Waste heat recovery on liquefied natural gas-fuelled ships

Final project report

Haglind, Fredrik; Baldasso, Enrico; Mondejar, Maria E. ; Pili, Roberto

Publication date:
2021

Document Version
Publisher's PDF, also known as Version of record

DTU Mechanical Engineering
Department of Mechanical Engineering

Waste heat recovery on liquefied natural gas-fuelled ships

Fredrik Haglind, Enrico Baldasso, Maria E. Mondejar*, Roberto Pili

*The co-author Maria E. Mondejar passed away on the 14th of February 2021.

Technical report

Waste heat recovery on liquefied natural gas-fuelled ships

Final project report

Project summary

The project "Waste heat recovery on liquefied natural gas-fuelled ships" aimed at deriving guidelines with respect to the optimal integration of organic Rankine cycle-based waste heat recovery units on board ships powered by liquefied natural gas. The project included the development of numerical models and methods to evaluate the performance of organic Rankine cycles, as well as the realization of an experimental setup at DTU Mechanical Engineering.

In the initial stages of the project the various heat sources available on board were screened in order to identify the most suitable solution to integrate the recovery unit on board. The evaluations indicated that the highest savings could be attained by harvesting the heat of the exhaust gases and that fuel saving up to 10 % could be achieved.

Secondly a novel method to integrate the organic Rankine cycle on board was developed. This method enables to account for the additional backpressure supplied to the engine, which has an influence both on the engine performance and the waste heat availability. The findings suggest that it can be convenient to accept a performance drop in the engine in order to maximize the efficiency of the overall propulsion system.

With respect to the optimal design of organic Rankine cycle for maritime applications, it emerged that units operating for larger amount of time and of larger nominal size lead to reduced payback times. In particular, it was estimated that payback times in the range from 5 to 10 years can be expected when installing organic Rankine cycle units on board vessels. For retrofit installations the availability of space on board is an essential parameter that needs to be evaluated.

A novel concept enabling emission-free power production on ferries during harbor stays was evaluated. The concept, featuring the use of an organic Rankine cycle in combination with a thermal energy storage system, was proved to be technically feasible and economically superior to the use of lithium batteries when considering time scenarios more than 10 years.

Lastly, an experimental test rig was built at DTU Mechanical Engineering, featuring a diesel engine and an organic Rankine cycle equipped with an axial-flow turbine. In particular, an effective method to start the turbine in a safe and efficient manner was proven experimentally. The setup can be used both for teaching and future research works.

Contents

1 Introduction

This report summarizes the findings of the project entitled “Waste heat recovery on liquefied natural gas-fuelled ships”. This project was running in the period from 1st of April 2017 to 31st of December 2020 and the partners of the project were DTU Mechanical Engineering, MAN Energy Solutions, Fjord Line, Alfa Laval, and Lloyd’s Register Marine. Den Danske Maritime Fond, Orients Fond and the project partners funded the project.

1.1 Background

Due to environmental and legislative incentives and low gas prices, gas-fuelled shipping is expected to increase significantly in the coming years. This is also supported by a recent report from DNV GL [1], which states that there are currently 247 liquefied natural gas (LNG)-fuelled ships and 110 LNG-ready ships, excluding LNG carriers, and indicates that these numbers are expected to increase in the near future.

Concurrently with the growing use of LNG as fuel for maritime applications, increasing efforts are devoted to the study and development of waste heat recovery systems, which enable the conversion of the waste heat released by the marine engines into power, and thus to reduce the fuel consumption of ships. Among the various waste heat recovery (WHR) systems, the organic Rankine cycle (ORC) power systems is considered one of the most promising technologies due to its simple layout and high energy conversion efficiency [2]. The ORC operates in principle similarly to the steam Rankine cycle, but uses an organic compound as working fluid, leading to higher conversion efficiencies when utilized to exploit low to medium temperature heat sources.

The installation of ORC-based WHR systems on board LNG-fuelled vessels is expected to lead to higher savings compared to the installation on board heavy fuel oil-powered vessels. This is mostly due to two reasons. First, LNG-fuelled vessels are characterized by a reduced need for fuel preheating and, as a consequence, a higher amount of waste heat can be harvested by the WHR unit. Second, the absence of sulphur in the LNG results in a relaxation of the WHR boiler design constraints, allowing for the attainment of higher power productions. Lastly, because LNG is stored on board in cryogenic conditions, the low temperature heat released during the fuel preheating process can be used to further improve the performance of the WHR units installed on board.

1.2 Objectives and deliverables of the project

The LNG-waste heat recovery project aimed evaluating the technical and economic feasibilities of installing ORC-based waste heat recovery systems on board vessels powered by LNG. The project included the development of numerical models as well as the realization of an experimental test rig featuring a diesel engine and an ORC unit. The test rig is meant to prove the feasibility of the proposed concepts. The main objectives of the project revolved around the definition of the optimal design, control and integration of WHR units on board LNG-fuelled vessels.

The deliverables of the project are the following:

1. Mapping of the heat sources/sinks on-board of LNG-fuelled ships and identification of the most suitable heat source to be utilized by the LNG-driven ORC unit (see section 2);
2. Novel design of ORC units utilizing multiple heat and/or cold sources (see section 2);
3. An experimental facility at DTU Mechanical Engineering for future use in research projects and for teaching purposes (see section 6);
4. Proposal on design, control and integration of ORC units as retro-fit solutions and in new-buildings of LNG-fuelled ships (see section 4);
5. A novel method to optimize the design and control of the main engine combined with the ORC unit (see section 3);
6. Estimation of fuel saving potentials for LNG-fuelled ships; estimation of payback time and net present value for the proposed ORC configurations (see section 4).

1.3 Outline of the report

Section 1 includes a brief explanation of the background of the project and its deliverables. The screening of the available heat sources on board a vessel and the description of novel ORC configurations rejecting heat to multiple heat sources is included in the section. Section 3 describes a novel approach to integrate the use of ORC-based WHR systems on board vessels by accounting for the backpressure effect on the engine performance and the waste heat characteristics. Guidelines with respect of the optimal design ORC units for LNG-fuelled vessels and indications regarding their economic feasibility are included in section 4. Section 5 describes a case study work carried out in collaboration with Fjord Line. Section 6 presents an overview of the experimental facility developed at DTU Mechanical Engineering. Lastly, the dissemination activities of the project are listed in section 7, and the main conclusions are summarized in section 8.

.

2 Waste heat sources and novel organic Rankine cycle layouts

This section describes the mapping of the available heat sources and sinks on board LNG-fuelled vessels (deliverable 1), and introduces novel ORC architectures rejecting heat to multiple heat sinks (deliverable 2).

2.1 Mapping of available heat sources and sinks

A vessel is characterized by a complex energy system and therefore waste heat recovery solutions can be implemented on board in different ways. As shown in Figure 1, four heat sources that can be used for waste heat recovery on board a vessel: exhaust gases, jacket water, lubricating oil and scavenge air.

The lubricating oil has commonly low temperature (around 60-75 °C) [3] and thus is not a particularly attractive source to be considered. The exhaust gases are available at high temperature (above 200 °C) and are commonly utilized for the production of service steam, used to fulfill the heat demands on board [4]. In most cases, the heat contained in the exhaust gases largely exceeds the requirements for service heat – especially when the vessels are operated using low sulphur fuels (i.e. liquefied natural gas), because in these cases there is no need to preheat the fuel [5]. The quality of the waste heat contained in the scavenge air varies significantly as a function of the engine load, both in terms of available energy and temperature level, making it a not so attractive heat source. Lastly, the jacket water is available at a temperature of 80 – 90 °C, independently of the load at which the main engine is operated. This heat source is suitable to be used for the generation of fresh water and offers the potential for additional utilization by means of low temperature organic Rankine cycle power systems [6].

Regarding the available heat sinks, seawater represents the most commonly preferred solution. Seawater is abundantly available and its temperature is generally in the range 5 – 30 °C, depending on location and time of the year. Another possibility is the use of air as cooling media. Nonetheless, the poorer heat transfer properties of air compared to water, and its higher temperature variability makes it a less interesting solution, except for particular cases – i.e. ships sailing in the arctic region [7]. An additional heat sink can be considered in LNG fuelled ships. LNG is stored on board at atmospheric pressure in a liquefied state at about -160 °C, making it necessary to heat it up to about 30 °C before injection in the engine. The heat that needs to be provided for evaporation and pre-heating of the LNG can be provided by the heat rejected by an ORC unit, leading to the implementation of high efficiency waste heat recovery units.

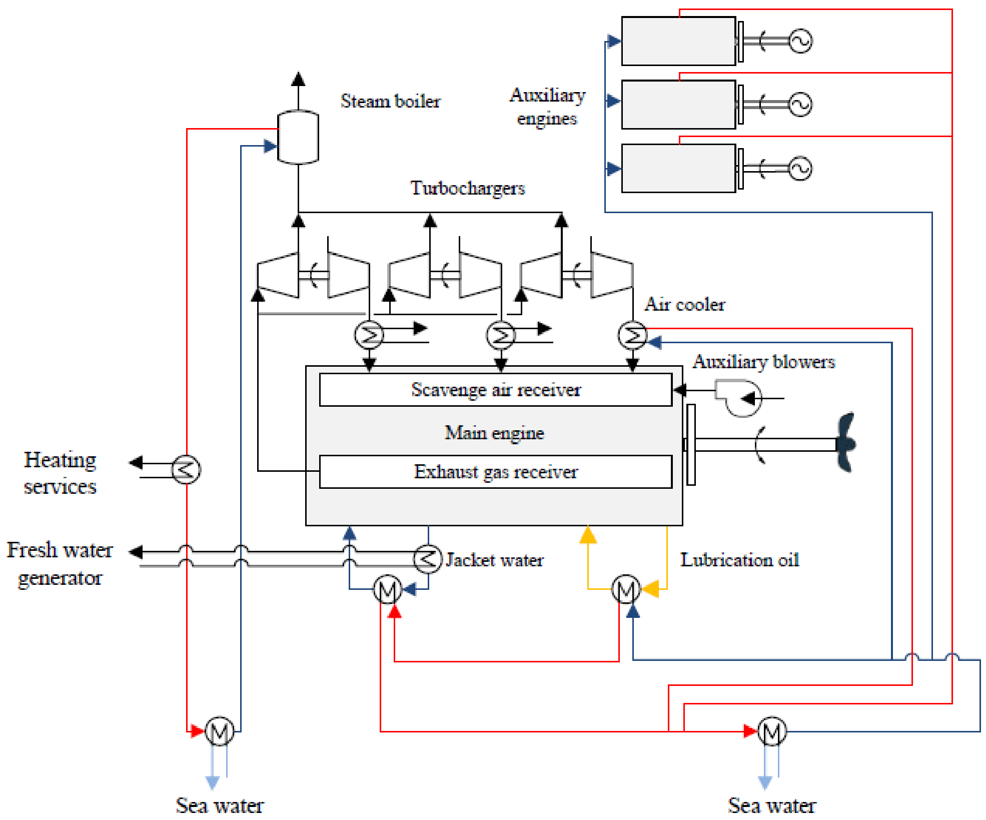


Figure 1: Simplified layout of a state-of-the-art machinery system aboard large ships [2].

The suitability of the various heat sources and sinks with respect to their potential for waste heat recovery was investigated through a study that considered a vessel powered by a 7G95ME-C9.5 MAN Energy Solutions dual fuel two-stroke marine engine with low pressure selective catalytic reduction tuning [8]. The CEAS engine calculation tool [9] from MAN Energy Solutions was utilized to retrieve the engine data, shown in Table 1.

Table 1: MAN 7G95ME-C9.5, performance and waste heat sources at different loads.

Load [%]	Power [kW]	SFC [g/kWh]	$\dot{m}_{ex}$ [kg/s]	T_{ex} [°C]	JW heat [kW]	T_{jw} [°C]	$\dot{m}_{jw}$ [kg/s]	$\dot{m}_{LNG}$ [kg/s]
100	36,820	135.8	79.1	261	4,380	85	69.47	1.39
75	27,615	129.7	60.9	253	3,570	85	69.47	0.99
50	18,410	127.1	42.6	268	2,760	85	69.47	0.65
25	9,205	129	22.4	285	1,940	85	69.47	0.33

The study assumed that a portion of the JW heat (400 kW) was used by the onboard fresh water generators, at all engine loads. Similarly, the requirements for service steam were neglected, as they are strongly reduced in LNG-fuelled vessels. Four ORC configurations were investigated. The first two configurations (case A) utilized the main engine exhaust gases and the jacket cooling water as heat sources, while the last two configurations (case B) harvested heat only from the engine jacked cooling water. Seawater and LNG preheating were considered as possible heat sinks. An overview of the considered heat sources and sinks in the various cases is show in Table 2.

Table 2: Selected heat sources and sinks for the considered configurations.

		Heat source	
		Exhaust gases + jacket water	Jacket water
Heat sink	Seawater	A1	B1
	LNG	A2	B2

Figure 2 shows a sketch of the considered ORC configurations. Simple ORC configurations were investigated for case B, while configurations of case A included also an internal recuperator and a jacket water preheater (see Figure 2). For this study, the boiler feed temperature lower limit was set to 110 °C (case A configurations only) as to avoid issues related to sulphuric acid corrosion (this corresponds to an LNG fuelled ships using a pilot oil containing sulphur). Taking into account the typical annual load profile of a containership, the four proposed ORC configurations were optimized, screening a variety of working fluids. The objective function of the optimization procedure was the ORC net power production when the engine was operated at 75 % load.

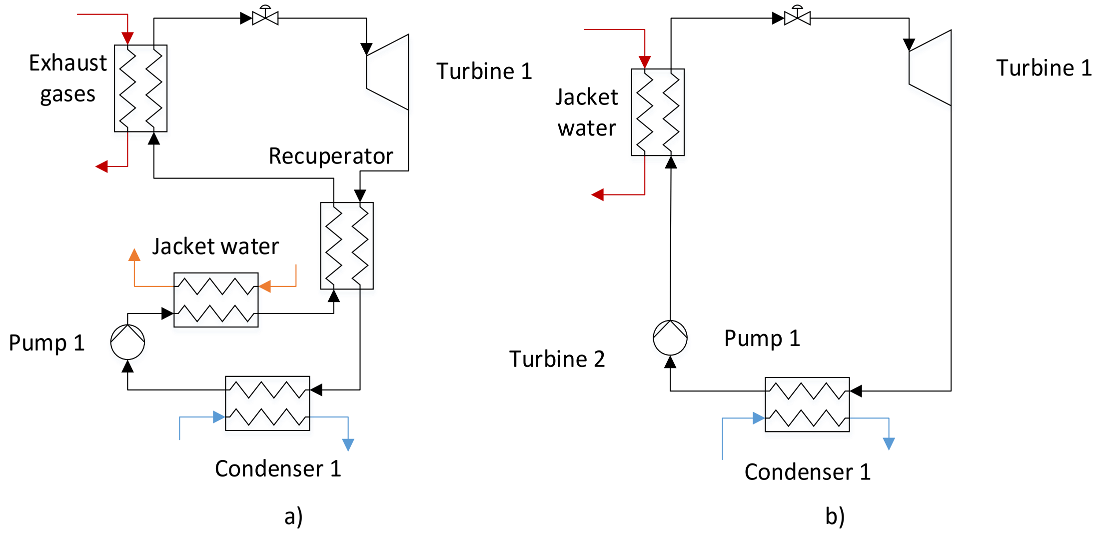


Figure 2: Sketch of the considered ORC configurations: a) cases A; b) cases B.

