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Design of organic Rankine cycles using a non-conventional optimization approach

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Abstract:

The organic Rankine cycle is a suitable technology for utilizing low grade heat for electricity production. Compared to the traditional steam Rankine cycle, the organic Rankine cycle is beneficial, since it enables the choice of a working fluid which performs better than steam at low heat input temperatures and at low power outputs. Selecting the process layout of the organic Rankine cycle and the working fluid are two key design decisions which are critical for the thermodynamic and economic performance of the cycle. The prevailing approach used in the design and optimization of organic Rankine cycles is to model the heat exchangers by assuming a fixed minimum temperature difference. The objective of this work is to assess the applicability of this conventional optimization approach and a non-conventional optimization approach. In the non-conventional optimization approach a total UA -value (the product of the overall heat transfer coefficient and the heat transfer area) is assigned to the cycle, while the distribution of this total UA -value to each of the heat exchangers is optimized. Optimizations are carried out for three different marine engine waste heat sources at temperatures ranging from 90 °C to 285 °C. The results suggest that the conventional optimization approach is not suitable for estimating the performance potential when the temperature profiles in the heat exchangers are closely matched. This is exemplified for the fluid MDM where the temperature profile of preheating aligns with the heat source fluid and for the zeotropic mixture R32/R134a where the temperature profile of condensation aligns with the cooling water. Furthermore, the conventional optimization approach shows weaknesses in evaluating the feasibility of using a recuperator, when the expander outlet temperature is high. In these cases the non-conventional optimization approach is the more suited methodology for designing organic Rankine cycles.

Keywords:

Waste Heat Recovery, Zeotropic Mixture, Working Fluid, Organic Rankine Cycle, Marine Diesel Engine.

1. Introduction

The organic Rankine cycle (ORC) is a heat engine which enables conversion of heat to mechanical work. The ORC can be used in many different applications, e.g., for renewable electricity production from geothermal heat or to increase the efficiency of existing energy systems by utilizing waste heat. The ORC is similar to the steam Rankine cycle, but uses an organic working fluid and a simpler cycle layout to improve the utilization of low-temperature heat at a smaller scale of power.

The heat exchangers in the ORC are typically modelled by assuming a minimum pinch point temperature difference. Maraver et al. [1] used this approach and maximized the second law efficiency of heat utilization from six different heat source and sink combinations. The range of heat source inlet temperatures was 100-300 °C, and three of the sources were restricted by a minimum outlet temperature while three were not. They assessed the feasibility of using an internal heat exchanger (recuperator) to recuperate heat at the expander exhaust for preheating at the pump outlet. The results indicated that the recuperator does not increase the power output of the ORC when there is no limit on the heat source outlet temperature, thus supporting conclusions from Dai et al. [2] and Li et al. [3].

Heberle et al. [4] used the conventional modelling approach and fixed the minimum pinch point temperature differences in the heat exchangers in their analysis of zeotropic mixtures. For utilization of a 120 °C geothermal source, a mixture of isobutane/isopentane (0.9/0.1_{mole}) achieved a second law efficiency increase of 8 % compared to pure isobutane. The constant minimum pinch point temperature difference resulted in variations in the UA -values, and the total UA -value of all heat exchangers in the cycle was found to be 40.4 % larger for isobutane/isopentane (0.9/0.1_{mole}) compared to pure isopentane. In a previous study [5] the present authors investigated a large group of pure fluids and mixtures, and found that the use of zeotropic mixtures enabled an increase in performance, but required larger UA -values compared to pure fluids when assuming constant minimum pinch point temperature differences in the heat exchangers. The larger UA -values suggest that larger heat transfer areas are needed for the heat exchangers, resulting in higher investment costs.

Baik et al. [6-7] used non-conventional modelling approaches in their studies of ORCs. In a comparison of a number of working fluids, R125 (transcritical), R134a (subcritical), R152 (subcritical) and R245fa (subcritical) [7], they fixed the total cycle UA -value and optimized the distribution of UA -value to each of the heat exchangers in the cycle. For low values of total cycle UA , the subcritical cycles performed best, while the transcritical cycle reached highest performance at higher total cycle UA -values.

Large marine vessels represent an interesting application area for the ORC, since vast amounts of waste heat are available from the diesel engine, e.g., from exhaust gasses, the scavenge air cooler and the jacket water cooler. Larsen et al. [8] compared the performance of three technologies: the steam Rankine cycle, the Kalina cycle and the ORC, for recovering the exhaust gas heat from a large marine diesel engine. The results indicated that the ORC is the preferred technology in terms of maximizing the power output, since it reached a power output of 7 % of the main engine power output, while the steam Rankine and the Kalina cycles achieved 5 % each.

In this paper we compare the outcome of ORC optimizations using the conventional optimization approach assuming constant minimum pinch point temperature differences, and a non-conventional optimization approach similar to that used by Baik et al. [7]. The novel contribution of the present paper is a detailed comparison of the outcomes from the two optimization approaches when applied in ORC design studies. The focus of the study is to identify situations where the two optimization strategies give different results, and to assess the applicability of the two methods in such cases. The analyses are based on the following three marine waste heat utilization cases: exhaust gas waste heat recovery (WHR), WHR from the scavenge air cooler and WHR from the jacket water cooler.

