IMPROVED ORC PROCESS FOR POWER PRODUCTION BY USING LOW TEMPERATURE HEAT

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ABSTRACT

Organic Rankine Cycles (ORC) are a modification of the classical water-steam process and are particularly suitable for electricity generation from low and medium temperature heat sources, e.g., industrial waste heat or geothermal energy. In contrast to the water-steam process, the ORC process uses organic fluids as working fluids. When using working fluids of the dry class (e.g. n-pentane), a recuperator is frequently installed in state-of-the-art ORC processes to increase the cycle efficiency. This paper analyses an improved ORC process design: A liquid working fluid stream is mixed with the vapour flow between the high-pressure stage and the medium-pressure stage of the turbine. Furthermore, the recuperator is replaced by a spray condenser. These two improvements were analysed by thermodynamic process simulations. As a use case, electricity production from clinker cooler waste heat at a temperature level of 275°C was simulated. The improved process as described would lead to an increase in the overall net efficiency up to 14%, compared to a state-of-the-art ORC process.

1 INTRODUCTION

1.1 State of knowledge in electricity generation by ORC processes

Electricity generation from low-temperature heat, such as waste heat, represents a significant contribution to fossil fuel substitution and a reduction in the CO_2 intensity of energy supply. According to a study by Oxford University, around 50% of global energy consumption is expected to end up as waste heat by 2030. Economically viable waste heat utilization will thus play a substantial position in the energy transition (Firth, Zhang & Yang 2019). A number of studies are devoted to estimating the existing and usable waste heat potential (Brueckner, Miró, Cabeza, *et al.* 2014; Miró, Brückner & Cabeza 2015; Su, Zhang, Xu, *et al.* 2021).

One of the most promising options for the conversion of waste heat to electricity is the Organic Rankine Cycle (ORC) process. In Europe, the waste heat that may be used by ORC plants is estimated to be around 20,000 GWh, which corresponds to a CO_2 saving of around 7.6 million tonnes (Campana, Bianchi, Branchini, *et al.* 2013). ORC processes are mainly used for electricity generation in the fields of geothermal energy, biomass, industrial waste heat or solar thermal energy. The annual installed capacity of ORC plants has steadily increased over the last 15 years. At the end of 2016, 1,754 plants with a total installed capacity of around 2,701 MW were installed worldwide (Tartière & Astolfi 2017). There are suggestions for improvement of this technology, but according to the current state of knowledge, they require more complex and sophisticated power plant cycles (Lecompte, Huisseune, van den Broek, *et al.* 2015). To expand the usable potential of low-temperature heat, the market also requires, among other things, cost efficiency, the utilization of low-exergy sources, acceptance by the operator and an entry-level technology (Loni, Najafi, Bellos, *et al.* 2021).

The ORC technology can be adapted to different heat sources by selecting the appropriate working fluid. This results in a wide range of applications, with a minimum off-heat temperature of approximately 80°C. Output power ranges from small plants with an electrical output of a few kW to power plants in the double-digit MW range (Feng, Hung, Greg, *et al.* 2015; Exergy International Srl 2021; Center for promotion of sustainable energy 2018; Li, Hung, Wu, *et al.* 2020)

The subcritical ORC process is well established in electricity generation from waste heat and also in geothermal plants. (Vaccaro & Franco 2016) More recent ORC technologies propose the following efficiency-enhancing developments:

