Comparison of different optimisation procedures for an organic Rankine cycle based on ship operational profile

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Abstract

At a time of strong challenges in relation to economic and environmental performance, the potential of waste heat recovery for ships fuel consumption has proved a considerable potential.

This paper presents the comparison of four different procedures for the optimisation of an organic Rankine cycle based on an increasing level of accounting of the ship operational profile and on the inclusion of engine control parameters to the optimisation procedure. Measured data from 2 years of operations of a chemical tanker are used for the application of the different procedures. The results suggest that for the investigated case study the application of a optimisation procedure which takes the operational profile into account can increase the savings of the installation of an organic Rankine cycle from 2.8% to 11.4% of the original yearly fuel consumption. The results of this study further suggest that i. simulating the part-load behavior of the ORC is important to insure its correct operations at low engine load and ii. allowing the engine control strategy to be part of the optimisation procedure leads to larger fuel savings than the optimisation of the waste recovery system alone.

Keywords:
Waste heat recovery, Marine Propulsion System, organic rankine cycle, low carbon shipping

Nomenclature

Acronyms

DP	Design point optimisation procedure
DP_+	Design point optimisation procedure (improved)
OP	Operational profile optimisation procedure
OP_+	Operational profile optimisation procedure (improved)
BSFC	Brake specific fuel consumption [g/kWh]

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DLS	Data logging system
EEDI	Energy Efficiency Design Index
ORC	Organic Rankine cycle
SEEMP	Ship Energy Efficiency Management Plan
WHR	Waste heat recovery

Subscripts

d	Design
eva	Evaporator
exp	Expander
i	Inlet conditions
is	Isentropic
ME	Main engine
o	Outlet conditions
prop	Propeller
pump	Pump
rec	Recuperator
S	Shaft
SG	Shaft generator

Variables

Δ_{ml}	Mean logarithmic temperature difference [K]
Δ_{PP}	Pinch point temperature difference [K]
$\dot{m}$	Mass flow [kg/s]
$\dot{Q}$	Heat flow [W]
$\dot{V}$	Volume flow $[m^3/s]$
η	Efficiency
λ	Load
λ_{in}	Max prop. system load for WHR on (cut-in load)
λ_{out}	Min prop. system load for WHR on (cut-out load)

λ_{switch}	Engine operational switch load
C_{exp}	Expander coefficient
F_{cu}	Electric generator copper loss factor
A	Heat exchange surface $[m^2]$
h	Specific enthalpy [J/kg]
P	Power, [kW]
T	Temperature [K]
t	Time [s]
U	Global heat exchange coefficient $[W/m^2K]$

1. Introduction

In a world where trade is at the heart of human development, it is estimated that 80% to 90% of the goods are transported by sea (UNCTAD, 2012). However, the shipping industry is at present subjected to a challenging transition. Fuel prices have increased three-fold compared to the 80's (Mazraati, 2011), and coming restrictions regarding sulphur oxides emissions are expected to further augment fuel prices (DNV, 2012). Furthermore, the recently released regulations on CO_2 emissions from ships (EEDI, SEEMP) have been recently issued will require an additional effort from the industry for reducing its impact on the climate (Devanney, 2011). Several fuel saving solutions for shipping have been subject of research and development under the aforementioned forces. Operational measures include improvements in voyage execution (Armstrong, 2013), engine monitoring (Sala et al., 2011), reduction of auxiliary power consumption, trim/draft optimisation (Armstrong, 2013), weather routing (Shao et al., 2011), hull/propeller polishing (Khor and Xiao, 2011) and slow-steaming (Armstrong, 2013). Design measures can relate to the use of more efficient engines and propellers, improved hull design, air cavity lubrication (Mäkiharju et al., 2012; Slyozkin et al., 2014), wind propulsion (Schwab, 2005), fuel cells for auxiliary power generation (Sattler, 2000), pump frequency converters, cold ironing (Peterson et al., 2009), and waste heat recovery systems (DNV, 2012). This study focuses particularly on waste heat recovery systems, and in particular on organic Rankine cycles.

Despite their high thermal efficiency, Diesel engines waste large amounts of energy to the environment. Part of the heat released to the environment in the form of exhaust gas is recovered to fulfil auxiliary heat demand; this demand is however relatively small and leaves potential for further utilisation of the available waste heat for other purposes (Baldi et al., 2014). In particular, WHR systems for the conversion of waste heat to electric power are widely employed in other industrial sectors and their application to shipping has been extensively studied in the scientific literature. Steam cycles are mostly used when four-stroke engines are employed since they are most suitable for relatively high temperatures of the available waste heat ($\sim$ 320°C , compared to $\sim$ 275°C for two-stroke engines (Mollenhauer and Tschoeke, 2010)).

Theotokatos and Livanos proposed a techno-economic evaluation of the application of a single-pressure steam cycle to bulk carriers (Theotokatos and Livanos, 2013) and to ferries (Livanos et al., 2014). In spite of the lower temperature of the exhaust gas, steam-based Rankine cycles have also been proposed for application on two-stroke engines. Ma et al. evaluated the part-load peformance of a single-pressure design (Ma et al., 2012). Dimopoulos et al. proposed the thermo-economic optimisation of a steam-based WHR system for a containership powered by a two-stroke engine (Dimopoulos et al., 2011, 2012). Grimmelius et al. proposed a modelling framework for evaluating the waste heat recovery potential of Diesel engines and tested it to marine applications (Grimmelius et al., 2010). Steam distribution systems are widespread in the shipping industry, which makes steam the most ready-to-use solution for the application of WHR on ships. Steam based WHR systems for both four- and two-stroke engines are available commercially, among others by MAN, Wärtsilä, Mitsubishi and Alfa Laval. Most of the proposed solutions also involve the use of a power turbine in connection with a turbocharger bypass (Dimopoulos et al., 2011).