The paper begins with a description of the methodology in section 2, and in section 3 the results of the analyses are presented. The results are discussed in section 4 and conclusions are given in section 5.

2. Methodology

Numerical models of the ORC process (see Fig. 1) were developed in Matlab[®] 2014b [9] using the open-source software CoolProp version 4.2.3 [10] (exhaust gas and scavenge air cases) and the commercial software REFPROP[®] version 9.0 [11] (jacket water case) for thermodynamic property data. Figure 1 presents the layout of the simple and recuperated ORC. The simple ORC consists of an expander, a condenser, a pump and a boiler, where the boiler comprises a preheater, an evaporator and a superheater section. In the recuperated ORC, excess heat at the outlet of the expander is used to preheat the working fluid after the pump.

Table 1 lists the conditions for the three waste heat sources analyzed in this study and the considered working fluids. The data is based on a MAN 12K98ME engine at full load conditions. For the exhaust gas case the outlet temperature ($T_{hf,o}$) is limited to 160 °C in order to avoid acid corrosion problems, and for the jacket water the outlet temperature is set to 75 °C, since the jacket water temperature cannot be too low when it is returned to the engine. For each of the three cases

working fluids are selected based on the studies by Larsen et al. [12-13], Quoilin et al. [14] and Bao and Zhao [15]. The main selection criteria are zero ozone depletion potential, high performance and a subcritical optimum pressure. Additionally, fluids that are commonly used by ORC manufacturers are also included in the selection. The linear siloxanes hexamethyldisiloxane ($C_6H_{18}OSi_2$) and octamethyltrisiloxane ($C_8H_{24}O_2Si_3$) are abbreviated MM and MDM, respectively. The objective of the optimizations is the maximization of the net power output from the ORC,

$$\dot{W}_{NET} = \dot{m}_{wf} [h_5 - h_6 - (h_2 - h_1)], \quad (1)$$

where \dot{m}_{wf} is the mass flow of working fluid in the ORC and h is mass specific enthalpy.

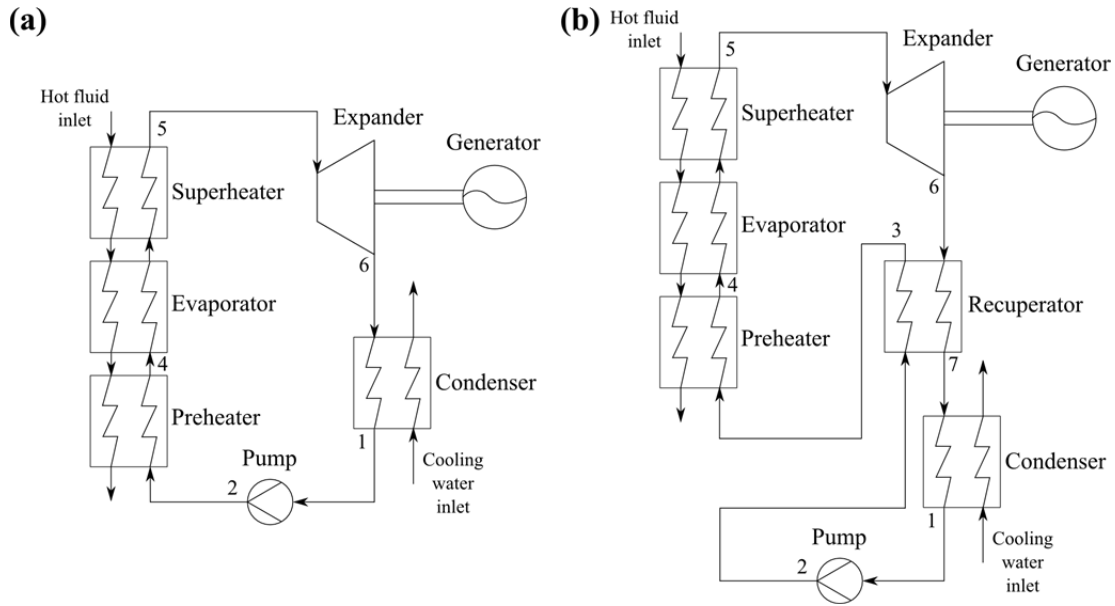


Fig. 1. A sketch of a) the simple ORC and b) the recuperated ORC

Table 1. Hot fluid conditions and working fluids for the three cases

Cases	$T_{hf,i}$ [°C]	$T_{hf,o}$ [°C]	\dot{m}_{hf} [kg/s]	P_{hf} [bar]	Working fluids
1. Exhaust gases	285	160	184.0	1.5	Toluene, benzene, MDM and ethanol
2. Scavenge air cooler	191	-	180.7	1.5	Cyclopentane, solkatherm and MM
3. Jacket water cooler	90	75	150.0	1.5	R32, R134a, R32/R134a and R245fa

The modelling conditions are listed in Table 2. The cooling water mass flow is chosen as a fixed parameter in order to compare solutions which require cooling water pumps of similar capacities and costs. The pinch points for the heat exchangers are only used in the optimizations with constant minimum pinch point temperature difference. Additional assumptions are the following: no pressure loss in piping or heat exchangers, no heat loss from the system, steady-state conditions and homogeneous flow in terms of thermodynamic properties.

