TALLINN UNIVERSITY OF TECHNOLOGY
SCHOOL OF ENGINEERING
Department of Mechanical and Industrial engineering

EVALUATING ENERGY EFFICIENCY AND EMISSION REDUCTION POTENTIAL OF WASTE HEAT RECOVERY FOR BALTIC SEA RO-RO PASSENGER VESSEL

LÄÄNEMERE RO-RO REISILAEVA HEITSOOJUSE TAASKASUTAMISE ENERGIATÕHUSUSE JA HEITKOGUSTE VÄHENDAMISE POTENTSIAALI HINDAMINE

MASTER THESIS

Student: Ilja Makarov

Student code: 204257MATM

Supervisor: Kristjan Tabri, Tenured Associate Professor

Co-supervisor: Martin Jõgeva, MSc

Tallinn 2024

(*On the reverse side of title page*)

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Department of Mechanical and Industrial engineering

THESIS TASK

Student: Ilja Makarov, 204257MATM

Study programme: MATM02/18 - Product Development and Production Engineering

main speciality: 5 - shipbuilding (Aalto University)

Supervisor(s): Tenured Associate Professor, Kristjan Tabri, +3725092585

Consultants: Martin Jõgeva, Project Manager

LTH-Baas AS, +37253063866, martin.jogeva@lth-baas.com

Thesis topic:

(in English) Evaluating Energy Efficiency and Emission Reduction Potential of Waste Heat Recovery for Baltic Sea Ro-Ro Passenger Vessel

(in Estonian) Läänemere Ro-Ro reisilaeva heitsoojuse taaskasutamise energiatõhususe ja heitkoguste vähendamise potentsiaali hindamine

Thesis main objectives:

1. Investigate the energy efficiency of Ro-Ro Passenger Vessel
2. Analyze various Waste Heat Recovery solutions available for ships
3. Develop and validate a case study ship's power plant simulation model
4. Evaluate the potential of Waste Heat Recovery solutions using the model
5. Identify the optimal energy and cost-efficient solution for the case study ship

Thesis tasks and time schedule:

No	Task description	Deadline
1.	Ship machinery system and energy flow analysis	23.02.2024
2.	Ship Waste Heat Recovery solutions market research	08.03.2024
3.	Ship's power plant simulation model in Matlab & Simulink	05.04.2024
4.	Case study ship Waste Heat Recovery solutions simulations	26.04.2024
5.	Case study ship optimal solution payback time calculation	10.05.2024

Language: English **Deadline for submission of thesis:** "20" May 2024

Student: .... "......."...................................20….a

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Supervisor: .... "......."......................20….a

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Consultant: .... "......."......................20….a

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CONTENTS

PREFACE

The thesis combines the University's academic knowledge with professional experience gained from the industry. The topic was made possible through the partnership of LTH Baas AS and the shipowner. Their support allowed me to get real ship experience, learn new skills, and explore one of the many solutions for future sustainable shipping. Energy efficiency was, is, and will always be important.

I express my gratitude to the thesis supervisor Kristjan Tabri for the encouragement. I thank the thesis co-supervisor and my colleague Martin Jõgeva for new ideas and support during the ship visits. I would also like to thank the LTH Baas management and my colleagues for their support. I am grateful with all my heart to my Mother and Father, my family, and my Tanya for inspiration.

Ilja Makarov

ship, energy efficiency, emissions reduction, waste heat recovery, master's thesis

List of abbreviations and symbols

CAC	Charge air cooler
CII	Carbon Intensity Indicator
CO2	Carbon dioxide
DCS	Data Collection System
EEA	European Economic Area
EEDI	Energy Efficiency Design Index
EEXI	Energy Efficiency eXisting ship Index
EG	Exhaust gas
EGCS	Exhaust gas cleaning system
EGE	Exhaust gas economizer
ETS	Emissions Trading System
EU	European Union
GHG	Greenhouse Gas
GT	Gross Tonnage
HFO	Heavy fuel oil
HT	High-temperature
IAMCS	Integrated Alarm, Monitoring and Control System
IMO	International Maritime Organization
LNG	Liquefied natural gas
LO	Lubricating oil
LT	Low-temperature
MARPOL	International Convention for the Prevention of Pollution from Ships
MCR	Maximum Continuous Rating
MGO	Marine gas oil
OFB	Oil-fired boiler
ORC	Organic Rankine cycle
ROPAX	Ro-Ro Passenger Ship
SECA	Sulphur Emission Control Area
SEEMP	Ship Energy Efficiency Management Plan
SRC	Steam Rankine cycle
ST	Steam turbine
WHR	Waste heat recovery

1 INTRODUCTION

Water covers 70% of Earth. Since ancient times, mankind has been using ships to transport people and cargo around the globe. Nowadays, most goods and consumables are transported by sea. The electronic device where you read this thesis most probably reached your continent or country on a container ship. Seaborne trade is growing, inevitably causing a rise in shipping emissions if there is no regulatory framework.

The International Maritime Organization is a unique regulatory body with a mission to set rules for international shipping safety and pollution prevention. The organization is committed to reducing ships' greenhouse gas emissions to address climate change by adopting new mandatory ship energy efficiency measures. The measures aim to stimulate shipowners to improve the energy efficiency of their ships using technical or operational solutions. One among others potential solutions being on the table is waste heat recovery.

Over half of the consumed fuel energy onboard a ship is converted into heat. The amount of fuel burnt in ships' engines and consequently created heat depends on ship type, size, and speed. Utilizing heat created for ship heating needs would be most efficient, but the heat available might significantly exceed the actual demand. A potential solution could be to recover thermal energy in a process that could convert it into mechanical energy to drive an electric generator to produce electricity.

An identical situation developed on a ship studied as part of the thesis. The reference ship is a large, relatively fast ro-ro passenger vessel operating in the Baltic Sea with surplus heat created at sea. The shipowner's commitment to efficient and sustainable operations provided an opportunity to study energy flows on board an operating ship. The study's outcome should determine an optimal waste heat recovery solution to improve ship energy efficiency and reduce emissions.

The problem of evaluating waste heat recovery potential is that every ship is unique. The case study ship's energy system design and operation will be investigated to understand the existing energy flows, the amount of heat available for recovery, and the current energy efficiency level. Various ship waste heat recovery solutions on the market will be analysed to find potential solutions suitable for the case study ship operations. The ship's power plant calculation model will be developed and validated to evaluate the selected solutions' energy efficiency and emission reduction potential. The optimal energy and cost efficiency solution for the case study ship will be identified.

In the background study chapter, the task is to familiarize with international shipping emissions and energy efficiency regulations, which can influence the analysis. The case study investigates ship peculiarities such as local shipping regulations, sea and weather conditions, and ro-ro passenger ship speed. The ship waste heat recovery solutions market research is carried out, analysing previous related works, articles, and information from the industry. The ship power plant and the Rankine cycle heat recovery theory are studied to define opportunities and limitations.

The evaluation methodology chapter covers the tasks related to the case study ship analysis. Based on the design and operational data from the reference ship, the ship energy system calculation model is developed and validated. The steam and organic Rankine cycle calculation model is developed to evaluate the waste heat recovery potential on board the case study ship. The models are implemented using MATLAB and Simulink software.

The results and conclusions identify the optimal waste heat recovery solution for the case study ship retrofit. The selection is based on the solutions' operational, regulatory, and financial results. The study considers solutions' risk and safety matters, the effect on the mandatory ship energy efficiency indexes, and installation complexity. The solutions' potential payback time and profit are estimated to support the decision-making process.

2 BACKGROUND STUDY

2.1 Emissions from ships

2.1.1 Global emissions

Ships play a significant role in the world economy, carrying around 80 % of global trade by volume and over 70 % by value [1]. In 2023, the United Nations Conference on Trade and Development forecasted seaborne trade to grow above 2 % annually through 2028 [2]. Even though ships are the most CO_2 emission-efficient mode of cargo transport (measured in g CO_2 per ton·km) [3], the combination of global cargo volume and further growth will not go unnoticed and will leave its footprint – the carbon footprint.

The shipping industry accounts for 2,9 % of global man-made CO_2 emissions, according to the Fourth IMO GHG Study 2020 [4]. As illustrated in Figure 2.1, during the previous years, when seaborne trade growth was positive, the sector's CO_2 emissions were up more than 10 % from 2012 levels. In the business-as-usual scenario, when no new shipping carbon intensity or energy efficiency regulations are adopted, 2050 shipping CO_2 emissions could increase by up to 50 % over 2018 levels [4]. To control the shipping impact on climate change and reach net-zero emissions by 2050, the International Maritime Organization has adopted strategies and regulations aimed at improving energy efficiency and reducing Greenhouse Gas emissions from ships across the globe.

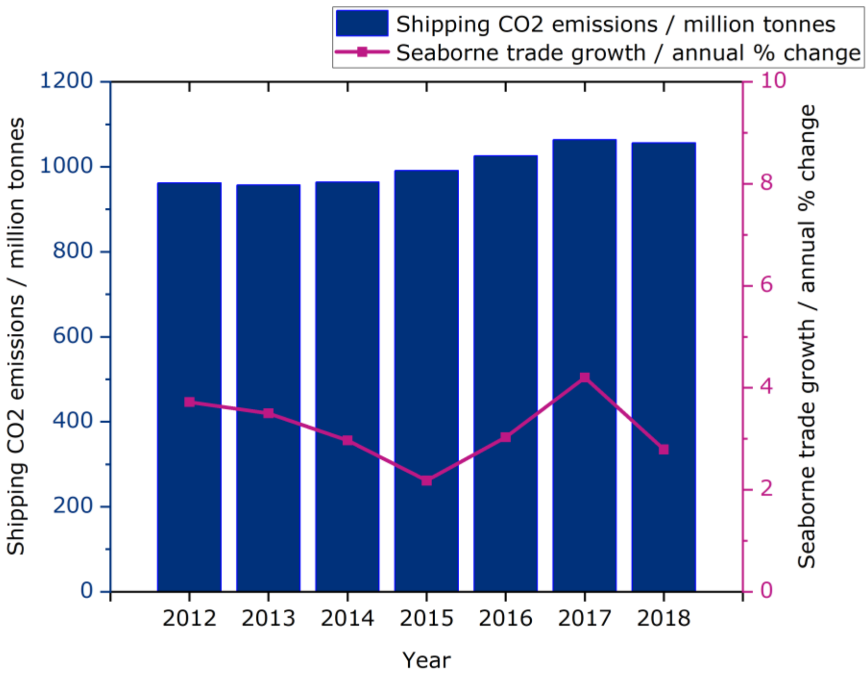

Figure 2.1 Shipping CO_2 emissions and Seaborne trade growth between 2012 and 2018

2.1.2 Emissions in Europe

In international maritime trade, Europe holds 14 % of exports and 16 % of imports [5]. 74 % of EU external trade by weight is carried by ships, which makes maritime transport the pillar of the EU economy. The share of EU internal freight transport by sea is 27,2 % after road transport with 54,3 % [6]. According to MarineTraffic's density map of global shipping in Figure 2.2, where red represents very dense shipping routes, European seas, including the Baltic Sea, have very intensive maritime traffic.

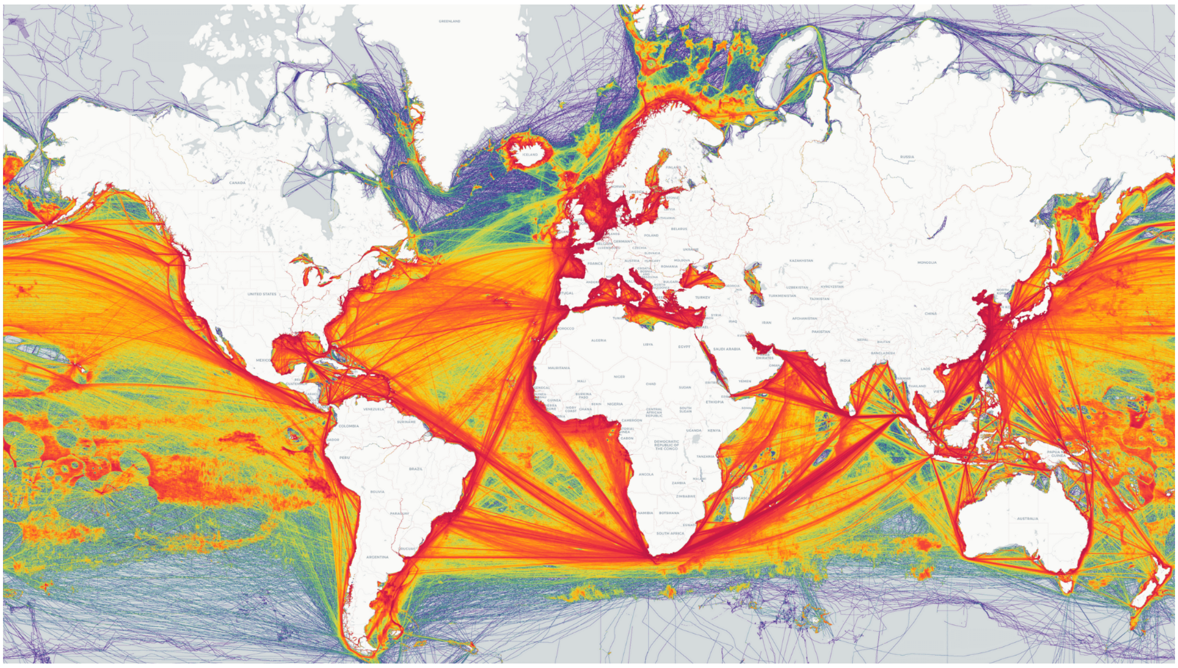

Figure 2.2 Density map of global shipping 2020 [7]

According to the European Commission report [8], maritime transport represent 3 to 4 % of total EU CO_2 emissions The rise of shipping GHG emissions, expected maritime transport demand growth, and the goal of achieving climate neutrality in Europe by 2050 have driven the European Commission to legislate measures to control ships' climate impact and promote the transition to green shipping in the European Economic Area.

2.2 Ship Energy Efficiency Regulations

2.2.1 International

The International Convention for the Prevention of Pollution from Ships (MARPOL) regulates ships' environmental impact. GHG emissions impact on climate change is covered in the Annex VI Prevention of Air Pollution from Ships [9]. The convention controls the utilization of ozone-depleting substances, sets allowable limits on nitrogen oxides NO_x and sulphur oxides SO_x emissions of marine diesel engines, and establishes mandatory technical and operational energy efficiency measures aimed at reducing GHG emissions from ships.

The first regulations to improve ships' energy efficiency were enforced in 2013, including the Energy Efficiency Design Index and the Ship Energy Efficiency Management Plan, which are mandatory measures. The EEDI regulation is an energy efficiency technical measure for newbuilds to encourage the use of innovative energy efficiency technologies, more efficient engines and propulsion systems, improved hull designs, and larger ships. The required EEDI value that a newbuild must meet gets reduced every 5 years. IMO estimated that in 2030, the impact of the EEDI regulation on GHG reduction will be between 180 and 240 million tonnes annually [10]. The SEEMP is an operational measure for ships to improve energy efficiency by optimising operations, considering new efficient technologies and practices, and monitoring ship performance over time. Each ship shall keep on board a ship-specific Ship Energy Efficiency Management Plan. To have the necessary data to decide on further ship energy efficiency measures, the Data Collection System was established. From 2019 all ships of 5000 GT and above are obliged to record and report fuel consumption and distance travelled. The data from DCS is then used to calculate the ship's operational Carbon Intensity Indicator, which is a part of the short-term measures along with the Energy Efficiency Existing Ship Index developed according to the Initial IMO strategy on reduction of GHG emissions from ships [11]. The EEXI and CII measures entered into force in 2023 and will be used in this work to assess the potential of proposed energy efficiency technologies to support the ship's compliance with the measures.