In the case of two-stroke engines, however, low exhaust temperature makes it challenging to obtain high efficiency with steam-based cycles, which makes organic Rankine cycles (ORCs) a competitive alternative. ORCs are Rankine cycles where water is substituted by an organic fluid whose evaporation temperature better fits the available heat source; in addition, some organic fluids have a positively-sloped vapour saturation curve, which makes them attractive for avoiding the formation of droplets in the last stages of the expander.

Larsen et al. proposed a methodology for the simultaneous optimisation of the ORC process design layout, working fluid and process parameters depending on the temperature of the heat source (Larsen et al., 2013b); Choi and Kim analysed the performance of a dual-loop ORC system for a medium-sized containership under operational conditions (Choi and Kim, 2013), while Yang et al. analysed the performance at part-load and transient conditions for a larger vessel (Yang et al., 2013). A comparison of conventional steam cycles with ORCs have been proposed by Hountalas et al. (Hountalas et al., 2012), while Larsen et al. also included Kalina cycles in the analysis (Larsen et al., 2013a, 2014). These studies are of particular relevance since two-stroke engines are by far the most employed prime mover in the shipping industry in terms of installed power (Haight, 2012).

With reference to different types of technologies, case studies, and designs, the previously mentioned works witness quite significant possibility for energy saving when WHR systems are employed, ranging from around 1% for single-pressure steam cycles applied to two-stroke engines (Theotokatos and Livanos, 2013) to more complex systems based on ORCs (up to 10% (Hountalas et al., 2012)) or including the cooling systems as a source for waste heat (over 10%

(Dimopoulos et al., 2012)).

However, to the best of our knowledge, only few studies have been published accounting for the complexity of ships operational profiles in the selection of the design. Dimopoulos et al. identified 4 operational conditions, and took them into account in the steam-based WHR optimisation problem (Dimopoulos et al., 2011). Choi et al. analysed the operational profile of a case study and identified 2 main operational conditions of particular relevance for the recovery cycle, and optimised a dual-loop ORC system on those conditions (Choi and Kim, 2013). Kalikatzarakis et al. took the full operational profile complexity into account when optimising an ORC, and showed that different operational profiles have a large impact on the expected economic performance of the system (Kalikatzarakis and Frangopoulos, 2014). In previous work, we proposed the accounting of the operational profile in a feasibility analysis of WHR systems for ships, focusing on the required system performance rather than on how to achieve it in terms of optimal cycle parameters (Baldi and Gabrielii, 2015). None of the studies above, however, explicitly dealt with the comparison of optimising a WHR system at its design point and based on the full operational cycle.

The aim of this paper is to systematically investigate the influence of taking the operational profile into account in the process of optimising WHR design parameters. We aim at achieving our aim by testing various optimisation methods which have different levels of accuracy in accounting for the operational profile of the ship and by the fuel savings achieved by the different optimal systems relate to the chosen optimisation method.

2. Methodology

2.1. Description of the case study

The proposed comparison of different optimisation procedures is applied to a Panamax product/ chemical tanker. The ship is equipped with a data-logging system (DLS) which provides measurements of propulsion power and auxiliary power demand with a 15 minutes frequency. The ship is powered by two MaK 8M32C four-stroke Diesel engines rated 3,840 kW each. The two main engines (ME) are connected to a common gearbox (GB), which in turn is connected to a controllable pitch propeller and to the shaft generator (SG) which provides 60 Hz current to the ship. Auxiliary power during port stays is generated by two auxiliary engines (AE) rated 682 kW each. In this study we only consider operations during sea passages; therefore, AEs are not considered in this study. A conceptual representation of the ship propulsion system is represented in Figure 1.

The main engine power is calculated according to Equation 1.

$$P_{ME} = \frac{\frac{P_{prop}}{\eta_S} + \frac{P_{SG}}{\eta_{SG}}}{\eta_{GB}} \quad (1)$$

Where the variables P and η refer to power and efficiency and subscripts prop and S respectively refer to the propeller and the propeller shaft. P_{SG} and

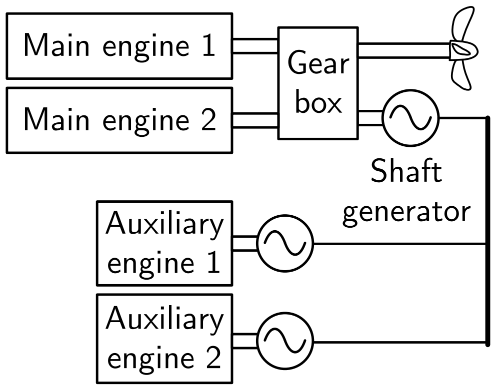


Figure 1: Conceptual representation of ship propulsion system

P_{prop} are available from the continuous monitoring system installed on board; η_S is assumed equal to 0.99 (Shi et al., 2009); η_{GB} is assumed equal to 0.983 as reported by the shipyard where the ship was built. Since on the case study ship the SG is dimensioned for high power demand when unloading the cargo, it often operates at low load. The expression proposed by Haglind (Haglind, 2010) for modelling large ships generator part-load efficiency based on the design point efficiency and the copper loss fraction of the total losses is used:

$$\eta_{SG} = \frac{\lambda \eta_{SG,d}}{\lambda \eta_{SG,d} + (1 - \eta_{SG,d})[(1 - F_{cu}) + F_{cu}\lambda_{SG}^2]} \tag{2}$$

where λ, $\eta_{d,e}$ and F_{cu} respectively represent the load, the design efficiency and the copper loss factor of the electric generator. A value of 95% is assumed for SG design efficiency in accordance with technical specifications, while a value of 0.43 for F_{cu} was assumed (Haglind, 2010). Figure 2 shows the distribution of the combined propulsion and auxiliary power demand for the case study ship over one year of operation.