2.1. Pinch point based optimization

For the cases where there is no limit on the hot fluid outlet temperature the optimization parameters are the expander inlet pressure and temperature, while when the hot fluid outlet temperature is constrained the expander inlet temperature is the only optimized parameter.

The boiler pinch point is assumed to be at the saturated liquid point (state 4), while an additional check on the pinch point is made at a temperature $T = (T_4 + T_3)/2$. The additional check is necessary for MDM, since the preheating temperature profile is slightly curved for this fluid. In the recuperator the pinch point is assumed to be at the cold inlet. For the condenser the pinch point is

checked at the saturated vapor point and at the inlet and outlet. The validity of the assumptions on the pinch point locations are verified for all optimizations.

Table 2. Modelling conditions

Parameter description	Symbol	Value	Unit
Cooling water			
Cooling water inlet temperature	$T_{cool,i}$	25	°C
Cooling water mass flow	\dot{m}_{cool}	400	kg/s
Cooling water pressure	P_{cool}	4	bar
Condenser			
Minimum pinch point temperature difference	ΔT_{cond}	5	°C
Outlet vapor quality	x_1	0	-
Pump			
Isentropic efficiency	$\eta_{s,pump}$	0.8	-
Recuperator			
Minimum pinch point temperature difference	ΔT_{recu}	5	°C
Boiler			
Minimum pinch point temperature difference	ΔT_{boil}	10	°C
Expander			
Isentropic efficiency	$\eta_{s,exp}$	0.8	-
Minimum outlet vapor quality	$x_{exp,min}$	1	-

When there is a limit on the hot fluid outlet temperature, this temperature is being fixed at its minimum value, and the expander inlet temperature is then determined from the hot fluid outlet temperature. In all cases the condensation pressure is found using an iterative solving procedure such that the specified condenser pinch point is satisfied. For the simple ORC with fixed hot fluid outlet temperature, the expander inlet temperature can be directly found using the specified pinch point in the boiler, the hot fluid outlet temperature and the energy balance equation.

For the recuperated ORC the heat transfer in the recuperator is dependent on the conditions at the expander inlet, and the expander inlet temperature is therefore determined using an iterative solving procedure such that the minimum hot fluid outlet temperature is reached. Table 3 lists the solution strategies for the analyzed cases. For the low-temperature case (with a lower limit on the hot fluid outlet temperature) it is necessary to have the expander inlet temperature as an optimization parameter, while the boiler pinch point is set free to vary. This is further elaborated on in Section 3.3.

Table 3. Optimization and solving parameters used in the four different pinch point based models

	Optimizing parameters	Solving parameters
Simple ORC, no lower limit for $T_{hf,o}$	P_5 and T_5	P_{cond}
Recuperated ORC, no lower limit for $T_{hf,o}$	P_5 and T_5	P_{cond}
Simple ORC, lower limit for $T_{hf,o}$	P_5	P_{cond}
Recuperated ORC, lower limit for $T_{hf,o}$	P_5	P_{cond} and T_5

2.2. UA-value based optimization

When UA -values are used as modelling parameters for the heat exchangers, a total cycle UA -value is defined, and the distribution of UA -value to each of the heat exchangers is optimized. In addition to the optimization parameters used in the pinch point models, UA_{boil} is included as an optimization parameter for the simple ORCs, while UA_{boil} and UA_{cond} are included as optimization parameters for the recuperated ORCs. The UA -values are calculated by discretizing the heat exchangers and summing the UA -values for all control volumes

$$UA = \sum_{j=1}^n \frac{\dot{Q}_j}{\Delta T_{lm,j}}, \quad (5)$$

where \dot{Q}_j is the heat transfer and $\Delta T_{lm,j}$ is the log mean temperature difference in control volume j .

30 control volumes are used in the discretization of the boiler and 10 are used for the condenser and the recuperator. In order to ensure approximately linear temperature profiles in all control volumes, saturated liquid and vapor points are always located on a control volume boundary. The calculation of the UA -value of a heat exchanger requires that the mass flows and the inlet and outlet states are known. Iterative solving procedures are therefore implemented to find the process states which satisfy the UA -values of the heat exchangers. The condensation pressure is solved according to the UA -value of the condenser and the hot outlet temperature (T_7) is solved according to the UA -value of the recuperator. For the boiler the hot fluid outlet temperature is solved to satisfy UA_{boil} when there is no limit on the hot fluid outlet temperature, and the expander inlet temperature is used when there is a limit on the hot fluid outlet temperature. Table 4 lists the optimization and solving parameters used in the UA -value based models.

Table 4. Optimization and solving parameters used in the four different UA -value based models

	Optimizing parameters	Solving parameters
Simple ORC, no lower limit for $T_{hf,o}$	P_5 , T_5 and UA_{boil}	P_{cond} and $T_{hf,o}$
Recuperated ORC, no lower limit for $T_{hf,o}$	P_5 , T_5 , UA_{boil} and UA_{cond}	P_{cond} , $T_{hf,o}$ and T_7
Simple ORC, lower limit for $T_{hf,o}$	P_5 and UA_{boil}	P_{cond} and T_5
Recuperated ORC, lower limit for $T_{hf,o}$	P_5 , UA_{boil} and UA_{cond}	P_{cond} , T_5 and T_7

3. Results

3.1. Waste heat recovery from exhaust gases

Results from the pinch point based optimizations of recuperated ORCs (*PP recu*) and UA -value based optimizations of recuperated ORCs (*UA recu*) are displayed in Table 5. For each fluid the UA -value based optimizations use a total cycle UA -value which is equal to that found in the pinch point based optimization. For MDM an additional optimization is carried out, which uses the total cycle UA -value obtained for Toluene (*PP recu*)¹ in order to provide a comparison of toluene and MDM based on equal total cycle UA -value.