The EEXI regulation is a technical measure of a ship's energy efficiency adopted to improve the technical performance of existing ships of 400 GT and above, especially the older ships that are not in the scope of the EEDI regulation. The ship's attained EEXI is measured in gram CO_2 per ton·nautical mile. A detailed formula is provided in the guidelines on the method of calculation of EEXI [12]. The formula's simplified concept is presented in formula (2.1).

$$Attained\ EEXI = \frac{f_j \cdot CO_{2_{ME}} + CO_{2_{AE}} + f_j \cdot CO_{2_{PTI}} - f_{eff} \cdot CO_{2_{AEeff}} - f_{eff} \cdot CO_{2_{eff}}}{f_i \cdot f_c \cdot f_l \cdot Capacity \cdot f_w \cdot V_{ref} \cdot f_m}, \quad (2.1)$$

where $CO_{2_{ME}}$ – CO_2 emissions of main engines, g/h,

$CO_{2_{AE}}$ – CO_2 emissions of auxiliary engines, g/h,

$CO_{2_{PTI}}$ – CO_2 emissions of shaft motor, g/h,

$CO_{2_{AEeff}}$ – CO_2 emissions reduction from innovative mechanical energy-efficient technology for auxiliary engine, g/h,

$CO_{2_{eff}}$ – CO_2 emissions reduction from innovative mechanical energy-efficient technology for main engine, g/h,

f_j – ship-specific design elements factor,

f_{eff} – factor of each innovative energy efficiency technology,

f_i – capacity factor,

f_c – cubic capacity correction factor,

f_l – factor for general cargo ships equipped with cranes and other cargo-related gear,

f_w – factor for speed reduction at sea,

f_m – factor for ice-classed ships having IA Super and IA,

Capacity – deadweight or gross tonnage depending on ship type, t,

V_{ref} – ship speed, knot.

For the ship to comply with the regulation, the attained EEXI must be equal to or below the required EEXI, which is calculated using formula (2.2), as required in MARPOL Annex VI [9].

$$Attained\ EEXI\ \leq Required\ EEXI = \left(1 - \frac{y}{100}\right) \cdot EEDI\ reference\ line\ value\ , \qquad (2.2)$$

where y – the required reduction factor depending on ship type,

EEDI reference line – the reference line value defined as $a \cdot b^{-c}$, where a, b and c are the parameters depending on ship type.

The CII regulation is an operational measure of the ship's energy efficiency adopted to improve the operational performance of existing ships of 5000 GT and above. The CII rating is updated and verified annually based on fuel consumption and data from DCS, compared to the EEXI, which has a one-time certification. The ship's attained CII is measured in gram CO_2 per ton·nautical mile. The annual operational CII of a ship is calculated as in formula (2.3) from the guidelines [13]. The annual operational CII is subject to voyage adjustments, e.g., scenarios that may endanger the safe navigation of ship or navigation in ice conditions, and correction factors, e.g., energy consumption related to cargo handling, ship ice class, and capacity correction [14].

$$attained\ CII_{ship} = M/W\ , \quad (2.3)$$

where M – the total mass of CO_2 emissions emitted in a given calendar year, g,

W – the total transport work undertaken in a given calendar year, t·nm.

The total mass of CO_2 M is the sum of CO_2 emissions from all the fuel consumed on board a ship in a given calendar year from data reported to the DCS. M is calculated using formula (2.4).

$$M = FC_j \times C_{F_j}\ , \quad (2.4)$$

where j – the fuel type,

FC_j – the total mass of consumed fuel, g,

C_{F_j} – the fuel mass to CO_2 mass conversion factor.

The total transport work W could be actual transport work or a proxy using the ship's capacity and the distance travelled in a given year reported to the DCS. W is calculated using formula (2.5).

$$W_s = C \times D_t\ , \quad (2.5)$$

where C – the ship's capacity, depending on ship type deadweight or gross tonnage, t,

D_t – the total distance travelled, nm.

The CII rating calculation is based on the required annual operational CII defined in formula (2.6).

$$Required\ annual\ operational\ CII = (1 - Z/100) \times CII_{ref}\ , \quad (2.6)$$

where Z – the annual reduction factor as defined in Table 2.1,

CII_{ref} – the reference value defined as $a{\cdot}C^{-c}$, where a and c are the parameters depending on ship type from the guidelines [15].

Table 2.1 Reduction factor Z [16]

Year	**Reduction factor relative to 2019**
2023	5 %
2024	7 %
2025	9 %
2026	11 %
2027-2030	TBD

Ratings A, B, C, D, and E are assigned based on boundaries determined by the required annual operational CII and the ship type rating boundaries [17]. Figure 2.3 presents an example of the CII rating scale.

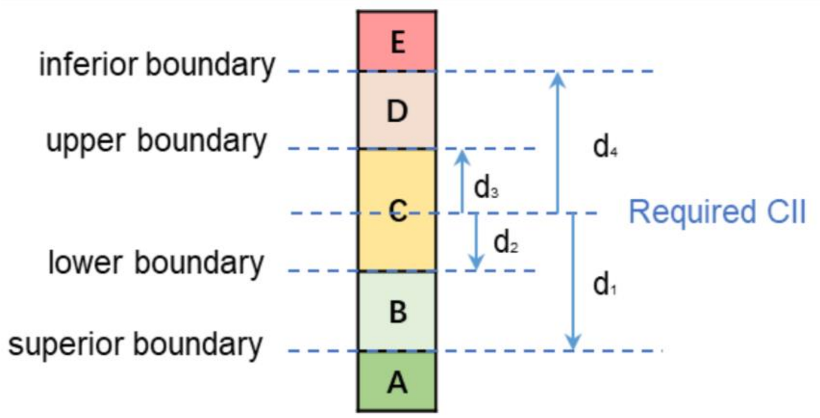

Figure 2.3 The CII rating scale [17]

Ships rated A, B, or C have a major superior, minor superior, or moderate performance, respectively, and are not obliged to take actions to improve the rating. Nevertheless, administrations, port authorities, and other stakeholders are encouraged to motivate shipowners to strive for ratings A and B. A ship rated D for three consecutive years or E shall undertake the developed plan actions to achieve the required annual operational CII [9].

Following the adoption of the Initial IMO strategy on the reduction of GHG emissions from ships in 2018, in 2023, the International Maritime Organization adopted the 2023 IMO Strategy on Reduction of GHG Emissions from Ships with new mid-term measures [18]. The new measures include a marine fuel standard aimed at the phased reduction of marine fuel's GHG intensity and a maritime GHG emissions pricing mechanism. At the same time, EU authorities have developed and implemented equal measures as part of the European Green Deal.

2.2.2 European

In 2021, to reduce EU greenhouse gas emissions by at least 55 % by 2030, the 'Fit for 55' package was proposed by the European Commission, which included actions addressing maritime transport's climate impact [19, 20]. The package consisted of the following measures:

- EU Emissions Trading System extension to the maritime sector;
- FuelEU maritime legislation to promote sustainable maritime fuels;
- Revision of the Directive on Deployment of Alternative Fuels Infrastructure;
- Revision of the Renewable Energy Directive;
- Revision of the Energy Taxation Directive.

The EU ETS is a measure to stimulate energy efficiency, deploy low-carbon intensity solutions, energy-efficient technologies, and reduce the price difference between alternative and traditional maritime fuels. Since January 2024, 5000 GT and above ships entering EEA ports are obliged to purchase and surrender emission allowances for each reported GHG ton. The amount of emissions subject to taxation is based on the data reported under the EU Monitoring, Reporting, and Verification (MRV) Regulation. In 2024, 40 % of CO_2 emissions are included in ETS, in 2025, it will increase to 70 % and reach 100 % in 2026, adding CH_4 (methane) and N_2O (nitrous oxide) emissions to the regulation. The ETS money will support innovative maritime projects striving for zero-emission shipping. Figure 2.4 by DNV shows the EU ETS coverage of the emissions from international and intra-EU/EEA voyages. 100 % of intra-EU/EEA voyage GHG emissions and 50% of into or out of EU/EEA are subject to the EU ETS. There are exceptional cases in the regulation on excluding emissions from voyages, e.g., voyages to the outermost regions and island connections until the end of 2030. The IA or IA Super or an equivalent ice class ships may surrender 5 % fewer allowances until the end of 2030 [21]. The EU ETS measure will be used in this work to assess the potential of proposed energy efficiency technologies to reduce ship's emission payments.

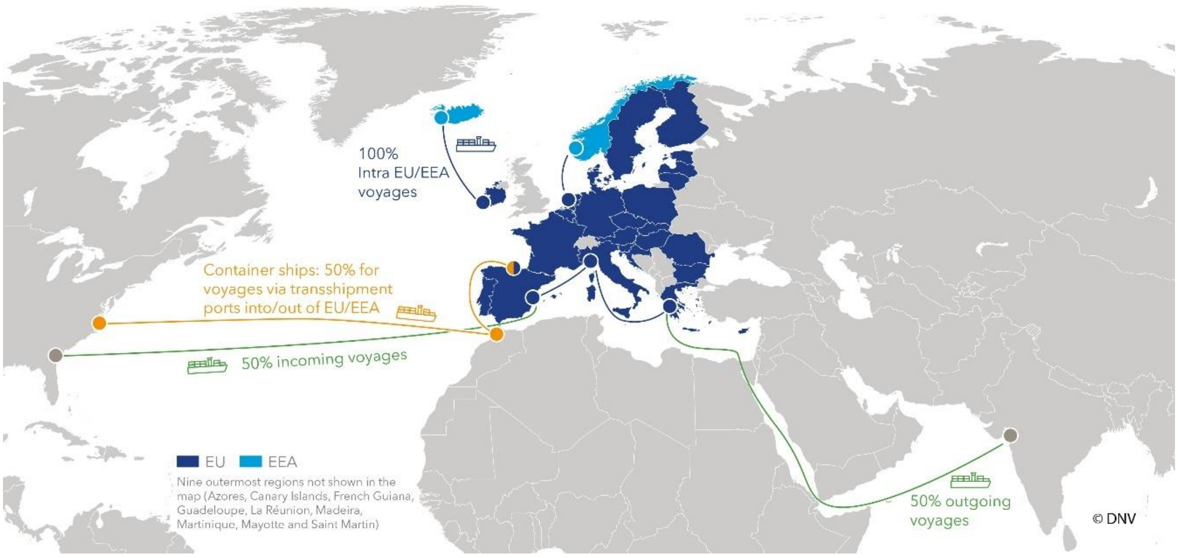

Figure 2.4 Geographic EU ETS emissions coverage [22]

The FuelEU maritime regulation is a measure to increase the utilization of renewable and low-carbon fuels and clean energy technologies for ships. The regulation sets maximum limits for the annual GHG intensity of the fuel consumed on board 5000 GT and above ship calling at European ports. The maximum limits will decrease every year starting from 2025. From 2030, the use of on-shore power supply (OPS), also named cold ironing, or alternative zero-emission technologies in EU ports will be mandatory for passenger ships and containerships to reduce air pollution in ports. However, the FuelEU regulation does not consider most energy efficiency technologies except wind assisted propulsion [23]. The revision of the Directive on Deployment of Alternative Fuels Infrastructure will support OPS port infrastructure and alternative fuel supply in the EU. The revision of Renewable Energy Directive should stimulate the increase of more expensive renewable energy use in the EU transport sector. The revision of the Energy Taxation Directive will remove tax exemptions for marine fuel in the EU [20]. The regulatory forced transition to alternative fuels will increase ships' operating costs, emphasizing energy efficiency and amplifying financial indicators of energy-saving technologies.

2.3 Baltic Sea shipping

The Baltic Sea is shipping-intensive region almost enclosed by EU countries and Russia. The 2022 Baltic Sea shipping CO_2 emissions were 11,4 % of EU total shipping CO_2 emissions [24]. Europe's modelled shipping CO_2 emissions intensity in 2011 is depicted in Figure 2.5. The narrow Baltic Sea is one of Europe's CO_2 emissions-intensive shipping regions after the North Sea, with a CO_2 emissions density of 36 t per km^2 sea surface area according to the 2011 data [25]. The EU ETS, IMO EEXI, and CII should motivate to improve ships' energy efficiency to reduce the region's shipping GHG emissions intensity. The Baltic Sea has been Sulphur oxides (SOx) Emission Control Area (SECA) since 2015, obliging ships to use maximum 0,1 % sulphur, content fuels which are more expensive than residual fuel oils or treat exhaust of residual fuel oil combustion with Exhaust Gas Cleaning Systems (EGCS). The ETS carbon tax and fuel requirement, hence increased ship operational cost, should improve the financial attractiveness of proposed energy efficiency technologies to control the costs.

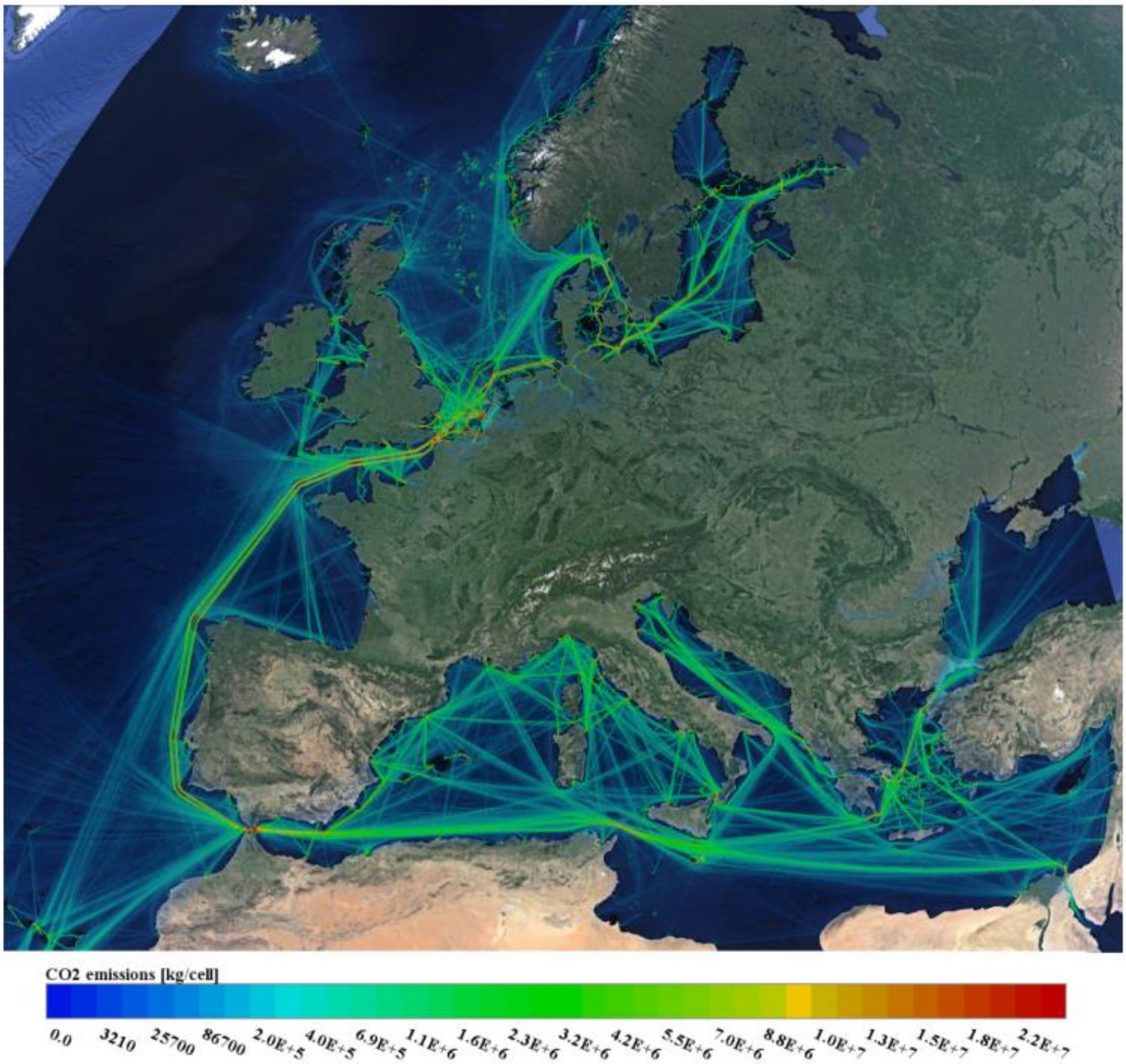

Figure 2.5 Europe's modelled shipping CO_2 emissions in 2011 [25]

The Baltic Sea climate is changing. The winters are becoming milder and ice conditions lighter due to rising air temperature [26]. The annual air temperatures variation is presented in Figure 2.6, measured on the Gotska Sandön island located in the center of the Baltic Proper [27]. The measurements' geographical location is halfway of the shipping routes between the North and South regions of the sea. During warmer winters, the ship's heating demand is not expected to increase significantly, whereas the propulsion load increases in waves or ice conditions up north. The highest waves occur during winter, caused by intense winds [26]. The estimated Baltic Sea weather and sea conditions' effect on the ship's fuel consumption is 7,9 %. An additional 2,0 % fuel consumption is estimated in the Baltic Sea winter navigation. The operation in ice conditions increases the fuel consumption by 4,6 % [24]. Again, increased ship fuel-related operational costs should highlight energy efficiency technologies' benefits.