The engine is modelled using a validated zero-dimensional, single zone model, based on the need to properly simulate the impact of engine operations on efficiency and exhaust flow. The model uses a double Wiebe curve for the modelling of heat release, the Woschni correlation for heat losses, and the Chen correlation for friction modelling, as suggested in available literature on the subject (Asad et al., 2014; Kumar and Kumar Chauhan, 2013; Scappin et al., 2012). Engine validation results are provided in Figure 3. Results related to engine brake specific fuel consumption (BSFC) and turbine outlet temperature are of particular relevance as they influence the efficiency of the combined cycle, and Figure 3 suggests how model output is in agreement with experimental measures (values for the root mean standard deviation are respectively 0.99% and 0.029% for BSFC and turbine outlet temperature). The use of the model allows producing simulating engine output in terms of BSFC, exhaust mass flow and temperature as a function of engine load, as shown in Figure 4.

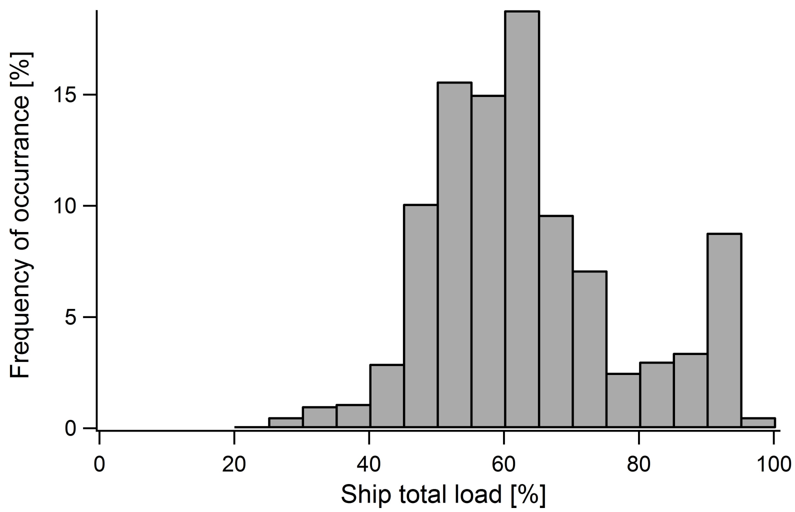


Figure 2: Combined propulsion and auxiliary power load distribution for one year of ship operations

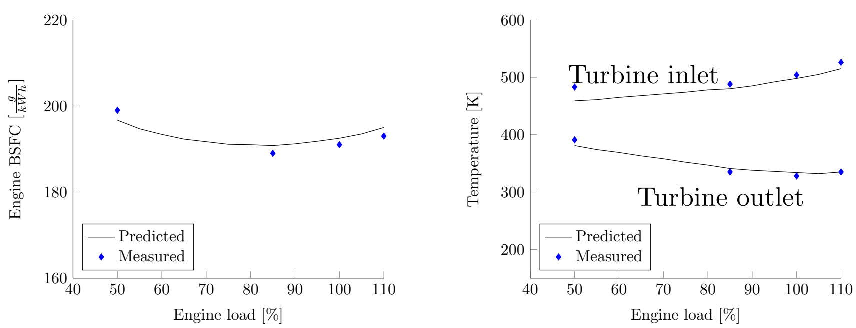
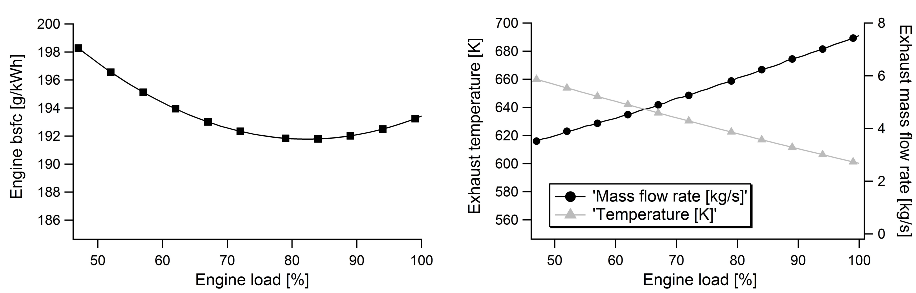


Figure 3: Engine model validation results



(a) BSFC (b) Exhaust, mass flow and temperature

Figure 4: Diesel engine model output for engine BSFC and exhaust gas

It should be noted that since the propulsion system is composed of two equally-sized engines, the definition of the load at which the switch between one-engine and two-engine operation happen is of primary importance. This can be observed in Figure 5 where the engine efficiency and the heat flow in the exhaust gas are plotted versus the load of the whole propulsion system for different switching loads. Under current conditions, where no WHR system is installed, this shift is performed at 47.5% load in order to maximise fuel efficiency keeping a safe engine load margin. When a WHR system comes into the picture, however, the trade-off between more efficient engine operations and a higher energy flow to the WHR system should be analysed more in detail, as discussed by Larsen et al. (Larsen, 2014). Figure 4a shows how decreasing the switch-load involves increasing both the efficiency of the engine and the energy flow in the exhaust gas.

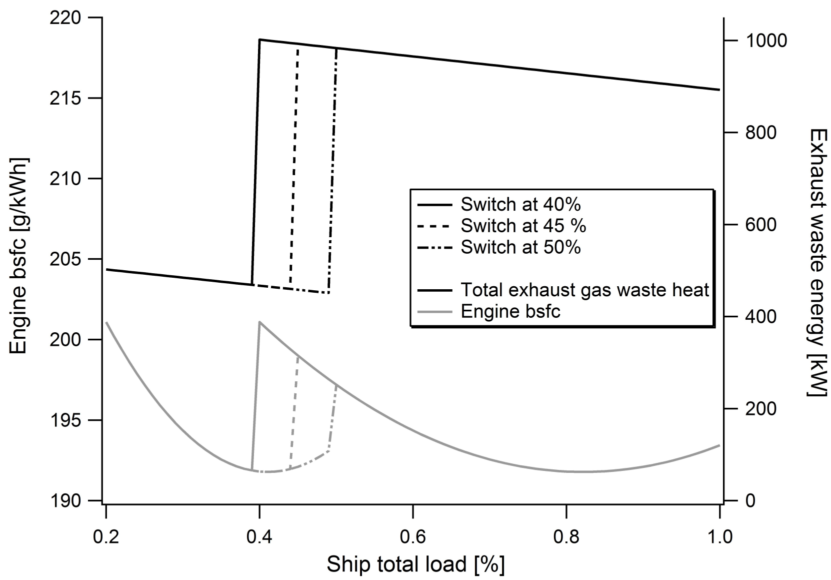


Figure 5: Engine efficiency and total exhaust waste energy as a function of the load for one-to-two engines operation switch

2.2. ORC systems modelling

A general ORC system is composed of 4 types of components: pump, expander, heat exchangers, and electric generator. The typical structure of an ORC system is presented in Figure 6.