- Selection of the process fluid:
  - selection procedures (White & Sayma 2020; Darvish, Ehyaei, Atabi, *et al.* 2015)
  - zeotropic fluid mixtures (Zhou, Wu, Li, *et al.* 2016; Zhi, Hu, Chen, *et al.* 2020)
- Process design:
  - dual-pressure processes (Wang & Yuan 2020; Wang, Zhang, Zhao, *et al.* 2019)
  - dual-loop processes (Boodaghi, Etghani & Sedighi 2021; Emadi, Chitgar, Oyewunmi, *et al.* 2020; Valencia, Fontalvo & Duarte Forero 2021)
  - novel ORC-architectures for waste heat recovery (Lecompte, Huisseune, van den Broek, *et al.* 2015)
  - trans- and supercritical process design (Song, Li, Wang, *et al.* 2020; Mohammed, Mosleh, El-Maghlany, *et al.* 2020; Hassani Mokarram & Mosaffa 2020).
- Optimisation of the process components (Witanowski, Klonowicz, Lampart, *et al.* 2020; Chatzopoulou, Lecompte, Paepe, *et al.* 2019) and the use of novel process components (Lecompte, Huisseune, van den Broek, *et al.* 2015; Xu & He 2011)
- Advanced control strategies (Hernandez, Desideri, Ionescu, *et al.* 2014; Baccioli & Antonelli 2017; Zhang, Li & Xu 2019; Imran, Pili, Usman, *et al.* 2020)

1.2 Improved ORC process design

The improved cycle aims at a relative increase of the overall efficiency beyond 10 % though using common components. The basic idea of the invention can be applied to a wide variety of designs, from highly efficient to cost-optimised, multi-purpose systems, in small to large power ranges. Unconventional ways of removing condensing heat and of extracting useful heat are central features of the proposed thermodynamic cycle. Despite the novelty, well-known process steps (e.g. de-superheating by means of liquid injection) are used at the component level.

The improved ORC process design (Figure 1, right) is characterized by the fact that preheated working fluid is mixed to the vapour flow between the high-pressure and medium-pressure stage of the turbine (MI MP - mixing chamber, medium pressure stage). In a further step, mixing takes place also after the medium pressure stage of the turbine, before the condenser (MI LP - mixing chamber, low pressure (exhaust vapour)). This renders unnecessary the recuperator heat exchanger that is usually part of a state-of-the art ORC process (component R in Figure 1, left). Furthermore, the mass flow of the working fluid increases in the preheater and in the medium pressure stage of the turbine, which leads to a higher energy output and an increase of the overall cycle efficiency.

The expenditures, compared to a common ORC, are:

a. The use of a turbine with two casings (similar to a steam reheat turbine)
b. The introduction of de-superheating at the intermediate pressure level to increase efficiency (spray de-superheating is a widely used component in steam power plant to control temperatures)
c. The additional introduction of de-superheating at the low-pressure level, before condensation, to enhance heat transfer at the condenser
d. The provision of a preheater with a somewhat larger heat transfer surface
e. The optional replacement of the tube to shell condenser by a spray condenser (an occasionally used technology in steam power plants in connection with dry cooling tower system), doing the job of de-superheating and condensing in one step.

Evidentially the point a. seems to be the most severe expenditure: on the other hand, a turbine with two casings eases the way to a combined heat and power generation in waste heat recovery. The use of condenser-side options may adversely lead to a simplification.

A patent application has been filed for this improved process design, and a national (Austrian) patent (Beckmann & Krail 2019) has already been granted; the international patent application (Krail & Beckmann 2019) is under review.

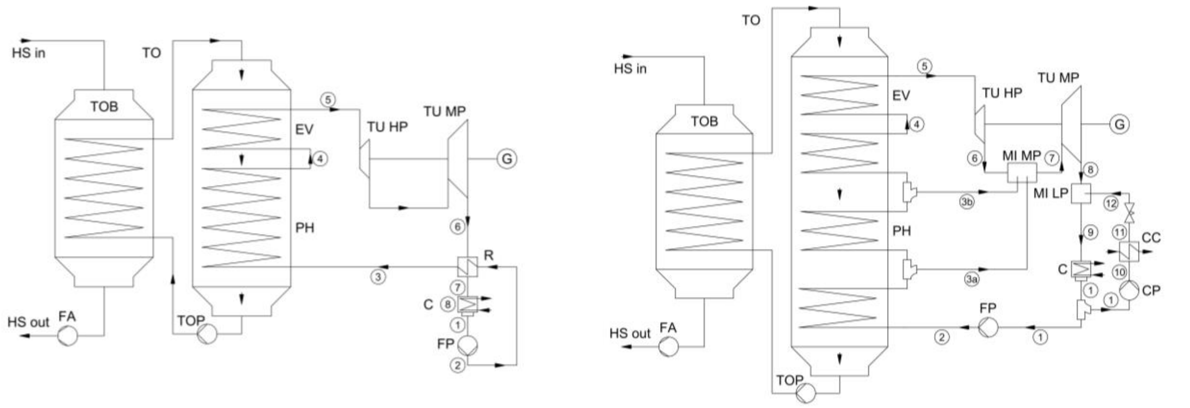

Figure 1: Flow diagram of a state-of-the-art ORC process (left), and of the improved ORC process (right), both applied to waste heat recovery from hot air from a clinker cooler