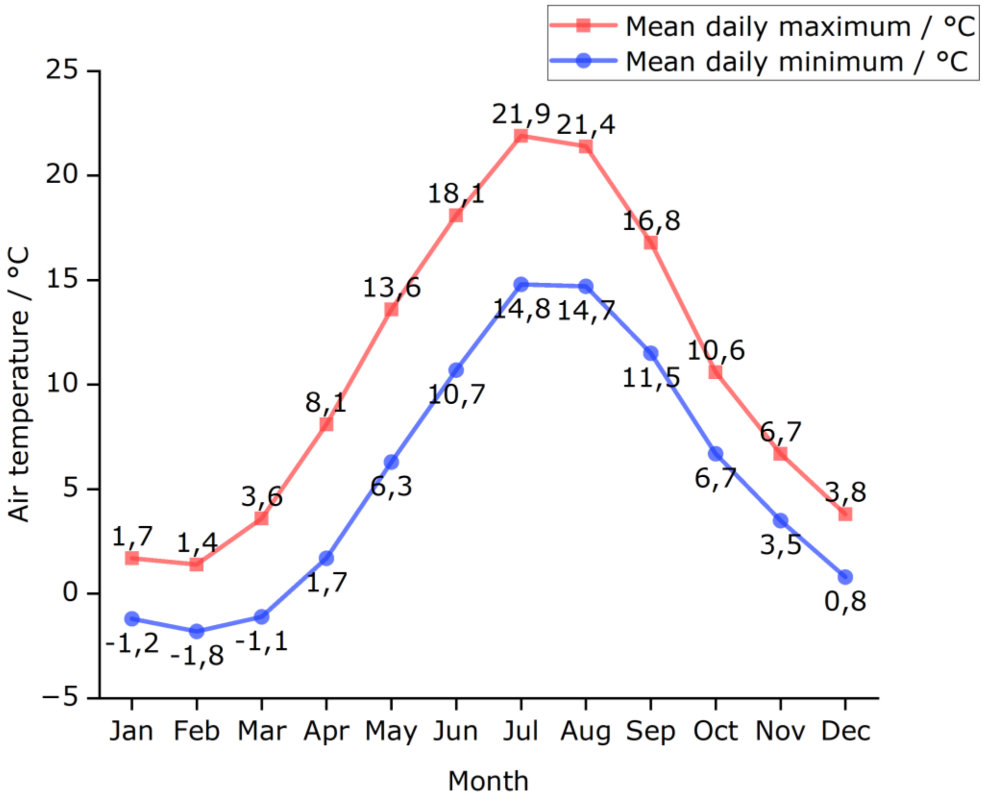

Figure 2.6 Air temperature mean daily values on Gotska Sandön island in the Baltic Sea

One of the important characteristics of the proposed heat recovery technology, which distinguishes the Baltic Sea from other EU regions, is seawater temperature. The Baltic Sea is relatively cold. Surface water temperature varies from just below zero to 4 °C during the winter. Summer temperatures can reach 20 °C at the beginning of August [28]. Figure 2.7 shows the Europe mean sea surface temperature from January 2023, where it can be seen that the seawater temperature in the Baltic Sea was close and below 4 °C. Seawater temperatures close to freezing will maximize the heat recovery technology efficiency and the ship's energy efficiency.

The largest part of shipping CO_2 emissions in the Baltic Sea in 2022 had 217 ro-ro passenger ships with a share of 27,2 %. In the second place were 2093 tankers with 19,1 %. The third place had 157 ro-ro cargo ships with 12,7 %. Next were general cargo ships, bulk carriers, and container ships with CO_2 emissions shares ranging from 9 to 11 % [24]. Based on this, the ro-ro passenger ships are the most carbon-intensive vessel type in the Baltic Sea, and they should have the highest energy efficiency improvement potential. The Baltic Sea ro-ro cargo ships could also benefit from the same energy efficiency technologies as their operation is similar to that of the ro-ro passenger ships but without the hotel services.

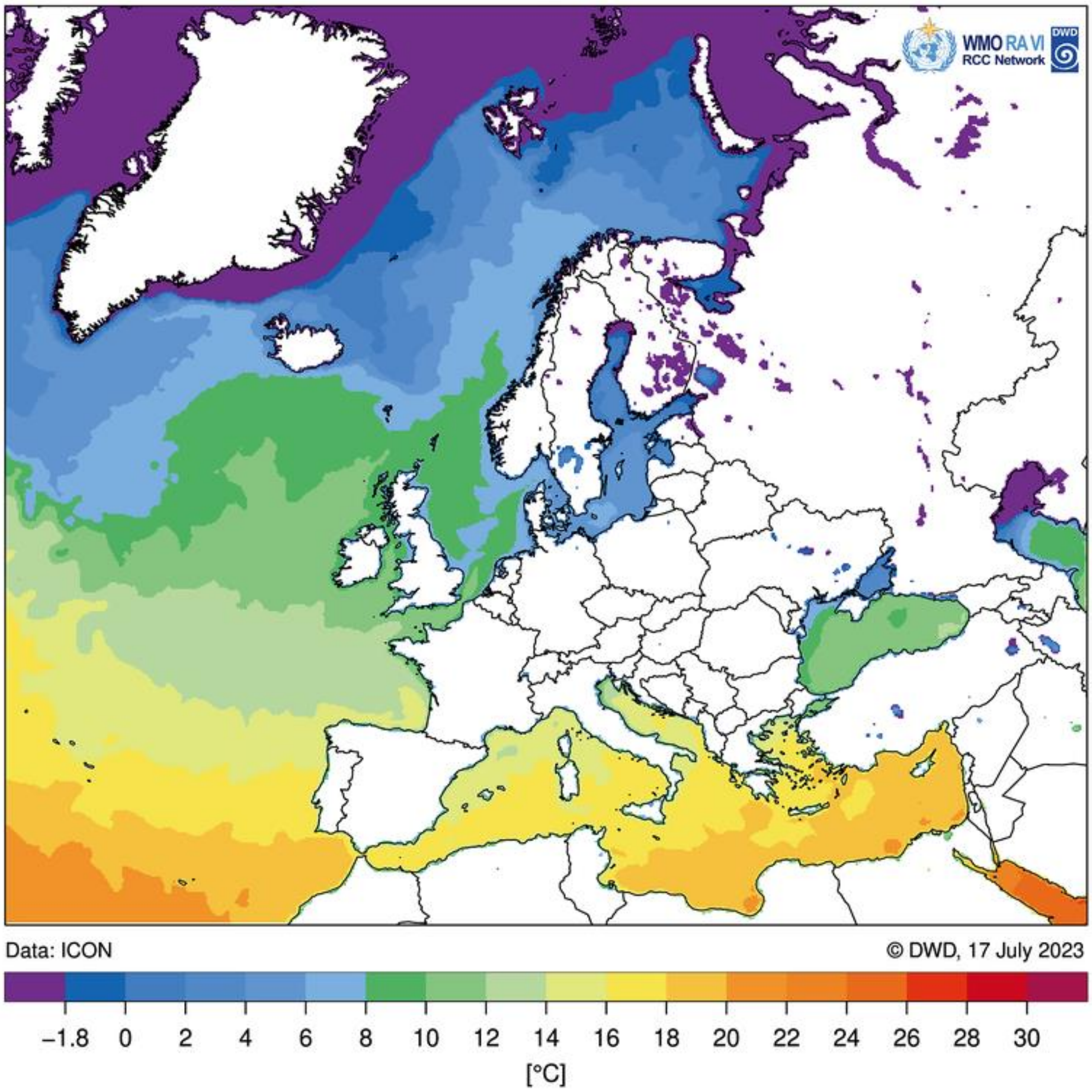

Figure 2.7 Mean sea surface temperature in January 2023 [29]

2.4 Ro-Ro Passenger Vessel

The Ro-Ro Passenger Vessel type, also named ROPAX, combines features of ro-ro cargo ship and passenger ship. The ROPAX example is shown in Figure 2.8. The common ro-ro cargo is passengers’ cars, motorcycles, trucks, trailers, and buses. The ship’s passenger spaces must provide adequate comfort level depending on the day or overnight operation profile. The ship is designed to maximize the utilization of space suitable for carrying cargo in a safe manner. Hence, the engine room layout is efficient, with little space left to install any new complex technologies that impose technical challenges. An economic challenge of retrofitting with new technologies could be ROPAX ships’ age, which in the EU is, on average, 27 years, according to European Maritime

Safety Agency 2023 data [30]. The retrofit investment must be financially sound, considering the ship's operating time left.

Figure 2.8 Ro-Ro Passenger Ship [31]

ROPAX ships are one of the fastest cargo-carrying ship types in the world, along with container ships, according to the Fourth IMO GHG Study 2018 average design speed data [4]. However, in reality, design speed does not necessarily mean operating speed. The ship's required propulsion power P_{ship} to maintain the desired speed could be estimated using the cubic relationship shown in formula (2.7). According to the estimation by Wärtsilä [32], a 10 % speed reduction would reduce required engine power and fuel consumption by 30 %, whereas a 20 % speed reduction reduces required engine power and fuel consumption by 60 %. Under the record high oil prices in 2008, one of the biggest container shipping companies, Maersk Line, to control the rising fuel cost, reduced the operating speed of its ships, creating the concept of slow steaming [33]. A positive side effect of slow steaming is reduced GHG emissions from shipping.

$$P_{ship}(v) = c_{ship} \cdot v^3 , \qquad (2.7)$$

where c_{ship} – the ship's dependent coefficient,
v – the ship's speed.

Slow steaming could not apply to the ROPAX ships in the same manner as to other cargo ships because of tight schedules between the destinations to satisfy passengers and business customers (logistics companies) seeking time-efficient connections. Hence, in the EU, ROPAX ships are the fastest ship type with high power demand and high GHG emissions. Their average time at EU seas is one of the longest, chasing ro-ro cargo ships at the top [8]. In conditions of high operating speeds, fast port operations, and low heat consumption, plenty of waste heat could be available [34]. Considering all the inputs, the waste heat recovery technology was selected to evaluate the potential for improving energy efficiency and GHG emission reduction of the ro-ro passenger vessel operating in the Baltic Sea region.

2.5 Waste heat recovery on ships

The aim of waste heat recovery for ships is to improve the ship's energy efficiency by recovering the otherwise lost thermal energy to perform useful work and, hence, to reduce the ship's required energy input by the amount of thermal energy recovered. The reduced ship's energy input reduces fuel consumption, emissions, and operating costs. Different waste heat recovery methods have different energy conversion efficiency and forms of converted energy. The recovered thermal energy can be used directly to meet the ship's heat demands, such as technical systems, freshwater production and heating, and space heating. Such waste recovery systems have high energy conversion efficiency. However, they require complex updates of existing ship systems design, and their effect on the ship's energy efficiency depends on recovered heat utilization, which varies with ambient and operational conditions, and the utilization is not always high. The same principles apply to waste heat recovery systems, which convert heat into cold for air conditioning and refrigeration. A waste heat recovery solution that is expected to be always fully utilized by ship systems is a converter of thermal power to mechanical and further to electrical power. Mechanical power could be used in mechanical propulsion system, and electrical power is always required for lights, fans, pumps, motors, communication, and navigation equipment of the ship. Most of waste heat recovery mechanical power generators are based on bottoming power cycles, such as the Rankine cycle, with potential energy savings of up to 15 %. Direct conversion of thermal power to electric power using thermoelectrical generators is not considered due to the technology's low efficiency and high cost. The described waste heat recovery technology analysis used in this work is based on Chapter 2, Energy systems on board ships, and Chapter 4, Waste hear recovery on ships, from the book Sustainable Energy Systems on Ships [35, 36].

Taking into account the lack of data describing the case study ship's real operating heating and cooling demands and, therefore, unknown recovered energy utilization, the focus is on electric power generation using the bottoming power cycles. This work studies the potential of the Rankine cycle waste heat recovery technologies application to the existing Baltic Sea ROPAX ship to improve energy efficiency, reduce fuel consumption, GHG emissions, and operating cost.

2.5.1 Case studies of the Rankine cycle waste heat recovery on ships

The Rankine cycle waste heat recovery on ships is a well-researched subject. Vainio studied heat recovery in a cruise ship being built. He optimized the ship's energy system through improved engine high temperature (HT) cooling water heat utilization onboard, a dual-pressure steam system, a high-pressure steam turbine, and an ORC. The HT cooling water ORC was not found effective in the studied cruise ship due to a lack of HT water heat after the consumers and warm seawater in the ship's operating region, which limits the ORC efficiency. The dual-pressure steam system requires significant modification of exhaust gas economizers, which increases their size so much that the ship's engine casing has to be modified to fit them. The most efficient and feasible measure for the studied cruise ship was the specific matching of the ship's heating demands with the generated heat flows [37].

El Geneidy studied potential waste heat recovery technologies to the improve energy efficiency of the cruise ship, the same ship whose design was studied by Vainio, based on the ship's real operating data. The novelty presented was engine low temperature (LT) cooling water heat recovery using a heat pump. The three most energy-efficient configurations all included the heat pump heating LT cooling water to be used as HT water. Their efficiency increased by 1 % to 1,4 % from the reference case. However, these configurations increased the overall fuel consumption from the reference case due to the increased electric power consumption by the heat pump. The three most fuel-efficient configurations all included an ORC. The ORCs were using excess HT cooling water heat available, meaning that the built ship HT system was not optimized as suggested by Vainio. The two most fuel-efficient configurations included a steam turbine [38].

Söderholm studied the optimization of another cruise ship operating in the Caribbean Sea. The waste heat recovery technologies considered were a steam turbine, an ORC for electric power generation, and an absorption chiller for chilled water production. The steam turbine power generation depends on steam availability, which depends on the ship's operating profile and primary steam consumers. When steam is used for freshwater generation, the steam turbine is not profitable. The warm Caribbean seawater limits the ORC efficiency. The ship's high cooling demands and the absorption chiller's higher energy conversion efficiency demonstrated the shortest payback time being the preferred waste heat recovery solution for the case studied [39].

Eronen studied the application of an ORC and an absorption chiller to a semi-small imaginary cruise ship during its design phase for operation in warm climates out of US ports. The ORC installation would have reduced the ship's annual fuel consumption by just less than 1 %, with a payback time of 6 years. Again, in the warm climate, the absorption chiller demonstrated the best performance, reducing the ship's annual fuel consumption between 1,4 % to 2,8 %, reaching a payback time of 1 to 2 years. From the ship's installation space and weight analysis, the ORC and the absorption chiller were considered possible to implement onboard [40].

The reviewed master's theses [37-40] focused on cruise ships operating in warm climates like the Caribbean Sea. The cruise ships produce fresh water on board, which is a large thermal power consumer, and there could not be enough quality heat left after the consumers to be used for a steam turbine or an ORC [35]. The ship's cooling demand is high. Thus waste heat recovery solutions for cooling as an absorption chiller should be prioritized because of their higher energy conversion efficiency. The warm seawater limits the efficiency of an ORC [36]. Therefore, efficient application of the Rankine cycle waste heat recovery is ship's type, operating environment, and profile specific.

More literature was reviewed to provide a broader view of the waste heat recovery potential on different ships operating worldwide. Ma et al. studied retrofitting a steam turbine generator combined with an exhaust gas power turbine generator on a 10000 TEU container ship. The steam system was single pressure, and the steam turbine was of condensing type. At an engine load of 85 %, the thermal efficiency increased from 50,6 % to 53,8 % (efficiency improvement of 6,3 %) using the steam turbine [41]. The manufacturer of combined waste heat recovery systems (WHRS) MAN stated that the electrical energy recovery potential of single-pressure steam turbine generator is 4 % to 7 %. At the same, MAN stated that the dual-pressure steam system would bring an additional 1 % to the recovery potential [42]. Similar WHRS products could be found in portfolios of ABB and Mitsubishi Heavy Industries [43, 44]. It is important to note that manufacturers use energy efficiency improvement relative values and not the absolute fuel energy savings, which in Ma et al. case study was 3,2% [41].