The pump was modelled as suggested by (Quoilin et al., 2011), where the coefficients of the regression (see Equation 3) were adjusted to match the higher efficiency due to larger size (see pump characteristics from commercially available centrifugal pumps described by (Manolakos et al., 2001)[31]). The isentropic efficiency of the pump (η_p) relative to its design value ($\eta_{p,d}$) can thus be

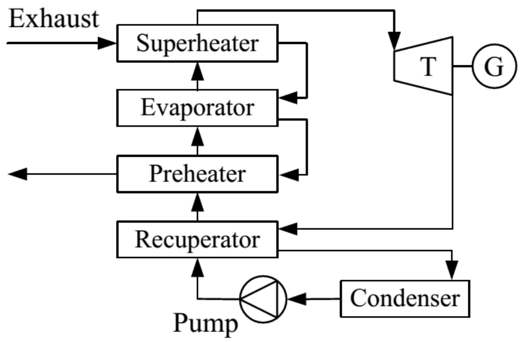


Figure 6: Sketch of the ORC process

calculated as a function of the pump volumetric flow ($\dot{V}$):

$$\frac{\eta_p ump}{\eta_{pump,d}} = a_3 \left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}} \right)^3 + a_2 \left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}} \right)^3 + a_1 \left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}} \right)^3 + a_0 \qquad (3)$$

where constants a_3, a_2, a_1 and a_0 equal to -0.168, -0.0336, 0.6317 and 0.5699.

The expander was modelled as suggested by Schobeiri et al. (M., 2005) who based his work on multistage axial steam turbines, as also reported by Manente et al. (Manente et al., 2013) in their study on a large scale geothermal ORC power plant. The isentropic efficiency of the expander ($\eta_{exp,is}$) relative to its design value ($\eta_{exp,is,d}$) can be defined at any load as follows:

$$\frac{\eta_{exp,is}}{\eta_{exp,is,d}} = 2\frac{\Delta h_{exp,is,d}}{\Delta h_{exp,is}} - \left(\frac{\Delta_{exp,is,d}}{\Delta h_{exp,is}} \right)^2 \qquad (4)$$

where h represents the enthalpy, subscript is and d the isentropic and design point. Δ represents the difference between inlet and outlet. The relationship between expander pressures, temperatures and mass flow rates was modelled according to the law of the ellipse as proposed by Stodola (Cooke, 1985):

$$C_{exp} = \frac{\dot{m}_{exp}\sqrt{T_{exp,i}}}{\sqrt{p_{exp,i}^2 - p_{exp,o}^2}} \qquad (5)$$

where C_{exp} is an expander constant parameter.

The product of the global heat exchange coefficient and of the surface of the heat exchangers is calculated at the design point (UA_d) starting from the knowledge of exchangers' temperature profiles as obtained from the optimisation of the recovery cycle and using Equation 6:

$$\dot{Q} = UA\Delta_{ml} \qquad (6)$$

where Δ_{ml} represents the mean logarithmic temperature difference. The efficiency of the heat exchange at part-load was assumed to be a function of the mass flow in the heat exchanger:

$$UA_{eva} = UA_{eva,d} * \left(\frac{\dot{m}_{eva}}{\dot{m}_{eva,d}}\right)^m \quad (7)$$

Different authors provide different estimations for the value of the exponent m. Haglind assumed a value of 0.58 for the HRSG of a gas-turbine based combined cycle (Haglind, 2011). Manente et al. based their results on the use of Aspen ©for the study of the part-load performance of a geothermal ORC (Manente et al., 2013); they assumed values of 0.15 for the preheater and vaporiser and 0.67 for the recuperator. Lee and Kim assumed a value of 0.3 based on studies of a real recuperator for ORC systems (Lee and Kim, 2006). In the present work a value of 0.6 is chosen to represent the behavior of shell-and-tube heat exchangers. In the case of the boiler, the gas side heat transfer coefficient is dominant (Haglind, 2011) and therefore the exhaust gas mass flow is used in Equation 7.

The part-load performance of the ORC depends on the applied control strategy; in the present case a sliding-pressure mode was adopted. The part-load evaporation pressures are thus governed by the Stodola equation (Eq. 5), by the heat transfer processes and by the pump characteristic curve. The latter component is here equipped with a variable frequency motor. This feature allows to investigate different operational modes, for example, keeping the turbine inlet temperature constant; however, results suggested that keeping the boiler exhaust gas outlet temperature constant, at the minimum allowed temperature (160°C), lead to the highest combined cycle work outputs. More importantly, this strategy ensures that sulphuric acid condensation in the heat exchangers is effectively prevented, particularly at low loads.

2.3. Optimisation procedures

In this work we compare four optimisation procedures for the selection of the optimal design parameters for an ORC system applied to the case study ship: two based on the performance of the system at design point (DP,DP_+) and two based on a detailed description of the operational profile (OP,OP_+). The generic optimisation procedure can be summarised in the following steps:

1. Initial conditions for exhaust gas mass flow rate and temperature are fixed
2. The optimisation procedure is initiated by assigning a tentative value to each optimisation parameter. The number of optimisation parameters depends on the chosen optimisation procedure. The optimisation parameters employed in each procedure are summarised in Table 1, while boundary values for each parameter are provided in Table 2.
3. The performance of the ORC is calculated at design point allowing the calculation of heat exchangers parameters (UA value) and the expander characteristic performance.