Two scenarios were investigated regarding the use of the energy produced by the ORC unit. In the first case, the ORC energy production was used for propulsion and the fuel savings were calculated as:

$$Fuel\ saving\ (\%) = 1 - \frac{Main\ engine\ annual\ consumption\ (with\ ORC)}{Main\ engine\ annual\ consumption\ (without\ ORC)} \tag{1}$$

In the second scenario it was assumed that the electricity produced by the ORC unit was used to replace the consumption on the on-board electricity generators, whose average fuel consumption was assumed to be 160 g/kWh [10]. Here the equivalent fuel savings were calculated as:

$$Equivalent\ \ fuel\ saving\,(\%) = \frac{Annual\ saving\ in\, auxiliary\, engines}{Main\, engine\, annual\ consumption\,(without\ ORC)} \tag{2}$$

The results of the optimizations are depicted in Table 3 and suggest that the exhaust gases are the most promising heat source available on board, leading to the highest fuel savings. The use of the jacket cooling water results in significantly lower savings – always below 1 %. Lastly, looking at the configurations utilizing the LNG preheating as heat sink, it appears that this option, despite the possibility of designing high efficiency ORC units (the estimated efficiency reached 35 % when using the exhaust gases as heat source), yields limited fuel savings. This is because of the limited mass flow rate of the LNG fuel that needs to be preheated, which practically sets a limit to the ORC working fluid mass flow rate and thus to the maximum attainable power output. For further information regarding the screening of the available heat sources and sinks, see Ref. [8].

Table 3: Results of the annual simulations for the two selected scenarios.

	Use for propulsion			Use for auxiliary generators		
Configuration	ORC production [MWh]	Fuel saving [ton]	Fuel saving [%]	ORC production [MWh]	Fuel saving [ton]	Equivalent fuel saving [%]
A1	8,048	1,075	6.9	8,283	1,325	8.5
A2	981	131	0.8	960	154	1.0
B1	923	124	0.8	898	144	0.9
B2	511	68	0.4	498	80	0.5

2.2 Novel organic Rankine cycle architectures

As emerged from the results described in the previous section, the exhaust gases represent the most promising heat source to be utilized for waste heat recovery, because it leads to the highest fuel saving potential. The heat contained in the jacket cooling water does not allow to obtain fuel savings above 1 % the ship annual fuel consumption. The investigations that aimed at assessing the prospects to use the cold energy contained in the LNG fuel as a way to obtain high efficiency ORC units lead to the following conclusions:

i. The use of the low temperature heat available in the LNG enables the design of high efficiency ORC units (cycle efficiencies up to 23 % and 35 %, when using jacket water and exhaust gases as heat sources, respectively);
ii. The LNG mass flow rate is limited and therefore poses a constraint on the maximum power output that can be produced by implementing such high efficiency units;

New ORC cycle configurations were therefore proposed as a way to take advantage of the low temperature of LNG mass flow rate, while getting over the limitations on the maximum power production [8]. The proposed ORC configurations are shown in Figure 3. Two configurations were proposed: in the first case (Figure 3a) the ORC unit harvests heat both from the exhaust gases and the jacket cooling water, while in the second case the jacket cooling water is the only considered heat source (Figure 3b).

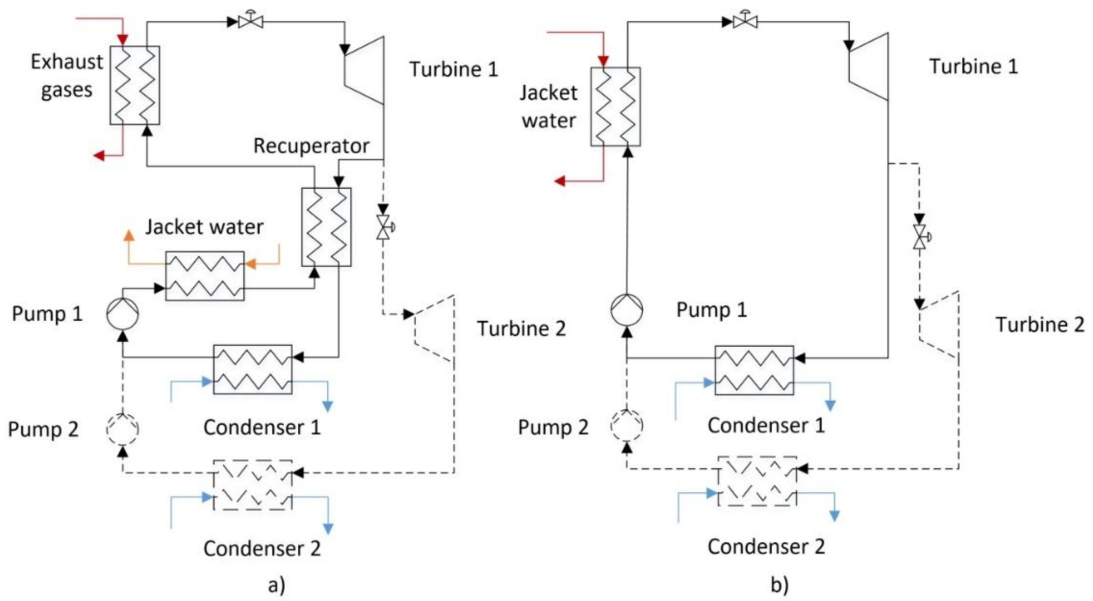


Figure 3: Sketch novel ORC configurations that were proposed in order to utilize multiple heat sinks: a) using exhaust gases and jacket water as heat sources; b) using only jacket water as heat source. The dotted lines represent the additional components required to realize the novel configurations.

The novel proposed configurations builds on the concept of a traditional ORC cycle, while including additional components as a way to exploit the low temperature LNG mass flow rate. In practice, a fraction of the working fluid mass flow rate is supplied to a second expander instead of going through the seawater condenser. The second expander ensures the production of higher net power outputs compared to the cases featuring only the seawater condenser and enables the exploitation of the energy released by the LNG during the preheating process. An optimization procedure based on the case presented in section 2.1 was conducted to assess the potential of the newly proposed configurations in comparison with the traditional cycle layouts.

The results of the optimizations are depicted in Table 4, where the novel configurations are named A3 and B3, for the case using the exhaust gases and the jacket cooling water, respectively. The results suggest that the implementation of the novel configurations leads to an increase of the attainable fuel saving potential. This increased fuel saving potential was estimated to be of in the range 42-66 tons/year, when using the ORC energy production to replace the consumption of the onboard auxiliary generators. The increased complexity of the proposed ORC configurations, and the need to include an additional expander unit, makes it however challenging to foresee that the proposed configurations will outperform the traditional layout in terms of economic attractiveness. For further information regarding the novel ORC architectures, see Ref. [8].

Table 4: Results of the annual simulations for novel ORC configurations in comparison with the traditional cycle layouts.

	Use for propulsion			Use for auxiliary generators		
Configuration	ORC production [MWh]	Fuel saving [ton]	Fuel saving [%]	ORC production [MWh]	Fuel saving [ton]	Equivalent fuel saving [%]
A1	8,048	1,075	6.9	8,283	1,325	8.5
A3	8,497	1,136	7.3	8,691	1,391	8.9
B1	923	124	0.8	898	144	0.9
B3	1,191	160	1.0	1,163	186	1.2

2.3 Summary of findings

The exhaust gases and the jacket water are identified to be the most attractive heat sources on board vessels. The use of seawater is recommended as heat sink, while air can be an interesting solution for ships sailing in the arctic region.

The installation of an organic Rankine cycle using the exhaust gases as heat source and seawater as cold sink can lead to equivalent fuel savings up to 10 %, when considering the use of the produced electricity to replace the consumption of the onboard auxiliary generators. Savings up to 1 % can be attained when using the jacket water as heat source.

The use of the low temperature heat released by the liquefied natural gas during its preheating phase before injection to the engine as a cold sink for an organic Rankine cycle can result in cycles characterized by high thermal efficiencies but low net power outputs. Novel organic Rankine cycle architectures featuring two condenser units were presented and enable the use of both seawater and liquefied natural gas preheating as cooling media for the organic Rankine cycle. The novel layouts result in increased power outputs in comparison with the traditional cycle configurations, but are characterized by a higher degree of complexity and are expected to be more expensive.

3 Design of organic Rankine cycle units accounting for engine backpressure

This section describes a method to design ORC units tailored for maritime applications accounting for the backpressure effect on the main engine performance and waste heat availability. The integrated design, which considers both the engine performance and the ORC production enables the attainment of an increased performance of the overall machinery system (deliverable 5).

3.1 Background and motivation

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.

The impact of the increased backpressure on the performance of turbocharged marine diesel engines has been previously investigated in both numerical and experimental studies. Nonetheless, these studies were limited to the investigation of the variation of the engine performance as a function of the supplied backpressure, while there is no clear indication regarding how to optimally integrate ORC units on board vessels by accounting for the additional backpressure supplied to the ship’s engine. This section describes a new approach to design ORC units tailored for maritime applications which accounts for both phenomena.

3.2 Method

In collaboration with MAN Energy Solutions, a novel approach to integrate ORC units on board vessels was derived. The method is based on the use of performance maps for the ship engine, and numerical models for the ORC unit and the waste heat recovery boiler which absorbs the heat contained in the exhaust gases and releases it to the working fluid of the ORC unit.

Figure 4 displays the impact on increasing the backpressure level to the engine MAN MAN 6S80ME-C9.5-GI engine with part-load tuning. The data has been provided by MAN Energy Solutions [11], and refers to the engine operated at full load. The main characteristics of the considered engine are reported in Table 5. The values refer to the engine operated with a backpressure of 3 kPa.

As it emerges from the figure, 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 indicated SFC increases as a function of the backpressure, and it includes both the LNG and the pilot fuel oil consumptions.

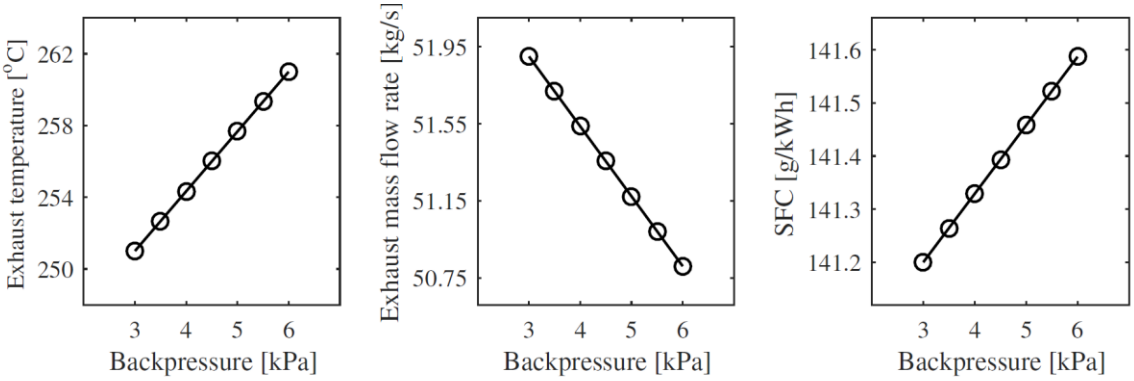


Figure 4: Engine performance as a function of the backpressure. Engine load = 100 %. Source: MAN Energy Solutions [11].

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 [11].

Table 5: The characteristics of the engine 6S80ME-C9.5 at full load and available waste heat in the exhaust gases.

Parameter	Value
Nominal power output [kW]	23,000
Nominal speed [r/min]	74.0
Exhaust gas temperature [°C]	251
Exhaust gas flow rate [kg/s]	51.9

The engine performance maps were integrated with the numerical models following the approach displayed in Figure 5. For every considered backpressure level, an ORC unit was optimized and the WHR boiler model was used to ensure that the additional backpressure supplied to the engine was matching the one assumed when retrieving the engine data from its performance maps.The overall consumption of the system comprising the engine and the ORC was estimated as:

$$SFC_{system} = \frac{SFC_{engine} \cdot \dot{W}_{engine}}{\dot{W}_{engine} + \dot{W}_{ORC}} \tag{3}$$

where SFC refers to specific fuel consumption (g/kWh) and $\dot{W}$ refers to power output (kW).The SFC of the engine was adjusted for every backpressure level according to the performance maps displayed in Figure 4. The evaluations were carried out considering different values for the boiler minimum pinch point temperatures, to ensure that the optimal matching between the ORC and the engine is not affected by this parameter. The evaluations were carried out considering a simple non-recuperated ORC configuration using cyclopentane as working fluid.

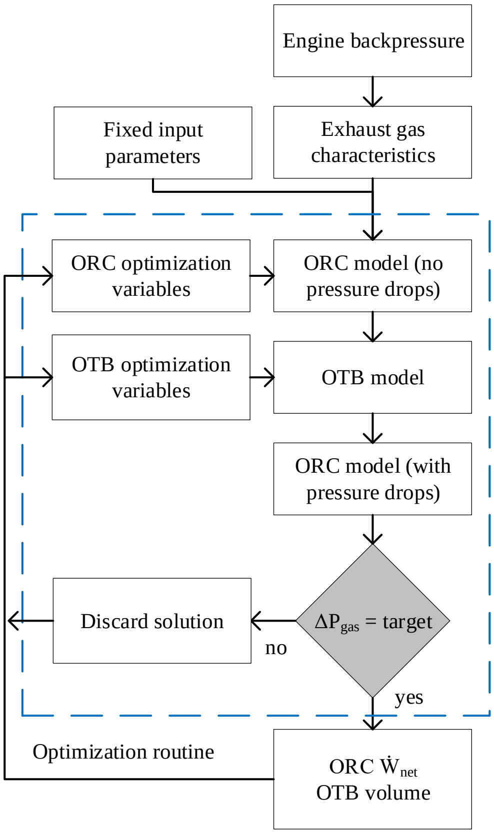


Figure 5: Schematic representation of the approach used to integrate the engine performance maps with the numerical models of the ORC and the WHR boiler.

3.3 Results

Figure 6 shows the impact of varying the backpressure supplied to the engine on the overall systems SFC.