Livanos et al. investigated the steam Rankine cycle application on a typical ferry or ro-ro ship. The study steam system was of single pressure with a superheater. The ship steam consumers were not considered. The energy efficiency increase from the steam turbine was from 2,5 % to 3,5 % [45]. Altosole et al. compared single-pressure saturated and superheated steam waste heat recovery systems for an existing cruise ferry. The superheated system steam turbine improved energy efficiency by 2,34 %. The saturated steam system Rankine cycle application improved energy efficiency by 2,31 % [46].

The real-world applications of the steam turbines on board passenger ships include the largest cruise ships and a ROPAX. According to the online Vessel Register for DNV [47], Royal Caribbean's Icon of the Seas has a steam turbine for auxiliary power generation. In 2024, Icon of the Seas is the largest LNG-powered cruise ship in the world, with installed engine power of 89310 kW. The manufacturer of the steam turbine installed on the Icon of the Seas is Fincantieri Marine Systems [48]. The product name suggests that the turbine design power is 2500 kW (2,8 % of installed engine power), and the generator voltage is 11 kV. The high voltage generator power leads to the conclusion that the steam turbine power could support the electric propulsion, which on diesel-electric ships is powered through the high voltage network. The steam turbine produces 6 % of the electrical power needed aboard the ship [49]. According to the online Vessel Register for DNV [47], Royal Caribbean's Oasis class ships (except Oasis of the Seas and Allure of the Seas) also have steam turbines installed.

The ROPAX equipped with a steam turbine is Viking Glory. Viking Glory is LNG-powered ROPAX/cruise ferry operating in the Baltic Sea. The ship's installed engine power is 33000 kW [50]. The waste heat recovery package included two steam turbines with a design power of 150 kW each, which were delivered by Climeon (0,9 % of installed engine power). The steam turbines' operating temperature is 180 °C. Climeon's waste heat recovery package for Viking Glory also included ORC turbines with a design power of 600 kW (1,8 % of installed engine power) [51]. The total waste heat recovery power on board Viking Glory is 900 kW (2,7 % of installed engine power).

The steam Rankine cycle waste heat recovery on board ships is feasible. The newbuild passenger ship applications suggest that the onboard steam turbine generator system design and installation is possible. However, the technical and economic feasibility of existing ships' steam turbine retrofits has to be analyzed on a ship-specific basis. The marine steam turbines may require additional steam superheaters, increasing retrofit costs. For the existing ships, an organic Rankine cycle waste heat recovery can be considered as an alternative.

The important distinction between the ORC application in the articles from the reviewed master's theses [37-40] is the ORC heat source. In the articles, the aim is on the engine exhaust gas heat recovery to reach the highest possible efficiency. In contrast, in the theses, the heat recovery is targeted to the engine cooling water heat, which is more practical since the engine exhaust gas heat is often already recovered for steam production. Larsen et al. compared a steam turbine, an ORC, and the Kalina cycle potential application on board a container ship with a capacity of 2500 TEU. The exhaust gas heat recovery with the ORC demonstrated 2,6 % fuel energy recovery. An intermediate heat transfer fluid loop was suggested for safety reasons despite its adverse effect on the heat transfer efficiency between the exhaust gas and the organic fluid. The steam turbine energy recovery was 1,7 %. The efficiency of the Kalina cycle, a Rankine cycle variation with ammonia solution as the working fluid, was comparable with the steam Rankine cycle efficiency. The ammonia toxicity and cycle relative complexity discourage the Kalina cycle application on board [52].

Mondejar et al. simulated a regenerative ORC for a small cruise ship operating in the Baltic Sea. The ORC used only the engine exhaust gases directly as a heat source to improve efficiency despite the flammability of the working fluid. The estimated power output of the ORC rose to almost 400 kW, representing 22 % of the total electric power consumption on board [53]. The advantage of the Baltic Sea cold seawater was not revealed since the ORC condenser was using freshwater cooling with temperature close to 30 °C.

de la Fuente et al. compared the SRC and the ORC application on an Aframax tanker operating between the North Sea and the Baltic Sea with a seawater temperature of 5 °C. The SRC generated mechanical power was 2 %, and the ORC generated power was 2,2 % from the ship's installed engine power. The ORC power generation represented 30 % of the diesel generator's electric power output. The SRC had the fastest discounted payback time, just over 2,5 years, with water mass flow rates and heat transfer areas of up to 8,6 times and 3,2 times smaller than the ORC [54].

Uusitalo et al. investigated excess steam utilization for electrical power production on cruise ships. The waste heat recovery system used available superheated steam after the consumers in a 1 MW 4-stage radial outflow turbine. The turbine exhaust steam was condensed using a heat transfer circuit to heat the low-temperature ORC. The investigation demonstrated that the SRC alone could produce 1 % to 1,5 % of the ship's engine energy production. Combining the ORC fed only with condensate heat did not improve the efficiency [55].

Elg et al. optimized the energy efficiency of an environmentally sustainable cruise ship by introducing SRC, ORC, and a battery system. The case ship represents a typical 4000-passenger cruise ship with an installed engine power of 75,6 MW. The optimization case with a backpressure steam turbine and the battery system could save up to 1,3 % fuel. Combining the steam and Rankine cycles and the battery system revealed maximum fuel savings of 3,9 %. The combination investment cost was very high. According to the article, the best combination would be an increased capacity condensing steam turbine and an HT engine cooling water ORC [56].

The real-world applications of ORC on board ships, in addition to the previously mentioned ROPAX Viking Glory, include a container ship, a cruise ship, and a catamaran ferry. The container ship Arnold Maersk was retrofitted with 125 kW ORC developed by Calnetix Technologies and Mitsubishi Heavy Industries. The system working fluid is R-245fa, heated by engine high temperature 85 °C – 95 °C cooling water. The condenser design seawater temperature was limited to 27 °C. The ORC actual average output was in the range of 110-115 kW during ocean crossings. The ship's electrical energy savings were 9 % from ORC electric power generation from main engine waste heat. The ORC electric power output dropped by 12 % when seawater temperature increased by 10 °C [57].

The cruise ship with ORC installed on board is Scarlet Lady of Virgin Voyages. Two other Virgin Voyages have the same ORC systems on board. Each Virgin Voyages ships has an installed engine power of 48000 kW. The ORC system supplier is Climeon. The ORC total design power output is 900 kW (1,9 % of the ship's installed engine power). The heat source is engine cooling water or exhaust gas. The ships offer cruises to destinations around the world [58].

The catamaran ferry Willem Barentsz and her sister ship have ORC installed on board. The ferry has LNG-powered engines with an installed power of 2984 kW. The ORC efficiency PACKs from Orcan Energy will provide a power output of 154 kW (5,2 % of the ship's installed engine power) when the engines operate under full load [59].

The overview of the case studies of the Rankine cycle waste heat recovery on ships is presented in Table 2.2. In cases where operational energy recovery data is unavailable, the energy recovery rate is provided based on the ship's installed engine power. In case engine efficiency is 50 %, the energy recovery fuel equivalent could be roughly estimated by dividing installed engine power by two. The review's main criteria is the case study's potential energy recovery. Depending on the case study data availability, the energy recovery is presented using installed engine power or fuel equivalent energy value. The Rankine cycle waste heat recovery on a container ship, a ro-ro cargo ship, a ro-ro passenger/ferry, and an advanced cruise ship demonstrated fuel energy recovery from 2 % to 4 %.

Table 2.2 The Rankine cycle ship waste heat recovery case studies review

Ship type	**Cycle**	**Energy recovery / fuel equivalent or installed engine power**	**Reference**
Cruise ship (Mein Schiff 3)	SRC	8200 MWh	[37]
Cruise ship (Mein Schiff 3)	SRC	0,4 %	[38]
	ORC (HT)	0,6 %	
	SRC + ORC (HT)	0,7 %	
Cruise ship	SRC	0,6 %	[39]
	ORC (HT)	0,5 %	
Cruise ship	ORC (HT) (Climeon)	1 %	[40]
Container ship	SRC	3,2 %	[41]
Ferry / Ro-Ro ship	SRC	2,5 % – 3,5 %	[45]
Cruise ferry	SRC	2,3 %	[46]
Cruise ship (Icon of the Seas)	SRC (Fincantieri)	2,8 % of installed engine power	[47]
RoPax / Cruise ferry (Viking Glory)	SRC + ORC (Climeon)	2,7 % from installed engine power	[51]
Container ship	SRC	1,7 %	[52]
	ORC (EG)	2,6 %	
Cruise ship (Birka Stockholm)	ORC (EG)	1,2 % of installed engine power	[53]
Tanker	SRC	2 % of installed engine power	[54]
	ORC (EG)	2,2 % of installed engine power	
Cruise ship	SRC	1 % - 1,5 % of operating engine power	[55]
Cruise ship	SRC (incl. battery)	1,3 %	[56]
	SRC + ORC (HT) (incl. battery)	3,9 %	
Container ship (Arnold Maersk)	ORC (HT) (Calnetix & MHI)	0,2 % of main engine power 4 % fuel for electric generators	[57]
Cruise ship (Scarlet Lady, Valiant Lady, Resilient Lady)	ORC (HT+EG) (Climeon)	1,9 % of installed engine power	[58]
Catamaran ferry	ORC (Orcan Energy)	5,2 % of installed engine power	[59]

2.5.2 Main Engine

The ship main engine type studied in this work is Diesel engine, named after creator Rudolf Diesel. Turbocharged medium-speed four-stroke diesel engines dominate the ro-ro passenger ship engine sector. The medium-speed engines offer high power-to-weight and volume ratios, crucial for increasing ships' useful capacity [60]. In 2024, the world's most efficient four-stroke diesel engine is Wärtsilä 31. At 85 % MCR Wärtsilä 31 ME energy conversion efficiency is 49,5 % [61]. The studied ro-ro passenger ship's engine Wärtsilä 46D ME energy conversion efficiency at 85 % MCR is 47,4 % [62]. A decade after the case study ship was built, the engines' energy efficiency has improved by 2,1 %. Nevertheless, half of the fuel energy is still lost without the waste heat recovery. Figure 2.9 shows the energy balance of the case study ship's engine Wärtsilä 9L46D with 10395 kW MCR.

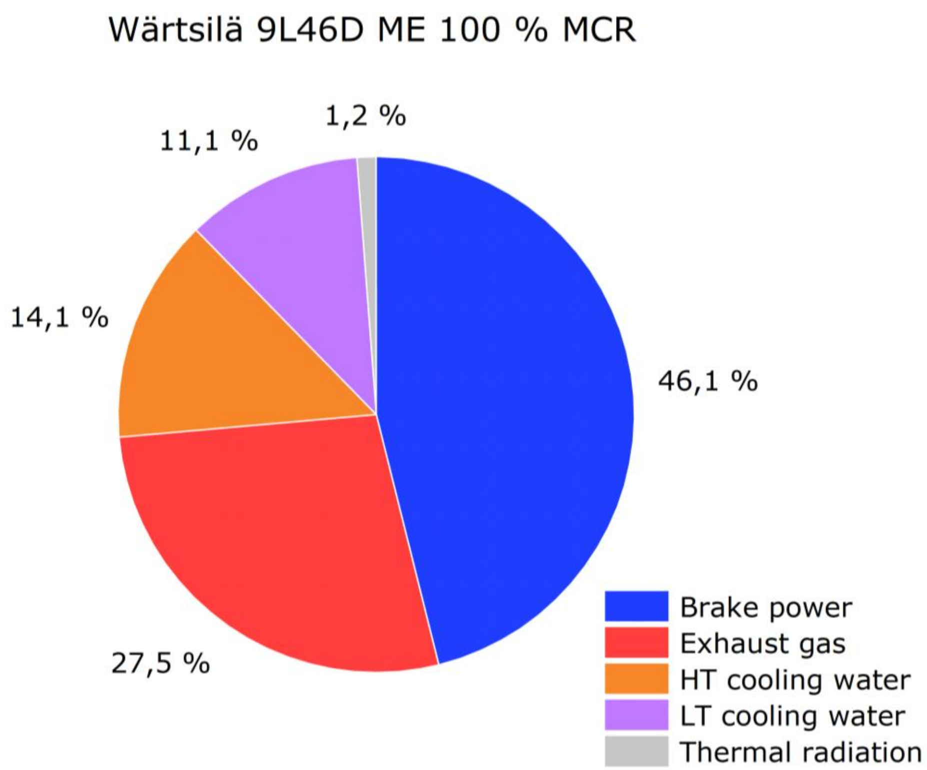

Figure 2.9 The case study ship's engine energy balance

Most energy is lost through exhaust gas (EG). The exhaust gas temperature and mass flow depend on the engine load. The average operating exhaust gas mass flow rate of the case study ship's 9-cylinder engine is 10 kg/s to 20 kg/s. At average operating load, exhaust gas temperature is above 320 °C and can reach 360 °C and 390 °C at part load and full load conditions, respectively. The engine exhaust gas temperature rises further if the engine suction air temperature is high as it is during summer. When the engine room cools down as it is during winter, the suction air temperature and the exhaust gas temperature drop down [62]. The engine exhaust gas is a high-quality heat source. Usually, an exhaust gas economizer or boiler is a part of a ship's exhaust gas system, which recovers heat to produce steam. Economizer exhaust gas waste heat recovery is a common technology that has its limitations depending on the operating conditions. The limitations are discussed under 2.5.3 Economizer item. Figure 2.10 shows the engine's simplified thermal energy flows.

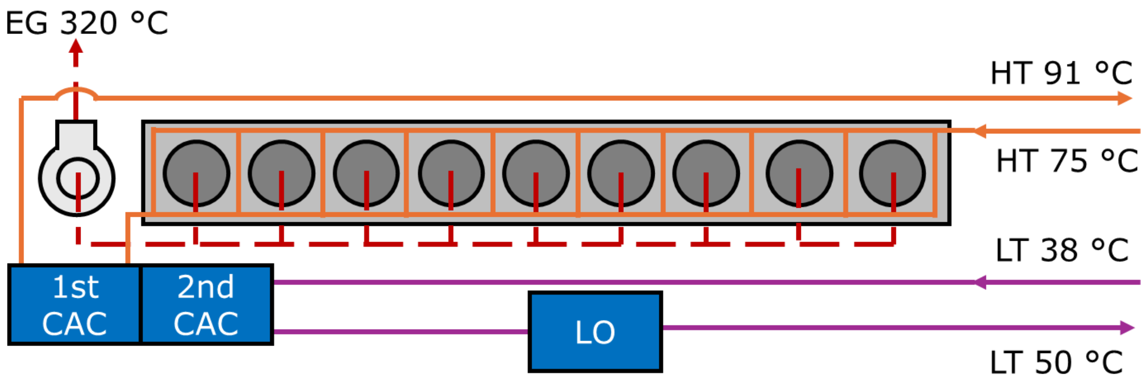

Figure 2.10 The case study ship's engine simplified thermal energy flows

The engine high-temperature cooling water is the second largest thermal energy source. The quantity of HT cooling water energy depends on the engine load. The engine cooling water temperature set-points are fixed values derived by the engine's manufacturer for safe and efficient operation in all design conditions. The HT water energy content increases when the engine suction air temperature is higher. The regulating valves control the amount of cooling. The common intake HT cooling water temperature set-point is 75 °C. The HT water cools cylinder jackets, cylinder heads, and the first stage of the charge air cooler (1st CAC). The engine outlet HT water temperature set-point is 91 °C [62]. The engine HT cooling water could be considered as a medium-quality heat source. Usually, a freshwater evaporator uses HT water heat to produce freshwater in a vacuum. The HT water energy can be used for the Organic Rankine cycle waste heat recovery. If the HT water heat recovery is considered, special attention is required to the circuit's temperature control to prevent engine overheating or undercooling, which may lead to engine damage.

The engine low-temperature cooling water takes away one-tenth of the fuel energy. Just like the HT water, the quantity of LT water energy depends on the engine load, and cooling is controlled by the regulating valves. The common intake LT cooling water temperature set-point is 38 °C. The LT water cools the second stage of the charge air cooler (2nd CAC) and the lubricating oil (LO). The LT water outlet temperature is around 50 °C [62]. The engine LT cooling water is a low-quality heat source usually not recovered. The LT water can potentially be used for preheating or matching low-temperature consumers' heat demand.