4. The objective function which outputs the fuel consumption of the Engine+ORC system is calculated. The objective function involves the calculation of the efficiency of the whole system at different loads depending on the optimisation procedure. The loads at which the objective function is calculated for each optimisation procedure are summarised in Table 1. When more than one propulsion system load is taken into account, the contribution of each load to the objective function is weighted depending on how frequently the ship is operated at that specific load based on the operational profile shown in Figure 2.
5. The genetic algorithm iterates trying to minimise the objective function.

The optimised system is finally tested for the case study vessel. The expected yearly fuel consumption over one year of ship operations is calculated according to the following equation:

$$m_{fuel,tot} = \sum_i BSFC_{tot,i} t_i P_i \quad (8)$$

where the subscript i refers to the i-th propulsion system load and $BSFC_{tot}$ represents the load of the combined propulsion system.

The four alternative optimisation procedures are defined with an increasing degree of required computational effort and are summarised in Table 1.

The DP (design point) optimisation represents the "baseline" type of optimisation procedure. According to a DP optimisation procedure the objective function is only calculated at the propulsion system's design point. The ship's operational profile is not taken into account. This optimisation procedure only requires one cycle calculation for each evaluation of the objective function.

The DP_+ represents an improved version of the DP procedure. In this case the objective function is also calculated at part-load for the minimum load of the propulsion system at which the ORC system is required to operate (we fixed this value to 50% load for this optimisation procedure in this study). Designs where the equation system does not converge or converges to a thermodynamically incorrect solution are discarded, while the objective function is still calculated based only on performance at design load. This optimisation procedure only requires two cycle calculations for each evaluation of the objective function, while ensuring the ability of the system to perform at low load.

The OP (operational profile) optimisation also accounts for the performance of the ORC system at part-load in the evaluation of the objective function. In particular, the objective function is based on the ORC performance at loads from 50% to 100% of the original propulsion system with 5% intervals, for a total of 11 cycle calculations per objective function evaluation. The contribution of the fuel consumption for each load to the objective function is weighted based on the measured frequency of ship operations at that specific load.

The range of engine load at which the ORC system is operated is a has a large influence on the whole system efficiency, as previously mentioned and further discussed in the Results section. For this reason we performed an additional optimisation (OP_+), in which these variables (ORC cut-in load, ORC cut-out load, engine switch-load) were also optimised. This optimisation procedure requires 11 function evaluations but involves three more optimisation parameters and therefore requires a larger number of function evaluations in order to converge to an optimal solution.

The following fluids were analysed in this study: 1-butene, benzene, cyclopropane, cyclopentane, ethylbenzene, ethylene, isobutane, isobutene, isohexane, isopentane, octamethyltrisiloxane (MDM), decamethyltetrasiloxane (MD2M), dodecamethylpentasiloxane (MD3M), hexamethyldisiloxane (MM), neopentane, propylene, propyne, R134a, R218, R227EA, R236EA, R236FA, R245fa, R365MFC, RC318, toluene, water, cis-2-butene, m-xylene, n-butane, n-decane, n-dodecane, n-heptane, n-hexane, n-pentane, trans-2-butene. These fluids were selected based on previous work for high-temperature ORCs (Lai et al., 2011; Larsen et al., 2013a). Six of these fluids showed promising performance for the specific temperature range and where therefore shown in the results: R236ea, R245fa, MM, MDM, benzene, toluene and cyclopentane.

<table>
<tr><th>Procedure name</th><th>Loads in f_{obj}</th><th>Parameters</th></tr>
<tr><td>Design Point (DP)</td><td>100%</td><td rowspan="3">Design evaporation pressure (p_{ev})
Recuperator $\Delta T_{PP,rec}$
Evaporator $\Delta T_{PP,eva}$
Fluid</td></tr>
<tr><td>Design Point (DP_{+})</td><td>100%, 50% (check)</td></tr>
<tr><td>Operational Profile (OP)</td><td>100% to 50%, 5% intervals</td></tr>
<tr><td>Operational Profile plus (OP_{+})</td><td>Variable range</td><td>Design evaporation pressure
Recuperator Pinch point ($\Delta T_{PP,rec}$)
Evaporator pinch point ($\Delta T_{PP,eva}$)
Fluid
Engine operational switch load (λ_{switch})
Max prop. system load for WHR on (cut-in load, λ_{in})
Min prop. system load for WHR on (cut-out load, λ_{out})</td></tr>
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Table 1: Summary of the analysed optimisation procedures

Parameter name	Unit	Range
p_{ev}	bar	1 - 0.9p_{crit}
$\Delta T_{PP,rec}$	K	10 - 250
$\Delta T_{PP,eva}$	K	10 - 250
λ_{switch}	%	27.5 - 57.5
λ_{in}	%	70 - 100
λ_{out}	%	55 - 30

Table 2: Boundary values for optimisation parameters

3. Results

Figure 7 shows a comparison between the four different optimisation procedures and the baseline case for the case study ship. The power generated at each propulsion system load by each optimised design is shown in Figure 8. The cumulative savings versus the propulsion system load generated by the different optimised designs are shown in Figure 9. Finally, the results of the economic analysis are presented in Figures 12 and 13. The optimal parameters, together with a summary of systems performance, are displayed in Table 3 for the evaluated optimisation. Table 4 shows the part-load performance of the optimal OP_+ ORC system.

