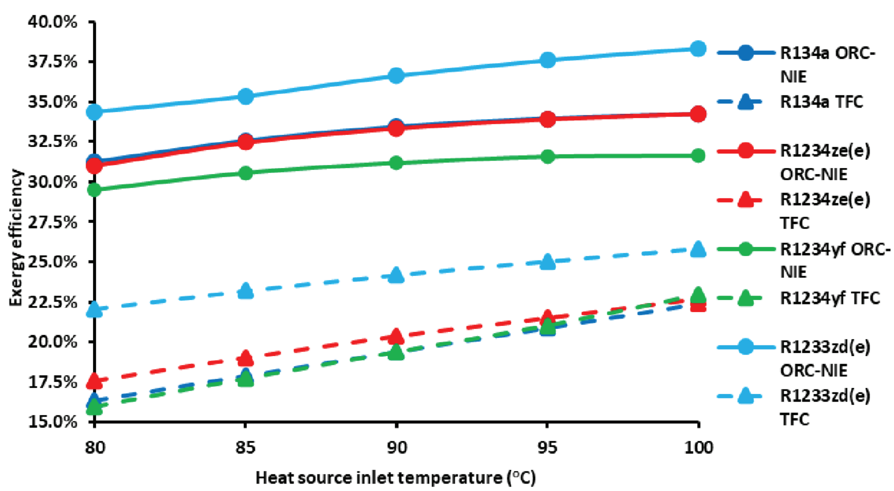


**Figure. 4.** Exergy efficiency depending on the heat source inlet temperature for both cycles and different organic working fluids.

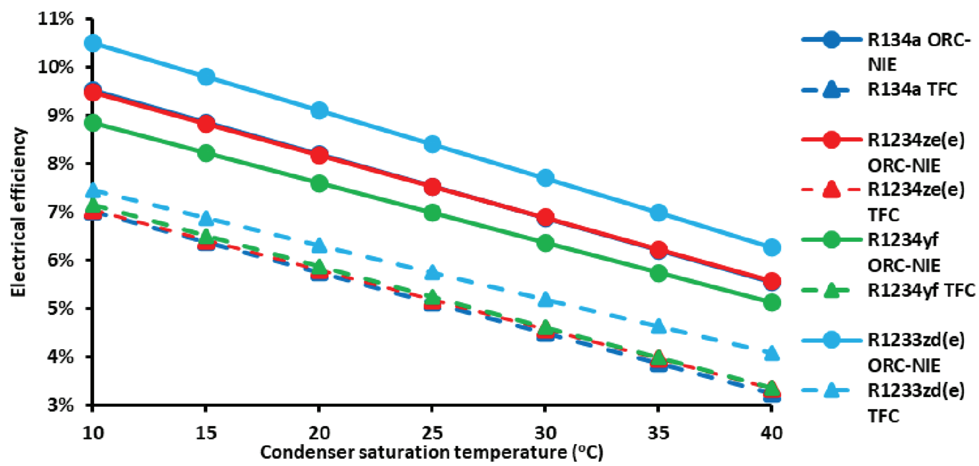


**Figure. 5.** Electrical efficiency depending on the condenser saturation temperature for both cycles and different organic working fluids.