Engine thermal radiation is not considered for heat recovery even though technologies such as the thermoelectrical generators are available due to low efficiency and high cost. In cold climates, engine thermal radiation heats up the engine room, which is important to prevent the engine from under cooling.

2.5.3 Economizer

The economizer is a shell-and-tube heat exchanger used to recover engine exhaust gas heat to produce steam. The most common type is the water tube type. The exhaust gas flows inside the shell, heating the water circulating in the tubes. The economizer does not have a steam drum, and the steam produced is transferred to an oil-fired boiler’s steam drum. The oil-fired boiler is used to support steam production if needed and during port stays.

The ship steam system pressure is often equal to 8 bar absolute (steam temperature 170 °C) [63]. The case study ship’s steam system design pressure is 9 bar absolute (steam temperature 175 °C). The increased system pressure helps transfer steam to the consumers around the ship. The steam used on board is in its saturated state, however, some water may be present in the system. If a blade turbine is installed to recover energy from the excess steam, an economizer with a superheater section must be installed. Superheating is required to prevent steam condensation and turbine blade damage. A dual-pressure steam system improves the steam Rankine cycle heat recovery efficiency, but the system is more complex and requires more space [42]. To profitably recover the high-pressure steam energy on an existing ship with a saturated steam system a solution with minimum modifications is required.

Figure 2.11 shows the simplified arrangement of ship’s steam system. The feed pump (FP) raises the water pressure, transferring the water from the hot well to the oil-fired boiler (OFB). The circulating pump (CP) is used to continuously transfer the water to the economizer. The steam collecting in the OFB steam drum flows through the ship’s steam system to the consumers and returns in the hot well as condensate. The surplus steam energy is dumped in the condenser. The condensate collects in the hot well, and the steam production cycle repeats.

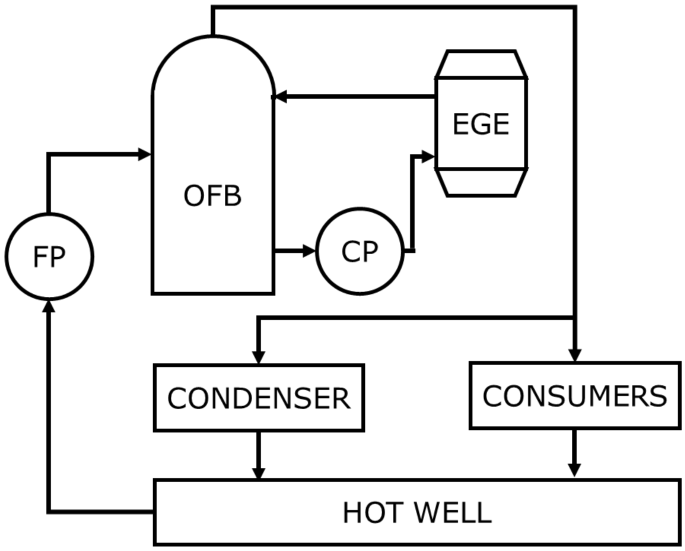

Figure 2.11 Ship steam system simplified arrangement

The exhaust gas heat recovery has operational and design constraints. The operational constraint is related to the fuel sulphur content. As the exhaust gas cools down in the economizer, condensation may occur. In moist conditions metal corrode. If the exhaust gas sulphur oxides condense and sulphuric acid forms, it will significantly speed up corrosion, leading to exhaust gas leakages and repair costs. Residual heavy fuel oil (HFO) has the highest sulphur content. Distillate fuel, such as marine gas oil (MGO) has lower sulphur content. Fuel blends such as very low sulphur fuel oil (VLSFO) are between them. Each fuel has its own sulphuric acid dew point temperature, but in general, a temperature of 165 °C can be used as an exhaust gas cooling limit to prevent condensation [63]. Burning cleaner fuels such as LNG or methanol should reduce the temperature limit and allow more waste heat to be recovered. Ship economizers' cold side exhaust gas temperature is well above the 165 °C limit due to the design constraints.

The exhaust gas heat recovery design constraint is the pinch point. The pinch point determines the economizer's efficiency. It is the lowest temperature difference between the exhaust gas and the steam. The economizer with a lower pinch point has higher heat recovery efficiency. However, the heat transfer area of a low pinch point economizer is larger, which requires more installation space and adds material weight and cost. The minimum recommended pinch point to prevent economizer fires due to soot deposits is 20 °C [63]. The case study ship's economizer design pinch point is 75 °C, which is demonstrated in Figure 2.12. The studied economizer is co-current with a high pinch point. The design could be dictated by material selection and space restrictions in the engine casing.

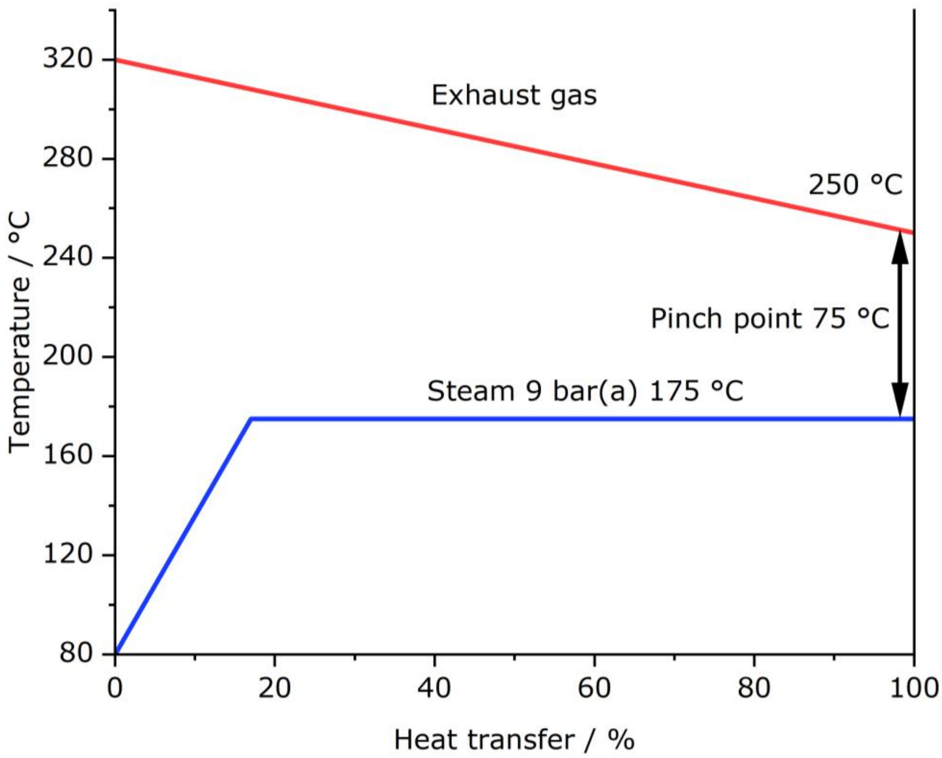

Figure 2.12 The case study ship's economizer heat transfer diagram

2.5.4 Steam Rankine cycle

The Rankine cycle, named after its creator, William Rankine, is a thermodynamic power generation cycle where mechanical power is generated by working fluid expanding through a turbine. In the steam Rankine cycle (SRC) the working fluid is water. The steam power cycle was widely used in the past in ship propulsion before Diesel engines, and it is still used on nuclear icebreakers and naval vessels, such as nuclear-powered aircraft carriers and submarines. High fuel prices and environmental regulations promote the steam Rankine cycle application for waste heat recovery on ships. The ship machinery manufacturers have commercially available waste heat recovery systems based on the steam Rankine cycle [42-44].

The case study ship has most of the steam Rankine cycle machinery components on board in the steam system. In the ship steam system, consumers are different machineries using steam for heating, whereas in the Rankine cycle, the steam consumer is the turbine, which converts heat to work. The turbine shaft connected to an electric generator allows to produce electricity. Figure 2.13 shows the components of the Rankine cycle. The high-pressure vapor (state 1) after expansion in the turbine (turbine work W_t) becomes low-pressure vapor (state 2). The vapor is cooled in the condenser (heat out Q_{out}), so the fluid phase changes to liquid (state 3). The pump raises the pressure in the system (pump work W_p) and transfers the liquid to the boiler (state 4). In the boiler, the liquid is heated (heat in Q_{in}), so the fluid phase changes to vapor (state 1). The cycle repeats.

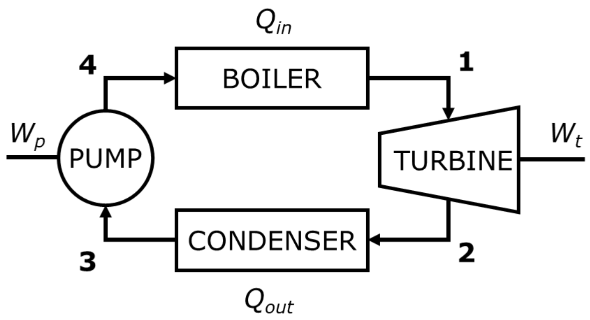

Figure 2.13 The components of the Rankine cycle

The initial assessment of the waste heat recovery potential of SRC on board the case study ship is done using the concept of ideal Rankine cycle. The graphical representation of the cycle is presented in Figure 2.14. The heat transfer processes 2-3 (condenser 1 atm 100 °C) and 4-1 (steam 9 bar(a) 175 °C) are isobaric. The expansion and compression processes are isentropic and adiabatic. All processes are internally reversible.

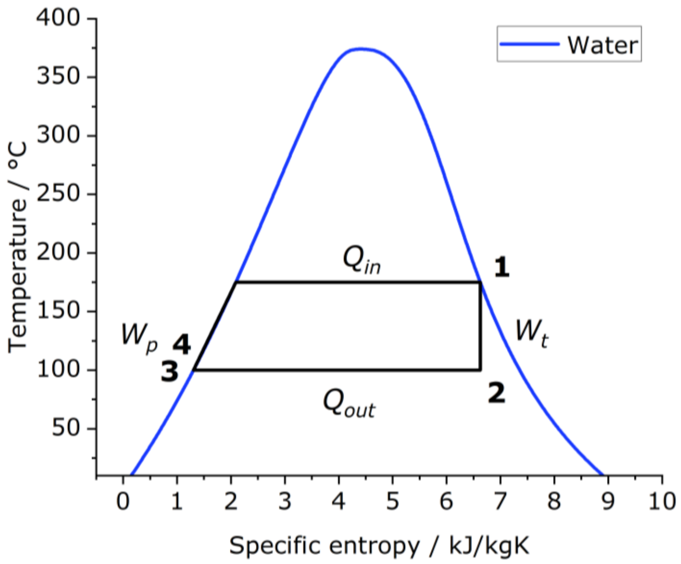

Figure 2.14 The case study ship's ideal steam Rankine cycle T-s diagram

The ideal cycle thermal efficiency η is calculated using formula (2.8) [64].

$$\eta = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}, \tag{2.8}$$

where h_i – is the specific enthalpy of working fluid in the respective state, J/kg.

The calculated heat-to-work conversion efficiency of the case study ship ideal steam Rankine cycle, converted to percents, is 15,8 %. The real cycle efficiency, including irreversibilities and losses, will be lower. The irreversibilities are caused by turbine and pump, pressure and heat losses. The cycle efficiency can be improved by increasing the steam pressure, superheating, and lowering the condenser pressure, which increases complexity and costs.

The wet expansion of steam in the turbine requires a special solution. Steam expanding between the states 1-2 under the saturation curve will have vapor and liquid present, as shown in Figure 2.14. The forming liquid droplets can damage the turbine blades rotating at high speeds. Therefore, installation of SRC on ships requires superheaters to prevent condensation in the turbine. The alternative solution considered for the wet steam expansion case study is a positive displacement screw expander. The screw expanders are designed for wet expansion. The expander consists of two matching rotors helical rotors, as presented in Figure 2.15. In medium-scale applications of up to about 1 MW, screw expanders are lower-cost and more compact alternatives to blade turbines. The heat-to-work adiabatic efficiency of the screw expanders is about 70 %, which is lower than that of the turbo expanders, 80 % to 90 % [65, 66].

Figure 2.15 The screw expander rotors for wet steam expansion [65]

2.5.5 Organic Rankine cycle

In the organic Rankine cycle (ORC), the working fluid is a fluid that contains carbon molecules, also called organic. The common organic fluids are hydrocarbons and refrigerants. The distinctive property of organic fluids from water is lower vaporization temperatures, which allow heat recovery from low-temperature sources such as the engine cooling water [36]. The ORC operating principles and components are the same as the steam Rankine cycle demonstrated in Figure 2.13. Based on the heat source properties, the most appropriate fluid is selected. In addition to wet fluids, dry and isentropic fluids are available, which properties help to prevent wet expansion in the turbine [36]. The high availability of low-temperature heat on board ships and potential business cases have promoted the ship-oriented ORC solutions, which are now commercially available [58, 59].

The ORC working fluid selection is an important step in designing efficient, regulation-compliant, and safe waste heat recovery system. The working fluids providing the highest cycle thermal efficiencies, such as the dry hydrocarbons, could be highly toxic and flammable. The ORC installation in the ship engine room should use non-toxic and non-flammable working fluid to minimize the risks and safety system design-related costs. The global warming potential (GWP) and ozone depletion potential (ODP) of the working fluids used on board have to be compliant with the MARPOL convention [9] and local regulations such as the EU F-gas Regulation, which prohibits use of refrigerants with GWP of 2500 or more [67].

Based on the existing ORC ship installation [57] and the review of ORC for maritime application by Mondejar et al. [68], the ORC working fluid selected for the recovery of the case study ship engine high-temperature cooling water is Genetron® 245fa (R-245fa). R-245fa is a non-flammable working fluid with low toxicity, noted as the ORC heat transfer fluid [69]. The fluid properties are presented in Table 2.3. The low boiling point and sufficient critical temperature suit the 90 °C engine HT cooling water heat recovery.

Table 2.3 R-245fa properties [69]

Property	**Value**
Boiling point at 1 atm	15,3 °C
Critical temperature	154,01 °C
GWP	1030
ODP	0

The initial assessment of the waste heat recovery potential of the ORC on board the case study ship is done using the concept of the ideal Rankine cycle. The graphical representation of the ideal R-245fa ORC is presented in Figure 2.16. The evaporator (boiler) and condenser generally used pinch point is 5 °C [68]. The R-245fa heat transfer processes 2-3 (25 °C solid line and 10 °C dash line) and 4-1 (85 °C) are isobaric. The expansion and compression processes are isentropic and adiabatic. All processes are internally reversible. As it can be seen, the turbine expansion process 1-2 is outside the saturation curve, eliminating the wet expansion.

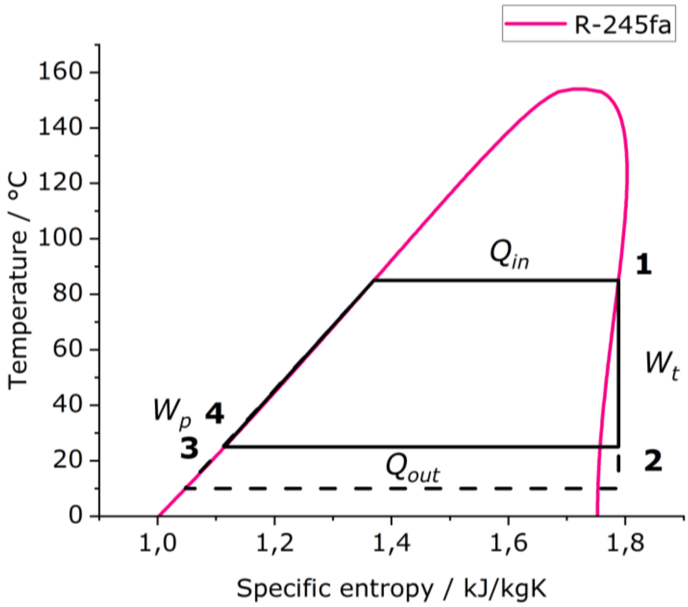

Figure 2.16 The case study ship’s ideal R-245fa organic Rankine cycle T-s diagram

The ideal ORC cycle thermal efficiency is calculated using formula (2.8). The calculated heat-to-work conversion efficiency of the case study ship’s ideal ORC is presented in Table 2.4. The ORC efficiency can be improved by applying regenerators or a double-pressure system. To achieve short payback periods simple arrangement systems are recommended for the engine cooling water heat recovery [68].