The performance of the DP optimisation procedure leads to estimated fuel savings of 7.3% compared to the baseline case with no ORC system installed. This result stems from the inability of the system to be operated when only one engine is running. The importance of low-load operations can be particularly observed in Figure 9where the amount of potential energy recovered that is lost when not operating at low load is shown

The performance of the DP_+ optimal systems brings yearly savings up to 9.9%. The improvement compared to the DP results is mainly connected to the ability of the system to run at lower loads, as ensured by the additional step in the optimisation procedure. This is shown in Figure 8. This advantage overcomes the lower ORC power output available at higher propulsion system load (see also Figure 9) because the ship under study operates at low load for long periods of time, as shown in Figure 2.

The performance of the OP optimised design showed no significant difference compared to the DP_+ design. The two optimisation procedures lead to the same set of optimal parameters and, consequently, to the same performance.

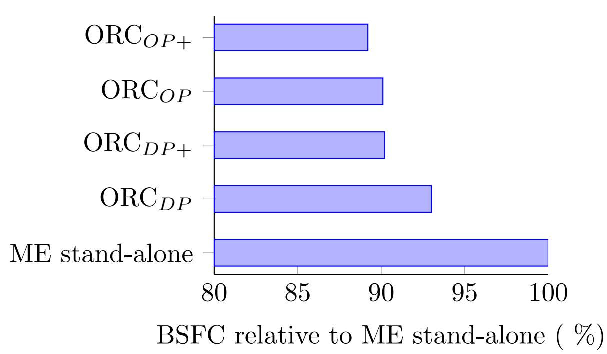

Figure 7: Total BSFC for the engine+ORC system relative to engine-alone baseline system

The first three optimisation procedures refer to the assumption of maintaining the existing approach of switching from one- to two-engines operation at 47.5% of the original propulsion system load and on the WHR system being designed for operating between 50% and 100% of such load. However, these three

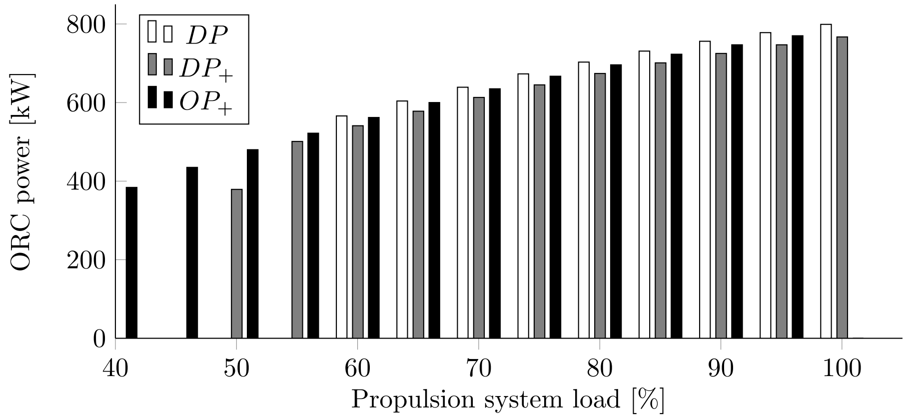

Figure 8: ORC outlet power versus whole propulsion system load for the different optimised systems

values have a large influence on the performance of the combined engine+WHR system (see Figure 10). It can be noted that the difference between the systems can be substantial (the yearly fuel consumption reduction can range from 6.7% to 9%); in addition, it appears that there is no well-defined trend in the influence of these parameters on the ORC system performance. We therefore concluded that the inclusion of these variables in the optimisation procedure should be investigated.

The results of the OP_+ optimisation procedure show that by manipulating the aforementioned engine control parameters it is possible to achieve savings of up to 10.8% from a value of 9.9% when they are not taken into account. This result is obtained for cut-in load, cut-out load and switch load respectively set at 95%, 40% and 32.5% (See Table 3. Note that the values for λ_{switch} and λ_{out} differ because a minimum difference between the two must be kept in order to prevent oscillations in the control system when the WHR is started). It should be noted that this improvement in the ORC performance is achieved without no additional investment cost when compared to the results of the DP and DP_+ procedures.

The results presented in the Figures 7 to 10 refer to the use of the fluid that provides the lowest combined fuel consumption in each case. Figure 11 shows the combined efficiency relative to baseline in the OP_+ optimal case for the 7 fluids showing the highest performance in terms of reduction of yearly fuel consumption. These results suggest that apart from cyclopentane, also benzene is a promising fluids to be used in this specific case.

The net present value (NPV) and the payback time (PBT) of the OP_+ optimised ORC system are shown in Figures 12 and 13 as a function of the expected fuel prices and specific investment cost, which was based on the investigations proposed by Quoilin et al. (Quoilin et al., 2013). For the NPV a time horizon of 10 years and an interest rate of 10% have been selected. It can be noticed that

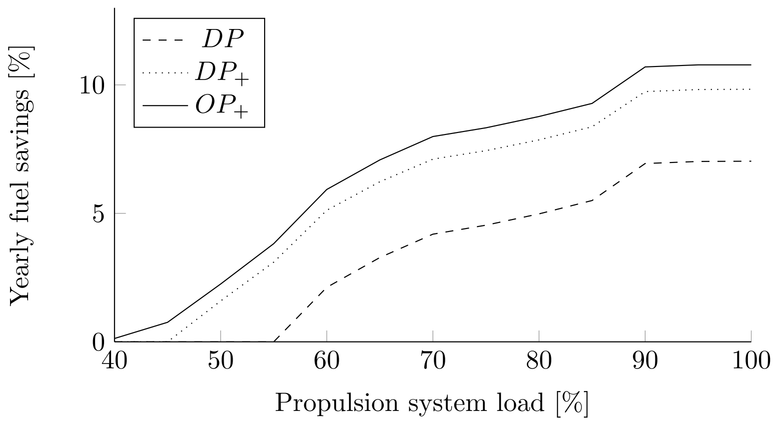

Figure 9: Cumulative fuel yearly savings for different optimisation procedures compared to the baseline

the NPV is positive for all scenarios, regardless the fuel price and the required investment cost. The payback time is in line with what normally estimated as an acceptable payback time for retrofitting projects (DNV, 2012), ranging from 1.5 to 5.2 years respectively in the best and worst case scenario.