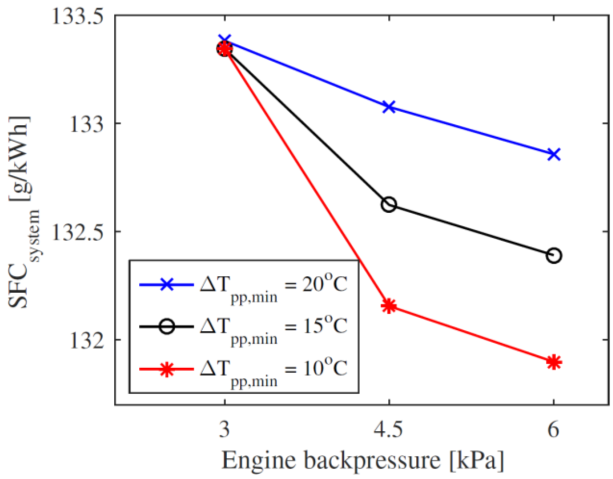


Figure 6: Impact of the selected engine backpressure on the overall system specific fuel consumption.

As it emerges from the figure, the overall system consumption can be minimized by increasing the allowed backpressure on the engine. In particular, 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.

This is the result of the fact that increasing the backpressure to the engine leads to an increment of the available waste heat that can be harvested by the ORC unit. Figure 7 shows the attainable ORC power output as a function of the boiler pinch point and backpressure level.

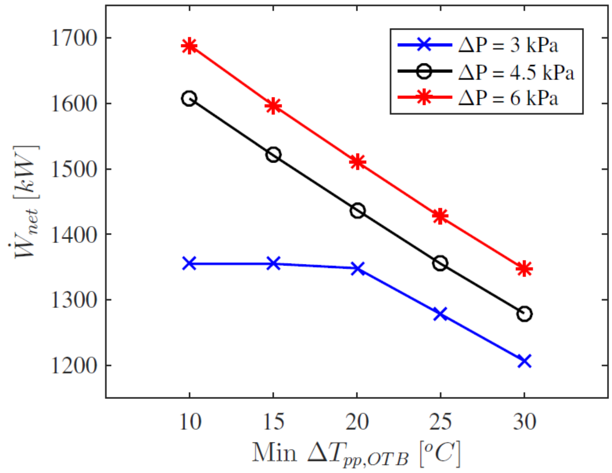


Figure 6: ORC power production as a function of the WHR minimum pinch point temperature and backpressure supplied to the engine.

As it emerges from the figure, an increase of the backpressure supplied to the engine by 1.5 kPa results in an increase of the ORC power output by 6 % (in average). Higher increases can be observed when the allowed boiler pinch point is below 20 °C. This increase in the production from the ORC unit more than compensates for the loss of efficiency of the main engine, indicating that for the considered case, it is suitable to consider an overall systems approach when designing ORC units for WHR recovery.

Finally, it should be pointed out that the sensitivity of the main engine to the applied backpressure and the selected engine turbocharging strategy have a large impact on the optimal design of the overall system. In some cases, like for the engine used in this work, the fuel consumption is lowered with an increasing allowable engine backpressure, whereas the opposite trend may be observed for another type of engine. For further details regarding the novel method to design ORC units, please refer to Ref [12].

3.4 Summary of findings

This chapter presented a new method to design organic Rankine cycles as part of the whole engine machinery system. The method allows to account for the effect of the additional backpressure supplied to the engine and therefore to attain a more accurate estimation of the attainable fuel savings. The use of the proposed method in a case study indicates that designing the organic Rankine cycle unit by accounting also for the engine performance can result in an increase of the attainable fuel savings in the range from 0.52 g/kWh to 1.45 g/kWh, compared to the traditional design approach, which assumes a fixed backpressure level to the engine.

4 Optimal design of organic Rankine cycle units

This section provides recommendations regarding the optimal design, integration and control of organic Rankine cycle units on board liquefied natural gas-fuelled ships (deliverable 4). In addition, indications are also provided with respects to the attainable fuel savings and economic attractiveness of the proposed solutions (deliverable 6). Considerations regarding the installation of waste heat recovery units in retrofit applications are also included.

4.1 Case studies and design considerations

In order to assess the prospects for ORC-based waste heat recovery on board liquefied natural gas-fuelled ships, two cases studies were evaluated. The first case study considers a long distance containership of middle size which operates in slow steaming mode in Tier II zones. The second case study revolves around a feeder ship operating in Tier III areas which utilizes exhaust gas recirculation in order to fulfill the requirements of reduced NO_x emissions. Table 6 provides an overview of the two considered ships, the installed engines and the annual fuel consumptions. The profiles of the exhaust gases temperatures and mass flow rates are reported in Ref. [13] and Ref. [14] for the feeder and the containership, respectively.

Table 6: Characteristics of the two considered reference vessels

Parameter	Feeder	Containership
Engine	MAN 7S60E-C10.5-GI	MAN 6S80ME-C9.5-GI
Engine rated power [kW]	10,500	23,000
NO_x emission abatement	Exhaust gas recirculation (Tier III)	Not installed (Tier II)
Annual operating hours [h]	4,380	6,500
Annual fuel consumption* [ton]	4,314	9,795
Engine backpressure [kPa]	3.0	3.0

**Propulsion only*

The ships not only have engines of different sizes (10.5 MW for the feeder and 23 MW for the containership) but are also operated according to different sailing profiles, as shown in Figure 8. The feeder is mostly operated at high engine loads, while the containership, as previously mentioned, is operated in slow steaming mode. With respect to the optimal design of ORC units suitable to be installed in the considered cases, the following recommendations are provided:

i. Simple non-recuperated ORC cycles are suggested, because of the reduced complexity which leads to reduced investment costs for the units;

ii. Given the negligible content of sulphur in the LNG, there is no need to preheat the working fluid before it enters the waste heat recovery boiler. This is especially true when considering low sulphur pilot fuels;
iii. The use of hydrocarbons as working fluid in the ORC unit is recommended as they are the working fluid candidates leading to the highest energy productions. Hydrocarbons are however flammable fluids, thus special attention should be used when designing and operating the ORC unit (such as using double piping with ventilation and gas leak detection systems);
iv. An off-design control strategy aiming at keeping a constant superheating at the inlet of the turbine is recommended, because it leads to a good off-design performance and it ensures the absence of droplets of fluid at the inlet of the turbine. The presence of fluid droplets would result in severe damage of the turbine.
v. Given that WHR units needs to be economically attractive, it is recommended that the optimization of the design includes economic indicators, ensuring the realization of a unit which is not only recovering high amounts of energy, but is also characterized by short payback times.

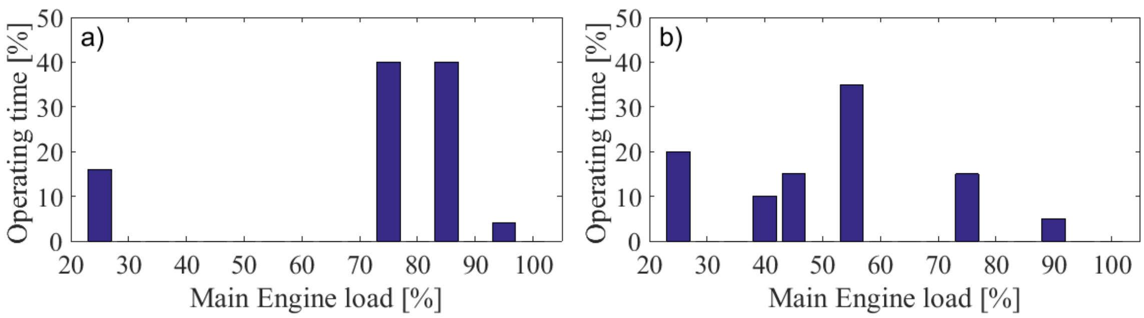


Figure 8: Considered sailing profiles: (a) feeder; (b) containership.

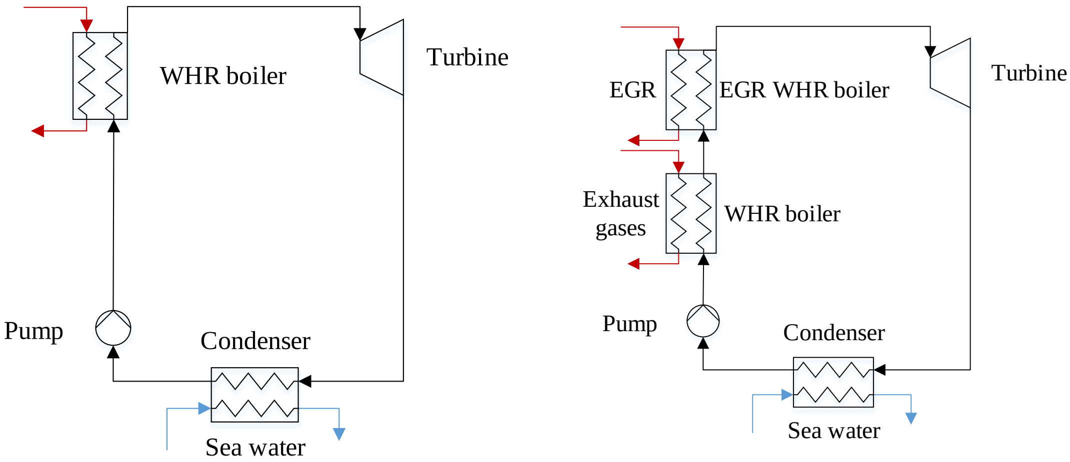
Figure 7 shows a sketch of the considered ORC configurations for the two cases. For the feeder case, the boiler is subdivided into two separate sections, one recovering heat from the portion of the exhaust gases which is recirculated in the EGR unit, and the other one recovering heat from the bulk of the exhaust gases. The validation of the ORC design and off-design models is described in Ref. [15] and Ref. [6], respectively.

The economic attractiveness of the proposed ORC configurations is evaluated by means of two economic indicators, the net present value (NPV) and the simple payback time (PB), calculated as follows:

$$NPV = -C_{tot} + \sum_{n=1}^{25} \frac{Annual\ savings}{(1+r)^n} \tag{4}$$

$$PB = \frac{C_{tot}}{Annual\ savings} \tag{5}$$

Where C_{tot} represent the total installation cost for the ORC unit and r is the discount rate, assumed to be 6 %. The total installation cost is estimated by using the procedure presented by Turton et al. [16], while the price of the LNG was fixed to 12 $/mmBTU. A 25-years lifetime was assumed when calculating the NPV and cyclopentane was chosen as working fluid for the ORC unit. For further information regarding the optimal design of ORC units tailored for LNG-fuelled ships, see Ref. [13]



(a) (b)

Figure 7: ORC configurations: (a) containership; (b) feeder.

4.2 Optimized organic Rankine cycle configurations

Table 7 shows the estimated performance of the ORC units optimized for the two considered case studies. As it emerges from the table, the ORC unit are capable of producing a significant amount of energy on an annual basis. If this energy is used to replace the consumption of the auxiliary generator, and assuming a specific fuel consumption of 160 g/kWh for the auxiliary generators, the resulting fuel savings is equal to 6.5 % and 8.4 % the annual consumption of the main engine, for the containership and the feeder, respectively. In both cases, the PB is estimated to be within 10 years, which should be compared with the 25 year life-time for the vessels.

Table 7: Results attained for the two considered vessels – ORC designs maximizing the NPV.

Parameter	Containership	Feeder
ORC design power [kW]	1,204	705
Volume heat exchangers [m^3]	17.00	13.54
Annual energy production [MWh]	4,014	2,270
Annual fuel saving [ton]	642.3	363.3
ORC specific cost [$/kW]	1,784	2,542
NPV [k$]	2,385	772.7
PB [years]	6.06	8.93

The higher economic attractiveness of the installation on board the containership is due to a combination of various aspects: i) the containership is operated for a significant higher amount of hours annually; ii) the higher ORC design power outputs results in lower specific investment costs; and iii) the ORC unit installed on board the feeder features two waste heat recovery boilers, and thus requires higher costs for the components and the installation.

Figure 9 shows the results of the sensitivity analysis on the impact of the variation of the fuel price on the economic attractiveness of the ORC units. As it emerges from the figure, the attainable NPV/PB is highly dependent on the fuel price and the units become more attractive with higher costs of the fuel. In both cases, positive NPVs are expected when the fuel price is at least of 10 $/MMBtu. Simple payback times lower than 5 year are estimated for the containerships when the fuel price is over 15 $/MMBtu. It should also pointed out that the ORC specific cost is a very uncertain parameter which has a significant impact on the estimated economic attractiveness of the units. Mass production of ORC units for maritime applications may result in specific ORC prices below 1,000 $/kW and thus to much shorter PB times.

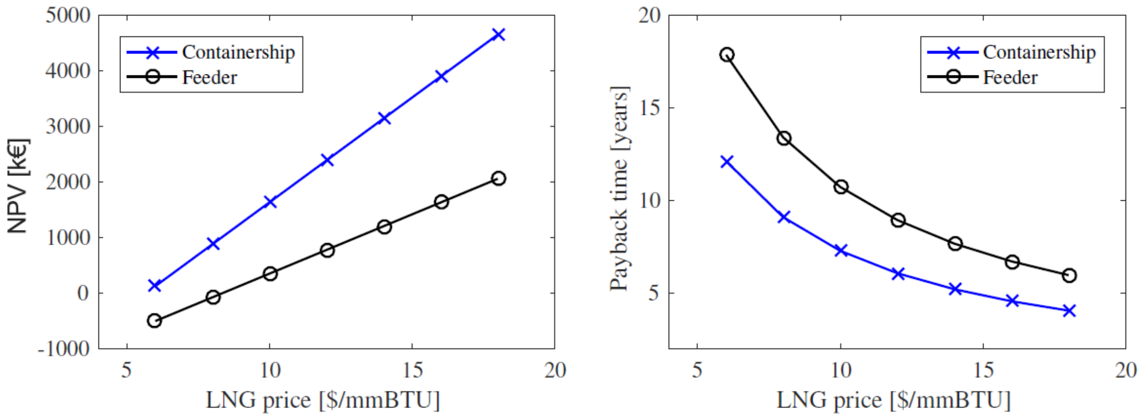


Figure 9: ORC power production as a function of the WHR minimum pinch point temperature and backpressure supplied to the engine.

4.3 Retro-fit installations

The retrofit installation of ORC units on board vessels represent a challenging engineering task, primarily with respect to availability of space. From a simulation point of view, the installation of an ORC unit in a retrofit case can be investigated by evaluating the impact of having a reduced size of the heat exchangers on the attainable fuel savings/NPV. For the NPV a 25 years scenario is considered, which could lead to optimistic results if the installed ORC is operated for a lower number of years.

Figure 10 shows the impact of imposing a reduced volume of the ORC heat exchangers on the attainable fuel savings/NPV for the case of the containership. As shown in Table 7, the optimal ORC configuration requires a volume of 17 m^3 for the heat exchangers.

Reducing space availability to 10 m^3, leads to a reduction of the fuel savings by 10 %, and in a reduction of the expected NPV by 8.4 %.

