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Design of organic Rankine cycle power systems for maritime applications accounting for engine backpressure effects

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HIGHLIGHTS

- A method to optimize organic Rankine cycles for maritime applications is described.
- The method accounts for the backpressure impact on the performance of the engine.
- Neglecting the boiler design constraints can lead to an overestimation of the savings.
- The optimal system design is dependent on the engine sensitivity to the backpressure.

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ABSTRACT

The installation of an organic Rankine cycle unit on the exhaust line of a marine engine imposes an increase in the backpressure on the engine, resulting in a decrease of the engine performance and a variation of the available waste heat. In this paper, a method is presented for the optimal design of organic Rankine cycle power systems for waste heat recovery in marine applications. The method is based on the use of performance maps for the engine and numerical models for the organic Rankine cycle unit and the waste heat recovery boiler, thereby enabling consideration of the effect of the increased backpressure on the performance of both the main engine and the organic Rankine cycle unit. The method is evaluated on a hypothetical containership fuelled by liquefied natural gas. The results of the study indicate that the overall system fuel consumption can be reduced by 0.52 g/kWh to 1.45 g/kWh by allowing higher backpressure levels on the engine. In addition, the results of the study indicate that for a fixed power output of the organic Rankine cycle unit, a reduction of the space requirement for the waste heat recovery boiler by up to 35% can be attained when increasing the maximum allowed engine backpressure from 3 kPa to 6 kPa.

1. Introduction

As confirmed in the study by Bouman et al. [1], the installation of waste heat recovery (WHR) units is one of the most promsing solutions to decrease the environmental impact of shipping. The prospects for WHR arise because nearly half of the fuel energy content is released to the environment as waste heat. This heat can be harvested in WHR units capable of generating heating, cooling or electricity [2].

Recent evaluations indicated the organic Rankine cycle (ORC) power systems as a suitable WHR system for on board installation [3]. The ORC technology is based on the Rankine cycle but uses an organic compound as working fluid, leading to higher conversion efficiencies when harvesting heat from low temperature heat sources [4]. The comparison studies carried out by Larsen et al. [5] and Andreasen et al. [6] showed that the installation of ORC units on board vessels can lead to higher power productions and increased off-design performances, compared to the use of steam Rankine cycle (SRC) units.

Previous studies addressing the optimal design of ORC units for marine applications focused on the identification of the most suitable working fluid [7], on the optimization of the unit's design given the ship sailing profile [8], on ensuring stable operation of the unit through the use of optimized control strategies [9], and on the definition of methods to carry out preliminary evaluations of the techno-economic feasibility of the installation [10].

Nonetheless, when evaluating the prospects for installing an ORC-based WHR unit on board a vessel, it is important to consider that this

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Nomenclature

Acronyms

ECA	Emission control area
IMO	International Maritime Organization
LNG	Liquefied natural gas
NO_x	Nitrogen oxides
Ntp	Number of tubes per pass
Ntr	Number of tube rows
ORC	Organic Rankine cycle
OTB	Once-through boiler
SFC	Specific fuel consumption
SO_x	Sulphur oxides
SRC	Steam Rankine cycle
WHR	Waste heat recovery

Symbols

l	Length, m
P	Pitch, m/ pressure, kPa
V	Volume, m^3
$\dot{W}$	Electrical or mechanical power, kW

Greek symbols

η	Efficiency
Δ	Difference

Subscripts and superscripts

exp	Expander
gear	Gearbox
gen	Generator
l	Longitudinal
net	Net
p	Pump
pp	Pinch point
sw	Seawater
t	Tube/transversal

will result in an increased complexity of the overall machinery system, higher risk hazards due to possible interaction between the unit's working fluid and the engine, and a decrement of the engine performance because of the increased backpressure [11]. The decrement of the engine backpressure due to the increased backpressure is documented by both numerical [12] and experimental [13] works.

There are only a few previous studies considering the interaction between the additional backpressure and the ORC/SRC unit design and its impact on the engine performance. All but one of these studies are related to the automotive sector, or consider stationary engines. Katsanos et al. [14] investigated the implementation of Rankine-based WHR units for truck applications and estimated that the installation of a SRC bottoming cycle would result in an increase of the main engine backpressure by 2.2 kPa. As a consequence, the authors suggested that the WHR unit would have a limited impact on the engine performance. Di Battista et al. [15] discussed the effects of the pressure losses produced by an ORC-based power unit mounted on the exhaust line of a turbocharged IVECO F1C engine, operated on a test bench and concluded that the use of plate heat exchangers could lead to a backpressure increase exceeding 25 kPa. Yamaguchi et al. [16] investigated two different boosting strategies to counterbalance the increased backpressure on the exhaust line of a six-cylinder heavy duty diesel engine and concluded that the installation of an ORC unit could lead to an improvement in fuel economy of 2.6%.

Only the work of Michos et al. [11] addressed the maritime sector. They numerically investigated the performance of advanced turbocharging techniques against the backpressure caused by fitting an ORC unit in the exhaust line of a 1.5 MW high-speed diesel engine, used as a generator set in maritime applications. The authors of this study focused on the assessment of the engine performance as a function of the backpressure caused by the ORC unit. However, the additional backpressure supplied to the engine due to the installation of the ORC unit was assumed, rather than estimated based on the design of the WHR heat exchanger.

This paper presents a method to design ORC power systems for WHR in maritime applications by accounting for the effect of increased backpressure on the performance of both the main engine and the ORC unit. This is accomplished by combining the use of numerical models for the ORC unit and the WHR boiler, and performance maps describing the behavior of the engine as a function of the additional backpressure caused by the ORC unit.

The major novel contribution of this paper to state-of-the-art is that the design of the WHR boiler is considered when estimating the additional backpressure to the ship engine caused by the installation of the ORC unit. The incorporation of the WHR boiler design in the performance analysis of the system comprising the ship's main engine and the ORC unit makes it possible: i) to estimate accurately the ORC power production; ii) to quantify the space requirements for the WHR boiler, and iii) to quantify the relationship between the backpressure supplied to the engine and the space requirements for the WHR boiler.