4. Discussion

In this paper, we investigated the optimisation of an ORC for the recovery of waste heat on board a product-chemical tanker. The aim of this paper was to investigate the influence of taking the operational profile into account in the process of optimising WHR design parameters. In order to fulfil this purpose, we proposed various optimisation methods and compared the extent of the potential fuel savings. Additionally, we investigated how additional optimisation parameters related to the behavior of the combined engine and ORC system influence the optimisation procedure.

The main contribution of this paper can be identified in showing the potential of including the operational profile in the optimisation of ORC-based WHR systems. In particular, our results suggest that:

- Ensuring efficient operations at low load is a good compromise for increasing the reliability of the optimisation while not significantly increasing the computational effort. An efficient ORC optimisation procedure should therefore also test the ORC performance at the lowest load at which the propulsion system is expected to operate for a significant amount of time.

- The operational parameters related to the combined engine and ORC system (in this case cut-in, cut-out and switch load) are important for the optimisation of the combined system and should therefore be taken into account in the optimisation procedure.

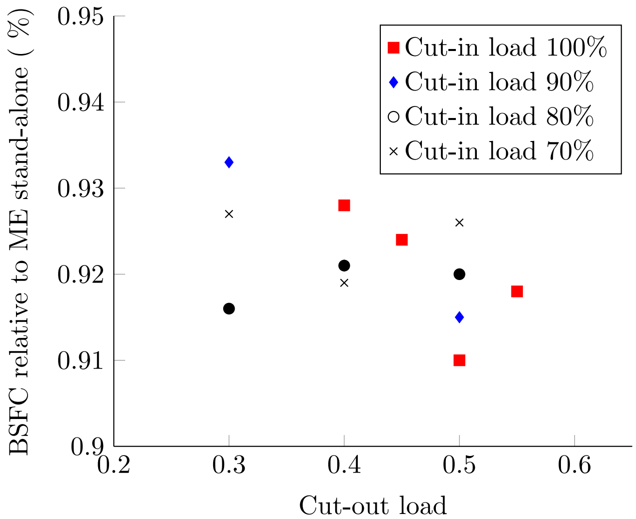

Figure 10: Influence of the cut-in and cut-out load of the ORC on the total BSFC for the engine+ORC system

The comparison of the optimisation procedures suggest that improvements in ORC system efficiency can be achieved when moving from an approach only based on design point performance (DP) to one that also takes into account part-load performance (DP_+,OP,OP_+). In the case studied in this paper, this lead to an increase from 7.0% to 10.8% yearly savings. In particular, savings could be improved from 7.0% to 9.0% using the DP_+ approach, which requires only one additional system simulation per evaluation of the objective function in the optimisation in order to ensure efficient operations at low load. This result has a twofold explanation. On the one hand, the switch between one- to two-engines operations brings a discontinuity in the boundary conditions for the ORC (exhaust mass flow rate and temperature) which the ORC system cannot handle if it has not been included in the design process. On the other hand, the relatively large amount of time the ship spends at low load increases the importance of the ORC power output in these conditions.

The complexity of the operational profile led us to test an optimisation procedure where the performance of the ORC at part-load was included in the evaluation of the objective function (OP). This required a larger number of system simulation per objective function evaluation (10 compared to 2 in the DP_+ case). Surprisingly, however, this led to the same optimal cycle as evaluated with the DP_+ procedure. This result can be explained by the monotonic dependence of the ORC system power output on propulsion system load. This result suggests that the use of a DP_+ procedure can be a good compromise between efficiency and computational time when designing ORC systems for WHR in shipping. This result partly confirms the validity of the approach presented by Choi and Kim (Choi and Kim, 2013), who identified two main operational

Variable	Unit	Optimisation procedure		
		DP	DP_+ and OP	OP_+
Engine switch load	[%]	47.5	47.5	32.5
ORC cut-out load	[%]	-	50	40
ORC cut-in load	[%]	100	100	95
ORC fluid			cyclopentane	
ORC des. evaporation pressure	[bar]	34.4	24.5	33.7
ORC des. evaporation temperature	[C]	218.1	193.6	216.7
ORC condensation pressure	[bar]	0.42	0.42	0.42
ORC des. working fluid flow rate	$[\frac{kg}{s}]$	4.25	4.34	4.08
ORC des. power	[kW]	799	767	763
ORC des. efficiency	[%]	30.7	26.6	30.0

Table 3: Optimal design parameters for the ORC WHR system using the OP and DP optimisation procedures

Variable	Unit	40% load	60% load	80% load	95% load
Power output	[kW]	387	563	697	763
Cycle efficiency	[%]	30.2	31.5	31.9	30.0
Evaporation pressure	[bar]	18.3	25.1	30.6	33.7
Evaporation temperature	[C]	176	196	210	217
Working fluid mass flow	$[\frac{kg}{s}]$	2.2	3.1	3.7	4.1

Table 4: Operational parameters and performance for the OP_+ optimised ORC system at part-load

modes of the original propulsion system based on the ship's operational profile and used them as the base for the optimisation of the WHR system.

In presence of two engines in the original propulsion systems, the switching load between one- and two-engine operations has an effect on both the efficiency of the Diesel engines and of the WHR system. We therefore tested the effect of taking it into account in the optimisation procedure. In addition, the minimum and maximum load at which the ORC system is required to be able to operate (cut-in and cut-out load) were included in the OP_+ procedure. This lead to an increase of the yearly fuel savings from 9.0% to 10.8% for the case study vessel. On the one hand, this improvement is achieved at the cost of a higher computational effort. On the other hand, being designed for a lower maximum load, the OP_+ optimised system is expected to be require a lower capital investment.