The net power outputs obtained in the UA -value based optimizations are higher than the net power outputs obtained in the pinch point based optimizations. This is because, the values of the pinch points chosen for the pinch point based optimizations are not optimized, while the UA -value distribution is optimized in the UA -value based optimizations.

For the pinch point based optimizations, the resulting UA -values for each fluid vary depending on the heat transfer rate and the mean temperature difference of the heat exchangers. The total cycle UA -values are the following for toluene, MDM, benzene and ethanol respectively: 3035 kW/K, 4889 kW/K, 2985 kW/K and 2818 kW/K. Toluene, benzene and ethanol obtain similar total cycle UA -values, whereas the total cycle UA -value for MDM is significantly larger than for the other fluids. It is primarily the UA -values of the boiler and the recuperator which contribute to the larger total cycle UA -value for MDM.

Figure 2 displays the solutions resulting from the pinch point based optimizations for toluene (a-c) and MDM (d-f) in Ts -diagrams and $\dot{Q}T$ -diagrams. For toluene, the heat transfer in the boiler is occurring at high temperature differences at the inlets and outlets, while the temperature difference is small around the saturated liquid point (pinch point location); see Fig. 2 b). For MDM, the temperature difference in the boiler is close to the minimum value during the entire preheating stage, while during evaporation the temperature difference is large; see Fig. 2 e). A large part of the heat transfer in the boiler for MDM is thereby occurring at a low temperature difference. This means that a high UA -value is necessary in order to achieve the desired pinch points. For the recuperator the higher amount of heat transferred for MDM necessitates higher UA -values.

Table 5. Results from optimizations of exhaust gas WHR with $T_{hf,i} = 285 \text{ }^\circ\text{C}$ and $T_{hf,o} = 160 \text{ }^\circ\text{C}$

Fluids	\dot{W}_{NET} [kW]	P_5 [bar]	T_5 [$^\circ\text{C}$]	P_{cond} [bar]	T_7 [$^\circ\text{C}$]	ΔT_{boil} [$^\circ\text{C}$]	ΔT_{cond} [$^\circ\text{C}$]	ΔT_{recu} [$^\circ\text{C}$]	UA_{boil} [kW/K]	UA_{cond} [kW/K]	UA_{recu} [kW/K]
Toluene											
<i>PP recu</i>	5862	7.91	217	0.081	45.8	10	5	5	767.7	1929	337.5
<i>UA recu</i>	5973	8.96	227	0.087	49.0	3.6	6.9	6.4	1144	1576	313.7
MDM											
<i>PP recu</i>	6246	4.78	224	0.013	45.3	10	5	5	1699	1931	1259
<i>UA recu</i>	6247	4.85	225	0.012	45.7	9.8	4.9	5.6	1733	1961	1196
<i>UA recu</i> ¹	5600	5.09	227	0.018	67.7	11.1	12.2	20.9	1453	1069	511.6
Benzene											
<i>PP recu</i>	5844	13.0	235	0.25	46.1	10	5	5	751.6	1928	305.1
<i>UA recu</i>	5966	14.6	246	0.27	49.7	3.5	6.8	6.9	1129	1578	277.4
Ethanol											
<i>PP recu</i>	5767	29.4	273	0.18	46.5	10	5	5	774.0	1927	117.2
<i>UA recu</i>	5876	31.3	283	0.20	46.1	5.9	6.9	2.7	1076	1572	170.0