Previous works did not fully capture the interconnection between the additional backpressure on the engine and the optimal design of the WHR unit. Katsanos et al. [14] numerically estimated the additional backpressure caused by installing a SRC on the exhaust line of a truck engine, but did not account for the effects of the additional backpressure on the exhaust gas mass flow rate and temperature, i.e., no engine performance map was considered. Michos et al. [11] numerically investigated the variation of the efficiency of a four-stroke marine engine due to increasing backpressure levels caused by an ORC unit, but did not address the design of the WHR boiler causing the additional backpressure, that is, Michos et al. [11] did not address the relationship between the pinch point temperature difference in the WHR boiler and the pressure drop caused by the WHR boiler, which may result in the consideration of infeasible WHR boiler designs and/or incorrect system performance estimations (see Section 3). In addition, Michos et al. [11] considered the use of an intermediate oil loop between the exhaust gases and the WHR boiler of the ORC unit, whereas no oil loop is considered in this work. Finally, to the best of the authors' knowledge, no previous study addressed the relationship between the backpressure supplied by the WHR boiler and its space requirements.

The method described in this work enables the estimation of the power output of an ORC unit installed on the exhaust line of a marine engine, the engine fuel penalty arising due to the increased backpressure to the engine itself, and the space requirement for the WHR boiler. Therefore, it supports both researchers and industry in the development of future waste heat recovery units tailored for maritime applications.

The paper is structured as follows. Section 2 explains the applied methods. The results are presented in Section 3 and discussed in Section 4. Finally, the conclusions are outlined in Section 5.

2. Methods

The proposed methodology is derived by integrating engine performance maps, an ORC simulation framework, and a WHR boiler design model. The following subsections present a description of the simulation models, their validation, and the procedure which was utilized to combine them for the purpose of this work.

2.1. Engine performance

This study considered the installation of an ORC unit on board a hypothetical vessel powered by a 23 MW MAN 6S80ME-C9.5-GI engine with part-load tuning. The engine is powered by LNG, and its main characteristics, retrieved from the MAN CEAS calculation tool [18], are listed in Table 1. The data for the engine exhaust gases are reported for the engine operated at full load.

The values in the table refer to the engine operated with a backpressure of 3 kPa [19]. The variation of the performance of the engine as a function of the backpressure supplied to the engine was provided by MAN Energy Solutions [20] and is provided in Table S1 in the supplementary material. An increase of the engine backpressure results in an increase of the exhaust gas temperature and the specific fuel consumption (SFC), and a decrease of the exhaust gas mass flow rate. The impact of varying the engine backpressure was limited to the range from 3 kPa to 6 kPa. The considered two-stroke engine needs to be operated with a maximum allowable design backpressure of 6 kPa, because higher backpressure levels would result in issues in the turbocharging matching procedure [20].

2.2. Organic Rankine cycle model

The present work considered the implementation of an ORC power system harvesting the waste heat from the selected marine engine operated at full load. Given that the engine is powered by LNG, no limit was set on the minimum temperature of the exhaust gases [3], as corrosion by sulphur condensation is not expected.

Both recuperated and non-recuperated configurations were considered, to assess the impact of including this additional component with the attainable ORC power output. In all cases, cyclopentane was selected as the working fluid, because it has been shown to lead to cost-effective ORC power systems for WHR in maritime applications [6,21]. The performance of the ORC unit was estimated using the numerical model previously described in Andreasen et al. [22], while the thermodynamic properties of the working fluid were retrieved from Coolprop 4.2.5 [23]. The ORC net power output was computed as

$$\dot{W}_{\rm net} = \dot{W}_{\rm exp}\eta_{\rm gear}\eta_{\rm gen} - \dot{W}_p - \dot{W}_{\rm p,sw} \quad (1)$$

where $\dot{W}_{\rm exp}$, $\dot{W}_p$ $\dot{W}_{\rm p,sw}$ represent the power of the ORC turbine, pump, and seawater pump. Fig. 1 shows the sketch of the ORC unit. The recuperator was considered only in the relevant cases.

When optimizing the ORC units, the maximum and minimum allowable pressures were set to 3000 kPa and 4.5 kPa, respectively, following the suggestions by Rayegan et al. [24], Dresher and Brüggerman [25], and MAN Energy Solutions [26]. Moreover, in order to avoid problems during operation near the critical point, the ORC unit was limited to a subcritical cycle configuration with a maximum reduced pressure of 0.8.

As the main objective of the work is to identify the impact of the WHR boiler constraints on the performance of ORC units, both the turbine and the pump were modelled with a fixed value of the isentropic efficiency, the recuperator was modelled with a minimum pinch point approach, and a fixed temperature for the working fluid condensation was imposed. The pressure drops in the recuperator and condenser were neglected.

2.3. Waste heat recovery boiler model

The waste heat recovery boiler was modelled as a finned tube once-through boiler (OTB) using the numerical model previously described and validated in Baldasso et al. [21]. The considered OTB has a staggered tubes layout, and features solid fins to enhance the heat transfer coefficient on the exhaust side. Fig. 2 shows the layout of the OTB, while Table 2 reports the heat transfer and pressure drop correlations that were used in the numerical model.

The thermodynamic properties of the exhaust gases were assumed equal to those of air at 100 kPa at the average temperature in the heat exchanger. In addition, a minimum gas velocity of 20 m/s of the exhaust gases in the reduced section between the tube banks was imposed, according to the recommendation of MAN Energy Solutions [19]. This constraint, initially developed for WHR boilers designed for ships operating on heavy fuel oil, minimizes the risk of soot fires in the OTB, because the high gas velocities ensure that the soot particles do not deposit in the tube banks of the OTB.