In addition to their importance in relation to the choice of the optimisation procedure, the results presented in this paper also show that the best efficiency for the combined Diesel engine+ORC WHR system is obtained when the load of the switch from one- to two-engines operations is set at only 32.5%. This result confirms what already observed by the authors in a previous study (Larsen et al., 2015), i.e. that when the trade-off between engine and ORC performance is considered, it is more efficient to penalise engine performance in order to

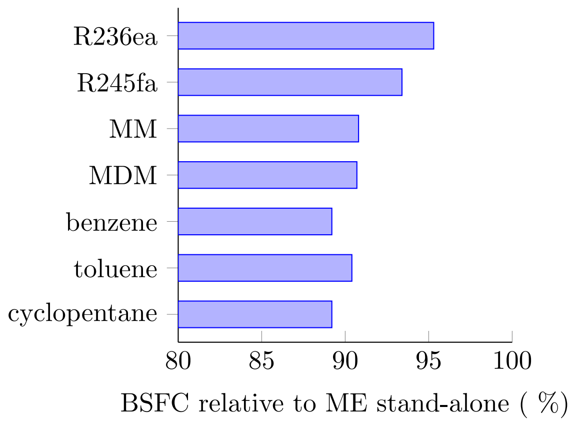

Figure 11: Influence of the working fluid on the total BSFC for the engine+ORC system

improve the power output of the bottoming cycle.

It should be noted, however, that low-load operations are known to generate higher stress on engine parts, which can cause an increase in maintenance costs and a decrease in engine lifetime. This aspect was not taken into account in the present study and should be a subject of further investigation in the future. Furthermore, the results obtained in this work are based on a number of assumptions:

- The control strategy for part-load operations of the ORC system is chosen to ensure a constant exhaust gas outlet temperature. This might not necessarily lead to the optimal part-load performance, although our initial investigations supported this hypothesis.

- It was assumed that an ORC system cannot be operated at a higher load than the design point. This could be avoided by means, for instance, of a bypass valve on the exhaust gas boiler, or by the utilisation of a different control strategy in the selection of the mass flow rate and evaporation pressure of the ORC system.

The optimisation of an ORC system at each load should therefore be further analysed in future studies on this subject.

In the case of ORCs the working fluid plays an important role. On the one hand, it influences the system's efficiency. On the other hand, organic fluids can be toxic or flammable and their use on board might not always be allowed or advisable. The results of this study confirmed what was observed in previous studies (Larsen et al., 2013b) identifying cyclopentane and benzene as the most suitable fluids for the application of ORCs to medium temperature heat sources (Lai et al., 2011; Larsen et al., 2013b).

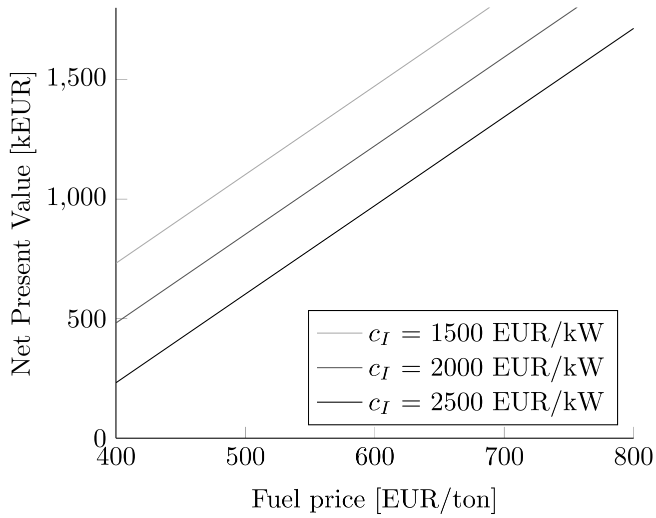

Figure 12: Expected net present value of the optimised ORC system versus fuel price for different specific investment costs

5. Conclusion

In this study we proposed the comparison of four optimisation procedures for the design of organic Rankine cycles for recovering waste heat from marine Diesel engines, where the operational profile of the existing system is taken into account with an increasing degree of accuracy. The four procedures were tested on a case study, a chemical tanker with two four-stroke engines rated 3840 kW each installed.

The results of the study suggest that accounting for the operational profile can lead to savings that would otherwise be lost if a simply design point-based optimisation was employed. Estimated savings ranged from 7.0% to 9.0% when increasing the level of accounting on the operational profile in the definition of the objective function; when parameters related to the interaction between the engine and the ORC system (engine load switching between one- and two-engines operation, minimum and maximum propulsion system load at which the ORC system is required to operate) were included in the optimisation procedure, the expected savings increased to 10.8%. In addition to the higher fuel savings, the proposed optimised system is designed for a lower maximum power output, therefore being smaller in size and less expensive to build.

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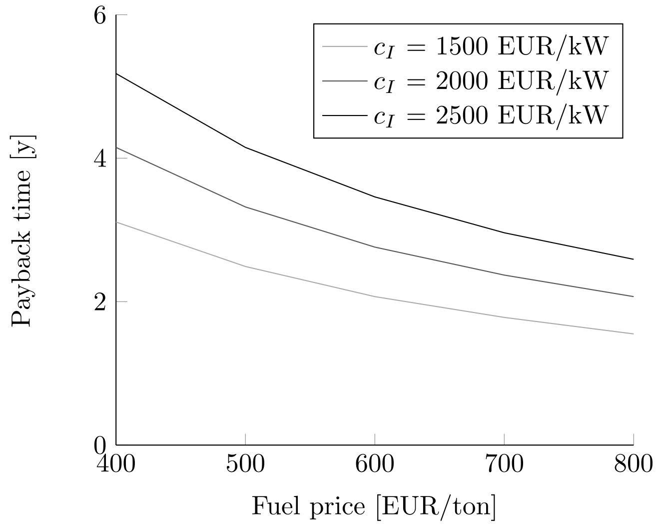

Figure 13: Expected payback time of the optimised ORC system versus fuel price for different specific investment costs

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