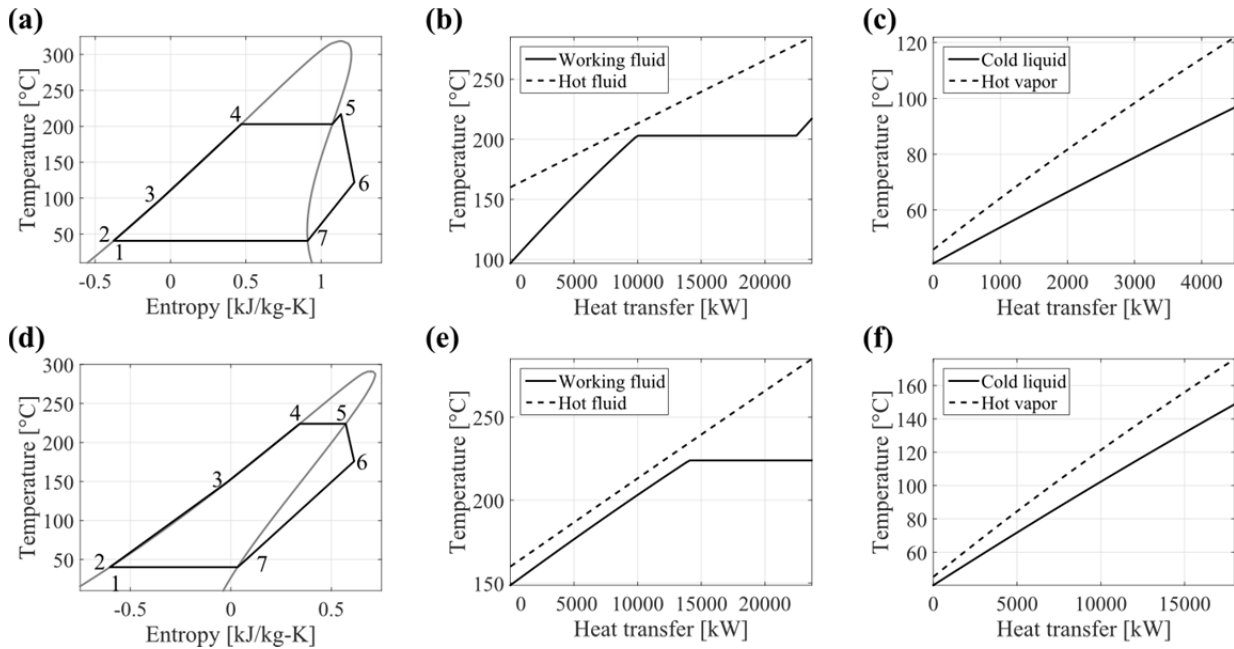


Fig. 2. Results from the pinch point based optimizations of toluene (a-c) and MDM (d-f) illustrated in a) and d) Ts -diagrams, b) and e) boiler $\dot{Q}T$ -diagrams and c) and f) recuperator $\dot{Q}T$ -diagrams

This comparison of toluene and MDM illustrates that by defining the minimum pinch point temperature as the fixed parameter it is possible that some fluids are allowed to have much higher UA -values than others. The results from the pinch point based optimizations suggest that the net power output of MDM is 6.6 % larger than toluene. However, when compared based on the same total cycle UA -value toluene reaches 6.7 % higher net power output compared to MDM¹.

3.2. Waste heat recovery from scavenge air cooler

Table 6 lists the results from optimizations of a scavenge air cooler WHR system. In addition to optimization results for recuperated ORCs (*PP recu* and *UA recu*), the table displays results from pinch point based optimizations of simple ORCs (*PP simp*) and UA -value based optimizations of simple ORCs (*UA simp*). In this case the hot fluid outlet temperature is not limited. The net power

¹ This optimization for MDM uses the total cycle UA -value obtained from the *PP recu* optimization for toluene

outputs obtained from the pinch point based optimizations are the same whether a recuperator is used or not. The use of a recuperator simply decreases the heat input to the cycle which results in an increase in the thermal efficiency of the cycle. This is in correspondence with the results obtained by Maraver et al. [1], Dai et al. [2] and Li et al. [3]. For the UA -value based optimizations, cyclopentane does not benefit from having a recuperator. Solkatherm (SES-36) and MM achieves slightly higher power outputs (less than 0.6 % higher) with a recuperator. The results exemplify that different conclusions are obtained depending on which optimization approach is applied. Table 6 also shows results for a hot fluid with the same mass flow and hot fluid inlet temperature as used for the exhaust gases (see Table 1), but with no constraint on the outlet temperature (MDM²). For these hot fluid conditions, MDM obtains a significant net power increase of 244 kW when a recuperator is used. This corresponds to a relative net power increase of 4.3 %.

Table 6. Results from optimizations of scavenge air WHR with $T_{hf,i} = 191$ °C

Fluid	\dot{W}_{net} [kW]	P_5 [bar]	T_5 [°C]	$T_{hf,o}$ [°C]	P_{cond} [bar]	T_7 [°C]	ΔT_{boil} [°C]	ΔT_{cond} [°C]	ΔT_{recu} [°C]	UA_{boil} [kW/K]	UA_{cond} [kW/K]	UA_{recu} [kW/K]
c.-pen.												
<i>PP simp</i>	2406	6.14	117	95.2	0.70	-	10	5	-	598.1	1736	-
<i>PP recu</i>	2406	6.14	117	99.6	0.70	43.7	10	5	5	582.4	1688	113.1
<i>UA simp</i>	2564	7.68	128	95.1	0.74	-	2.7	6.9	-	992.6	1391	-
<i>UA recu</i>	2564	7.77	129	97.5	0.74	59.6	2.7	7.8	19.2	979.6	1386	17.43
SES-36												
<i>PP simp</i>	2507	11.4	127	77.3	1.1	-	10	5	-	935.9	1895	-
<i>PP recu</i>	2507	11.4	127	93.8	1.1	44.5	10	5	5	842.3	1771	320.5
<i>UA simp</i>	2629	15.0	141	78.6	1.2	-	5.0	6.7	-	1398	1535	-
<i>UA recu</i>	2637	14.5	140	88.5	1.2	63.0	5.1	6.7	21.9	1344	1506	83.53
MM												
<i>PP simp</i>	2551	1.68	119	72.4	0.11	-	10	5	-	907.3	1896	-
<i>PP recu</i>	2551	1.68	119	98.1	0.11	43.4	10	5	5	735.9	1716	524.4
<i>UA simp</i>	2696	2.30	131	73.9	0.11	-	3.9	6.0	-	1344	1632	-
<i>UA recu</i>	2711	2.29	131	91.5	0.11	64.1	3.7	5.9	25.2	1271	1585	119.5
MDM²												
<i>UA simp²</i>	5620	3.20	203	73.9	0.017	-	21.6	10.1	-	1477	1554	-
<i>UA recu²</i>	5864	3.11	201	135	0.018	75.4	15.3	10.0	29.2	1223	1435	374.1