Figure 2 shows the state-of-the-art ORC process and the improved ORC process for the working fluid n-pentane in the T-s diagram, in order to illustrate the thermodynamic advantages of the improved ORC process design.
Dry class working fluids have the characteristic that the exhaust vapour state after expansion is in the superheated state. This has the advantage that the vapour does not have to be superheated before entering the turbine, which is particularly relevant for low-temperature processes. However, this overheating of the exhaust vapour at the turbine outlet has thermodynamic disadvantages, which are partly compensated by a recuperator heat exchanger in state-of-the-art processes, see Figure 1 (left) and Figure 2 (left).
In contrast to the state-of-the-art process, in the improved ORC process the superheating of the turbine exhaust vapour is reduced by an adding working fluid after the high-pressure stage of the turbine. This is carried out until the state of the vapour entering the medium pressure stage of the turbine is on the saturated vapour line. This increases the mass flow through the medium-pressure stage of the turbine, resulting in a higher turbine power output. Furthermore, working fluid is again added after the turbine's medium pressure stage in the form of a spray condenser, which has the effect of avoiding the recuperator in the improved ORC process. The absence of the recuperator's pressure loss allows expansion to a lower exhaust vapour pressure level, compared to the state-of-the-art ORC process, see Figure 1 (right) and Figure 2 (right). This leads to a higher enthalpy gradient at the turbine stages, which increases the turbine power output. For simplification, the improved ORC process is shown in Figure 2 with only one mixing step (3a) and without a condensate cooler (CC).

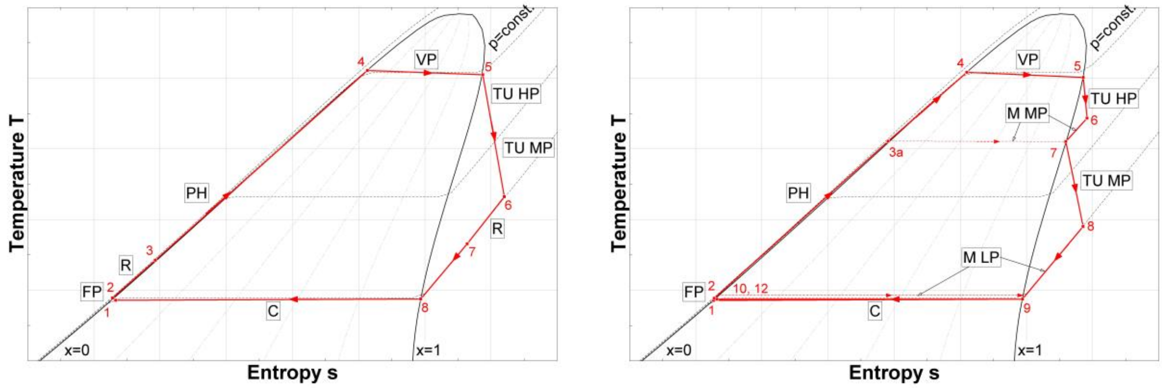

Figure 2: T-s Diagram of the state-of-the-art ORC process (left) and the improved ORC process (right)

2 MATERIALS AND METHODS

2.1 Energy balance and process efficiency

The energy balance of the investigated ORC processes was based on mass and energy balances for steady state flow process, according to equation (1) and (2), where the index "IN" stands for the mass and energy flows entering the system and the index "OUT" for the energy and mass flows leaving the system. In the following calculations, heat losses were neglected for all components (units and pipes). For heat exchangers, a pressure drop was calculated, but assumed to be zero for pipes.

$$\sum \dot{m}_{IN} = \sum \dot{m}_{OUT} \tag{1}$$

$$\dot{Q} + \dot{W} = \sum \dot{m}_{OUT} \times h_{OUT} - \sum \dot{m}_{IN} \times h_{IN} \tag{2}$$

The process evaluation was carried out on the basis of efficiency coefficients (Figure 3). In addition to the overall net efficiency η_{NET}, the individual efficiency coefficients of the vapour generator η_B and the thermodynamic cycle η_C were also calculated.