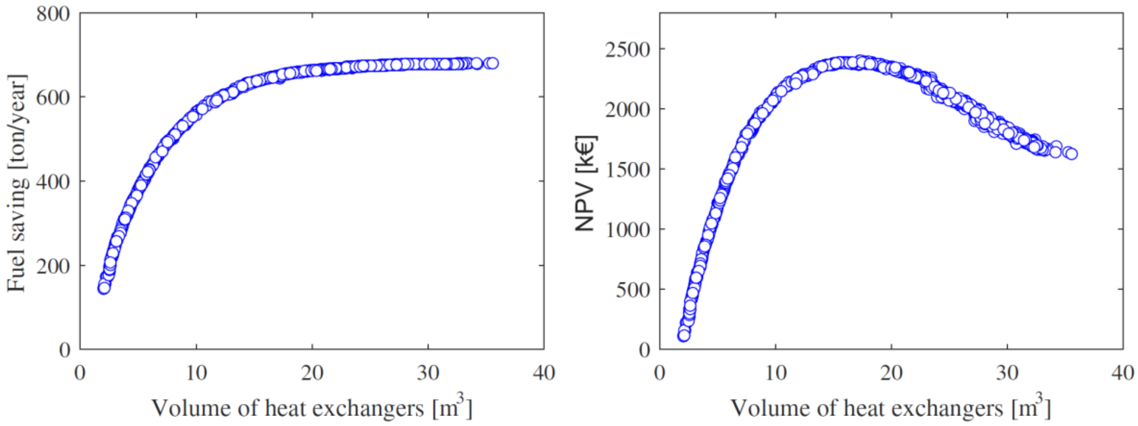


Figure 10: Impact of constraining the volume of the heat exchangers on the attainable fuel savings and NPV. The results are provided for the case considering the containership.

Figure 11 shows the impact of imposing a reduced volume of the ORC heat exchangers on the attainable fuel savings/NPV for the case of the feeder. As shown in Table 7, the optimal ORC configuration requires a volume of 13.5 m^3 for the heat exchangers. Reducing space availability to 7 m^3, leads to a reduction of the fuel savings by 13 %, and in a reduction of the expected NPV by 19.1 %. More stringent volume constraint are connected to higher reductions in the fuel savings and expected NPVs.

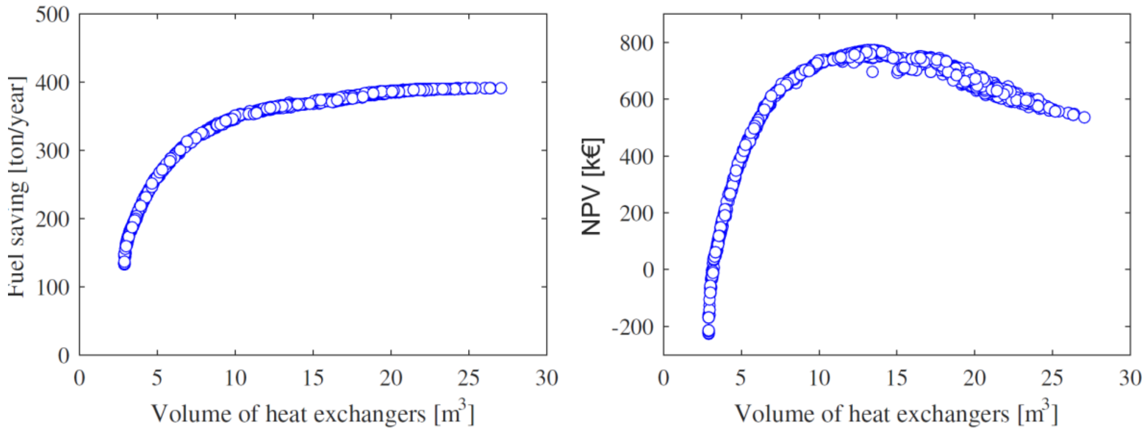


Figure 11: Impact of constraining the volume of the heat exchangers on the attainable fuel savings and NPV. The results are provided for the case considering the feeder.

4.4 Summary of findings

This chapters presented guidelines with respect to the optimal design of ORC units to be installed on board ships. The findings indicate that ORC units tailored for marine applications should be designed to maximize their economic effectiveness, which was estimated by means of the NPV of the installation.

The results of the study suggest that the economic attractiveness of installing ORC units increases with the ship's sailing time and engine power output (because larger unit have lower specific costs). Moreover, the evaluations carried out for two case studies based on a feeder ship operating in Tier III zone, and a containership operating in Tier II zone, indicate that payback times in the range from 5 to 10 years can be expected, depending on the fuel price. With respect to retrofit installations, the reduced space availability on board the vessel can result in a reduction of the economic attractiveness of waste heat recovery units.

5 A novel concept for emission-free power production on ships during harbour stays

This section of the report describes a test case evaluation carried out in collaboration with Fjord Line. The case study aims at evaluating the prospects for utilizing WHR solutions based on the use of an ORC unit in combination with a thermal energy storage system in order to supply zero-emission power on board a cruise during harbor stays. The technical and economic feasibilities of the concept were evaluated and compared with an alternative solution featuring the installation of lithium batteries.

5.1 Background and motivation

Although most of the emissions from shipping take place at sea, those released in harbor areas and port cities are the most noticeable for humans.

As a way to promote a reduction of the emission of pollutants in port areas, several incentive schemes were introduced in recent years. These incentive programs encourage ships calling in the various ports to reduce their emission levels below the requirements of the IMO and normally reward them with a discount on port dues [17]. The Environmental Ship Index (ESI) and the Clean Shipping Index (CSI) are among the most commonly applied incentive schemes. On the side of more widespread actions, individual incentive schemes were also introduced by single ports as a way to support green shipping; the harbor of Sandefjord in Norway, for example, rewards cleaner ferries by awarding them with the best departure times. Ship-owners are therefore facing an increasing demand for greener shipping, which is supported by significant economic benefits.

A reduction of the emissions during harbor stays can be achieved by using shore-power connections, but these are not always available and not suitable for cruise ships, which are characterized by short stays in ports. Therefore, in collaboration with Fjord Line a novel concept is investigated and compared to the alternative installation of lithium batteries tailored for fulfilling the onboard electricity needs during harbor stays.

5.2 The concept

The proposed concept features the use of a thermal energy storage system in combination with an ORC unit. A sketch of the concept is depicted in Figure 12.

During sailing, the exhaust gases are utilized to heat up a thermal oil, which is partly stored in the TES and partly utilized to run the ORC unit. During harbor stays, the TES is discharged in order to run the ORC unit. In defining the control strategy for the system, the first priority is to ensure the availability of the required power during the harbor stays. Auxiliary generators are utilized during sailing to provide the power demand that cannot be satisfied by the ORC unit. Both a stratified storage tank and a two-tank system were considered as TES. Therminol 66 was considered as thermal oil.

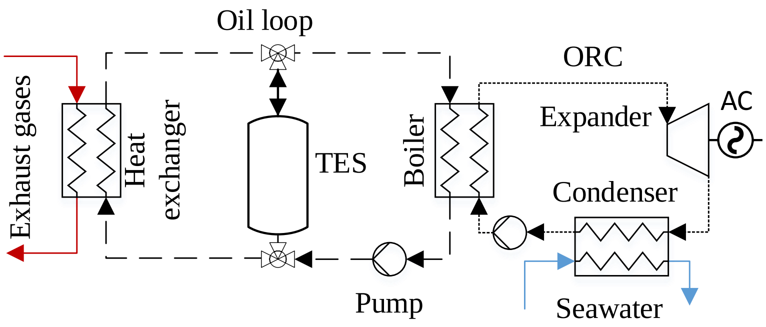


Figure 12: A sketch of the proposed system integrating the use of a thermal energy storage system and an organic Rankine cycle unit.

The technical feasibility of the proposed concept was verified considering a cruise sailing profile provided by Fjord Line and assuming that the cruise would require 1 MW of on board power both during sailing and port stays. Table 8 displays the considered cruise sailing profile. The cruise is powered by two 8 MW Wärtsilä 7L46DF engine, whose exhaust characteristics were retrieved from the manufacturer's product guide.

Table 8: Description of the considered sailing profile.

Phase	Sailing mode	Engine load [%]	Duration [min]
1	Maneuvring	15	20
2	Harbor stay	0	70
3	Sailing	85	150
4	Harbor stay	0	60
5	Sailing	85	150
6	Harbor stay	0	50
7	Sailing	85	150
8	Harbor stay	0	40
9	Sailing	85	150
10	Harbor stay	0	20
11	Maneuvring	15	20
12	Shore power	0	560

Because the use of the proposed concept enables not only the coverage of the energy demand during the harbor stays, but also produces electricity for on board use during the sailing phases, a new indicator was introduced. The 'Energy coverage factor' displayed in equation 6, describes the share of daily onboard energy which is supplied by the ORC unit.

$$Energy\ coverage\ [\%] = \frac{Annual\ energy\ supplied\ by\ ORC}{Annual\ energy\ required\ on\ board} \tag{6}$$

The economic feasibility of the concept was verified by comparing it to the installation of a battery system suitable to supply the required electricity during the harbor stays. The comparison is based on the estimation of the levelized cost of the electricity (LCOE) supplied for onboard use both during sailing and harbor stays. The LCOE was estimated as follows:

$$LCOE = \frac{I_0 + \sum_{y=1}^{n} \frac{I_y + O\&M_y + F_y}{(1+r)^y}}{\sum_{y=1}^{n} \frac{E_y}{(1+r)^y}} \qquad (7)$$

where the calculation considers that the system is operated for a number of years equal to n. The symbols $I_y, O\&M_y, F_y$ and E_y represent the investment cost, operation and maintenance costs, fuel expenditures, and the electricity generation at the year y. The symbol r is the discount rate considered for depreciation, assumed to be of 6 %, while I_0 represents the initial investment cost. The electricity generation represents the annual energy requirement of the cruise for onboard use, while the fuel expenditures are equal to the price of the LNG required to run the auxiliary generators. The consumption of the LNG generators was assumed to be 160 g/kWh [10].

5.3 Results

Figure 13 shows the estimated volume of the TES, the ORC energy coverage factor and LCOE as a function of the maximum allowed thermal oil temperature in the TES. The results are displayed for both the considered technologies for the TES system (e.g. the stratified tank case and the two-tank system case).

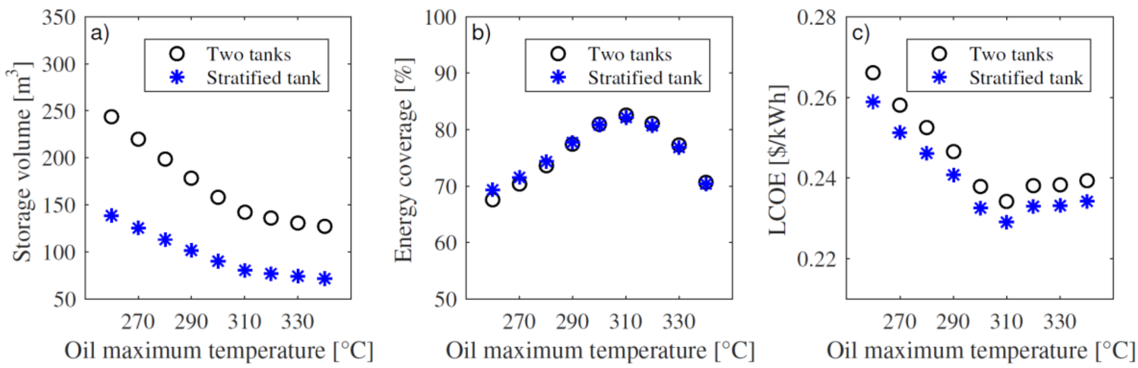


Figure 13: Impact of the oil maximum temperature on the performance of the two proposed storage concepts: a) storage volume; b) coverage of onboard energy demand; c) LCOE

The following findings emerge:

i. The selection of the stratified tank leads to a substantial reduction of the TES volume in comparison to the case where a two-tank system is considered.

ii. The attainable coverage factors are not significantly affected by the selection of the storage technology but rather by the oil temperature. The maximum coverage factor can be achieved by allowing a maximum oil temperature of 310 °C.

iii. The use of a stratified tank leads to lower LCOEs. In particular the minimum LCOE can be attained by selecting an oil temperature of 310 °C, which is the one which is also leading to the maximum energy coverage factor.

Figure 14 displays the ORC power production during the sailing profile for the two considered storage technologies.

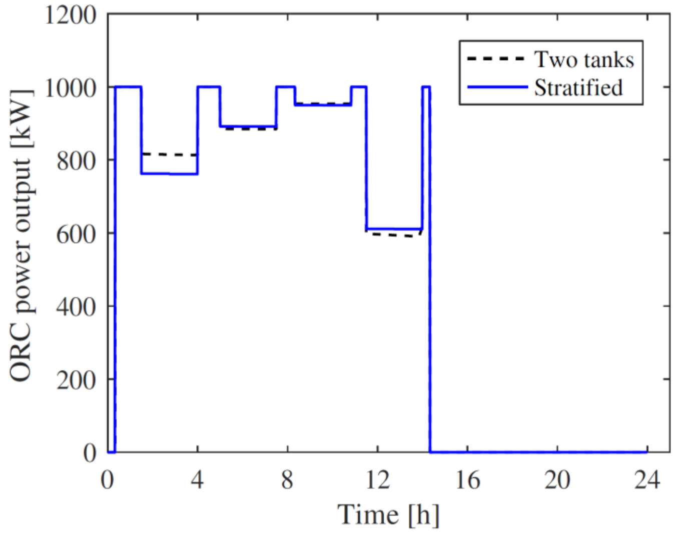


Figure 14: ORC power production throughout the sailing route for both proposed configurations

It clearly emerges that in both cases the concept is suitable to supply the required energy during the port stays and to additionally produce a significant share of the electricity required on board during the sailing phases. This proofs the technical feasibility of the concept. The attainable daily reductions of the ferry emissions of CO, CO_2, SO_2 and NO_x are around 15.8 kg, 5.4 ton, 2.9 kg, and 29.1 kg, respectively.

With respect to the economic comparison with the use of lithium batteries, Table 9 displays the attained LCOEs for the two considered cases (lithium battery and proposed concept using a stratified tank).

Table 9: The characteristics of the engine 6S80ME-C9.5 at full load and available waste heat in the exhaust gases.

Parameter	Battery system	ORC system
5 years LCOE ($/kWh)	0.141	0.2291
20 years LCOE ($/kWh)	0.140	0.1022
Investment cost (k$)	1,125	4,523
Annual fuel expenditures (k$)	472.88	84.80
Annual maintenance cost (k$)	16.88	67.85

Two time horizons are considered: i) 5 time horizon representing a retro-fit installation; ii) 20 years time horizon representing the installation in a new ferry. The results indicate that the battery system results in a lower LCOE when considering a five-year scenario, while the ORC LCOE is 30 % lower than the battery solution when a 20-year scenario is considered. The investment cost for the ORC system is roughly four times higher than the one of the battery system, while the annual expenditures (sum of fuel expenditures and maintenance costs) are lower for the ORC system ($84,800 compared to $472,880).