LNG is known to result in lower soot formation, because soot formation is favored by the presence of carbon – carbon double bonds, which are not present in the methane molecule [27,28]. Therefore, the attained results are expected to be on the conservative side. The minimum distance between the tip of the fins in adjacent rows was set to 6 mm [29]. The volume of the WHR boiler was estimated as follows:

$$V_{OTB} = l_t P_l P_t (Ntp + 1)(Ntr + 1) \quad (2)$$

where l_t, P_l and P_t represent the length of the OTB tubes, and the longitudinal and the transversal pitch. *Ntp* and *Ntr* stand for the number of tubes per pass and the number of tube rows.

2.4. Overall optimization routines

The engine performance map, and the ORC and OTB models were combined to create an optimization framework suitable to evaluate the impact of changing the maximum allowed backpressure level to the engine on the performance of the overall system in terms both of fuel consumption and volume requirements for the OTB. Fig. 3 depicts a sketch of the optimization routine that was used to investigate the impact of varying the allowed engine backpressure on the optimal design of the ORC unit, and on the performance of the overall system. With respect to the pressure drop calculations, the working fluid pressure at the outlet of the boiler was kept constant, while its pressure and temperature at the boiler inlet were updated according to the pressure drop estimated by means of the OTB model. Single-objective optimizations were conducted.

The backpressure level to the engine has a direct impact on the characteristics of the available waste heat, and therefore multiple optimization runs were carried out in order to investigate the optimal design of the ORC unit as a function of the selected backpressure level. For each optimization run, the backpressure level to the engine was fixed, and the designs of the ORC unit and of the OTB were optimized so as: i) to match the predefined pressure drop in the exhaust gases side, and ii) to fulfill a constraint on the minimum allowed boiler pinch point temperature difference (min $\Delta T_{pp,OTB}$). The outputs of the calculation routine were the ORC net power output and the volume of the OTB (V_{OTB}). It was decided to fix the backpressure level to the engine for each optimization run because this leads to two advantages: i) fixing the backpressure to the engine allows to fix the waste heat availability to the ORC unit, meaning that the heat source characteristics do not need to be updated during each iteration of the optimization procedure, and ii) attaining a range of optimal ORC designs as a function of the selected

Table 1
The characteristics of the exhaust gases of the engine 6S80ME-C9.5 at full load.

Parameter	Value
Nominal power output [MW]	23
Nominal speed [r/min]	74.0
Exhaust gas temperature [°C]	251
Exhaust gas flow rate [kg/s]	51.9

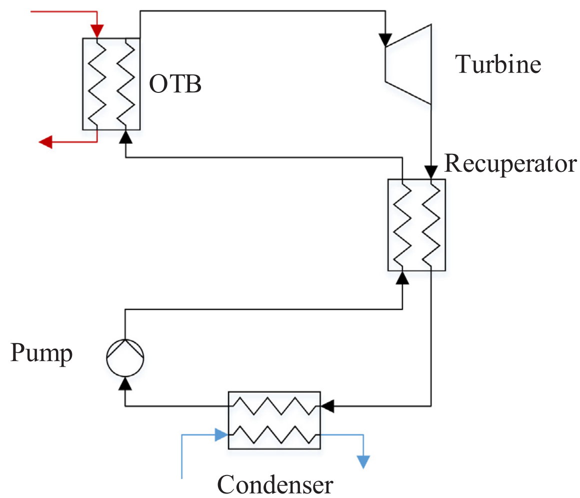

Fig. 1. Sketch of the ORC power system.

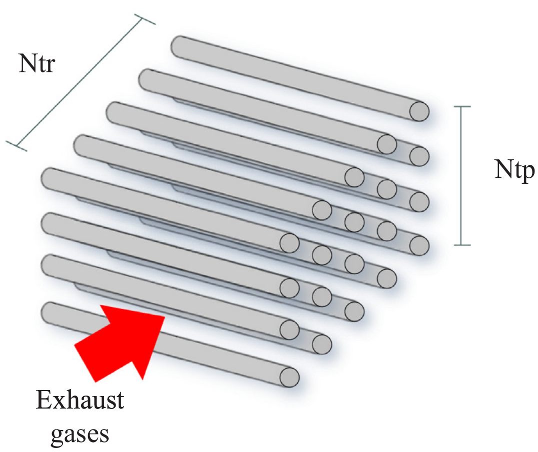

Fig. 2. Layout of the OTB.

Table 2
Heat transfer and pressure drop correlations used in the OTB model.

	Correlation
Gas side	
Heat transfer coefficient	ESCOA [30]
Pressure drop	ESCOA [30]
Fin efficiency	Weierman [31]
Fluid side	
Heat transfer coefficient (single-phase)	Gnielinski [32]
Heat transfer coefficient (two-phase)	Shah [33]
Pressure drop (single-phase)	Kern [34]
Pressure drop (two-phase)	Friedel [35], Rouhani and Axelsson [36]

backpressure allows to have a clear understanding of the relationship between the considered variables (backpressure, ORC power output, OTB volume).

In order to evaluate the impact of including the detailed OTB model in the overall optimization procedure, additional simulations were carried out where the OTB boiler calculation was by-passed and the feasibility of the ORC designs was checked only by evaluating the minimum pinch point temperature of the heat transfer process.