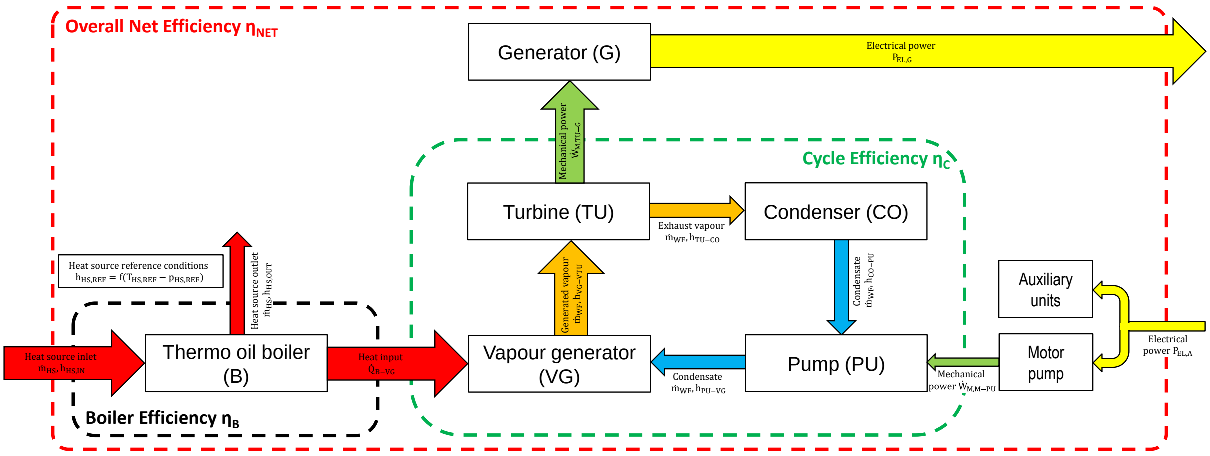

Figure 3: Energy balance and efficiency

The efficiency of the thermo oil boiler η_B was calculated according to equation (3). Due to the fact that waste heat serves as a heat source for the investigated processes, the heat input into the vapour generator is referred to reference conditions ($T_{HS,REF}$ = 25°C, $p_{HS,REF}$ = 1.013bar) that are commonly used in power plant engineering. The output is defined as the thermal power transferred to the cycle, with heat losses neglected for the vapour generator and for all heat exchangers and pipes.

$$\eta_B = \frac{\dot{Q}_{B-VG}}{\dot{Q}_{HS-REF}} = \frac{\dot{m}_{WF}(h_{VG-TU} - h_{PU-VG})}{\dot{m}_{HS}(h_{HS,IN} - h_{HS,REF})} \tag{3}$$

For the calculation of cycle efficiency η_C, the output of the expansion machine was considered as the useful power. The input power is the thermal power of the vapour generator, the power supplied to the cycle for the operation of the pump was considered, too (equation (4)).

$$\eta_C = \frac{\dot{W}_{TU-G} - \dot{W}_{M-PU}}{\dot{Q}_{B-VG}} = \frac{\dot{m}_{WF}(h_{VG-TU} - h_{TU-C}) - \dot{m}_{WF}(h_{PU-VG} - h_{CO-PU})}{\dot{m}_{WF}(h_{VG-TU} - h_{PU-VG})} \tag{4}$$