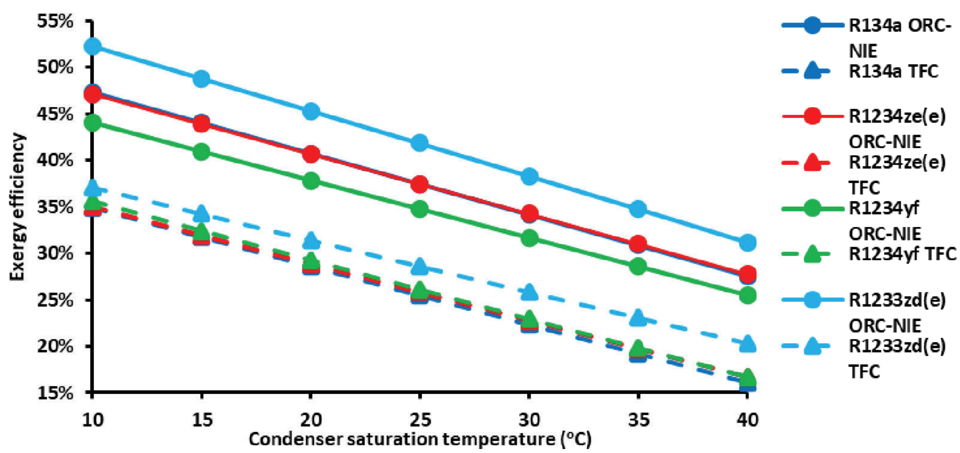


**Figure. 6.** Exergy efficiency depending on the condenser saturation temperature for both cycles and different organic working fluids.

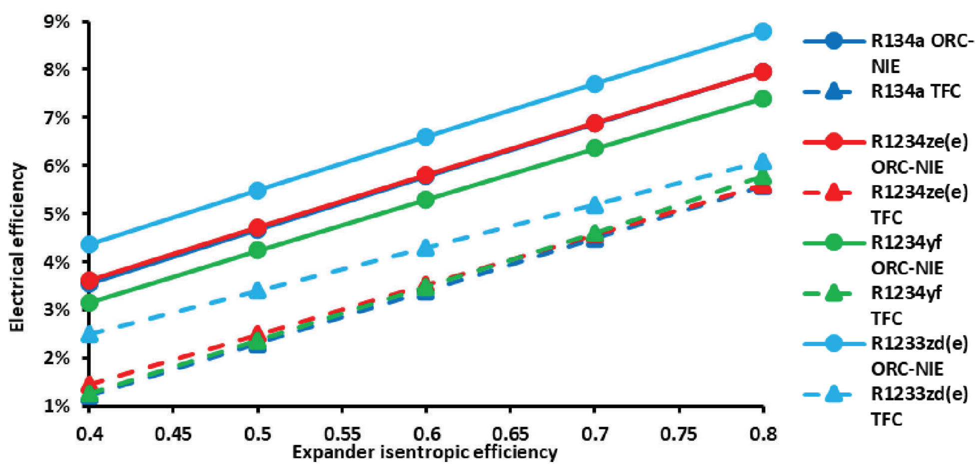


**Figure. 7.** Electrical efficiency depending on the expander isentropic efficiency for both cycles and different organic working fluids.


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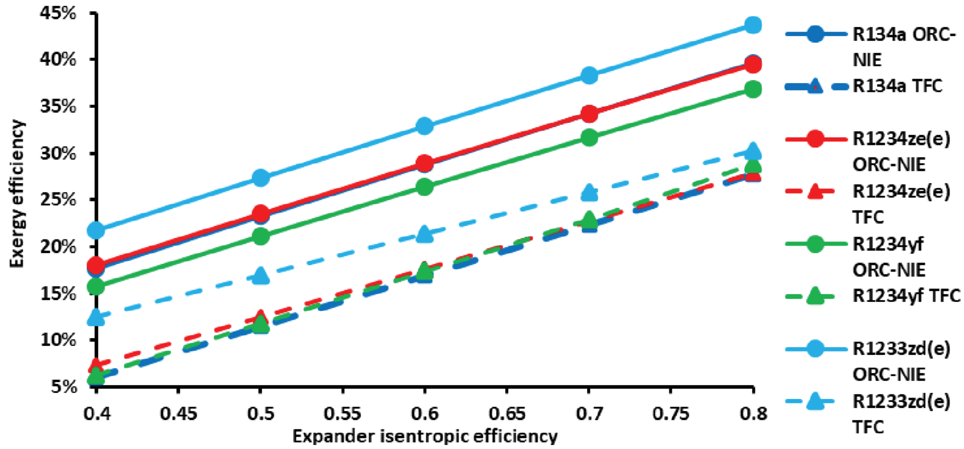


**Figure. 8.** Exergy efficiency depending on the expander isentropic efficiency for both cycles and different organic working fluids.