According to the recommendations from MAN Energy Solutions, it was assumed that the engine exhaust piping accounts for a pressure drop of 1.5 kPa [19]. Therefore, for every backpressure level imposed on the engine, the maximum allowed pressure drop of the exhaust gases in the OTB was computed as

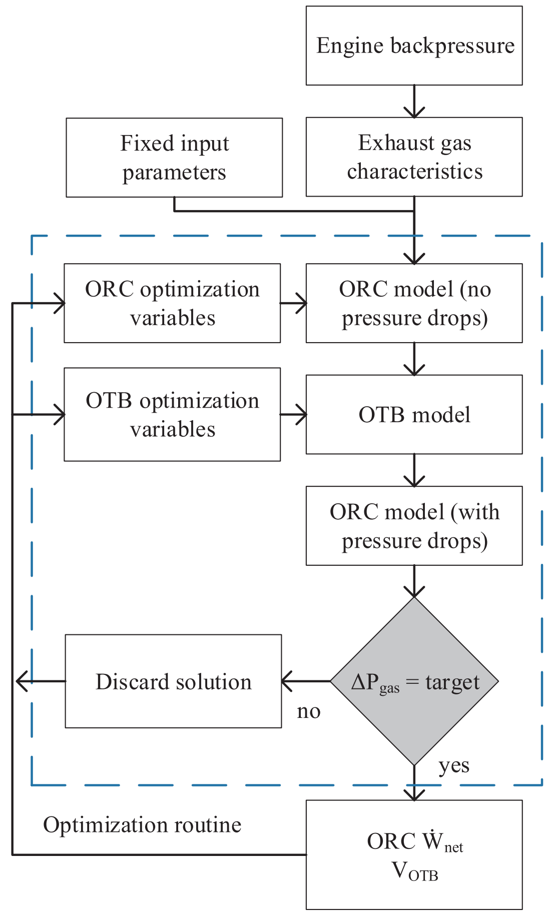

Fig. 3. Sketch of the ORC optimization routine.

Table 3
Fixed input parameters in the optimization procedure.

Parameter	Value
Electrical generator efficiency [%]	98
Gearbox efficiency [%]	98
Seawater pump efficiency [%]	70
Organic Rankine cycle unit	
Turbine isentropic efficiency [%]	85 [38,39]
Pump isentropic efficiency [%]	70 [39]
Fluid condensation temperature [°C]	30
Recuperator minimum pinch point [°C]	20
Seawater inlet temperature [°C]	15
Seawater outlet temperature [°C]	20
Once-through boiler/EGR once-through boiler	
Layout	Staggered (equilateral triangle)
Material	Carbon steel
Carbon steel thermal conductivity [W/ m K]	48
Tube thickness [mm]	2.0

$$\Delta P_{OTB,gasside} = \Delta P_{engine} - 1.5\text{kPa} \tag{3}$$

Tables 3 and 4 show the list of the fixed input parameters and the optimized variables which were considered in this study. The ranges of the geometrical parameters for the OTB fins were retrieved from ESCOA [30], while the allowed tube diameters/lengths were attained from Coulson et al. [37]

The overall optimization procedure was used for two purposes: i) to investigate the variation of the maximum ORC power production as a function of the backpressure level to the engine; and ii) to assess the minimum volume requirement for the OTB boiler as a function of the backpressure level to the engine and of the ORC power output.

In the first case, the ORC net power output ($\dot{W}_{net}$) was selected as the

Table 4
Optimized variables and their considered ranges.

Variable	Lower bound	Upper bound
Turbine inlet pressure [kPa]	100	0.8 P_{crit}
ORC superheating [°C]	5	90
ORC mass flow rate [kg/s]	1	60
OTB tube inner diameter [mm]	21.4	216
OTB superheater tube inner diameter [mm]	21.4	216
OTB tube length [m]	0.6	7.16
OTB fin height [mm]	6.4	31.8
OTB fin thickness [mm]	0.9	4.2
OTB fin spacing [mm]	3.6	25.6
OTB transversal pitch [mm]	42.85	114.3

objective for the optimization, and constraints were imposed on the exhaust gases pressure drop across the OTB and on the minimum pinch point temperature in the OTB. The optimization problem was therefore set as:

$$\begin{aligned} &\mathit{Maximize} \quad \dot{W}_{\mathrm{net}} \\ &\mathit{Subject\ to} \quad \begin{cases} \Delta P_{OTB,gasside} = \Delta P_{engine} - 1.5\,kPa \\ \Delta T_{pp,OTB} \geq Min\ \Delta T_{pp,OTB} \end{cases} \end{aligned} \tag{4}$$

The ORC designs attained by carrying out these optimizations give an indication of the impact of constraining the maximum allowed backpressure supplied to the engine on the power production attainable by installing a WHR unit.

In the second case, the OTB volume (V_{OTB})was selected as the objective for the optimization, and constraints were imposed on the minimum ORC power output and on the OTB pinch point and pressure drops:

$$\begin{aligned} &\mathit{Minimize} \quad V_{OTB} \\ &\mathit{Subject\ to} \quad \begin{cases} \Delta P_{OTB,gasside} = \Delta P_{engine} - 1.5\,kPa \\ \Delta T_{pp,OTB} \geq Min\ \Delta T_{pp,OTB} \\ \dot{W}_{\mathrm{net}} \geq \dot{W}_{\mathrm{net,target}} \end{cases} \end{aligned} \tag{5}$$

The ORC designs attained by carrying out these optimizations give an overview on the relationship between ORC power production and volume requirements for the OTB, as a function of the maximum allowed backpressure supplied to the engine.

The overall performance of the system comprising the ship engine and the ORC unit was evaluated by means of the overall system specific fuel consumption (SFC_{system}). This represents the specific fuel consumption of the system comprising both the engine and the ORC unit, and was calculated as follows:

$$SFC_{system} = \frac{SFC_{engine} \cdot \dot{W}_{engine}}{\dot{W}_{engine} + \dot{W}_{net}} \tag{6}$$

where $\dot{W}_{engine}$ and $\dot{W}_{net}$are the power outputs of the engine and the ORC unit, respectively. $SFC_{engine}$is the engine specific fuel consumption, which was varied as a function of the additional backpressure supplied to the engine (see Table S1 in the supplementary material).

The evaluations of the OTB volume requirements and the overall performance of the system were carried out only for the non-recuperated ORC unit, because the results indicated that the use of a recuperator resulted in a negligible increase of the maximum attainable power production, while requiring the use of a more complex and thus more expensive unit layout (see Section 3.1).