Although the use case investigated here is a process where the heat source is available as waste heat, it is appropriate for economic reasons to consider the overall net efficiency η_{NET} as the most relevant criterion for the process comparison with a conventional state-of-the-art ORC process. It relates the net electrical output of the process to the amount of waste heat based on the reference conditions mentioned above and represents the “yield” of electrical power from an existing / available waste heat source (equation (5)). For the net electrical output of the process, all outputs for operating auxiliary units were subtracted from the electrical output of the generator. The auxiliary units were the pump for the working fluid, the thermal oil pump, as well as the electrical power for the fan of the air condenser and the electrical power for the waste heat ventilator for covering the pressure loss of the thermo oil boiler.

$$\eta_{NET} = \frac{P_{EL,G} - \sum P_{EL,A}}{\dot{Q}_{HS-REF}} = \frac{P_{EL,G} - \sum P_{EL,A}}{\dot{m}_{HS}(h_{HS,IN} - h_{HS,REF})} \tag{5}$$

2.2 Process simulation

The simulation of the processes was carried out with the commercial simulation software IPSEpro (SimTech GmbH 2021). Figure 4 describes the steps of thermodynamic process simulation. In the model library, the individual units that are used to design the process are described with thermodynamic mass and energy balance equations. Furthermore, the thermodynamic fluid properties are also calculated in the model library. The database used for calculation is the "RefProp" database (NIST - National Institute of Standards and Technology 2013). The IPSEpro library for low-temperature processes "LTP-Lib - Low-Temperature Processes" (SimTech GmbH 2016) was used as model library. IPSEpro "PSE - Process Simulation Environment" (SimTech GmbH 2017a) offers a visual programming interface for the modelling, simulation and optimisation of thermodynamic cycles. The IPSEpro equation solver, which is part of the simulation environment, is based on a Newton-Raphson algorithm (SimTech GmbH 2014a). The optimisation module "PSOptimize" (SimTech GmbH 2014b) allows process multivariate optimisation. Free equations can be used to define additional variables, e.g. for the calculation of efficiency coefficients. By means of the additional module "PSXLink" (SimTech GmbH 2017b), parameter variations can be carried out in Excel.

In the simulation model, the processes to be analysed are designed based on units available in the model library, and then appropriate boundary conditions are added.

The simulation result delivers the results of the parameter variation and the process optimisation based on process-specific indicators, such as efficiency factors. Furthermore, simulation results can be displayed as diagrams (e.g. T-s diagram).

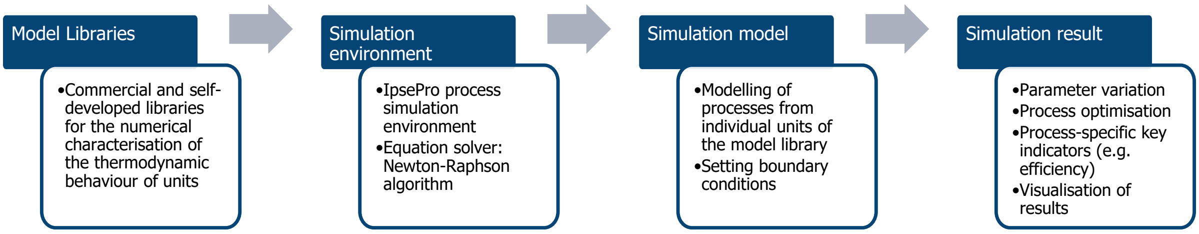

Figure 4: Structure of the simulation environment IPSEpro

2.3 Comparison of the improved ORC process with the state-of-the-art ORC process

The suitability and advantages of the improved ORC technology depends on boundary conditions and parameters that are specific for the project, process, and plant being studied.