**Table 4.** Main thermodynamic results of both cycles at the default scenario (R1233zd(E) as working fluid).

| Outputs | Values for ORC-NIE | Values for TFC |
|---|---|---|
| Condenser area | 210.2 $m^2$ | 261.4 $m^2$ |
| Condenser UA | 178.7 kW/K | 222.2 kW/K |
| Heat recovery system area | 143.7 $m^2$ | 78.5 $m^2$ |
| Heat recovery system UA | 122.2 kW/K | 66.7 kW/K |
| Additional heater area | 14.7 $m^2$ | - |
| Additional heater UA | 12.5 kW/K | - |
| Low pressure | 1.55 bar | 1.55 bar |
| High pressure | 6.58 bar | 8.33 bar |
| Refrigerant mass flow rate | 6.7 kg/s | 21.4 kg/s |
| Total heat input | 1686.2 kW | 1690.5 kW |
| Pump electricity consumption | 4.15 kW | 17.77 kW |
| Expander electricity production | 134.05 kW | 105.53 kW |
| Net electricity production | 129.9 kW | 87.76 kW |
| Electrical efficiency | 7.70% | 5.19% |
| Exergy efficiency | 38.32% | 25.82% |

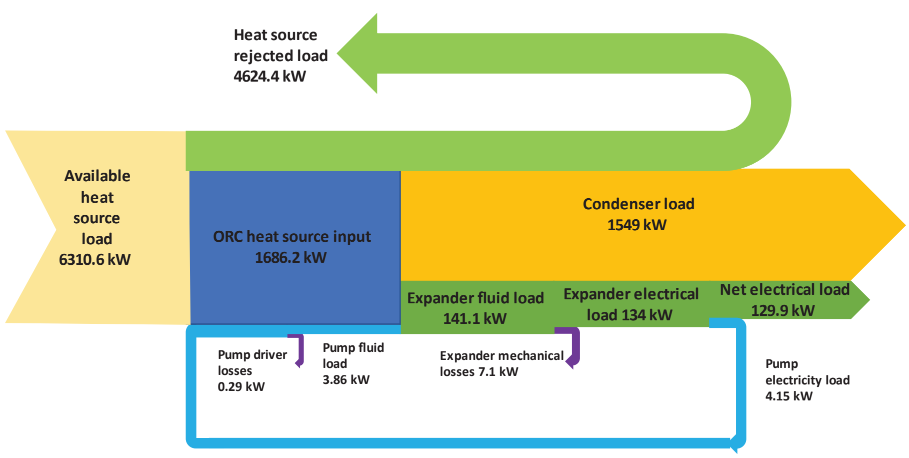


**Figure. 9.** Sankey diagram for the default scenario (R1233zd(E) as working fluid) of the ORC-NIE.

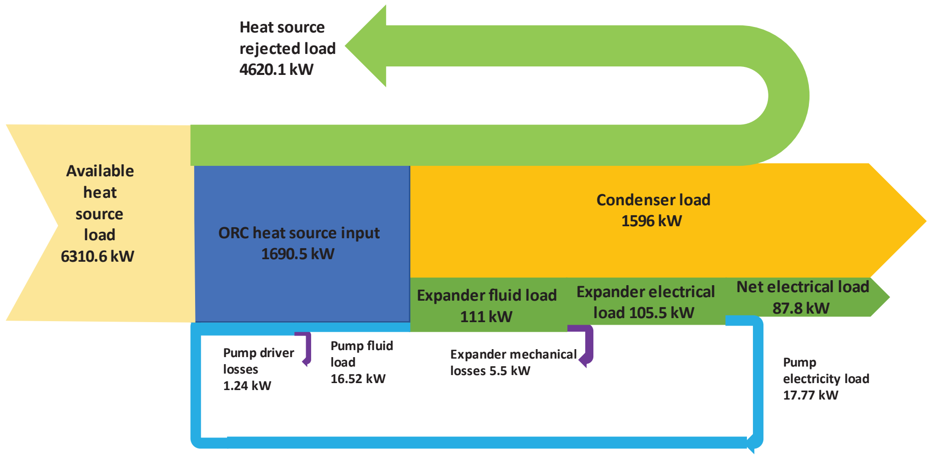


**Figure. 10.** Sankey diagram for the default scenario (R1233zd(E) as working fluid) of the TFC.

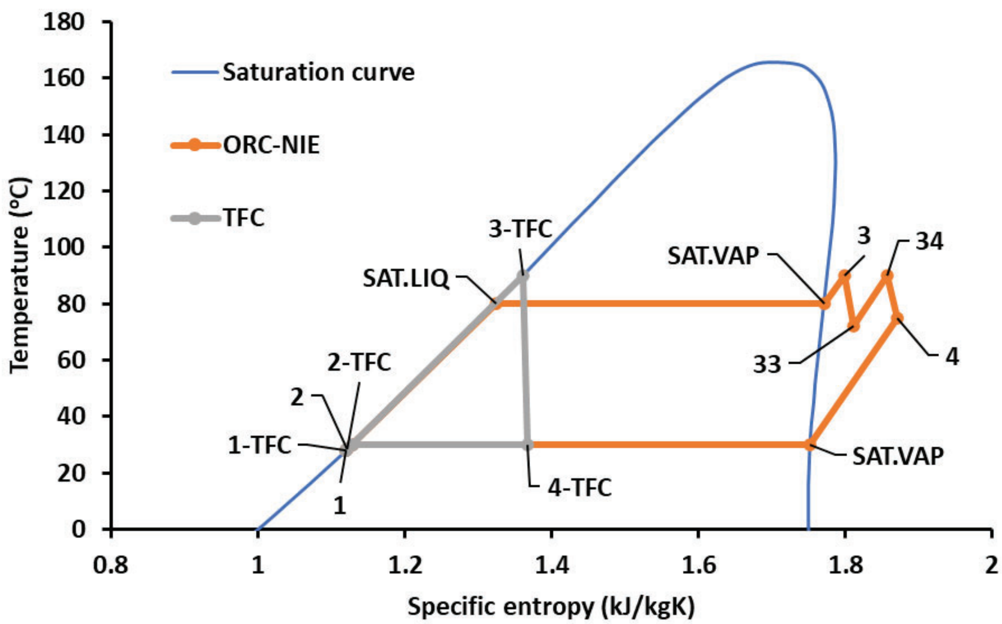


**Figure. 11.** Temperature-specific entropy (T-s) diagram for the default scenario (R1233zd(E) as working fluid) of the two examined cycles.

## 3.2. Techno-economic analysis

In this section, the results of the techno-economic study are presented, which have been calculated taking into account R1233zd(E) as working fluid and the thermodynamic results of the default scenario, from the previous section. As it is shown in Table 5, the investment cost of the ORC-NIE is greater compared