The overall system-specific fuel consumption was calculated also by using the engine data described in the work by Michos et al. [11], thereby demonstrating that the proposed method is capable of replicating previously published results. When carrying out the simulations with the data from the previous study, the pressure drops across the exhaust line pipes were set to zero in order to be consistent with the approach used in Ref. [11].

The optimization routines were carried out using a combination of pattern search and particle swarm optimizers, available in the Matlab optimization toolbox [40]. The use of evolutionary algorithms for the optimization routines follows the recommendations of Astolfi et al. [41], indicating their suitability to find global optima in design optimizations of ORC units. The solution attained through the evolutionary algorithm (particle swarm) was further refined by running the pattern search optimization routine using the solution of the evolutionary algorithm as a starting point. The particle swarm optimizer was run for 100 generations using a swarm size of 5000 individuals, while the pattern search optimization routine was executed for 500 iterations.

2.5. Validation of the numerical models

The numerical models used to perform design performance estimations for the ORC unit were previously verified [22]. The works from Larsen et al. [7] and Walraven et al. [42] were used for validating the calculations of the cycle first and second law efficiencies, respectively. The results of the validation indicated that the simulation code is able to predict the cycle's first and second law efficiencies with a maximum relative deviation of 3.3%.

The waste heat recovery boiler was previously validated in Baldasso et al. [21]. The procedure to size the boiler was verified with an example from Verein Deutscher Ingenieure [43] with a relative deviation of 0.66% and 0.75%, respectively, for the estimated heat transfer coefficient and the heat transfer area. For the gas side, the heat transfer coefficient estimation procedure was verified against examples from Weierman [31], indicating a deviation of 4.6%, due to roundings and unit conversion approximations. The estimated pressure drops indicated a relative deviation within 20%, compared to the estimations from Mon [44] (see Fig. 6 in Ref. [21]).

3. Results

3.1. Impact of backpressure constraint on the design of the organic Rankine cycle unit

Fig. 4 shows the maximum ORC power output attainable as a function of the engine backpressure and the minimum acceptable boiler pinch point temperature. Detailed information about the design of the optimized ORC units are provided in Table S2 in the supplementary material. The results are attained for the non-recuperated ORC and indicate that given a specific value for the engine backpressure, the ORC power output tends to increase linearly when decreasing the minimum acceptable pinch point in the OTB. Nonetheless, this appears

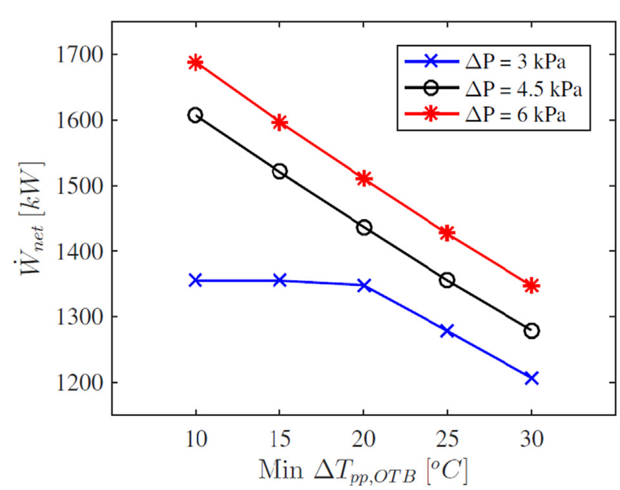

Fig. 4. Impact of backpressure and pinch point constraint on the attainable ORC net power output.

not to be true when considering an engine backpressure of 3 kPa. In this case, after some threshold value, the ORC power output remains constant. This happens because it is not possible to find a suitable design for the OTB that matches the backpressure constraint with the minimum allowed pinch point temperature. This suggests that the choice of the heat exchanger technology to be used for the OTB and the set of constraints for the exhaust gases (maximum allowed pressure drop and minimum velocity) result in a minimum attainable boiler pinch point. In particular, the minimum pinch point temperature in the OTB that could be attained with the considered set of constraints is equal to 18.8 °C.

Secondly, it is noted that by allowing a higher backpressure on the engine, it is possible to design ORC units with higher net power outputs. On average, the ORC net power output increases by 6% when increasing the allowed backpressure by 1.5 kPa. This results from the fact that an increase by 1.5 kPa in the engine backpressure leads to an increase in the exhaust gases temperature by around 5 °C, and a reduction of the exhaust gases mass flow rate by around 1.1%.

Fig. 5 depicts the maximum attainable ORC power output when setting the engine backpressure to 3 kPa. Three cases are investigated: i) non-recuperated ORC; ii) recuperated ORC; and iii) non-recuperated ORC simulated without the OTB model. The results indicate that the recuperated and non-recuperated ORC lead to a similar trend in the attainable power output. The use of a recuperated ORC leads to an increase in the attainable power output by 2.1%, when the minimum allowed boiler pinch point is lower than 20 °C. This is because the use of the recuperator allows for decreasing the minimum attainable OTB pinch point temperature from 18.8 °C to 17.4 °C.

On the other hand, significant differences appear when the ORC power output is estimated without accounting for the OTB model. In this case, the ORC power output increases linearly with the minimum allowed OTB pinch point constraint, leading to a considerable overestimation of the attainable power when the pinch point constraint is below 20 °C. In particular, the attainable power output is overestimated by 15%, when considering a minimum pinch point of 10 °C. This happens because when the OTB design is excluded from the analysis, the design constraint on the maximum pressure drop on the exhaust gas side is disregarded. The differences in the estimated power output when the pinch point temperatures are above 20 °C are due to the fact that the working fluid pressure drops in the OTB were set to zero when bypassing the OTB model.