Figure 15 depicts the results of the sensitivity analyses carried out for the estimated LCOEs.

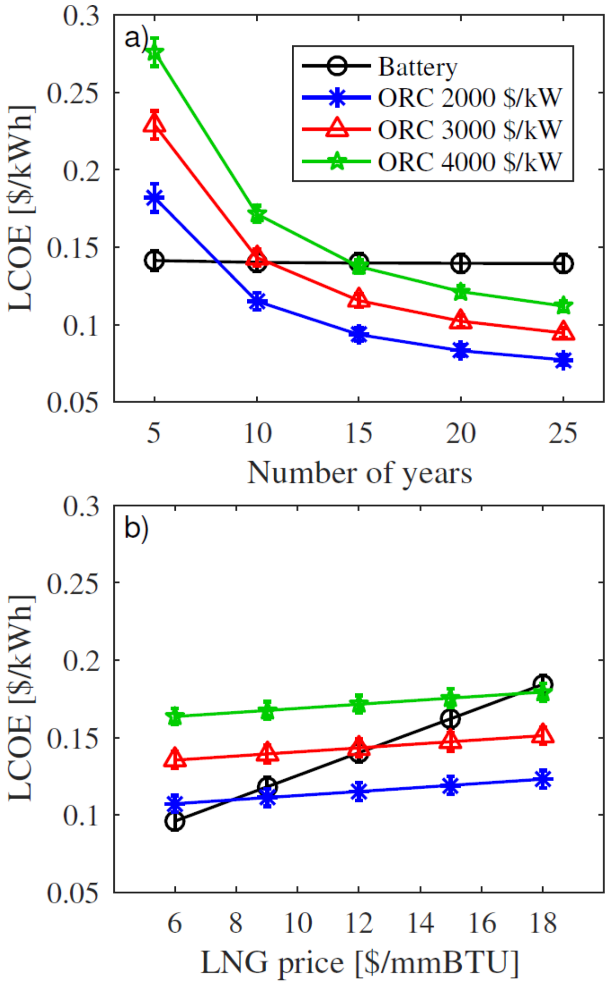


Figure 15: Results of the sensitivity analysis on the expected LCOE for the various systems: a) impact of number of years of operation; b) impact of LNG price on the 10-year LCOE. The error bars represent the standard deviations computed through the uncertainty analysis

Figure 10a shows that the estimated LCOE for the ORC system is highly affected by the number of years for which the system is assumed to be operating. The longer the considered period, the lower the resulting LCOE. On the contrary, no variation is seen with respect to the battery system. This is because the battery system needs to be replaced every five years. Moreover, all the ORC solutions result in a lower LCOE than the battery system if a period of 15 or more years is considered. Figure 10b depicts the impact of the LNG price on the estimated 10-year LCOE. The plot shows that the battery system is the one which is most affected by this parameter, and that the ORC system is more promising when the LNG price is high. This results from the fact that the LNG price affects linearly the annual fuel expenditures, which are a predominant factor in defining the LCOE of the battery system. The ORC system requires, on the contrary, lower amounts of LNG (around 20 % the amount required by the battery system), and therefore it is less affected by LNG price fluctuations.

Considerations regarding the required volumes were also carried out. Corvus Energy [32] supplies containerized battery systems for vessels, claiming that a standard 40-foot container (roughly 67.6 m^3) could be filled either with a battery package of 1,365 kWh or with a battery package of 819 kWh plus the required power electronics. Given that the considered ferry requires a battery system of 1,500 kWh plus the power electronics, it is expected that the battery system would require a space equivalent to almost two 40-foot containers. This means that the space requirement of the battery and ORC systems are similar. The volume requirements for the recovery heat exchanger and the stratified tank were estimated to be of 82.1 m^3 and 6.4 m^3, respectively. This indicates that the two compared solutions (lithium batteries and proposed concept using a stratified tank) would result in similar volume requirements on board the cruise. Further information regarding the proposed concept and the validation of the numerical models is included in Ref. [18].

5.4 Summary of findings

A novel concept to attain emission-free power production on board vessels during harbor stays was evaluated in collaboration with Fjord Line. The concept features the use of an organic Rankine cycle and a thermal energy storage system.

The techno-economic feasibility of the concept was evaluated by means of a case study featuring a ferry, whose sailing profile data was provided by Fjord Line. The evaluation indicates that the installation of the proposed concept on board a cruise ships enables the production of zero-emission power during harbor stays, and to a reduction of the daily CO_2 emissions by 5.4 ton. With respect to the economic performance, the evaluations suggest that the proposed concept is more cost-effective than the installation of lithium batteries when considering life-times of more than 10 years.

6 Test rig at DTU Mechanical Engineering

This section provides and overview of the test rig developed at DTU Mechanical Engineering (deliverable 3). The test rig features a diesel engine and an ORC unit.

6.1 Background and motivation

In addition to the numerical analyses, there is a need to experimentally evaluate the technical feasibility of using the ORC technology for waste heat recovery on combustion engines. Particularly, in order to ensure safe and efficient operation of an ORC unit, the operation and control strategies of the ORC unit need to be evaluated experimentally. Moreover, numerical models need to be validated based on results of experimental work. For these purposes an experimental facility was constructed. The test facility is currently used in on-going research and teaching activities and it will be used in future such activities.

6.2 Test rig: layout, automation and safety

The ORC test rig is installed inside a cabinet located at the Department of Mechanical Engineering at DTU. It consists of a 3-kW ORC unit equipped with plate heat exchangers, an axial-flow turbine and a positive-displacement pump. The piping and instrumentation (P&I) diagram of the test rig is shown in Figure 16, while a photo of the existing set-up is shown in Figure 17. The energy input to the ORC unit is provided by an 80-kW diesel engine, whose main parameters are summarized in Table 10.

Table 10: Description of the diesel engine providing the heat source to the test rig.

Parameter	Value
Engine type [-]	Ford DV6TED4
Cylinder volume [cm^3]	1560
Bore [mm]	75
Stroke [mm]	88.3
Compression ratio [-]	17.4
Maximum power [kW]	80
Maximum torque [Nm]	240
Maximum boost pressure [bar]	2.4

Three heat sources are available for the ORC unit: the exhaust gases leaving the engine turbocharger (in purple in Figure 16), the engine jacket cooling water (in red) and the charge air after the compression stage of the engine turbocharger (in yellow). The engine jacket

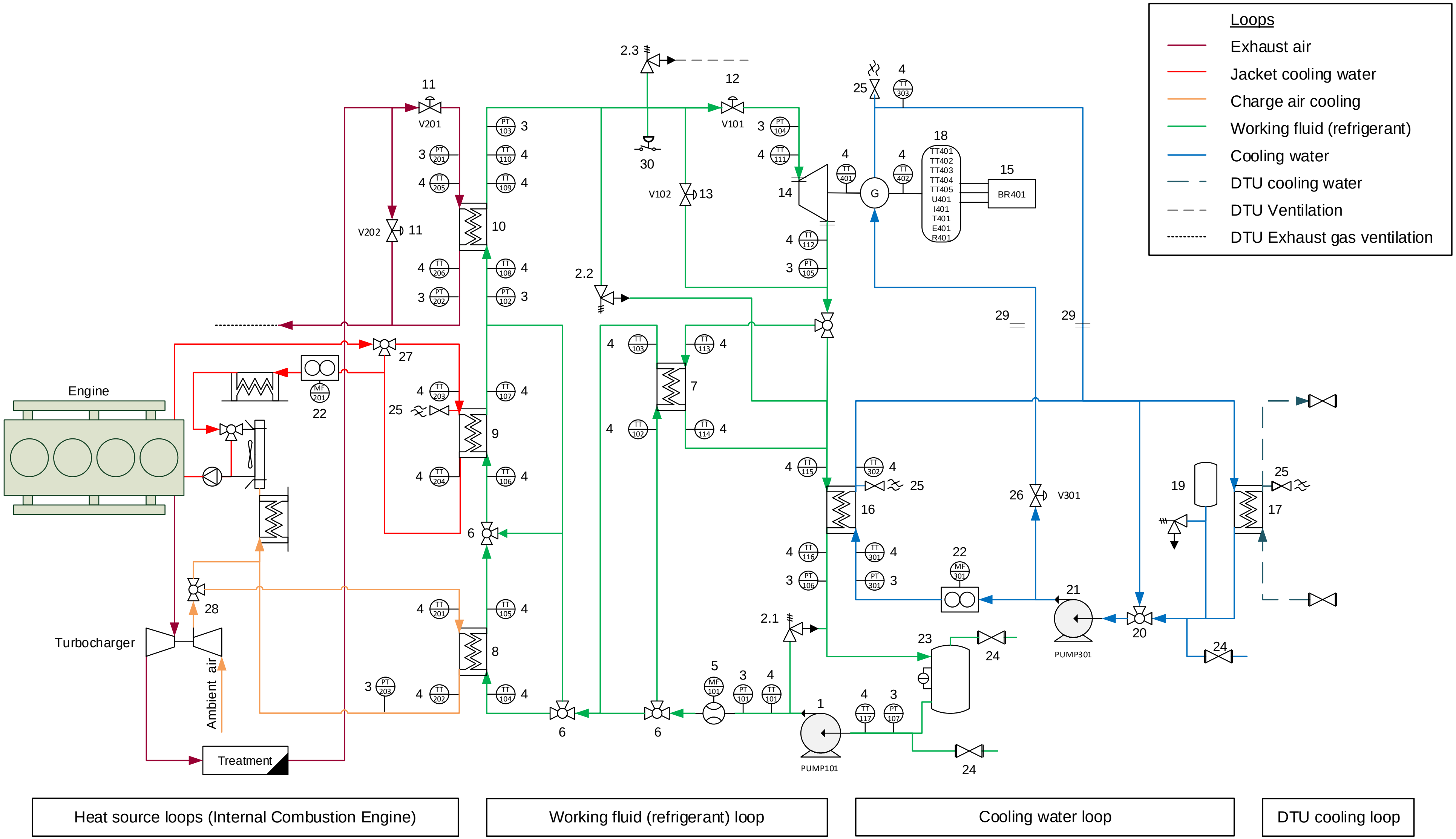


Figure 16: P&I diagram of the test rig at DTU Mechanical Engineering.

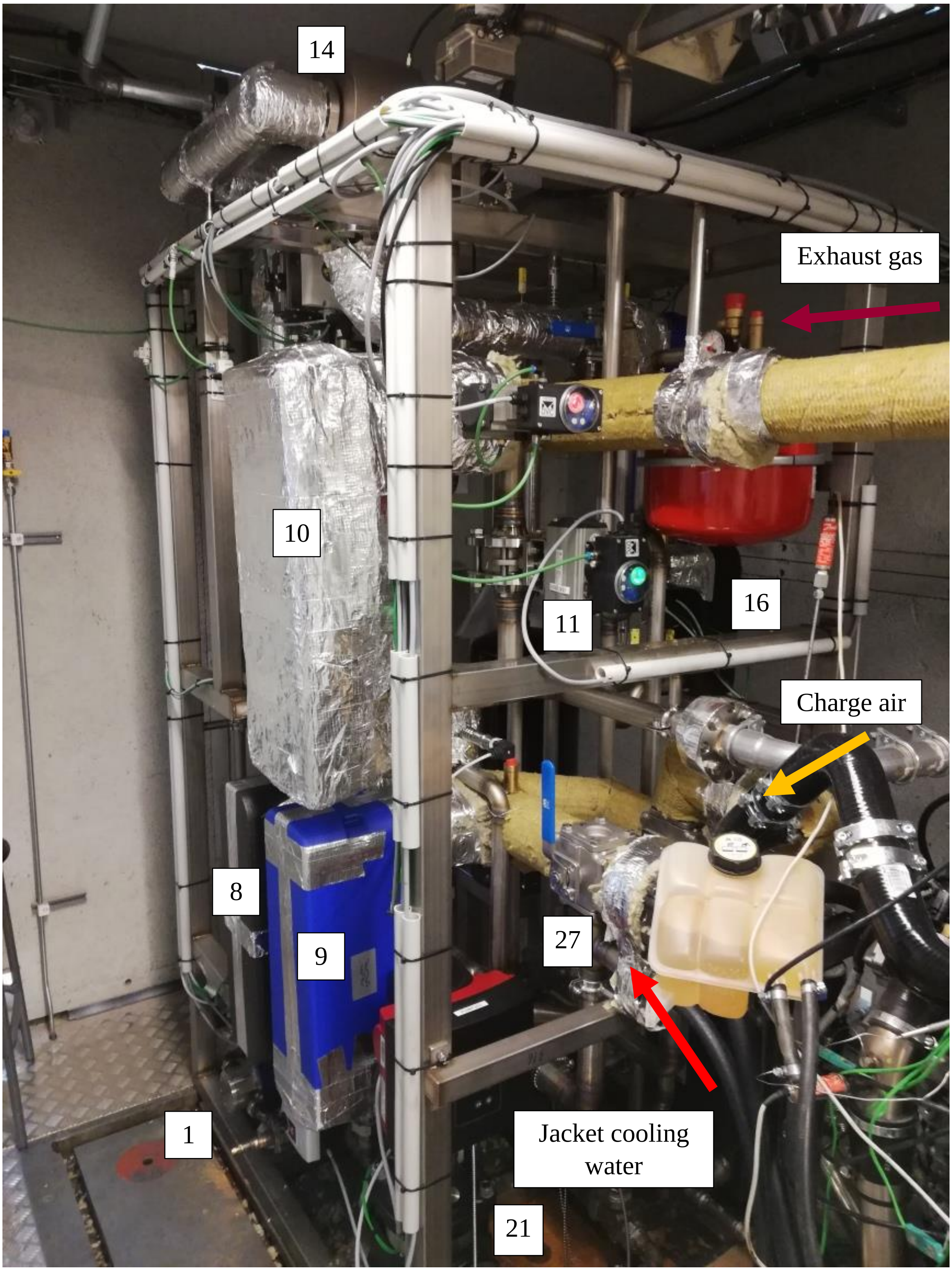


Figure 17: Picture of the ORC test rig.

cooling water and the charge air are cooled by external plate heat exchangers when the ORC test rig is not active, such that the heat can be rejected to the DTU central cooling water system. The exhaust gases can bypass the ORC unit by opening the pneumatic valve V202 and closing the pneumatic valve V201 (components no. 11). By means of two three-way valves (no. 27 and 28), the engine cooling water and the charge air can be directed to the ORC test rig. It is important to highlight that the heat sources are directly connected to the ORC unit without the usage of an intermediate oil loop. This is advantageous because it avoids exergy destruction and exergy losses in the intermediate loop and allows to maximize the net power output of the unit. On the other hand, it has to be ensured that the working fluid always operates below its maximum allowed temperature, otherwise thermal degradation will occur.