The recuperator can be beneficial, since it enables optimum usage of the heat transfer area. Without a recuperator, the boiler provides all the heat input to the cycle while the condenser provides all the cooling. By implementing a recuperator the heat loads on the boiler and the condenser are reduced. Whether it is beneficial to move some of the heat load to the recuperator depends on the temperature differences which are available for heat transfer. The heat required to preheat the liquid at the pump outlet (state 2) can be provided from the hot fluid through the boiler or from the vapor at the expander outlet through the recuperator. The boiler and the recuperator require different UA -values to deliver the heat for preheating depending on the temperatures of the hot fluid at the outlet of the boiler and the temperature at the expander outlet.

Figures 3 a) and b) display the Ts -diagrams for cyclopentane and MM resulting from the UA -value based optimizations, where the hot fluid is the scavenge air. Figure 3 c) displays the Ts -diagram for MDM where the hot fluid is the exhaust gases (with no limitation on the exhaust gas outlet temperature). For cyclopentane the temperature at the expander outlet is 66.8 °C and the hot fluid outlet temperature is 95.1 °C for the simple ORC. The heating of the liquid at state 2 can therefore be carried out with a larger temperature difference when the heat is provided by the hot fluid

² For these MDM optimizations the hot fluid characteristics of the exhaust gases are used without the limit on the exhaust gas outlet temperature. The total cycle UA -value is equal to that for toluene (*PP recu*) from Table 5.

compared to when the heat is provided by the hot vapor at the outlet of the expander. It is therefore more effective to transfer the heat through the boiler than transferring it through the recuperator. For MM the expander outlet temperature is 98.6 °C while the hot fluid outlet temperature for the simple ORC is 73.9 °C. In this case it is possible to use the UA -value more effectively by using a recuperator to provide the first preheating.

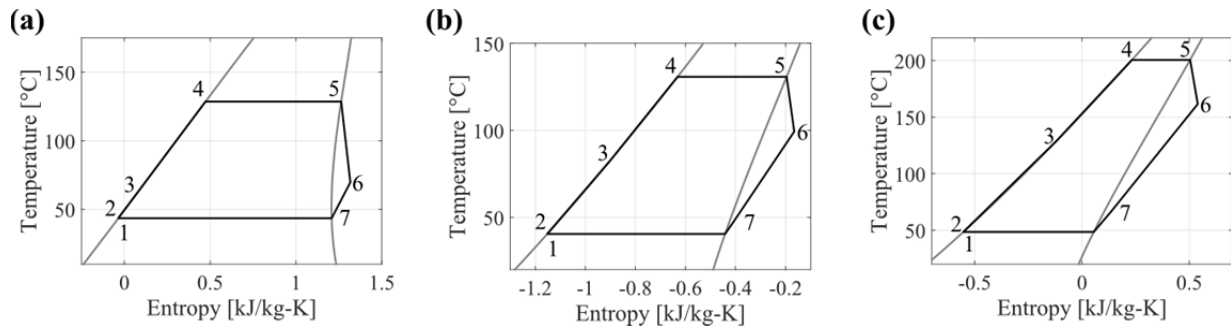


Fig. 3. Ts -diagrams for a) cyclopentane, b) MM and c) MDM resulting from the UA -value based optimizations of the recuperated ORC

It is important to note that in practice it is more expensive to implement a given heat transfer area in an additional heat exchanger (recuperator) than adding the heat transfer area to an already existing heat exchanger (boiler or condenser). The small increases in net power observed for MM and solkatherm when using the recuperator may not justify an implementation of this heat exchanger from an economical viewpoint. For MDM² (exhaust gas WHR with no outlet temperature limit) the benefits of using the recuperator are on the other hand significant. Figure 4 displays $\dot{Q}T$ -diagrams and Ts -diagrams of the optimized ORCs corresponding to the two MDM optimizations listed in Table 6. For the simple ORC the hot fluid outlet temperature is 73.9 °C (see Fig. 4 a)) while the outlet temperature of the expander is 161.3 °C (see Fig. 4 c)). By comparing the $\dot{Q}T$ -diagrams of the boiler and the condenser for the simple ORC in Fig. 4 a) and b) with the $\dot{Q}T$ -diagrams of the boiler, condenser and recuperator for the recuperated ORC in Fig. 4 d), e) and f), it is clear that the heat load is moved from the condenser and the boiler to the recuperator when a recuperator is included in the cycle. For the simple ORC the heat transfer in the boiler and the condenser make out 54 % and 46 % of the total heat transfer in the cycle, while the boiler, condenser and recuperator in the recuperated ORC transfer 44 %, 35 % and 21 % respectively. Furthermore, the total heat transferred in the two heat exchangers in the simple ORC is 14 % larger than the heat transfer in the three heat exchangers in the recuperated ORC.