First, there is the selection of the working fluid. As already mentioned, the thermodynamic advantages are predicated on organic working fluids of the dry class, but even within this class, the entropy reductions at the saturated vapour line during expansion can vary. Another determining factor is the waste heat source itself, with its yield and temperature. Current research is not known to allow generally valid and quantifiable statements about the thermodynamic advantages with the different parameters and parameter combinations. Therefore, the following comparison is based on a selected, actually implemented and comprehensively documented ORC plant, according to the state-of-the-art, for the conversion of hot air from the clinker coolers of a cement factory into electricity (Heidelberger Cement AG 2001; Umweltbundesamt 2010). The improved ORC process is designed for this same application. In order to ensure an equal comparison of performance data and yields between the two processes, the same boundary conditions were set in both cases (Table 1). The following simulations refer to an input waste heat stream at 275°C and a heat input of 14 MW, based on reference conditions. N-pentane is used as the working fluid in the well-documented state-of-the-art process, and so it was also chosen for the improved design. The isentropic efficiency of the turbine was calculated on the basis of design data from Heidelberger Cement AG (2001) and subsequently assumed to be 74.5% for both comparative processes in the respective turbine stages. On the heat sink side, an ambient temperature of 11°C was assumed in each case. The thermodynamic comparability of the heat exchangers was ensured by assuming the same minimum temperature difference between the hot and cold fluid of the heat exchanger. Since the recuperator is no longer required in the improved process, the turbine can expand to a lower condensation pressure by omitting the pressure drop of the recuperator.

Table 1: Boundary conditions

	State-of-the-art ORC process[1]	Improved ORC process	Unit
Waste heat input (hot air from clinker cooler)			
Waste heat input – temperature $T_{HS,IN}$	275	275	°C
Waste heat input – mass flow $\dot{m}_{HS}$	55	55	kg/s
Waste heat input $\dot{Q}_{HS-REF}$[2]	14,000	14,000	kW
ORC process			
Working fluid	n-Pentane	n-Pentane	
Thermo oil boiler – min. temp. difference $\Delta T_{TOB,MIN}$	37.9	37.9	K
Vapour generator – min. temp. difference $\Delta T_{VG,MIN}$	14.0	14.0	K
Recuperator – min. temp. difference $\Delta T_{R,MIN}$	28.9	-	K
Generator – electric efficiency $\eta_{EL,G}$	98.0	98.0	%
Generator – mechanic efficiency $\eta_{M,G}$	97.0	97.0	%
Motor – electric efficiency $\eta_{EL,MO}$	95.0	95.0	%
Motor – mechanic efficiency $\eta_{M,MO}$	97.0	97.0	%
Pump – isentropic efficiency $\eta_{IS,PU}$	78.5	78.5	%
Pump – mechanic efficiency $\eta_{M,PU}$	97.0	97.0	%
Turbine – isentropic efficiency $\eta_{IS,TU}$	74.5	74.5	%
Turbine – mechanic efficiency $\eta_{M,TU}$	98.0	98.0	%
Turbine – exhaust vapour pressure $p_{TU,OUT}$[3]	1.03	0.98	bar
Condenser (cooling medium ambient air)			
Condenser inlet/outlet – temperature $T_{CO,IN}$/ $T_{CO,OUT}$	11/33	11/33	°C

[1] State-of-the-art ORC process, boundary conditions based on Heidelberger Cement AG (2001) and Umweltbundesamt (2010)
[2] based on reference conditions $T_{HS,REF}$ = 25°C and $p_{HS,REF}$ = 1.013bar
[3] the lack of a recuperator-caused pressure loss in the improved process causes a lower exhaust vapour pressure at the turbine outlet

3 RESULTS

Parameter studies were carried out for the two processes, whereby both processes were investigated with a focus on maximising the overall net efficiency, using the IpsePro optimisation algorithm PSOptimize. While the process-external boundary conditions were kept constant, the vapour pressure entering the high-pressure stage of the turbine was varied. In the case of the improved process, for each pressure variation of the vapour pressure entering the high-pressure stage of the turbine, the pressure level of the intermediate pressure was also optimised with a focus on maximising the net overall efficiency. In both cases, the pressure variation of the vapour was carried out in the subcritical pressure range of the working fluid n-pentane.
Based on these simulations, an optimum vapour pressure was calculated for each individual process (Figure 5). The simulation results show that cycle efficiency increases with increasing vapour pressure entering the high-pressure stage of the turbine, which is to be expected thermodynamically. As the vapour pressure increases, the boiler efficiency decreases due to an increase in the output temperature of the waste heat source from the boiler. This effect is much more evident in the state-of-the-art ORC process than in the improved ORC process. The final result shows that with the state-of-the-art ORC process the maximum overall net efficiency is reached at a vapour pressure entering the high-pressure stage of the turbine of p_5=16.5 bar; the achieved overall net efficiency is η_{NET}=8.05%. With the improved ORC process the maximum overall net efficiency exceeds the respective values of the state-of-the art ORC at any pressure: at vapor pressure of p_5=20 bar the relative increase of the overall net efficiency of the 11.4 % is reached. At the thermodynamically optimum vapour pressure of p_5=32 bar the improved ORC process reaches an overall net efficiency of η_{NET}=9.17%, which means an increase