The ORC working fluid is R1233zd(E), which is a possible replacement of the commonly used R245fa, since it shows very similar thermodynamic properties, an analogous ozone depletion potential (very close to zero) and a much lower global warming potential than R245fa (7 vs 1030). Importantly for the safety of the test rig, it is a non-toxic and non-flammable fluid. In addition, it has a saturation pressure above 1 bar at ambient temperature, which prevents air leakage into the system. The working fluid loop is highlighted in Figure 16 with a green line.

The charge air and the engine jacket cooling water coolers (components no. 8 and 9, respectively) have the main task of preheating the working fluid up to saturation temperature. Depending on the operating conditions, the evaporation can already start in the engine jacket cooling water and/or in the charge air cooler. The task of completely vaporizing the working fluid and providing the desired degree of superheating is assigned to the exhaust gas cooler (no. 10). The hot working fluid at vapor state leaving the exhaust gas cooler is then expanded during normal operation in an axial-flow turbine (no. 14). The design conditions are summarized in Table 11. In front of the turbine a pneumatic control valve is installed (no. 12), which can regulate the pressure at the turbine inlet. During start-up, shutdown or whenever the degree of superheating is not sufficient, the turbine can be bypassed by means of a pneumatic valve (no. 13). The turbine drives a synchronous permanent-magnet electric generator, whose speed can be controlled by means of a variable speed drive. The excess electricity produced by the electric generator is dissipated through a brake resistor (no. 15), which is mounted on the outside of the cabinet in order to ensure sufficient cooling. The turbine exhaust vapor can be then condensed back to liquid state in a water-cooled condenser (no. 16) or it can be sent to the recuperator (no. 7), where the working fluid leaving the pump is preheated before receiving heat from the engine. The recuperator can be activated by means of a manual ball valve (no. 6).

The cooling water at the condenser flows in an intermediate closed loop (blue line) and is forwarded by a centrifugal pump (no. 21) driven by a 3-phase asynchronous motor. A variable speed drive is connected to the water pump motor in order to control its flow rate. A part of the cooling water is controlled by means of an electromagnetic valve (no. 26) to cool down the electric generator and the turbine bearings. The cooling water in the intermediate closed loop rejects the heat from the ORC unit into the DTU cooling water systems by means of an additional plate heat exchanger (no. 17).

Table 11: Design parameters of the ORC axial-flow turbine and the electric generator.

Parameter	Value
Generator power output [kW]	3
Working fluid mass flow rate [kg/s]	0.132
Working fluid pressure at inlet [bar]	10.77
Turbine and generator rotational speed [rpm]	22 000

The condensate working fluid from the condenser is then collected in the working fluid tank (no. 23), which has a volume of 30 l. A vane pump (no. 1) is then used to pump the working fluid to the coolers and close the ORC loop. The pump speed can be controlled by means of a variable speed drive.

The test rig is fully instrumented. The P&I diagram in Figure 16 also indicates the location of the measurement sensors. The accuracy of each sensor is reported in Table 12. In addition, a data acquisition and automation system has been developed in LabVIEW, which allows to monitor the test rig operation, log the measurement data and control the system actuators. The graphical user interface used to monitor and control the test rig is shown in Figure 18. The data acquisition interval is currently set to 0.5 s, but it can be reduced depending on the operator requirements. The data acquisition and automation system has also full access to the diesel engine onboard diagnostics, so that important information from the engine such as the charge air pressure at the outlet of the engine turbocharger or the mass flow rate of air and exhaust gas can be accessed.

Table 12: Description of the measurement sensors for the ORC test rig.

Measured parameter	Measurement principle	Measuring range	Accuracy of measurement	Sensors	Output signal
Pressure	Strain gauge	0 - 25 bar	±0.25 bar	PT1XX, PT301	4-20 mA
		0 – 2.5 bar	±0.025 bar	PT201-PT202	
		-1 – 34 barg	±0.11 barg	PT203	1-5 V
Temperature	Thermocouple (type K)	-200 - 1260 °C	±max(2.2 °C, 0.0075 MV)	TT1XX, TT2XX, TT3XX, TT401-TT402	0-48.8 mV
	PT100	0 - 150 °C	±(0.03 °C)	TT403-TT405	resistance
Mass flow rate	Coriolis sensor	0 – 0.3 kg/s	±(0.00006 kg/s)	MF101	4-20 mA
Volume flow rate	Magnetic sensor	0 – 6.1 m^3/h	±0.01 m3/h	VF301	4-20 mA
		0.1 – 3.6 m^3/h	±(0.015 MV ±0.011 m^3/h)	VF201	
Electric power	El. power meter	0 – 7.5 kW	±0.005 MV	E401	Profibus
		0 – 0.75 kW		E101	

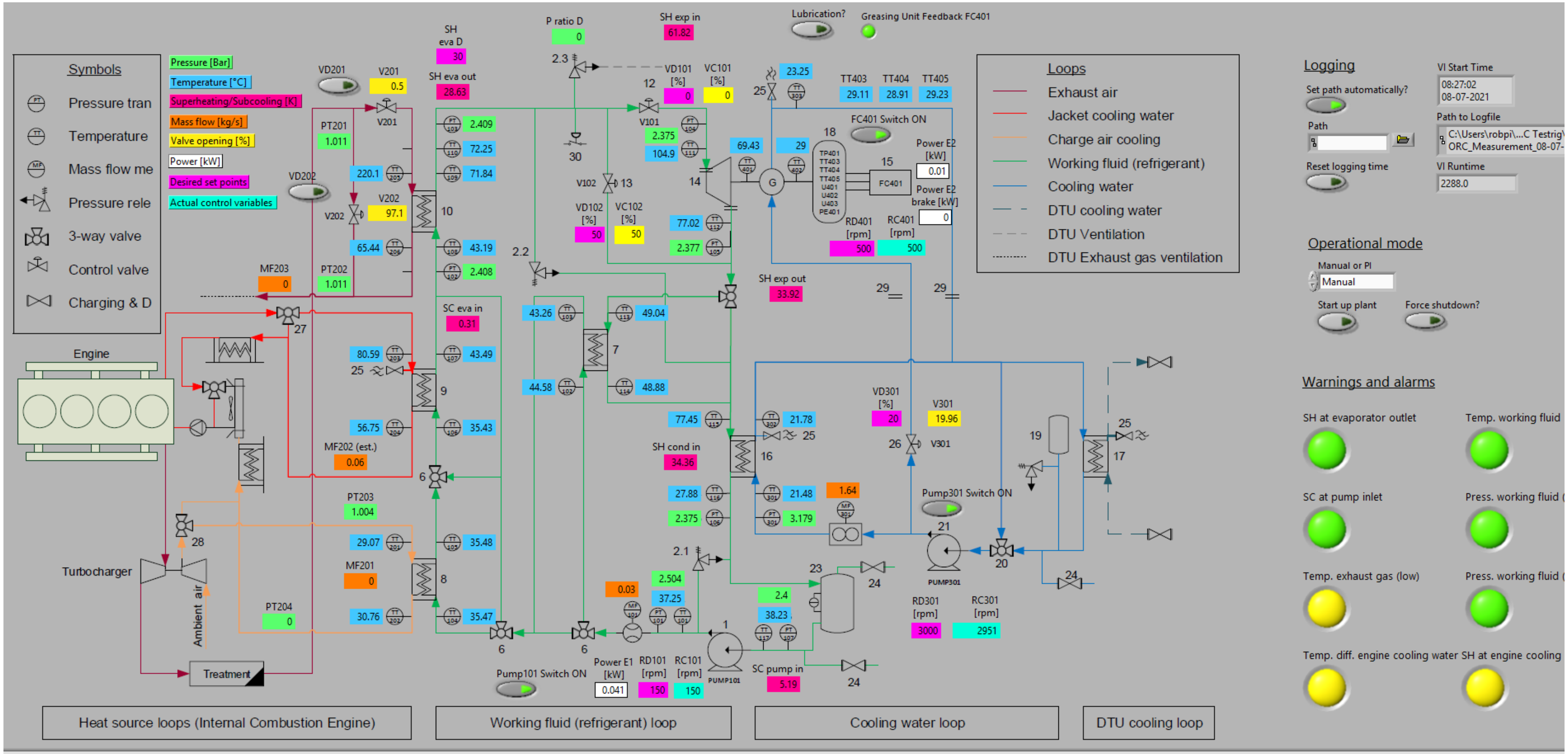


Figure 18: LabVIEW graphical user interface of the test rig at DTU Mechanical Engineering.

Several safety measures are implemented both in the hardware and the software and are reported in a risk assessment plan. In particular, the emergency loop of the ORC unit is subordinated to the emergency loop of the diesel engine, so that the system can immediately go to a safe state by pressing the same emergency stop button. The safe state is also reached if the engine reports malfunctioning, if the fuel supply system stops working, if the exhaust and the cabinet venting system are not active, if the compressed air is not available to drive the ORC pneumatic valves (the pressure should be at least 3 bar) or if the level of the working fluid liquid in the tank is too low. The safe state for the ORC unit forces the opening of the bypass of the exhaust gas and of the turbine. In addition, the variable speed drives force also the ORC pump and turbine and the cooling water pump to stop. Two prevent overpressures in the ORC loop, three safety pressure release valves are installed.

A first release valve (no. 2.1) opens when the pressure at the pump outlet exceeds 15 bar. In this case, the working fluid should be recirculated to the working fluid tank (no. 23). If this measure is not sufficient and the pressure exceeds 18 bar, a safety release valve at the exhaust gas cooler outlet (no. 2.2) directs the fluid to the condenser so that it can be cooled down (no. 16). As last safety measure, if the pressure reaches 20 bar, a release valve (no. 2.3) vents the working fluid to the exhaust vent system of the cabinet. Software-wise, the maximum temperatures and pressures of the working fluid are constantly monitored. Particular attention is paid to the temperature of the working fluid leaving the gas cooler, because thermal degradation of the working fluid occurs above a temperature of 150 ºC. A system shutdown is activated when this value is exceeded. The same holds when the degree of subcooling at the pump inlet is below 1 K to prevent cavitation of the pump. In addition, if the degree of superheating at the outlet of the exhaust gas cooler is not above 5 K, the bypass valve is opened and the turbine control valve is closed to prevent damage of the turbine.

6.3 Heat source maps

In order to make the waste heat recovery profitable, the ORC unit needs to maximize the net power output under any available waste heat from the diesel engine. The available waste heat is a function of the operating point of the diesel engine. At steady-state conditions, the engine operating point and the heat source conditions are univocally identified by the shaft torque and shaft rotational speed. When the engine operating point varies, a given time is required to reach a new steady-state operating point because of the engine control system and because of thermal delays occurring in the pipelines connecting the heat sources from the engine to the ORC unit. The time delay occurring in the engine is in general much smaller than the time response of the ORC unit, and for this reason, it is typically deemed of secondary importance and neglected here.

Figure 19 shows the brake efficiency of the diesel engine and the available waste heat from all the three heat sources referred to 25 ºC as a function of the shaft torque (also called "brake torque") and the shaft rotational speed of the diesel engine. The brake efficiency is the ratio of the shaft power to the fuel input power. For the efficiency and waste heat calculations, a lower heating value of 42.94 MJ/kg was assumed for the diesel fuel.

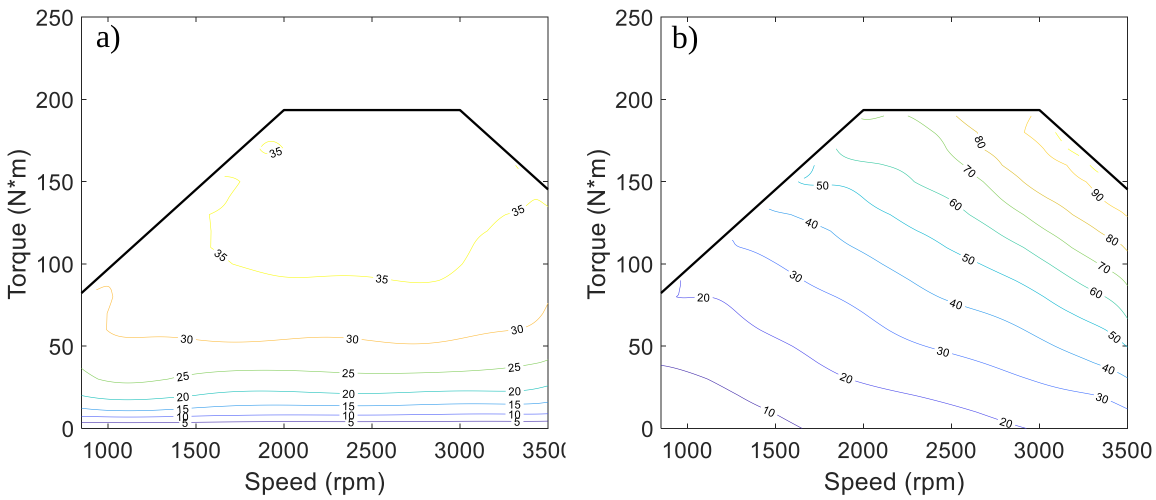


Figure 19: (a) Brake efficiency in % and (b) available waste heat in kW from the diesel engine.

It can be seen in Figure 19 that the engine rotational speed can vary between 800 rpm and 3500 rpm, while the torque varies between 0 and a maximum value depending on the shaft rotational speed. The maximum torque varies between 82 Nm and 193.5 Nm. The brake efficiency can vary between 0 % and approximately 37 %. The available waste heat is calculated from the difference between the fuel input power and the shaft power, and it ranges from 0 to approximately 95 kW.

The steady-state maps of the three heat sources for the ORC unit are shown in Figure 20. The axes are the same as those of Figure 19, while the contour lines represent the heat source conditions, i.e. its mass flow rate or temperature. The mass flow rate of the charge air varies between 0 kg/s and 0.11 kg/s, and is primarily a positive function of the engine speed. On the contrary, the temperature of the charge air at the outlet of the turbocharger mainly increases with the engine torque and varies between 45 ºC and 140 ºC. The mass flow rate of the exhaust gas is given by the sum of the mass flow rate of the charge air and the mass flow rate of the consumed fuel, and has a similar behavior with the engine speed as that of the mass flow rate of air. It varies between 0 kg/s and 0.12 kg/s, whereas the temperature of the exhaust gas ranges from 120 ºC to more than 400 ºC, and it is mostly dependent on the torque. It is important to highlight that the temperature of the exhaust gas exceeds the maximum allowable temperature of the ORC working fluid for most of the operating points (400 ºC vs 150 ºC). Therefore, particular caution is required to ensure that the flow rate of the working fluid is always sufficiently high to prevent hot spots in the working fluid.