Table 2.4 Ideal R-245fa ORC thermal efficiency with different condenser seawater temperatures

Seawater temperature / °C	**Thermal efficiency / %**
5	17,1
20	14,0

The ORC thermal efficiency is comparable to the SRC. With a lower seawater temperature efficiency is higher. Based on the heat cascade principle, the steam Rankine cycle condenser heat with temperatures 90 °C to 100 °C should be combined with the ORC to increase its output and total ship energy efficiency.

3 EVALUATION METHODOLOGY

3.1 Data collection

3.1.1 Case study ship

The waste heat recovery energy efficiency and emission reduction potential evaluation is applied to the fictional case study ro-ro passenger ship operating in the Baltic Sea. The case study reference ship side view is presented in Figure 3.1. The ro-ro decks for trucks, trailers, and reefers are located inside the hull and on the weather deck. The superstructure with passenger cabins, public spaces, and a garage for passenger cars is located forward. The engine room is located aft. The reference ship's main characteristics are presented in Table 3.1.

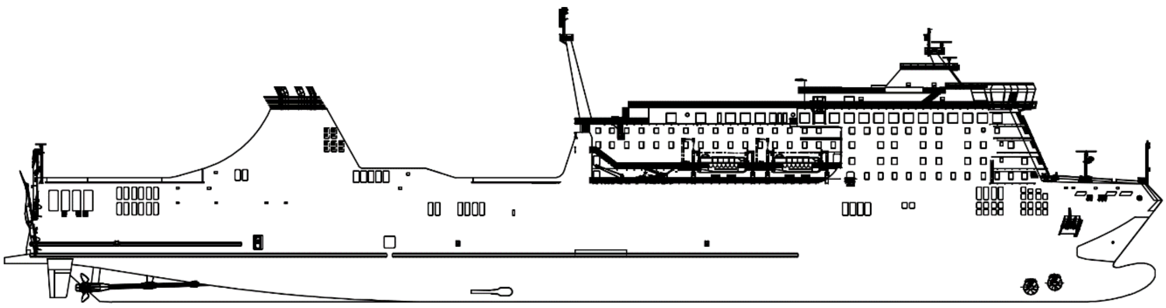

Figure 3.1 The case study reference ship side view

Table 3.1 The case study reference ship main characteristics

DIMENSIONS	
Length overall	218,8 m
Beam	30,5 m
Draught summer	7,1 m
Ice class	IA Super
PAYLOAD	
Total lane length	4215 m
Reefer unit capacity	192
Passengers	554 persons
TONNAGE	
Deadweight	9653 t
Gross tonnage	45923 t
SPEED	
Design speed	25 kn
Service speed	22 kn
MACHINERY	
Main engines	4 x 10395 kW, total 41580 kW
Auxiliary engines	3 x 1200 kW, total 3600 kW
Propulsion	2 x shaft with controllable pitch propeller
Shaft generators	2 x 2000 kW
Thrusters	2 x 2000 kW
BUILT	
Year	2007

The reference ship has such structure and engine output that she is assigned to ice class IA Super. A ship assigned ice class IA Super is capable of navigating in difficult ice conditions without the assistance of icebreakers [70]. The ship hotel services are designed for 554 passengers, including cabins, restaurant, café, sauna, and public spaces that require heating, cooling, and electrical energy. The ro-ro decks have to be illuminated and ventilated. The reefer cargo units are refrigerated trailers that are connected to the ship’s electrical network. The electrical energy can be produced by auxiliary engines and shaft generators at sea. The ship’s installed engine power allows her to reach a design speed of 25 knots. The service speed is reduced to 22 knots. The ship was built in 2007 before the EEDI regulation adoption. The case study ship operates on the intra-EU route between Helsinki (Finland) and Travemünde (Germany). The route is presented in Figure 3.2, which is about 611 nautical miles long.

Figure 3.2 The case study ship’s route [71]

Based on the timetable of the reference ship, travel to destination and return takes 72 hours. The time at sea is more than 85 % of the operation. Figure 3.3 shows the operation time share between port and sea, excluding maneuvering. When at sea, the main engines are running, rotating the propellers to propel the ship. Considering the main engine energy conversion efficiency discussed in 2.5.2 Main engine item, more than half of fuel energy is lost. The main engine’s continuous and steady operation should allow to recover heat from exhaust gas and cooling water. To have an in-depth understanding of the ship systems, machinery, and operation, the ship drawings package, and an interview were requested from the reference ship owner.

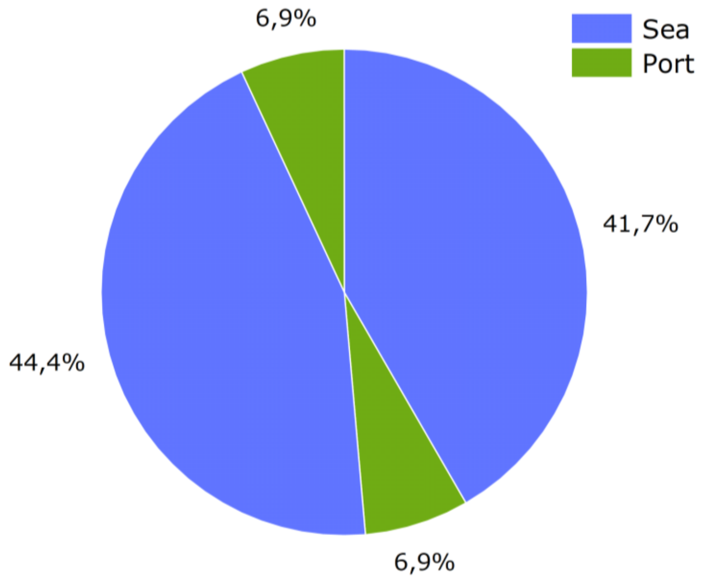

Figure 3.3 The operation time share based on the reference ship's timetable

3.1.2 Ship drawings

The reference ship drawings requested from the ship owner are the ship system diagrams, design load calculations, and equipment specifications. The system diagram represents the system's equipment, pipe connections, control and monitoring instruments, and interfaces with other ship systems. It describes how the system is built, the system process general flow, and which operating parameters can be monitored with the existing instruments. The system design load calculation shows the design parameters of the system's consumers in different design conditions. The equipment specification describes the equipment's design parameters and off-design behavior.

The ship drawings used in the study include:

- **Machinery Arrangement** maps machinery equipment location on board;
- **Steam and condensate diagram** represents steam flow between boilers and consumers, and condensate return;
- **Boiler feed water diagram** represents water supply to boilers and circulation in exhaust gas economizers;
- **Fresh and seawater cooling diagram** represents cooling water flow between heat exchangers and equipment;
- **Fuel oil diagram** represents fuel storage, preparation and supply to consumers;
- **Electric load analysis** shows the electrical equipment design load on ship electrical network;
- **General Arrangement** is ship's deck plan.

Based on the reference ship drawings, the case study ship's energy system layout is made, presented in Figure 3.4. The system's key components are four main engines. They convert the energy of fuel into mechanical energy. The engines are connected to two ship propellers via a gearbox on each shaft. The gearbox has a shaft generator connected to produce electrical energy. The electrical energy for the onboard consumers is also produced by three auxiliary engines coupled with electric generators. The exhaust gas heat is recovered in economizers installed on each engine to produce steam. Steam can also be produced in two oil-fired boilers to meet the steam consumers' demand. The excess steam is dumped in the dumping condensers. The main engine's high temperature cooling water heat is released to seawater in the cooling systems.

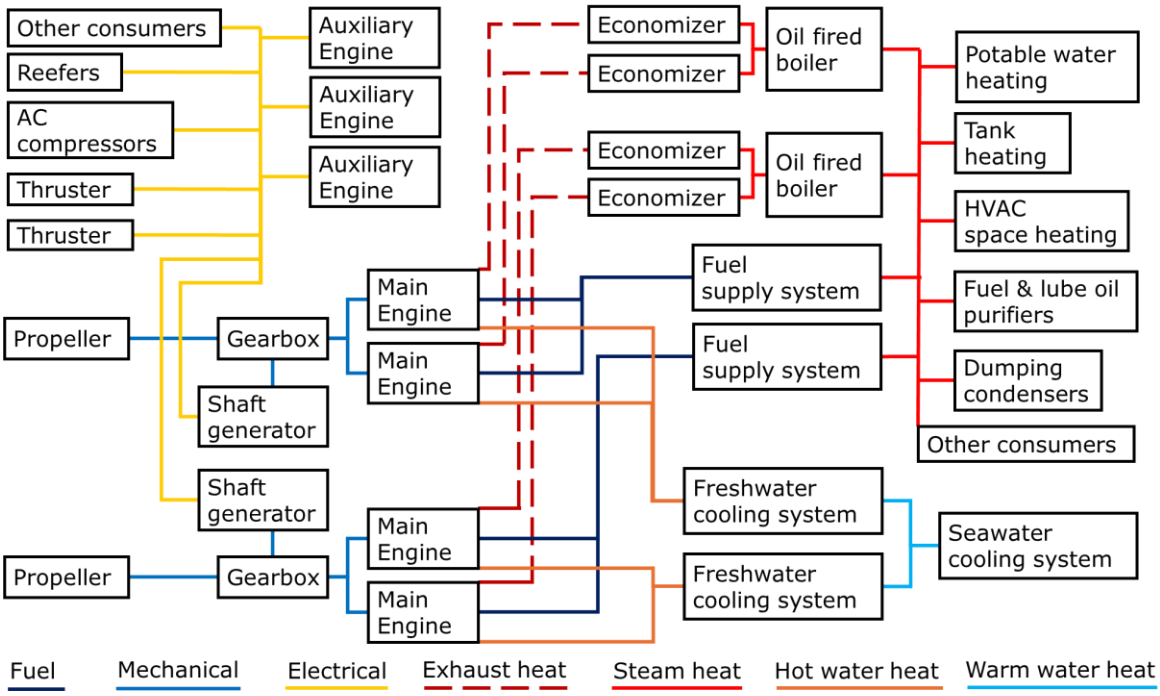

Figure 3.4 The case study ship's energy system layout

3.1.3 Shipowner communication

The drawings can provide details on the ship's design conditions but do not include information on the ship's real operating conditions. The ship's systems operating conditions can be provided by the engine department on board or the technical management in the office. As a first step, the case study ship's potential operating conditions were discussed with the reference ship's technical manager. He provided principal information on the ship's systems operating conditions and energy saving insights. The discussion memo is presented in Table 3.2.

Table 3.2 The case study ship’s operating conditions discussion memo

Topic	Description
Ship speed	20-23 kn
Fuel consumption	Main engines: 94 % HFO, 3 % MGO Auxiliary engines: 2 % MGO Oil fired boiler: 1 % MGO of the total fuel consumption per month
Heat demand at sea	Summer: 1 economizer Winter: 2 economizers
Heat demand in port	Summer: 1 oil fired boiler Winter: 2 oil fired boilers
Engine HT water heat recovery	Technical freshwater evaporator
Engine LT water heat recovery	None
Potable water source	Bunkering in port

The case study ship’s main engines are running on the high sulphur heavy fuel oil, and the sulphur is removed in the exhaust gas cleaning system, aka wet scrubber, to comply with the Baltic Sea Emission Control Area requirements, a part of the MARPOL convention [9]. The scrubbers are open-loop type, and their water discharge can be restricted in ports by authorities, therefore, for manoeuvring marine gas oil is used for manoeuvring.

The case study ship’s operating heating demands at sea could be met by one exhaust gas economizer during summer, and two economizers during winter. The heating demands in ports could be met by one oil-fired boiler during summer, and two-oil fired boilers during winter. The ship does not have a potable water production plant, and good-quality water is bunkered in the port. The main engine’s HT water heat is rarely used for technical freshwater production to maintain the system’s water level.

3.1.4 First ship visit

To have the latest updates on the reference ship operations, collect data logged on board, and access more ship technical documentation, the ship visit was agreed with the shipowner. The ship visit was carried out in March 2023, where during a three-day roundtrip, the ship energy system data available was collected. The ship’s Chief Engineer and Second Engineer fully supported the activities, confirmed the data previously received, and provided more information on the ship’s operations.

In the first place, the engine room safety briefing was carried out. Accompanied by Engineer, the ship engine room was inspected to understand the machinery arrangement, condition, and retrofit potential. More drawings and specifications of the ship systems and machinery were found on paper. Photocopies of the technical documents were made. The list of logged ship and machinery signals was obtained from the logging system. The logging system saves signals from the Integrated Alarm Monitoring and Control System (IAMCS). It allows continuous monitoring and reporting of ship machinery operating parameters. The list of logged IAMCS signals is defined by Engineer. The logging can be done to the main database with a frequency of 60 seconds for one year, or to the fast database with a frequency of 0,5 seconds for one week.

Based on the logging list, the signals related to the ship energy system measurements were selected and exported in text file format. Since the ship energy system operating parameters vary during summer and winter, the periods exported were July-August 2022 and January-February 2023 with 10-minute intervals from the main database. The signal history and technical documents collected during the ship visit included:

- **Ship speed** signal;
- **Main Engine**
    - fuel oil volumetric flow rate signal;
    - charge air pressure signal;
    - turbocharger rotating speed signal;
    - exhaust gas temperature after turbocharger signal;
    - HT cooling water temperatures on the inlet and outlet signals;
    - LT cooling water temperature on the inlet signal;
    - Factory test report documents;
- **Propeller shaft** mechanical power signal;
- **Shaft generator**
    - electric power signal;
    - instruction document;
- **Auxiliary engine** generator electric power signal;
- **Steam system**
    - exhaust gas economizer drawing;
    - steam balance document;
    - boiler feed water pump instruction document;
- **HVAC system** chilled and hot water flow and pumps documents;
- **Freshwater cooling system** balance, pumps and heat exchangers documents;
- **Seawater cooling system** pumps and heat exchangers documents.

It is important to note that the measurement instrument signal includes errorr and does not provide a true value. The instrument’s accuracy depends on its class, calibration, condition, and application.

3.2 Data processing

3.2.1 Data cleaning

The data from text files was imported to the spreadsheet software Microsoft Excel for cleaning [72]. The summer and winter data sets were processed separately. The work on 10-minute interval two-month data constituting around 9000 rows in Microsoft Excel did not cause any technical difficulties. The signals were grouped as a global ship signal (e.g., ship speed, propeller shaft mechanical power, shaft generator electric power, fuel volumetric flow rate) and local machinery signal (e.g., main engine systems pressure and temperature).

When the ship is at sea, and the machinery is running, the signals' values were assessed as realistic based on the information from communication with the ship engineers. When the ship is in port, and the machinery is not running, some of the signals, such as ship speed, main engine fuel flow rate, propeller shaft mechanical power) were not zero and showed single values. For the analysis, the signals of the switched-off machinery in ports were set to zero. The nonzero signals from the switched-off machinery are related to the signals' analogue nature. All the values were checked for the measurements' consistency.

3.2.2 Preliminary analysis

The collected data preliminary analysis was aimed at estimating heat generated by the main engines. The main engine cooling water temperature signal values were analysed. It was noticed that the engine HT cooling water temperature inlet value was higher than the set-point in the ship control system and the engine project guide [62]. According to the heat transfer formula (3.1), the heat power $\dot{Q}$, J/s, decreases as the fluid's temperature at the inlet increases. To estimate the heat amount, the fluid mass has to be known.

$$\dot{Q} = \dot{m} \cdot c_p \cdot (T_{out} - T_{in}) \,, \tag{3.1}$$

where $\dot{m}$ – is the fluid mass flow rate, kg/s,

c_p – is the fluid's specific heat capacity, J/kg,

T_{out} – is the fluid's temperature at the outlet, K,

T_{in} – is the fluid's temperature at the inlet, K.