of the overall net efficiency of 14% relative to the state-of-the-art ORC process (Figure 6). The detailed simulation results of both variants are shown in Table 2. The results apply to optimal vapour pressure in each case. In case of a realistic plant layout, the chosen vapor pressures is commonly lower than the thermodynamically optimum pressure. Generally, the idea of the improved cycle favours the application of higher pressure level on the working fluid side.

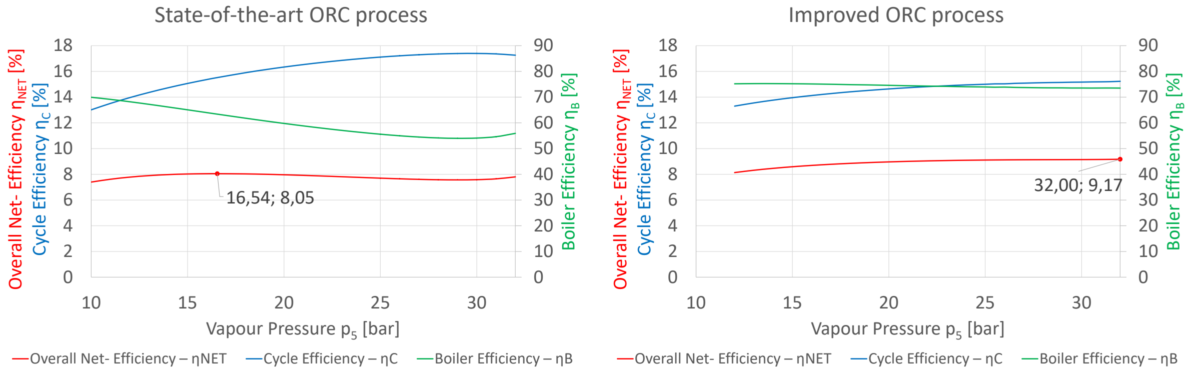

Figure 5: ***Parameter study of the state-of-the-art ORC process (left) and the improved ORC process (right) with focus on the boiler efficiency, cycle efficiency and net efficiency. The pair of values shown in both graphs denotes the optimization result for the overall net efficiency***

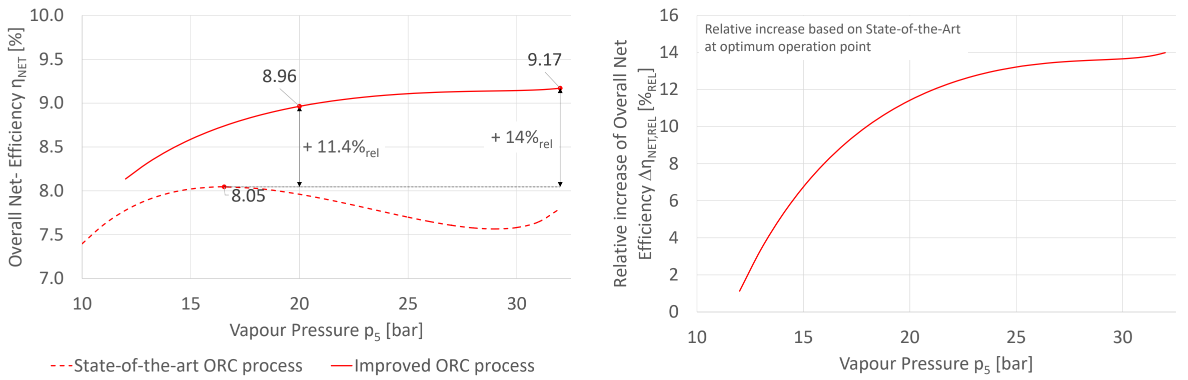

Figure 6: ***Comparison of the net efficiency of the state-of-the-art ORC process and the improved ORC process at varying vapour pressure entering the turbine (left) and relative increase of the net efficiency based the state-of-the-art ORC process at optimum operation point (right)***