to the corresponding value of the TFC. It is considered a reasonable result, as the higher electricity production requires a larger expander size. Additionally, the ORC-NIE requires heaters with larger heat-exchanging areas, as shown in Table 4. The greater size of these components strongly affects the investment cost, leading to a significant increase. Additionally, the ORC-NIE achieves greater values of NPV for all the examined operating hours. NPV is equal to 299 k€ in the case of 2,000 operating hours per year, and 2,417 k€ for 8,000 operating hours. However, for the TFC, slightly lower values of PBP are calculated, which are determined from 1.56 years for 8,000 operating hours per year up to 7.68 years when the operating hours are equal to 2,000. Consequently, the TFC is more economically viable in terms of initial cost and achieves slightly lower PBP, but in terms of NPV, the ORC-NIE performs better financially. For the case of 8,000


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operating hours per year, the main techno-economic results are presented in Table 5. The financial indexes for both cycles and different operating hours per year are shown in Figure 12.

**Table 5.** Results of techno-economic analysis

| Costs | Values for ORC-NIE | Values for TFC |
|---|---|---|
| Investment cost | 320,300 € | 202,000 € |
| Investment cost per $kW_{el}$ | 2,466 €/$kW_{el}$ | 2,300 €/$kW_{el}$ |
| Annual electricity production (8,000 operating hours) | 1,039.1 MWh | 702.1 MWh |
| Annual cash flow (8,000 operating hours) | 201.4 k€ | 136.4 k€ |
| Net present value (8,000 operating hours) | 2,417 k€ | 1,651 k€ |
| Payback period (8,000 operating hours) | 1.68 | 1.56 |