This example illustrates that in high temperature applications, where there is no limit on the hot fluid outlet temperature, the benefits of the recuperator should not be evaluated solely on a pinch point based optimization. Instead, the UA -value based optimization is a more suited methodology, since it considers the heat exchange process in greater detail with limited added effort.

3.3. Waste heat recovery from jacket water cooler

Table 7 lists the results for the jacket water WHR optimizations of simple ORCs. The UA -value based optimizations are carried out using the total cycle UA -values obtained from the pinch point based optimizations for R32/R134a.

In the pinch point based optimizations the specified pinch points are not reached in the boiler for R32 and for R32/R134a. These optimizations are limited by the constraint on the vapor quality at the expander outlet because R32 is a wet fluid. In order to find suitable solutions that respect the vapor quality constraint, the boiler pinch point temperature is kept as a free parameter, while the expander inlet temperature is optimized. R134a and R245fa are not constrained by this since R134a is close to an isentropic fluid and R245fa is a dry fluid (the expansion does not enter the two-phase region).

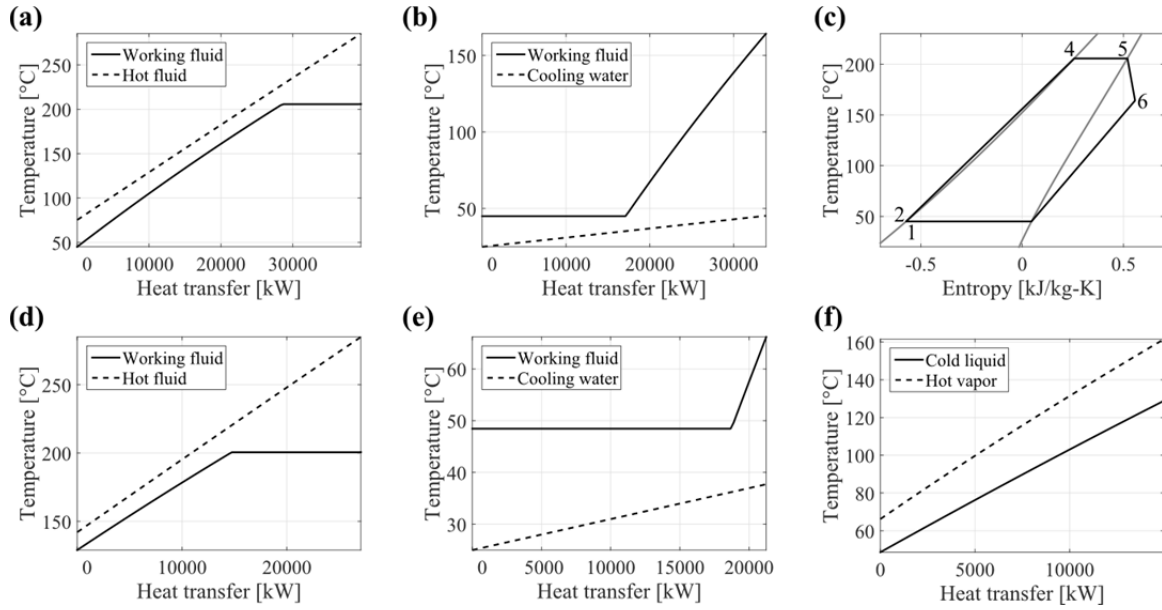


Fig. 4. a) $\dot{Q}T$ -diagram of the boiler, b) $\dot{Q}T$ -diagram for the condenser and c) Ts -diagram for MDM in the simple ORC, and d) $\dot{Q}T$ -diagram of the boiler, e) $\dot{Q}T$ -diagram of the condenser and f) $\dot{Q}T$ -diagram of the recuperator for MDM in the recuperated ORC

Table 7. Results from optimizations of jacket water WHR with $T_{hf,i} = 90$ °C and $T_{hf,o} = 75$ °C

Fluid	\dot{W}_{net} [kW]	P_5 [bar]	T_5 [°C]	X [-]	P_{cond} [bar]	ΔT_{boil} [°C]	ΔT_{cond} [°C]	UA_{boil} [kW/K]	UA_{cond} [kW/K]
R32									
PP simp	570.3	43.3	80	-	22.1	14.3	5	497.2	1210
UA simp	679.5	47.9	88.8	-	21.2	10.3	3.4	749.1	1557
R134a									
PP simp	643.7	20.7	80	-	8.8	10	5	603.6	1206
UA simp	741.3	24.1	87.2	-	8.8	4.0	4.8	1069	1237
R32/R134a									
PP simp	689.7	28.7	80	0.33	11.3	10.1	5	645.7	1660
UA simp	741.0	25.0	87.3	0.035	9.1	4.2	5.0	1072	1234
R245fa									
PP simp	652.4	5.82	69.9	-	2.1	10	5	585.2	1206
UA simp	755.6	7.09	88.8	-	2.1	3.0	5.6	1240	1066

A fluid comparison based on the pinch point based optimizations suggests that the mixture R32/R134a reaches the highest net power output. The total UA -value for the mixture is however also larger than those of the pure fluids. This is due to an alignment of the temperature profiles of condensation which minimizes the mean temperature difference of heat exchange [4-5]. In the UA -value based optimizations all four fluids are assigned the total cycle UA -value; i.e. the one resulting from the pinch point based optimization of R32/R134a. The results from the UA -value based optimizations suggest that there is no benefit of using the mixture R32/R134a compared to pure R134a. Instead the best performance is achieved when R245fa is used as the working fluid.