3.2. Impact of backpressure constraint on volume requirements

Fig. 6 shows the impact of the engine backpressure and ORC power output on the minimum required OTB volume for the non-recuperated ORC case. Detailed information about the design of the optimized ORC units is provided in Table S3 in the supplementary material. The results suggest that the OTB volume increases when designing ORC units with higher net power outputs. This increment is nearly exponential and results from the fact that: a) higher power outputs can be attained only by accepting a lower pinch point temperature in the OTB, and b) the velocity of the exhaust gases needs to be reduced in order to keep a constant backpressure on the exhaust gases side. The need to have lower a pinch point in the OTB, and the reduced velocity of the gases, which leads to a lower heat transfer coefficient, results in a more than linear increase in the required heat transfer area. Considering the impact of the imposed gas-side backpressure constraint, it appears that, for a given power output of the ORC unit, allowing higher pressure drops in the OTB results in a sharp decrease in the required OTB volume. In particular, considering an ORC net power output of 1,300 kW, the required OTB volume is of 7.37 m^3, 5.60 m^3, and 4.76 m^3, respectively, for an engine backpressure of 3 kPa, 4.5 kPa and 6 kPa. On a relative basis, the required volume decreases by 24% and 35%, when relaxing the backpressure constraint from 3 kPa to 4.5 kPa and 6 kPa, respectively. This suggests that allowing increased gas-side backpressures allows more compact designs for the OTB. From the engine perspective, increasing the backpressure from 3 kPa to 4.5 kPa and 6 kPa results in an increase of the SFC by 0.19 g/kWh and 0.38 g/kWh, respectively. In relative terms, the engine SFC increases by 0.13% and 0.27% when increasing the backpressure from 3 kPa to 4.5 kPa and 6 kPa, respectively.

3.3. Impact on the overall system performance

Figs. 7 and 8 show the impact of installing the non-recuperated ORC unit on the specific fuel consumption of the combined system. Fig. 7 is based on the engine data that was presented in this paper (see Table S1 in the supplementary material), while Fig. 8 is based on the engine data provided in Michos et al. [11]. Specifically, the plots show the overall system SFC as a function of the engine backpressure and minimum acceptable pinch point temperature in the OTB.

The results shows in Fig. 7 indicate that the installation of an ORC unit in the engine featured in this article leads to a reduction of the overall SFC in the range from 8.1 g/kWh to 9.4 g/kWh. Similarly, Fig. 8 suggests that the installation of an ORC unit in the engine described in Michos et al. [11] results in a reduction of the overall SFC in the range from 19.6 g/kWh to 20.7 g/kWh.

Looking at Fig. 7, it is possible to conclude that when the OTB minimum pinch point is set to 20 °C, there is an almost linear trend between the overall system SFC and the engine backpressure. Such linear trend was also found in the work of Michos et al. [11]. However, the results presented in the work of Michos et al. [11], which were successfully replicated in this work (see Fig. 8), indicated an increase of the overall system SFC when increasing the engine backpressure.

The difference in the trend is not due to the applied method (because the trend of the previous work was replicated using the method presented in this work) and can be explained by the fact that Michos et al. [11] considered a four-stroke auxiliary engine whose specific fuel consumption is more affected by the backpressure in comparison to the two-stroke engine considered in the present work. In addition, the engine turbocharging strategy utilized in Michos et al. [11] leads to lower variations in the exhaust gases temperature as a function of the additional backpressure, making the attainable ORC net power output less dependent on the engine backpressure.

In both cases, the selected backpressure level has a limited impact on the overall system SFC. For example, when considering a minimum pinch point temperature of 20 °C for the OTB, the installation of the ORC unit leads to a reduction of the overall system SFC (compared to the engine base SFC of 141.2 g/kWh) by 5.5%, 5.8% and 5.9%, when setting the engine backpressure to 3 kPa, 4.5 kPa and 6 kPa,

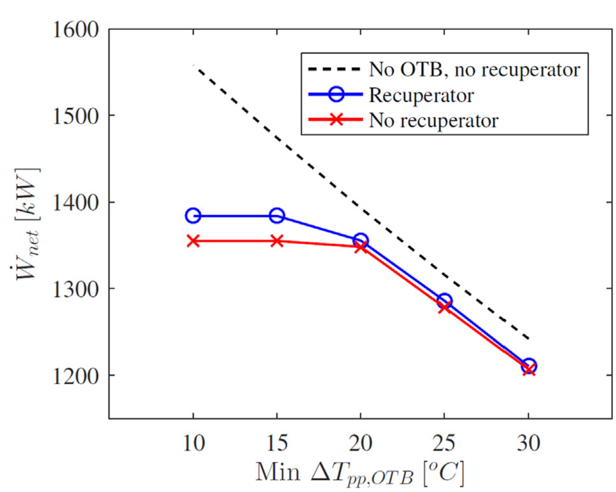

Fig. 5. Impact of using a recuperator on the maximum attainable ORC power output. The engine backpressure is set to 3 kPa.

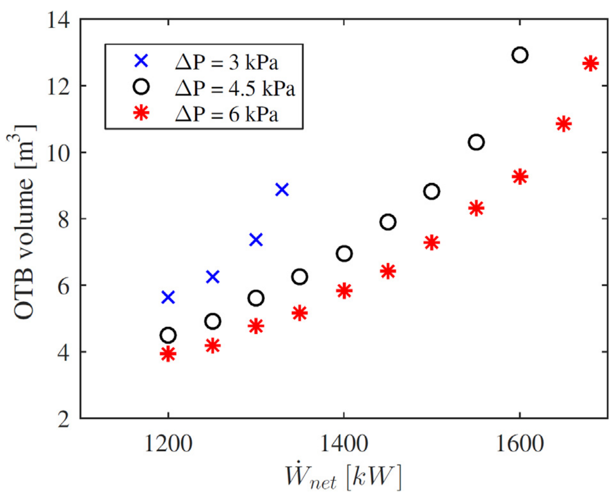

Fig. 6. Impact of backpressure and ORC power output on the minimum OTB volume.