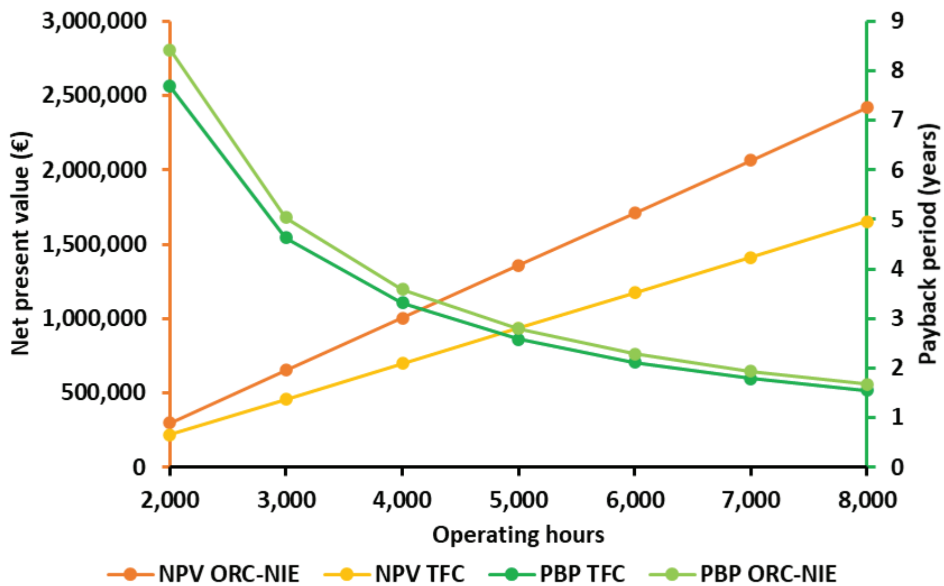


**Figure. 12.** Net present value and payback period for both cycles and different operating hours per year.

## 4. Conclusions

The present paper focuses on thermodynamic and techno-economic analysis and comparison of two organic cycle designs that are capable of operating at low-grade heat sources. For the comparison, the same available heat source conditions are assumed. The first one is similar to the ORC, but the expansion process approaches the isothermal conditions, and the second one is the TFC. The main conclusions are summarized below:

- R1233zd(E) seems to be the most proper organic working medium for both cycles in terms of thermodynamics.
- Both electrical and exergy efficiencies are enhanced with the increase of the heat source inlet temperature, and the expander isentropic efficiency while having a decreasing rate when the condenser saturation temperature increases.
- For the ORC-NIE, higher levels of electrical and exergy efficiencies are determined. The maximum achieved values are 10.51%, and 52.27%, respectively.
- For the case of TFC, lower investment costs and slightly lower payback period values are defined. The payback period can reach the value of 1.56 years when the annual operating hours are equal to 8,000.
- The ORC-NIE performs better in terms of net present value levels. On the other hand, TFC achieves net present values which are about 30% lower compared to the corresponding values of ORC-NIE.

The present work can be extended in the future, considering other organic cycle configurations, such as the recuperative ORC, and performing transient simulations.

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## Nomenclature

| | |
|---|---|
| CF | Cash Flow, € |
| E | exergy rate, kW |
| GWP | Global Warming Potential, - |
| i | Discount factor, % |
| IC | Investment Cost, € |
| K | Cost, € |
| N | Project lifetime, years |
| NPV | Net Present Value, € |
| ODP | Ozone Depletion Potential, - |
| P | pressure, bar |
| PBP | Payback Period, years |
| $P_{el}$ | electrical load, kW |
| Q | heat rate, kW |
| T | temperature, °C or K |
| UA | Heat transfer coefficient, kW/K |

**Greek symbols**

| | |
|---|---|
| η | efficiency |

**Subscripts and superscripts**

| | |
|---|---|
| 0 | reference |
| cond | condenser |
| crit | critical |
| el | electrical |
| ex | exergy |
| exp | expander |
| heater | heater |
| high | high |
| HRS | heat recovery system |
| in | inlet |
| is | isentropic |
| low | low |
| med | intermediate |
| net | net |
| O&M | Operation and maintenance |
| ORC-NIE | Organic Rankine Cycle with nearly isothermal expansion |
| pump | pump |
| TFC | Trilateral Flash Cycle |
| w | water |

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