The engine cooling water flow is defined by the engine manufacturer. The cooling water system design information is available in the engine project guide. The HT cooling water pump's nominal capacity at ISO standard conditions and 100 % engine load is 200 m^3/h. However, the real operating conditions and engine load differ from the conditions noted in the project guide. Cooling water flow can be estimated using pump curves, but the parameters in the project guide pump curves and on the ship are different, and the curves are not valid. The experimental engine HT cooling water heat estimation required field measurements of the inlet temperature and flow.

The theoretical HT engine cooling water heat estimation can be made using the heat balance diagram from the engine project guide using the real operating engine load and ambient conditions factor. However, the engine load was not available in the logged signal list. A method using engine charge air pressure and turbocharger speed signals was developed and tested based on the engines' factory test reports to obtain the real operating engine load. The engine load calculation methods were validated against the shaft mechanical power and shaft generators' electrical power signal values. The charge air pressure method underestimated the load by 10 %, and the turbocharger speed method overestimated the load by 20 %. The source of error was found to be different conditions during the factory test and real operation. The methods were not used. To include an ambient conditions factor, the engine suction air temperature is required, which was not available in the logged signal list. The theoretical engine HT cooling water heat estimation required the history of engine load values and suction air temperatures.

The main engine's exhaust gas heat estimation can be made using formula (3.1), where exhaust gas mass, specific heat capacity, temperature at the turbocharger expander outlet, and the engine intake air manifold temperature are used. It will reveal the amount of heat contained in the exhaust gas. Since the study investigates the recoverable heat using the ship economizers, formula (3.1) should be applied to the exhaust gas economizer. The mass and specific heat values remain the same, whereas the temperatures are changed to the exhaust gas temperatures at the economizer inlet and outlet.

For the estimation of exhaust gas heat recovered in the economizer, none of the parameters required could be used directly from the collected data without assumptions. The heat losses in the exhaust gas pipes have to be neglected. To obtain the heat load of the operating economizer, a method using theoretical exhaust gas calculations based on the engine project guide diagrams and the economizer technical specifications was developed. The theoretical exhaust gas economizer's heat load estimation requires the history of engine load values and suction air temperatures.

Based on the preliminary analysis findings, a second ship visit was needed to collect the required data and measure flows using portable measuring instruments for heat calculation methods development and validation. The preliminary analysis was reported to the shipowner including ship visit and measurements request. The ship visit was agreed.

3.2.3 Second ship visit

The second ship visit was carried out in October 2023, where during a three-day roundtrip, the measurements were taken, and required data was collected. The ship's Chief Engineer helped to find and export required signals from the scrubber logging system. The signal history was received for the same periods as data collected during the first ship visit. The ship visit's roundtrip data set was exported from the ship's IAMCS logging system to complement the field measurements. The list of new signals from the scrubber logging system included:

- Main engine fuel rack position;
- Exhaust gas temperature before scrubber;
- Seawater temperature.

Always in the first place, the engine room safety briefing was carried out. The field measurements scope and locations were agreed. During the first day, the measuring instruments data log and export functions were tested. The instruments used were:

- Portable ultrasonic flow measuring instrument with temperature input, volumetric flow rate uncertainty ±1 % of reading ±0,005 m/s, temperature input accuracy ±0,01 % of reading ±0,03 K;
- Thermocouple with log function, accuracy ±0,5 °C;
- Ambient temperature and humidity data logger, accuracy ±0,4 °C;
- Infrared thermometer, accuracy ±1,0 °C.

The ambient temperature data logger was installed near the main engine turbocharger air intake. The air temperature in this location is higher than in the engine room due to heat from the turbocharger's hot exhaust gas side. Unfortunately, uncertain technical issues with the data logger did not allow to obtain the main engine suction air temperature. When the data logger was visually monitored the temperature was 40 °C.

The main engine HT cooling water volumetric flow rate, engine inlet and outlet temperatures were measured at the locations as shown in Figure 3.5. The flowmeter (FM) and temperature transmitter (TT) were clamped to the HT outlet line, and a thermocouple logger was placed on the HT inlet line. Figure 3.6 displays the instruments during the measurements. In total about 24 hours of data, with-10 minute intervals, was obtained. The measurements will be used to validate the theoretical main engine's HT cooling water heat calculation method.

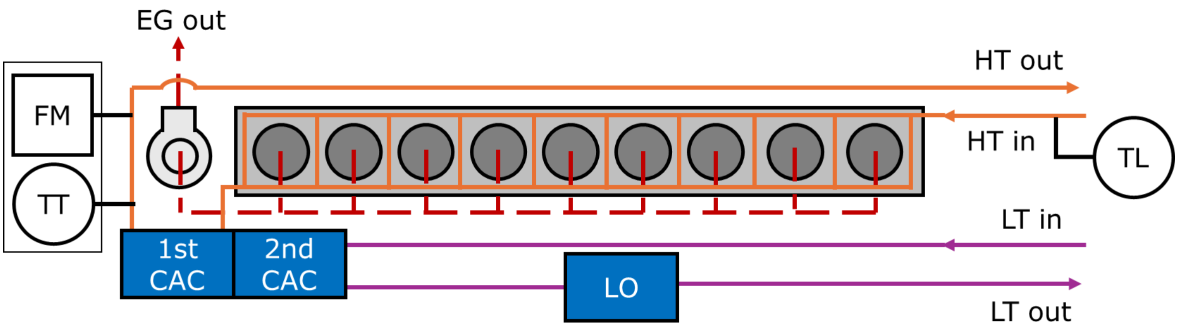

Figure 3.5 The main engine's HT cooling water measurement locations

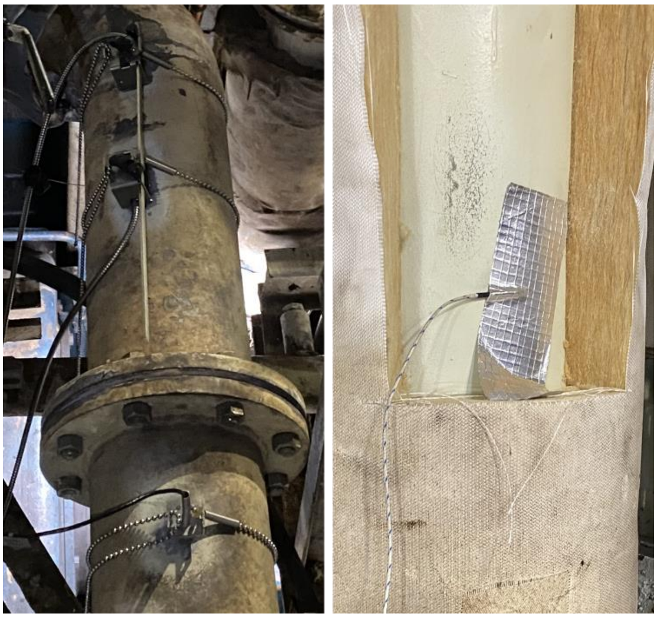

Figure 3.6 The main engine's HT cooling water measurement, FM and TT on the left, TL on the right

The volumetric flow rate and temperature of the boilers' feed water were measured to validate the developed method for the exhaust gas economizer's theoretical heat recovery calculation. Figure 3.7 demonstrates the location of the flowmeter and temperature transmitter measurement. It is assumed that at sea, steam is mostly produced in the economizers, and the oil-fired boilers' contribution to steam production is negligible. Therefore, the measured flow mass represents the economizer's steam mass flow. For about 15 hours, the flow was saved every 10 seconds. The logging frequency was increased to capture the feed pump's intermittent operation. Figure 3.8 displays the instruments during the measurements. After the measurements, the pipes were cleaned, and the insulation was restored.

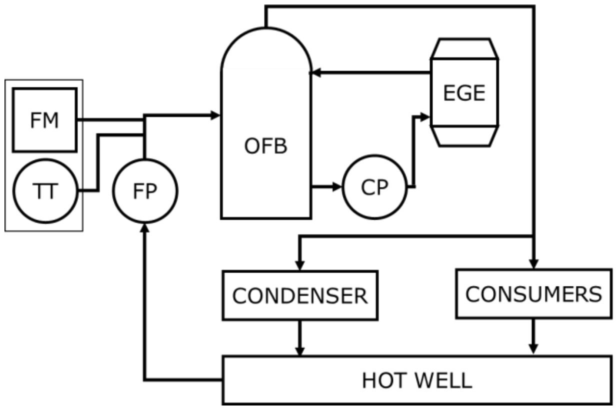

Figure 3.7 The boiler's feed water flow measurement location

Figure 3.8 The boiler's feed water flow measurement, FM and TT

3.2.4 Methods validation

During the second ship visit, it was confirmed that the suspiciously high engine HT cooling water inlet temperature previously obtained from the ship IAMCS logging system is not true. Since there were no other engine HT cooling water inlet temperature logged signals, the heat calculation based on the operating parameters was not possible. The main engine HT cooling water heat estimation had to be done using the previously mentioned theoretical heat estimation method and the engine load values collected during the second visit.

The engine HT cooling water heat calculation method is a black-box empirical model based on the engine project guide [62]. The advantage of the empirical model is its high accuracy compared to the physical model. On the other hand, the empirical model could have substantial errors when extrapolating [73]. The engine operating HT cooling water heat is determined based on the HT circuit heat dissipation diagram using the engine operating load. The heat dissipation diagram tolerance is ±10 % in ISO standard conditions. The influence of ambient temperature is included using the correction factor obtained from the respective diagram. The model's output is the engine HT cooling water heat dissipation measured in kW.

The engine HT cooling water heat calculation model is validated against the onboard measurements. The measured cooling water volumetric flow rate was converted to mass flow rate, and using formula (3.1), the heat dissipation was calculated. As the validation, the difference between calculated and measured values relative to measured was calculated. The difference is presented in Figure 3.9. The calculation model tends to underestimate the engine HT cooling water heat. Considering the calculation model source data tolerance, the model can be used in ship energy analysis.

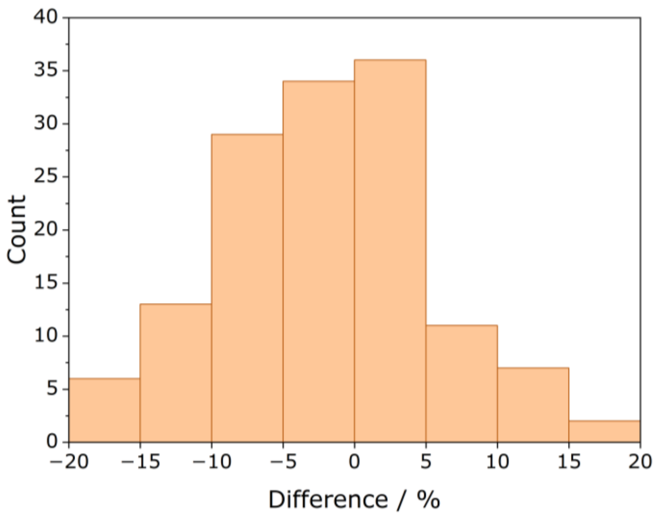

Figure 3.9 The difference between calculated and measured engine HT cooling water heat

The exhaust gas economizer heat recovery calculation method is a combination of empirical and physical models known as the gray-box model [73]. The economizer's heat recovery power $\dot{Q}$ is calculated using formula (3.1), where $\dot{m}$ is the engine exhaust gas mass flow rate, T_{in} is the exhaust gas temperature at the economizer inlet, T_{out} is the exhaust gas temperature at the economizer outlet. It is assumed that the exhaust gas temperature at the economizer inlet T_{in} is equal to the engine exhaust gas temperature after turbocharger. Heat losses to the environment are neglected. The exhaust gas temperature at the economizer outlet T_{out} is derived based on the exhaust gas temperature measurements from the scrubber system. The engine operating exhaust gas mass flow rate $\dot{m}$ and temperature after turbocharger are determined from the engine project guide respective diagrams using the engine operating load value.

The validation of the economizer's heat recovery power $\dot{Q}$ is two-step. First, the exhaust gas temperature at the economizer inlet T_{in}, determined from the guide diagram, is validated against the engine exhaust gas temperature measurements. The exhaust gas temperature after turbocharger diagram's tolerance is ±15 °C in ISO standard conditions. The engine suction air temperature correction has to be taken into account, since for a 10 °C rise in air temperature, the exhaust gas temperature increases by 15 °C. The difference between calculated and measured values relative to the measured was calculated. The difference is presented in Figure 3.10. The exhaust gas temperature calculation error is around 1 %, which is an acceptable result.

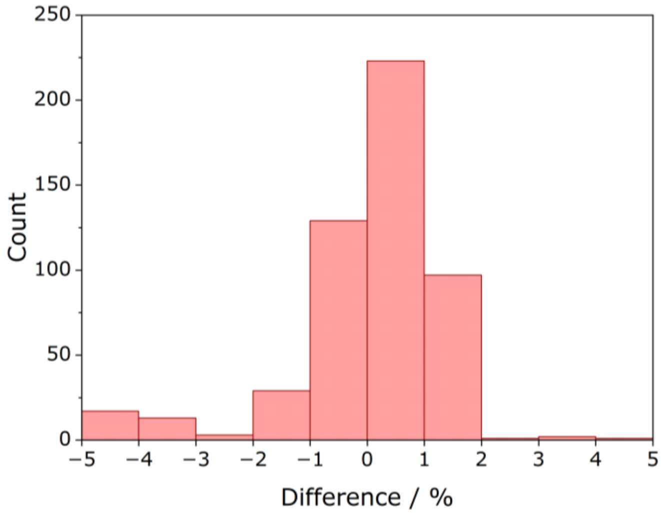

Figure 3.10 The difference between calculated and measured engine exhaust gas temperature after turbocharger

The validation second step is the diagram-based engine's exhaust gas mass flow rate $\dot{m}$ validation using the economizer's steam production $\dot{m}_{steam}$ formula (3.2) against the measured mass flow rate of water in the boiler feed system [74]. The exhaust gas mass flow rate diagram manufacturer's tolerance is ±5 % in ISO standard conditions. The exhaust gas temperature at the economizer outlet T_{out} is based on the measured operating exhaust gas temperature at the scrubber inlet. The exhaust gas temperature at the economizer inlet T_{in} is assumed to be the measured operating exhaust gas temperature after turbocharger.

$$\dot{m}_{steam} = \frac{\dot{m} \cdot c_p \cdot (T_{in} - T_{out})_{eg}}{(h_{steam} - h_{fw})}, \tag{3.2}$$

where h_{steam} – is the saturated steam specific enthalpy, J/kg,

h_{fw} – is the feed water specific enthalpy, J/kg.

The difference between calculated and measured fluid mass flow rate values relative to the measured was calculated. The difference is presented in Figure 3.11. The calculation model tends to significantly overestimate the exhaust gas mass flow during lower engine loads before the port. At sea, the model error is around 5 %. Considering the model inputs' possible errors, the economizer's heat recovery power calculation model could be used in ship energy analysis.

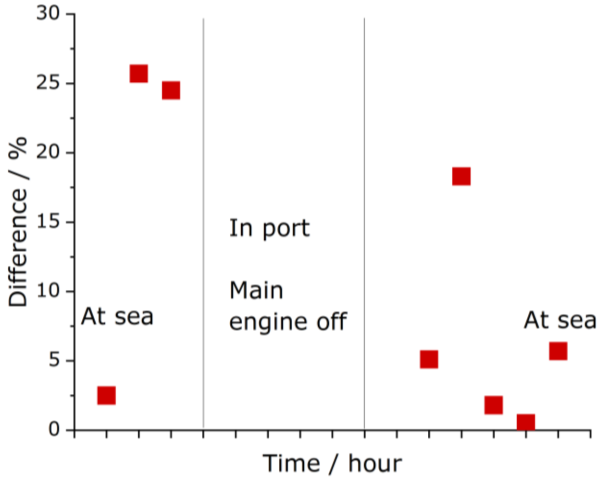

Figure 3.11 The difference between calculated and measured exhaust gas economizer steam production

3.3 Ship energy analysis

3.3.1 Systems analysis

The collected data was reviewed to determine which ship systems could be analysed using the operating parameters available in the data sets. It was found that most of the data signals are parameters of machinery operated at sea. It is not a problem for waste heat recovery analysis since most heat is produced during navigation. However, for a complete ship energy audit, data from all energy systems onboard will be required. Based on the data available, the case study ship's energy system layout was updated for the analysis. The layout is presented in Figure 3.12.