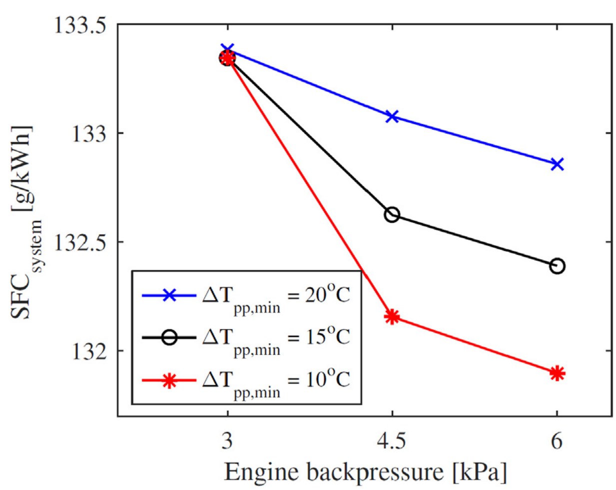

Fig. 7. Impact of backpressure and pinch point constraints on the overall machinery system performance. The engine SFC without the WHR unit is 141.2 g/kWh.

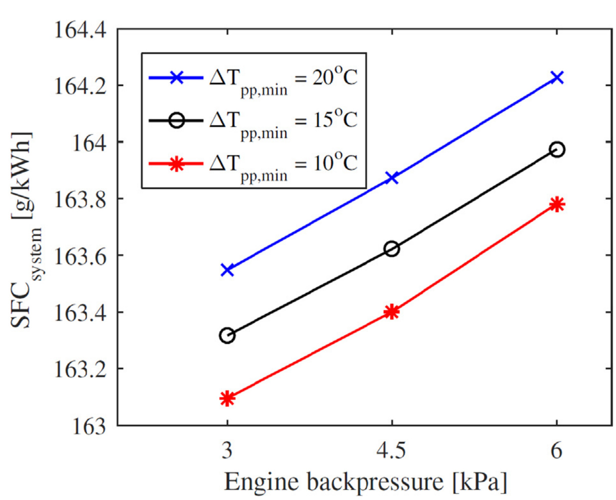

Fig. 8. Impact of backpressure and pinch point constraints on the overall machinery system performance. The results are attained using the engine performance data provided in Michos et al. [11]. The engine fuel consumption without the WHR unit is 183.8 g/kWh.

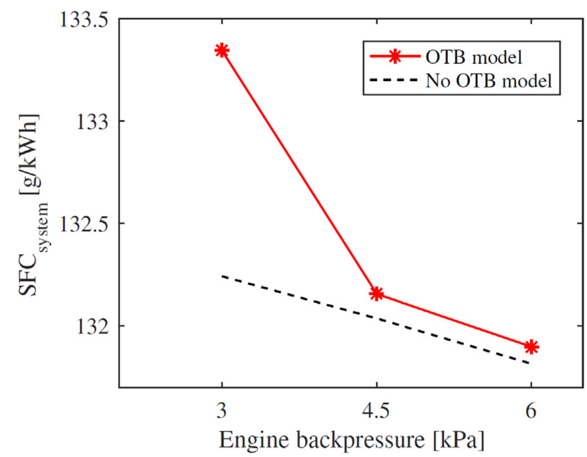

Fig. 9. Estimated overall machinery system performance with and without accounting for the OTB model. The results are presented for the case where the minimum allowed pinch point temperature in the OTB is set to 10 °C.

respectively.

Based on the results shown in Fig. 7, it is possible to conclude that the trends for the cases where the boiler pinch point is constrained to 15 °C and 10 °C show a sharp decrease when increasing the engine backpressure from 3 kPa to 4.5 kPa. These sharp decrements correspond to the sharp increases in the attainable ORC power output, which were reported in Section 3.1 (see Fig. 4).

In particular, the overall system SFC decreases by 0.72 g/kWh and 1.19 g/kWh when relaxing the backpressure constraint from 3 kPa to 4.5 kPa, with a boiler pinch point constraint of 15 °C and 10 °C, respectively. Similarly, relaxing the backpressure constraint up to 6 kPa results in a reduction of the overall system SFC by 0.52 g/kWh, 0.95 g/kWh and 1.45 g/kWh compared to the 3 kPa case, for a minimum boiler pinch point of 20 °C, 15 °C and 10 °C, respectively.

Fig. 9 shows a comparison of the estimated overall system SFC when including the OTB model in the overall methodology, and when estimating the ORC performance just by imposing a boiler minimum pinch point temperature. The presented case is the most extreme when the minimum OTB pinch point is set to 10 °C.

The figure indicates that by not considering the design of the OTB and its constraints, it is possible to overestimate the SFC savings that are attainable by implementing an ORC unit. This happens because, as previously discussed and shown in Fig. 5, the boiler design constraints hinder the attainment of the assumed minimum pinch point temperatures. As a consequence, the ORC net power output is lower compared with that of an approach not considering the design of the OTB and its constraints. In particular, when setting the engine backpressure to 3 kPa, the estimated overall system SFC was estimated to be 133.34 g/kWh and 132.24 g/kWh, when considering the two modelling approaches, respectively. In relative terms, the savings correspond to a reduction by 5.6% and 6.3% of the engine SFC, respectively. This indicates that the implementation of an approach which does not include a suitable OTB model can lead to a noticeable overestimation of the attainable fuel savings.

Fig. 10 depicts the impact of installing the ORC unit with a constrained OTB volume on the overall system SFC. As detailed in Fig. 6, constraining the OTB volume directly impacts the attainable ORC power output, because a larger OTB is capable of extracting higher amounts of heat from the exhaust gases. Fig. 10 indicates that allowing higher volumes for the OTB results in a decrease of the overall system SFC. Considering an engine backpressure level of 6 kPa, the overall system SFC decreases from 133.4 g/kWh to 132.7 g/kWh and 132.2 g/kWh, when the OTB volume is increased from 6 m^3, to 8 m^3 and 10 m^3, respectively.