This analysis exemplifies how a comparison of pure fluids and mixtures based on pinch points can give misleading results, since it does not take into account the larger UA -value required for mixtures when the temperature profile of the condensing working fluid is matched with the cooling water. When the pure fluids are allowed to have the same total cycle UA -value as the mixture, then R245fa reaches the highest net power output, and R32/R134a is no longer beneficial compared to pure R134a.

4. Discussion

Both the pinch point and UA -value based optimization approaches have downsides in terms of ensuring equal heat transfer areas of the heat exchangers in the ORCs. Two major disadvantages of the pinch point based approach are that the mean temperature difference can vary for different ORC designs, and that variations in heat transfer properties are not accounted for. The results of this paper illustrate that although the minimum temperature difference is fixed, large variations in UA -values and thereby also mean temperature differences, are possible when the pinch point based optimization approach is used. The UA -value based optimization approach does on the other hand represent a design methodology which enables an evaluation of ORC designs, where the mean temperature differences do not result in excessively large UA -values. As for the pinch point based approach, the UA -value based approach also has the drawback that it does not take into account variations in the heat transfer performance, i.e. variations in U .

Compared to the pinch point based approach, the UA -value based approach has the advantage that the UA -values are additive, which enables the optimization of the distribution of a total cycle UA -value. It is however important to note that U varies for the different heat exchangers in the cycle, meaning that the optimal distribution of UA -value does not represent the optimal distribution of heat transfer area. For example, for the two cases based on WHR from the exhaust gases and the scavenge air cooler, the hot fluid is a gas and it is therefore reasonable to expect that U is lower for the boiler than for the condenser, meaning that more area is required in order to achieve the same UA -value compared to the condenser. Also the total cycle UA -value which is assigned to the cycle is a parameter which affects the results from the UA -value based optimization.

The UA -value based approach does represent an optimization methodology, which is more elaborate than the pinch point based optimization approach. However the UA -value based optimization approach should not be considered as a replacement for detailed optimization methodologies combining the cycle and component design (e.g. [16]), but rather as a methodology used for an initial evaluation and comparison of power cycle designs. Compared to the pinch point based approach the UA -value based optimization approach is preferable for comparisons of ORC designs with different number of heat exchangers and for comparisons of fluids where alignments of the temperature profiles in the heat exchangers are possible.

5. Conclusion

In this work we present an assessment and comparison of a conventional optimization approach and a non-conventional optimization approach applied for ORC optimizations for three cases of marine diesel engine WHR. In the conventional optimization approach the minimum pinch point temperature differences of the heat exchangers in the ORC are kept constant, and in the non-conventional optimization approach the total UA -value assigned to the heat exchangers is kept constant. The study focusses on specific cases where the two optimization approaches results in different conclusions in terms of identifying the best performing working fluid and the best performing cycle configuration (recuperated or non-recuperated).

Results from a pinch point based optimization of toluene, MDM, benzene and ethanol for utilization of exhaust gases at 285 °C (with a minimum exhaust gas outlet temperature at 160 °C) indicates that MDM achieves a 6.6 % higher net power output than toluene. The high performance of MDM is caused by a perfect alignment of the temperature profiles of the exhaust gases and the temperature profile of preheating. When evaluated based on equal UA -values, toluene achieves 6.7 % higher net power output than MDM.

A pinch point based optimization of WHR from the scavenge air cooler (no limit on the scavenge air outlet temperature) indicates that the use of a recuperator does not increase the net power output for cyclopentane, solkatherm and MM. For MDM, utilizing the 285 °C exhaust gases (in this case without a minimum temperature limitation on the exhaust gas outlet) the UA -value based optimization indicated a 4.3 % net power increase when a recuperator is implemented in the cycle.

For the utilization of a 90 °C jacket water stream the pinch point based optimization indicates that a mixture of R32/R134a achieves higher net power output than R32, R134a and R245fa. When the UA -value based optimization is used, R245fa achieves the highest net power output, while the mixture optimization converges toward pure R134a.

The presented results suggest that the UA -value based method, compared to the conventional pinch point based method, is a more consistent approach for comparing working fluids and designs of ORCs, because it ensures that the mean temperature difference of the heat exchangers are comparable for all fluids and cycle configurations.

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Nomenclature

A heat transfer area, m²

h specific enthalpy, kJ/kg

\dot{m} mass flow rate, kg/s

P pressure, bar

\dot{Q} heat transfer rate, kW

s specific entropy, kJ/(kg K)

T temperature, °C

U average overall heat transfer coefficient, kW/(m² K)

\dot{W} mechanical power, kW

x vapor quality

Greek symbols

Δ difference

η efficiency

Subscripts

boil boiler

cond condenser

cool coolant

exp expander

hf hot fluid

i inlet

lm log mean

min minimum

NET net

o outlet

pp pinch point

pump pump

recu recuperator

s isentropic

tot total
wf working fluid

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