Table 2: ***Simulation results at optimum operation point***

	State-of-the-art ORC process	**Improved ORC process**	**Unit**
Energy balance			
Waste heat input/output – temperature $T_{HS,IN}/T_{HS,OUT}$	275/117	275/93	°C
Turbine inlet - vapour pressure p_5	16.5	32.0	bar
Thermo oil boiler – transferred heat $\dot{Q}_B$	8,875	10,259	kW
Vapour generator – transferred heat $\dot{Q}_{VG}$[1]	8,911	10,294	kW
Recuperator – transferred heat $\dot{Q}_R$	1,030	-	kW
Condenser – transferred heat $\dot{Q}_{CO}$	7,528	8,724	kW
Generator – electrical power $P_{EL,G}$	1,353	1,586	kW
Auxiliary units (Pumps + fans) – electr. power $P_{EL,A}$	227	304	kW
Electrical net power output $P_{EL,NET}$	1,126	1,282	kW
Efficiency			
Boiler efficiency – η_B	63.4	73.3	%
Cycle efficiency – η_C	15.5	15.3	%
Overall net efficiency – η_{NET}	8.05	9.17	%
Relative increase of Overall net efficiency – $\Delta\eta_{NET,REL}$	0	14	$\%_{REL}$

[1] The transferred heat of the vapour generator is higher than the transferred heat of the thermo oil boiler, as energy is supplied to the thermo oil circuit by the pump.

4 CONCLUSIONS

The improved process design leads to adaptations in the process layout. A fluid flow is mixed in after the high-pressure stage of the turbine via a branch from the preheater of the vapour generator. This increases the mass flow through the preheater as well as the mass flow of the medium-pressure stage of the turbine. The recuperator, which is frequently used for dry class working fluids, can be omitted and is replaced by a spray condenser after the intermediate pressure stage of the turbine. Consequently, the condensation pressure decreases due to the elimination of the pressure drop of the recuperator and a higher enthalpy drop at the expansion turbine can be achieved. Although the cycle efficiency of the improved ORC process decreases slightly, but the boiler efficiency increases significantly, the heat source can be better utilised. In total, the measures mentioned lead to an increase in the overall net efficiency up to 14% relative to the state-of-the-art ORC design, as determined at the optimal operating point of each process.
Detailed simulations with varying boundary conditions, such as different temperatures of the heat source and the heat sink or the influence of different working fluids, are planned.

NOMENCLATURE

Δ	difference	$\dot{Q}$	thermal power (kW)
η	efficiency (%)	$\dot{W}$	mechanical power (kW)
h	enthalpy (kJ/kg)	s	entropy
$\dot{m}$	mass flow (kg/s)	T	temperature (°C)
P	electrical power (kW)	x	vapour quality
p	pressure (bar)		

ABBREVIATIONS

A	auxiliary units (electric motors for fans and pumps)	MI	mixing chamber
B	boiler	MIN	minimum
C	cycle	MO	motor
CO	condenser	MP	medium pressure stage
CC	condensate cooler	OUT	outlet stream
CP	condensate pump	PH	preheater
EL	electric	PU	pump
EV	evaporator	R	recuperator
FA	fan	REF	reference conditions
FP	feed pump	REL	relative
G	generator	TO	thermo oil
IN	inlet stream	TOB	thermo oil boiler
IS	isentropic	TOP	thermo oil pump
HP	high pressure stage	TU	turbine
HS	heat source	VG	vapour generator
M	mechanic	WF	working fluid
NET	net value	WHS	waste heat source
LP	low pressure (exhaust vapour)	WHR	waste heat recovery

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