Moreover, the results shown in Fig. 10 suggest that when

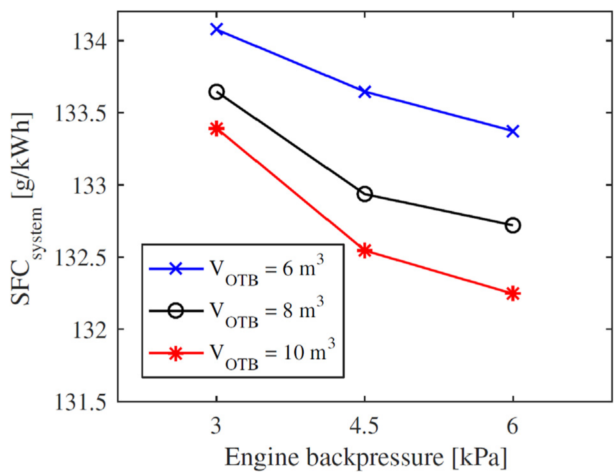

Fig. 10. Impact of backpressure and OTB volume constraints on the overall machinery system performance. The engine SFC without a WHR unit is 141.2 g/kWh.

considering the performance of the overall system, the ORC unit should be designed for the maximum allowed engine backpressure, regardless of the considered volume constraint for the OTB. This is in agreement with the trends identified in Fig. 7, and suggests that for the considered case study, the optimal system performance is attained when designing the ORC unit for the largest allowed engine backpressure (6 kPa).

4. Discussion

The proposed methodology to assess the performance of the system comprising a ship's main engine and an ORC WHR unit was applied both to a two-stroke dual fuel engine and to the four-stroke diesel engine previously investigated in the work of Michos et al. [11]. The results indicate that when considering the two-stroke dual fuel engine, the SFC of the overall system can be minimized by accepting higher backpressure levels on the engine. The opposite result is found when using the engine data for the four-stroke engine. It appears therefore that the optimal design point should be evaluated from case to case, and that the most crucial parameters to consider are the engine sensitivity to the backpressure level and the engine turbocharging strategy, which directly affects how much the ORC power output is dependent on the engine backpressure level.

The ORC volume requirements are an important parameter to consider when designing WHR units for marine applications. The results presented suggest that accepting higher backpressures on the engine allows realizing more compact OTBs. The increased compactness is beneficial, especially because the OTB is generally installed in the exhaust stack where the space availability may be limited or constrained in one of the dimensions (i.e. length or width available for the OTB).

With respect to the OTB design, only finned tube heat exchangers with solid fins and staggered tube layouts were considered in this study. Different layouts of the tubes/fins could be considered, including parallel tube layouts, serrated fins [30] and H-type finned tubes [45]. A dedicated study could be carried out to identify the optimal boiler design that maximizes the heat transfer while maintaining the exhaust gases pressure drops within the considered constraints.

As detailed in the work of Pili et al. [46], the weight of the ORC unit is also an important parameter in defining the economic feasibility of WHR units in the transportation sector. In fact, the weight of the ORC adds to the total transported weight and thus leads to an increase of the overall fuel consumption of the truck/vessel. However, when focusing on the maritime sector, the weight of the ORC can be considered negligible compared to the weight of the ship and its payload. Therefore, it was assumed that the additional weight due to the installation of the ORC unit has no impact on the ship's fuel consumption.

With respect to the working fluid selection, all the evaluations were carried out considering cyclopentane as working fluid. The selection of the working fluid is not expected to have a significant impact on the relationship between the attainable power output and backpressure level supplied to the engine by the ORC unit

Lastly, it should be mentioned that in many cases, intermediate oil loops are used to transfer the heat from the engine exhaust gases to the ORC fluid. In this case, the heat exchanger affected by the backpressure constraint would be the one transferring the heat from the exhaust gases to the thermal oil. Additional studies need to be carried out to quantify the impact of using an intermediate oil loop while allowing for an additional engine backpressure. Nonetheless, in order not to limit the ORC power output, the thermal oil needs to receive the heat from the exhaust gases at a higher temperature compared to that of the ORC fluid, suggesting the need for a larger heat transfer area (because the overall heat transfer coefficient is practically dominated by the exhaust gas side). The engine performance is therefore expected to be affected to a larger extent when increasing the backpressure compared to the case where no intermediate oil loop is used. However, the use of an intermediate oil loop facilitates the control of the minimum temperature in the exhaust gas heat exchanger, thus minimizing the risk of its failure.

5. Conclusions

The present study investigated the optimal design of organic Rankine cycle-based waste heat recovery units for maritime applications accounting for the effect of the increased backpressure on the engine, both on the design of the organic Rankine cycle unit and on the performance of the overall machinery system.

The findings of the study suggest that the combination of the maximum engine backpressure and minimum exhaust gas velocity constraints, result in a constraint in the maximum amount of heat that can be extracted by the waste heat recovery boiler. As a consequence, designing organic Rankine cycle units with a minimum pinch point temperature approach could result in the attainment of unfeasible designs for the waste heat recovery boiler.

When considering the space requirement of the waste heat recovery boiler, the evaluations indicate that, for a given design power output of the organic Rankine cycle unit, the space requirements decreases when allowing higher backpressure levels to the engine.

Analyzing the efficiency of the overall system including the main engine and the waste heat recovery unit, it emerges that it is necessary to include detailed calculations for the waste heat recovery boiler in order to identify the backpressure level to the engine that maximizes the overall performance. The evaluations considering the engine described in this paper indicate that increasing the backpressure level to the engine leads to a reduction of the overall fuel consumption. On the contrary, a minimization of the engine backpressure is required to maximize the efficiency of the system in case the engine described in Michos et al. [11] is considered. The contradiction in findings suggests that the optimal design point is dependent on the engine sensitivity to the engine backpressure level and, hence, should be evaluated for each specific case.

Lastly, the results presented enlightens the importance to consider design approaches including the detailed the design of the waste heat recovery boiler, because the omission of suitable evaluations for such component can result in an overestimation of the attainable fuel savings.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.applthermaleng.2020.115527.

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