The mass flow rate of the engine jacket cooling water is internally controlled by the diesel engine such that its upper temperature does not exceed 83 ºC. For this reason, the flow rate of the cooling water depends on the return temperature from the ORC unit at engine inlet. Figure 20e shows the mass flow rate of the jacket cooling water for a return temperature of 73 ºC and a temperature increase in the engine of 10 K. The mass flow rate

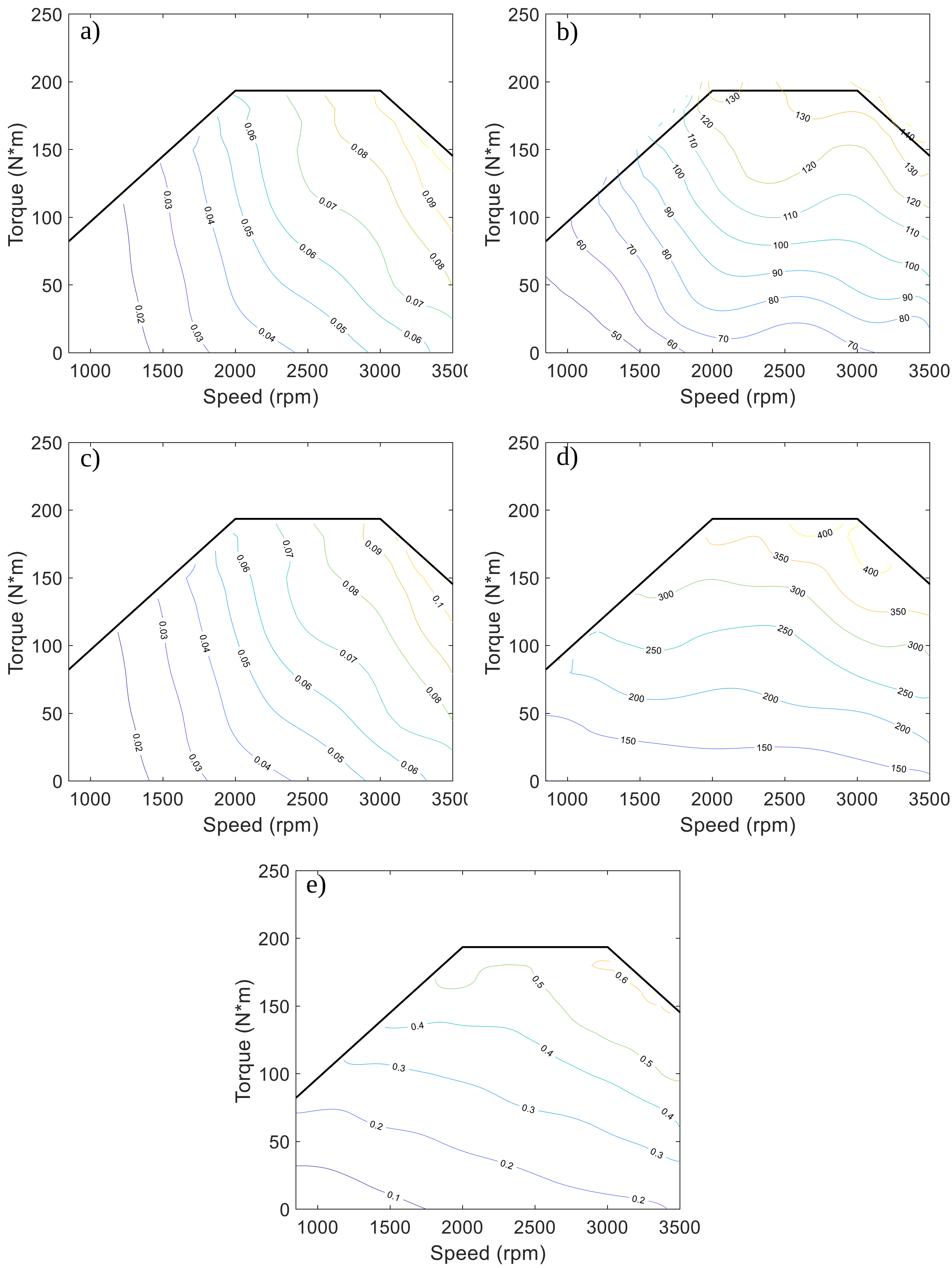


Figure 20: Heat source maps as a function of the engine operating point: (a) charge air mass flow rate in kg/s; (b) charge air temperature in ºC; (c) exhaust gas mass flow rate in kg/s; (d) exhaust gas temperature in ºC; (e) jacket cooling water mass flow rate in kg/s.

varies between 0 kg/s and 0.62 kg/s. For temperature increases larger than 10 K, the maximum mass flow rate of jacket cooling water decreases.

6.4 Preliminary experimental analysis: turbine start-up and estimation of turbine throat area

6.4.1 Turbine start-up

The start-up of the ORC unit and especially of the turbine is a crucial operational phase, because the operational conditions vary rapidly and over a broad range before they can reach the desired values. During the start-up, particular effort needs to be set on the control and automation system, so that the ORC system can reach a stable operational state in the shortest amount of time possible. In this way, the produced electricity can be maximized together with the economic performance of the ORC unit. A coordination of the system actuators is necessary in this phase (cooling water system, speed of the ORC pump, bypass and control valves, etc.) such that the thermodynamic quantities do not exceed their maximum or minimum allowed values. For instance, it is crucial that the degree of superheating at the turbine inlet stays positive whenever the turbine control valve is open, otherwise the liquid droplets in the working fluid could lead to the erosion of the turbine blades and considerably reduce the turbine performance and lifetime. Nonetheless, especially during the start-up, when the turbine control valve is opened, it can be very challenging to ensure a positive degree of superheating, as discussed in the following.

Figure 21a shows the opening of the turbine control (VD101) and turbine bypass valve (VD102). During this experiment, the operating conditions of the diesel engine are kept constant. It can be seen that the closing of VD101 and the opening of VD102 are coordinated, in order to avoid sudden drops in temperature and pressure at the evaporator outlet. Considerable temperature drops can occur when the cross-sectional area of the turbine is much larger than the cross-sectional area of the bypass valve V102. The temperature drop can lead to a sudden decrease of the degree of superheating at the turbine inlet and thus the start-up process needs to be repeated again, extending the turbine downtime.

The key for a successful start-up of the ORC turbine is to keep the pressure at the turbine inlet moderate at the moment of opening the valve VD101. This can be done by keeping a high opening of the bypass valve VD102 before the start-up of the turbine. It can be seen in Figure 21a that the VD102 is 50 % at the start of the experiment. This results in a low pressure at the outlet of the exhaust gas cooler (approximately 2.3 bar, see Figure 21b). In this way, the temperature of the working fluid at the outlet of the exhaust gas cooler is not negatively affected by the opening of VD101, as shown in Figure 21c., and hence, no sudden drop of the outlet temperature of the exhaust gas cooler occurs. From the moment that VD101 reaches the desired value (t = 200 s in Figure 21a), the control system can then regulate the system as desired, and the power production can take place.

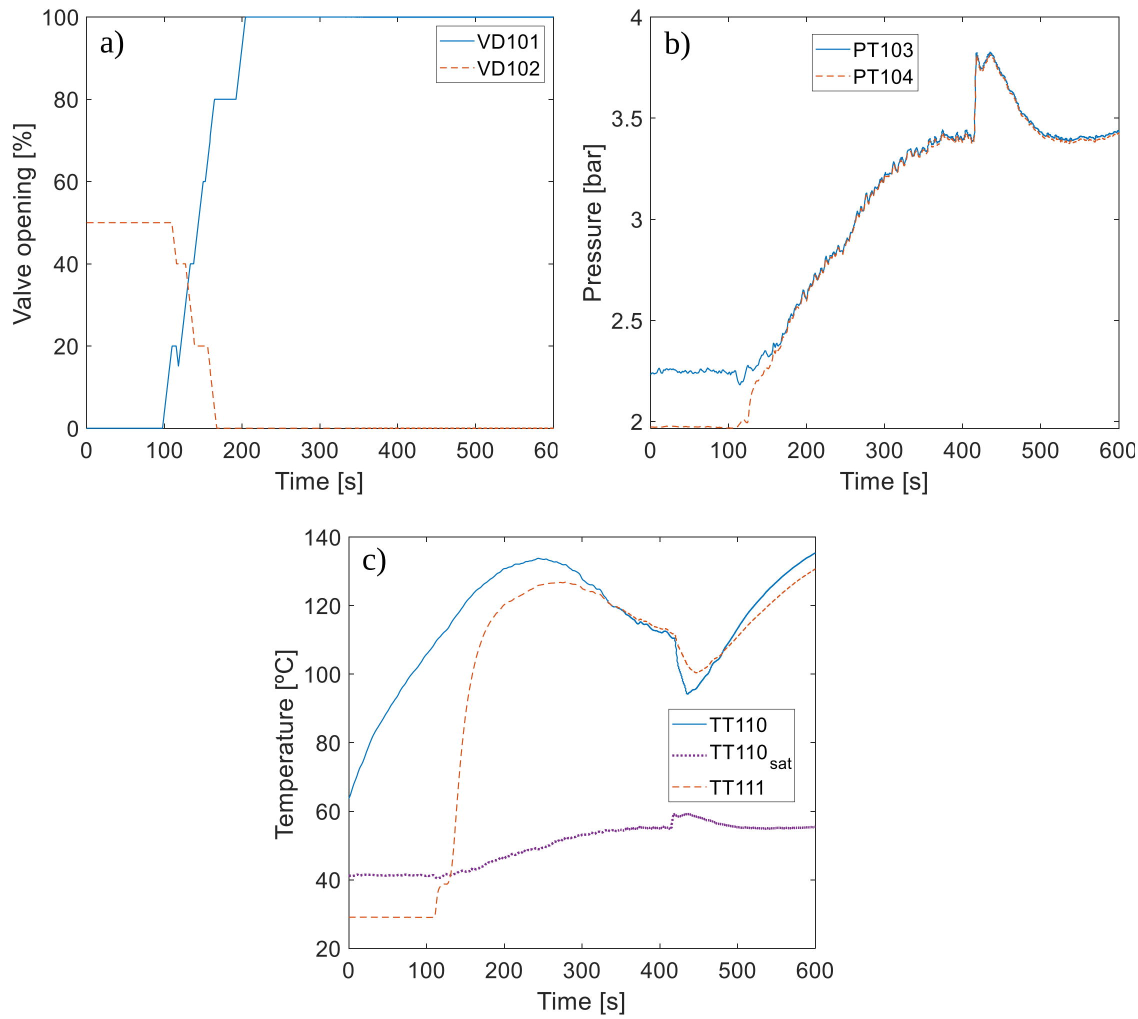


Figure 21: Successful start-up of the ORC turbine: (a) opening of turbine control and bypass valve; (b) pressure at inlet and (c) temperature at inlet.

6.4.1 Estimation of the turbine throat area

The turbine throat area is of major importance to determine the part-load behavior of the ORC unit, since this defines the relationship among the mass flow rate, the pressure and the inlet temperature of the working fluid at turbine inlet. The turbine consists of a static blade cascade (also called "stator" or "nozzle") and a rotating blade cascade (also called "rotor"). In ORC applications, given the low speed of sound of the working fluid, supersonic conditions are reached in one of the cascades. Typically, the first cascade to reach supersonic conditions is the stator, and in particular these conditions first appear in the smallest cross-section (also called "throat"). When the sonic conditions are reached in the stator throat, the part-load behavior of the turbine can be described by assuming isentropic conditions from the inlet to the throat of the turbine stator, so that the following relationships hold:

$$h_{T,in} = h(T_{T,in}, p_{T,in}) \tag{8}$$

$$s_{T,in} = s_{T,t} = s(T_{T,in}, p_{T,in}) \tag{9}$$

$$a_{T,t} = a(h_{T,t}, s_{T,t}) \tag{10}$$

$$h_{T,t} = h_{T,in} - 1/2\, a_{T,t}^2 \tag{11}$$

$$\rho_{T,t} = \rho(h_{T,t}, s_{T,t}) \tag{12}$$

$$\dot{m}_{T,in} = \dot{m}_{T,t} = \rho_{T,t} A_{T,t} a_{T,t} \tag{13}$$

where $h, T,\ p,\ s,\ a,\ \rho$ and $\dot{m}$ are respectively the specific enthalpy, temperature, pressure, specific entropy, speed of sound, density and mass flow rate at the turbine inlet ('T,in') or at the turbine stator throat ('T,t'). If experimental data about the mass flow rate, temperature and pressure at the turbine inlet are available, the stator throat area can be estimated. The estimation of the throat area is carried out through the experiment results shown in Figure 22. During this experiment, the turbine control valve was fully open (VD101 = 100 %) while the turbine bypass valve was closed (VD102 = 0 %). It can be seen that the pump rotational speed RC101 is varied stepwise from 300 to 340 rpm (corresponding to an overall change in rotational speed of 13.3 %). Consequently, also the mass flow rate MF101 and the pressure at the turbine inlet PT104 change. In order to estimate the throat area, the calculation the density and the speed of sound at the throat is performed using equations 8-12. Then, the throat area that minimizes the relative root mean square error (RRMSE) between equation 13 and the measured mass flow rate MF101 is found:

$$RRMSE = \sqrt{\frac{1}{n} \sum_{l=0}^{n} \left(1 - \frac{\rho_{T,t} A_{T,t} a_{T,t}}{MF101}\right)^2} \tag{14}$$

The result is a throat area of $3.1263 \cdot 10^{-5}$ m^2, for a RRMSE = 2.8 %, which is a reasonable value if the dynamic effects between the point where MF101 is measured (at the outlet of the ORC pump) and the turbine inlet are considered. The estimated mass flow rate using equation 13 is also highlighted in Figure 22.

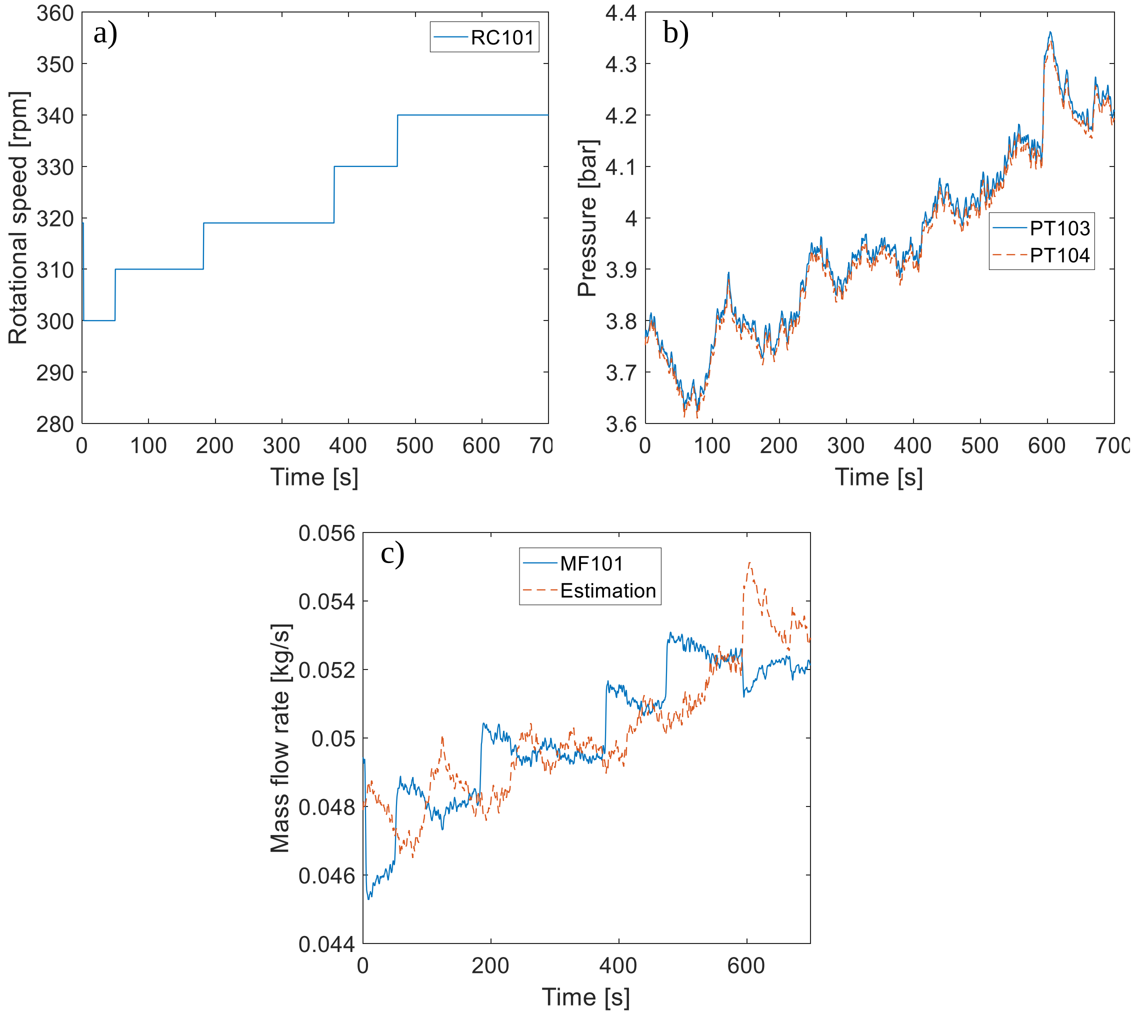


Figure 22: Estimation of the throat area of the ORC turbine: (a) rotational speed of the ORC pump; (b) pressure and (c) mass flow rate of the working fluid.

7 Conclusions

The main conclusions of the project are listed in this section, particular focus is posed on the project's deliverables.

This final report details the main activities carried out throughout the project "Waste heat recovery on liquefied natural gas-fuelled ships" and summarizes the main results and achievements. The project aimed at providing guidelines for the optimal design and integration of organic Rankine cycle-based waste heat recovery units on board liquefied natural gas-fuelled ships. The project included the development of numerical tools as well as the realization of an experimental test rig at DTU Mechanical Engineering. The following main conclusions can be drawn from the project:

i. Among the different heat sources available on board, the most attractive solutions feature the use of either the main engine exhaust gases or jacket cooling water. With respect to the cooling sink, the use of seawater is recommended, while the use of external air can represent an interesting solution for ships sailing in the arctic regions. The installation of an organic Rankine cycle using the exhaust gases as heat source and seawater as cold sink can lead to equivalent fuel savings up to 10 %, when considering the use of the produced electricity to replace the consumption on the onboard auxiliary generators. The use of the main engine jacket water as an heat source for an organic Rankine cycle can lead to fuel savings up to 1 % of the main engine annual fuel consumption;

ii. The use of the low temperature heat released by the liquefied natural gas during its preheating phase before injection to the engine as a cold sink for an organic Rankine cycle can result in cycles characterized by high thermal efficiencies but low net power outputs. Novel organic Rankine cycle architectures featuring two condenser units were presented and enable the use of both seawater and liquefied natural gas preheating as cooling media for the organic Rankine cycle. The novel layouts result in increased power outputs in comparison with the traditional cycle configurations, but are characterized by a higher degree of complexity and are expected to be more expensive;

iii. A novel method to design organic Rankine cycles as part of the whole engine machinery system was presented. The method enables to account for the effect of the additional backpressure supplied to the engine and hence to have a more accurate estimation of the attainable fuel savings. The use of the proposed method in a case study indicates that designing the organic Rankine cycle unit by accounting also for the engine performance can result in an increase of the attainable fuel savings in the range from 0.52 g/kWh to 1.45 g/kWh;

iv. It is recommended that organic Rankine cycle to be installed on board vessels are designed to maximize their economic effectiveness. In particular, this increases when with the ship's sailing time and engine power output (because larger unit have lower specific costs). Two case studies based on a feeder ship operating in Tier III zone, and a containership operating in Tier II zone indicate that payback times in the range from 5 to 10 years can be expected, depending on the fuel price. With respect to retrofit installations, the reduced space availability on board the vessel can result in a reduction of the economic attractiveness of waste heat recovery units;

v. In collaboration with Fjord Line a novel concept integrating an organic Rankine cycle and a thermal energy storage system was evaluated. The installation of the proposed concept on board a cruise ships enables the production of zero-emission power during harbor stays and therefore to a significant reduction of the impact of the ship on the harbor areas. The proposed concept was found to be more cost-effective than the installation of lithium batteries when considering life-times of more than 10 years.

vi. An experimental test rig was built at DTU Mechanical Engineering, featuring a diesel engine and an organic Rankine cycle. The setup can be used both for teaching and future research works. Heat source maps were developed to define the heat input to the organic Rankine cycle unit as a function of the diesel engine load. In the initial investigations particular focus was set on the development of a safe and effective start-up concept for the turbine of the organic Rankine cycle unit. This concept was proven experimentally and an estimation of the turbine throat area was performed.

8 Dissemination

The results of this project were disseminated to both scientific community and industry by means of publications in high impact factor journals, contributions to conferences, and presentations in workshops of relevance in the shipping sector. These contributions are listed in this chapter.

Scientific journals

[1] Baldasso E., Mondejar, M. E., and Haglind, F., *'Regression models for the evaluation of the techno-economic potential of organic Rankine cycle-based waste heat recovery systems on board ships using low sulfur fuels'*, Energies, vol 13, issue 6, 2020.

[2] Baldasso, E., Mondejar, M. E., Mazzoni, S., Romagnoli, A., and Haglind, F., *'Potential of liquefied natural gas cold energy recovery on board ships'*, Journal of Cleaner Production, vol 271, 2020.

[3] Baldasso, E., Gilormini, T.J.A., Mondejar, M.E., Jesper, J.G., Larsen, L.K., Fan, J., and Haglind, F., *'Organic Rankine cycle-based waste heat recovery system combined with thermal energy storage for emission-free power generation on ships during harbor stays'*, Journal of Cleaner Production, vol 271, 2020.

[4] Baldasso, E., Mondejar, M.E., Andreasen, J.G., Rønnenfelt, K.A.T., Nielsen, B.Ø., Haglind, F., *'Design of organic Rankine cycle power systems for maritime applications accounting for engine backpressure effects'*, Applied Thermal Engineering, vol 178, 2020.

[5] Baldasso, E., Andreasen, J. G., Mondejar, M. E., Larsen, U., and Haglind, F., *'Technical and economic feasibility of organic Rankine cycle-based waste heat recovery systems on feeder ships: Impact of nitrogen oxides emission abatement technologies'*, Energy Conversion and Management, vol 183, pp. 577-589, 2019.

[6] Baldasso, E., Elg, M., Haglind, F., and Baldi, F., *'Comparative Analysis of Linear and Non-Linear Programming Techniques for the Optimization of Ship Machinery Systems'*, Journal of Marine Science and Engineering, vol 7, issue 11, 2019.

Conference contributions

[1] Baldasso, E., Andreasen, J. A., Meroni, A., and Haglind, F., *'Performance analysis of different organic Rankine cycle configurations on board liquefied natural gas-fuelled vessels'*, in Proceedings of ECOS 2017: the 30th International conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy systems, San Diego, California, USA.

[2] Baldasso, E., Mondejar, M. E., Larsen, U., and Haglind, F., *'Prediction of the annual performance of marine organic Rankine cycle power systems'*, in

Proceedings of ECOS 2018: the 31th International conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy systems, Guimarães, Portugal.

[3] Baldasso, E., Elg, M., Haglind, F., and Baldi, F., *'A comparison of linear and non-linear programming for the optimization of ship machinery systems',* in Proceedings of MOSES 2019: the 2nd International Conference on Modeling and Optimization of Ship Energy Systems, Glasgow, United Kingdom.

[4] Zhang, J., Baldasso, E., Mancini, R., Elmegaard, B., and Haglind, F., *'Evaluation of Heat Transfer Correlations for Flow Condensation in Plate Heat Exchangers and Their Impact on The Design of Organic Rankine Cycle Systems',* in Proceedings of the 5th International Seminar on ORC Power systems, Athens, Greece, 2019.

Presentations at workshops

Baldasso, E., Haglind, F., Mondejar, M. E., Andreasen, J. A., Meroni, A., and Imran, M., *'Waste heat recovery on liquefied natural gas-fuelled ships'*, in DNV GL Nordic Maritime University Workshop 2018, Kgs. Lyngby, Denmark.

Multimedia

Project webpage: http://www.whrmaritime.mek.dtu.dk/

Teaching

Results of the project have been disseminated to students at DTU through a course and projects. In the course "41422 Applied Engineering Thermodynamics", the students were provided with an overview of the project and its main findings during the lecture regarding the utilization of low temperature heat sources for power generations.

In addition, knowledge was transferred to the students through the following projects:

[1] *'Design and optimization of flexible organic Rankine cycle unit for waste heat recovery on board liquefied natural gas-fuelled vessels'*, MSc thesis.

[2] *'Performance comparison of Power cycles for low grade heat utilization'*, special course.

[3] *'Techno-economic evaluation of the implementation of an organic Rankine cycle unit in biomass power plants'*, MSc thesis.

[4] *'Exploration and investigation of novel solutions to integrate an organic Rankine cycle on board vessels'*. BSc thesis.

[5] *'Waste heat recovery on passenger vessels for zero-emission power production in harbor'*, MSc thesis.

[6] *'Optimization of organic Rankine cycle units for waste heat recovery in liquefied natural gas-fuelled tankers equipped with exhaust gas recirculation'*, special course.

Bibliography

[1] DNV GL. Brief status on uptake of LNG as fuel. 2018.

[2] Mondejar ME, Andreasen JG, Pierobon L, Larsen U, Thern M, Haglind F. A review on the use of organic Rankine cycle power systems for marine applications. Renew Sustain Energy Rev 2018;91:126–51. doi:10.1016/j.rser.2018.03.074.

[3] Soffiato M, Frangopoulos CA, Manente G, Rech S, Lazzaretto A. Design optimization of ORC systems for waste heat recovery on board a LNG carrier. Energy Convers Manag 2015;92:523–34. doi:10.1016/j.enconman.2014.12.085.

[4] Theotokatos G, Livanos G. Techno-economical analysis of single pressure exhaust gas waste heat recovery systems in marine propulsion plants. Proc Inst Mech Eng Part M J Eng Marit Environ 2012;227:83–97. doi:10.1177/1475090212457894.

[5] Andreasen JG, Meroni A, Haglind F. A comparison of organic and steam Rankine cycle power systems for waste heat recovery on large ships. Energies 2017;10:1–23. doi:10.3390/en10040547.

[6] Haglind F, Mondejar ME, Andreasen JG, Pierobon L, Meroni A. Organic Rankine cycle unit for waste heat recovery on ships (PilotORC) - Final Report, Tech. rep. Department of Mechanical Engineering, Technical University of Denmark (DTU). Copenhagen, Denmark: 2017.

[7] Suárez de la Fuente S, Larsen U, Pierobon L, Kærn MR, Haglind F, Greig A. Selection of cooling fluid for an organic Rankine cycle unit recovering heat on a container ship sailing in the Arctic region. Energy 2017;141:975–90. doi:10.1016/j.energy.2017.09.125.

[8] Baldasso E, Andreasen JG, Meroni A, Haglind F. Performance analysis of different organic Rankine cycle configurations on board liquefied natural gas-fuelled vessels. ECOS 2017 30th Int. Conf. Effic. Cost, Optim. Simul. Environ. Impact Energy Syst., San Diego, California: 2017.

[9] MAN Energy Solutions. CEAS calculation tool 2017. https://marine.man-es.com/two-stroke/ceas (accessed December 12, 2018).

[10] Kristensen HO. Energy Demand And Exhaust Gas Emissions of Marine Engines, Tech. rep. Copenhagen, Denmark: 2012.

[11] Kari Anne Tveitaskog Rønnenfelt, MAN Energy Solutions. Private communication 2018.

[12] Baldasso E, Mondejar ME, Andreasen JG, Rønnenfelt KAT, Nielsen BØ, Haglind F. Design of organic Rankine cycle power systems for maritime applications accounting for engine backpressure effects. Appl Therm Eng 2020;178:115527. doi:10.1016/j.applthermaleng.2020.115527.

[13] Baldasso E, Andreasen JG, Mondejar ME, Larsen U, Haglind F. Technical and economic feasibility of organic Rankine cycle-based waste heat recovery systems on feeder ships : Impact of nitrogen oxides emission abatement technologies. Energy

Convers Manag 2019;183:577–89. doi:10.1016/j.enconman.2018.12.114.

[14] Baldasso E, Mondejar ME, Larsen U, Haglind F. Regression Models for the Evaluation of the Techno-Economic Potential of Organic Rankine Cycle-Based Waste Heat Recovery Systems on Board Ships Using Low Sulfur Fuels. Energies 2020;13. doi:10.3390/en13061378.

[15] Andreasen JG, Larsen U, Knudsen T, Pierobon L, Haglind F. Selection and optimization of pure and mixed working fluids for low grade heat utilization using organic rankine cycles. Energy 2014;73:204–13. doi:10.1016/j.energy.2014.06.012.

[16] Richard Turton, Richard C. Bailie, Wallace B. Whiting JAS. Analysis, Synthesis and Design of Chemical Processes. 3rd ed. Pearson Education; 2013.

[17] Becqué R, Fung F, Zhu Z. Incentive schemes for promoting green shipping - Discussion paper. 2018.

[18] Baldasso E, Gilormini TJA, Mondejar ME, Jesper JG, Larsen LK, Fan J, et al. Organic Rankine cycle-based waste heat recovery system combined with thermal energy storage for emission-free power generation on ships during harbor stays. J Clean Prod 2020;271.

DTU Mechanical Engineering
Section of Thermal Energy
Technical University of Denmark

Nils Koppels Allé, Bld. 403
DK-2800 Kgs. Lyngby
Denmark

Phone: +45 4525 4131
Fax: +45 4525 1961

www.mek.dtu.dk

July 2021