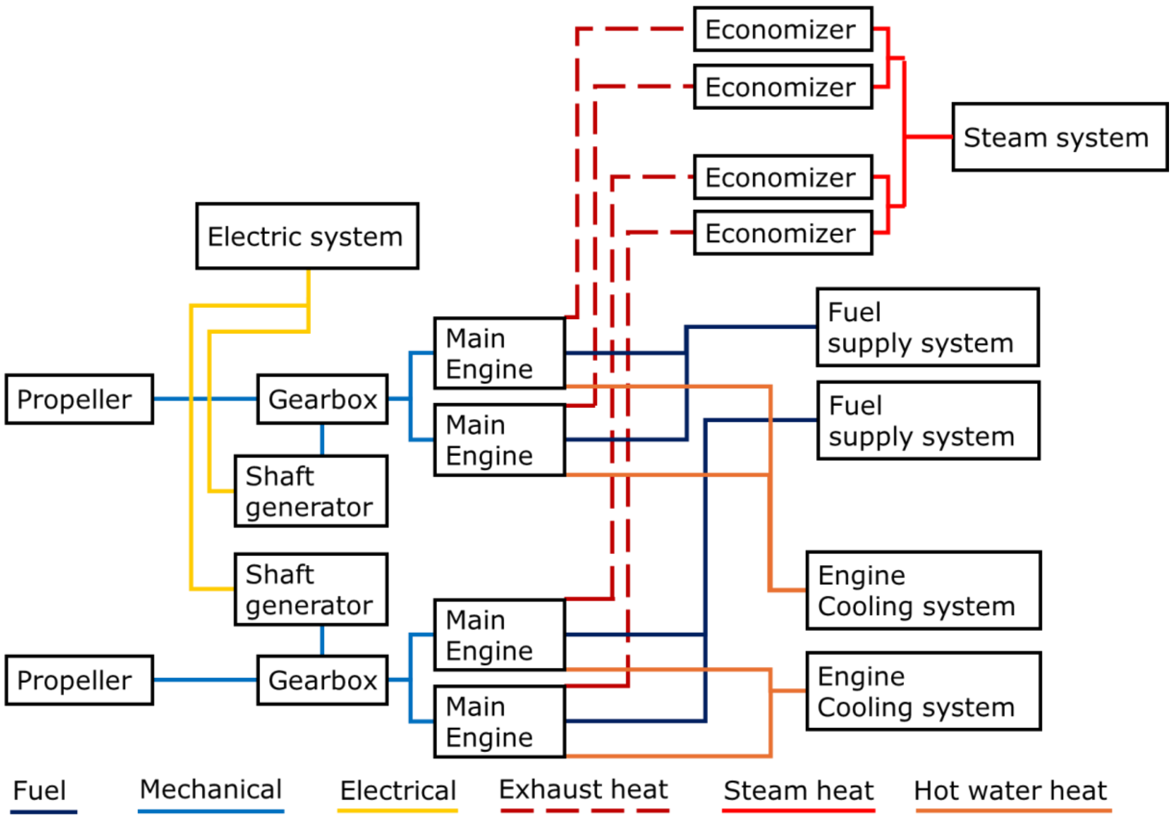

Figure 3.12 The ship's energy system layout used in the energy analysis

The systems governing the ship's energy consumption at sea are the propulsion and the electric systems. The thermal energy demand at sea is usually covered by waste heat. The propulsion system energy consumption increases with the ship speed as defined in formula (2.7). The electric system energy consumption depends on electrical consumers number and power demands. Both systems require mechanical energy to do work. The main engines convert fuel chemical energy to mechanical energy used in the propulsion and the electric systems. More than half of the fuel energy dissipates through the exhaust gas and the cooling water systems. As previously mentioned, part of the fuel energy in the form of exhaust gas heat is recovered to produce steam to cover the ship's demands. The fuel energy in the form of cooling water heat is usually dissipated into the sea, except for rare heat recovery with the technical freshwater evaporator to maintain the water level.

3.3.2 Operational profile

The ship's operational profile describes how the ship is operated over time. It is common to use a ship speed profile to describe it. The operational profiles differ depending on ship route, loading, weather, and sea conditions. In the studied ro-ro passenger ship case, the route is fixed. The loading and sea conditions are considered included in the speed profile. The weather effect is studied using different speed profiles for the summer and the winter operations. The case study ship operational profiles were derived and modified based on the reference ship's speed signal history from the summer and winter periods. The derived operational profiles should represent the typical operational profile of the fictional case study ship during summer and winter and could not be related to any real ship. The case study ship operational profiles are presented in Figure 3.13.

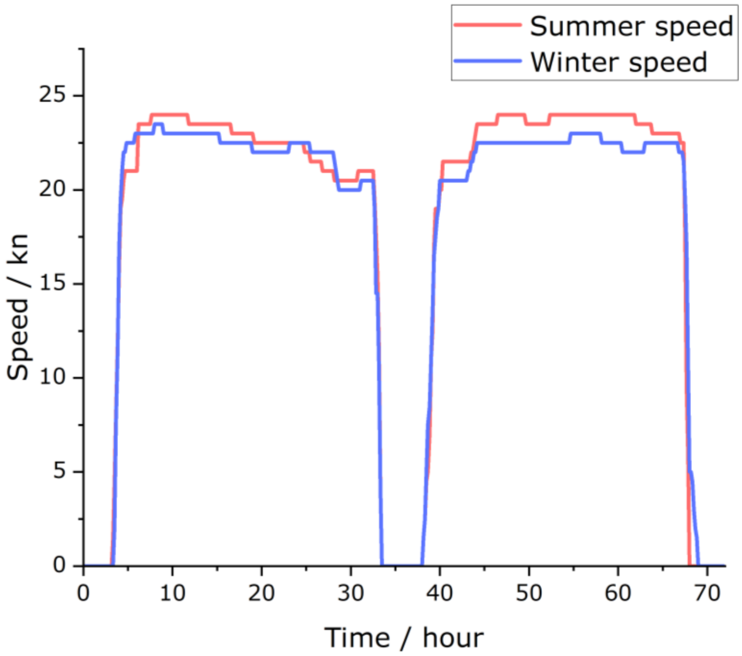

Figure 3.13 The case study ship's summer and winter speed profiles

The three-day operating profile represents one roundtrip between Helsinki and Travemünde. The first part of the roundtrip is almost identical between summer and winter profiles. In the return part during summer, the ship navigates at a higher speed and arrives earlier. The ship's average summer speed is 22,1 knots, whereas the average winter speed is 21,2 knots. The lower winter speed may be caused by the Baltic Sea storms, which are more frequent and stronger during winter [26]. Based on the speed profile, the operational modes were defined, which are presented in Table 3.3. The operational time distribution by modes presented in Figure 3.14 is similar for summer and winter.

Table 3.3 The case study ship's operational modes by speed

Operational mode	**Speed**
Port	0
Manoeuvring	below 5 knots
Acceleration and deceleration	between 5 and 20 knots
Service speed	above 20 knots

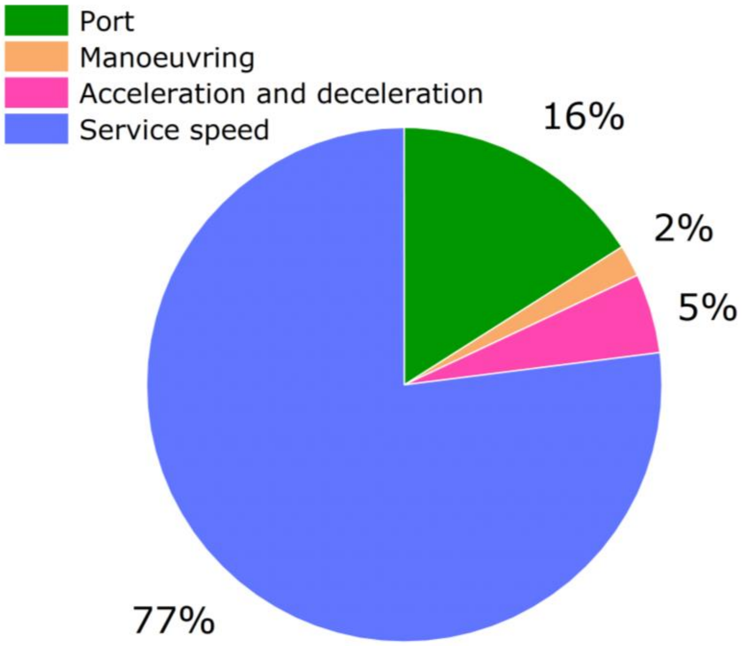

Figure 3.14 The case study ship's operational time by mode

3.3.3 Energy model

The ship energy system calculation model was developed to evaluate the energy efficiency of the case study. The steady-state gray-box model calculates ship energy flows at sea in a fixed number of states using physical and empirical equations. It is based on the First Law of Thermodynamics, which states that the energy can neither be created nor destroyed, the energy is converted from one form to another, which in the ship energy analysis is represented in formula (3.3) [73].

$$\dot{H}_{fuel} + \dot{H}_{air} = \sum_{waste} \dot{H}_{waste} + \dot{W}_{el} + \dot{W}_{mech} + \dot{Q}_{heating}\,, \tag{3.3}$$

where $\dot{H}_{fuel}$ – is the fuel energy consumed on a time rate basis, J/s,

$\dot{H}_{air}$ – is the air used in combustion process on a time rate basis, J/s,

$\dot{H}_{waste}$ – is the heat discharged into the environment on a time rate basis, J/s,

$\dot{W}_{el}$ – is the electrical power, J/s,

$\dot{W}_{mech}$ – is the mechanical power, J/s,

$\dot{Q}_{heating}$ – is the heat consumed on a time rate basis, J/s.

The calculation model was developed in MATLAB Simulink software [75]. Simulink is a well-known modelling environment with functionalities required for the model development.

The ship energy system model layout in the program is the same as in Figure 3.12. The energy flow is calculated every minute for the ship’s three-day roundtrip analysis based on the specified input variables. The model input signals are ship speed, electric and thermal power needs, and air temperature. The signals are connected to different function blocks and components representing ship machinery.

To model the ship propulsion system, the relationship between ship speed and required ship propulsion power was derived based on the reference ship’s modified operational data. A third-order polynomial was fitted to summer and winter data sets to obtain the cubic ship speed-power curve. The derived relationship also allows the inclusion of the effect of the sea state on the ship's energy system during summer or winter.

Power to the propeller is transferred via a shaft and the gearbox. The gearbox is used to transfer engine mechanical power to the propeller shaft and the shaft generator. The gearbox model transfers energy transmission losses. The shaft generator uses electric power needs model input signal to take off the required engine’s mechanical power to provide electricity on board. The shaft generator model considers electricity generation efficiency.

The ship engine operational logic is modelled using the Stateflow chart. Based on the ship speed, mechanical power required by the propellers, and the shaft generators, the engine loading is determined. At the service speed, the ship’s propulsion and electric power demand could be met with three engines. To explore the possibility of three-engine operation for improving ship energy efficiency, the three-engine operational logic was developed in parallel with the common four-engine operation.

The ship’s main engine operating parameters are calculated based on the load input from the engine’s control logic state chart component and the engine manufacturer’s performance maps. The corrected engine exhaust gas temperature and mass flow rate were validated as described in item 3.2.4 and considered to have enough accuracy for the analysis. The exhaust gas economizer’s heat recovery calculation method was also previously validated and proved suitable for sea conditions. The number of operating economizers required to meet the ship’s thermal power needs is provided as model input. The validated high-temperature cooling water heat dissipation calculation method was used in the analysis. The low-temperature cooling water heat dissipation is calculated using the same principles as high-temperature water. The remaining energy in the energy balance calculation is assigned to the engine heat radiation.

The engine fuel consumption is calculated using the specific fuel consumption curve from the engine project guide. The fuel consumption calculation include correction on the suction air temperature and fuel type as per ISO engine standard [76]. The ship fuel consumption calculation was used to validate the developed energy model.

The ship energy model was validated against the reference ship average fuel consumption at sea data, which was not used in the model training. The calculated ship fuel consumption at sea per roundtrip error is ±2 % compared to the reference ship. The proposed three-engine operation could achieve 2,2 % fuel savings per roundtrip through lower specific fuel consumption of the engines at higher loads. During winter navigation, the ship's roundtrip fuel consumption could increase by up to 3 %.

3.3.4 Summer analysis

The case study ship's summer navigation energy analysis is presented in Figure 3.15. The mechanical energy produced by the main engines is 44,5 % of the fuel energy consumed. The energy used for propulsion is 39,1 %, and 3,5 % is converted to electrical energy. The mechanical energy losses are, in total, 1,9 %. 55,5 % of the fuel energy is transformed into heat. Most of the heat is in the exhaust gas form, containing 32,3 % of the fuel energy. 2,4 % of energy is recovered in one exhaust gas economizer to produce steam to meet ship heat demands. An additional 7,3 % of energy could be potentially recovered in three other economizers to produce steam. 22,6 % of energy in the form of exhaust gas is lost into the environment. The HT cooling water heat is 10,4 of the fuel energy. The HT cooling water temperature is 90 °C. The LT cooling water heat is 10,5 % of the fuel energy. However, the LT cooling water temperature is only 50 °C. The leftover 2,3 % of fuel energy in heat is added to the heat Radiation category. The category includes not only the engine thermal radiation but also energy consumed by the engine-driven cooling water and lubrication pumps.

The case study ship's energy efficiency at sea η_{ship} is calculated using formula (3.4). The ship's summer navigation energy efficiency is 45 %.

$$\eta_{ship} = \eta_{propulsion} + \eta_{electrical} + \eta_{steam}\,, \tag{3.4}$$

where $\eta_{propulsion}$ – is the propulsion energy share of fuel energy, %,

$\eta_{electrical}$ – is the electrical energy share of fuel energy, %,

η_{steam} – is the steam energy share of the fuel energy, %.

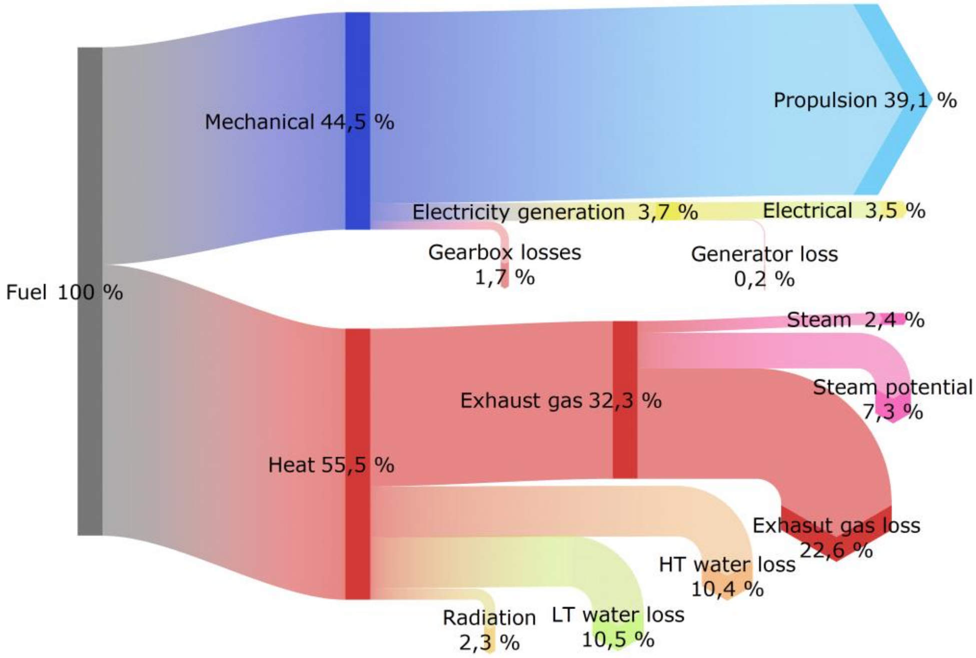

Figure 3.15 Sankey diagram of the case study ship's summer navigation energy analysis

3.3.5 Winter analysis

The case study ship's winter navigation energy analysis is presented in Figure 3.16. The ship's winter navigation energy efficiency calculated using formula (3.4) is 46,6 %. The winter navigation energy distribution is almost identical to the summer. The winter navigation improved efficiency is achieved through higher utilization of the exhaust gas economizers to produce steam to meet the ship's heat demand.

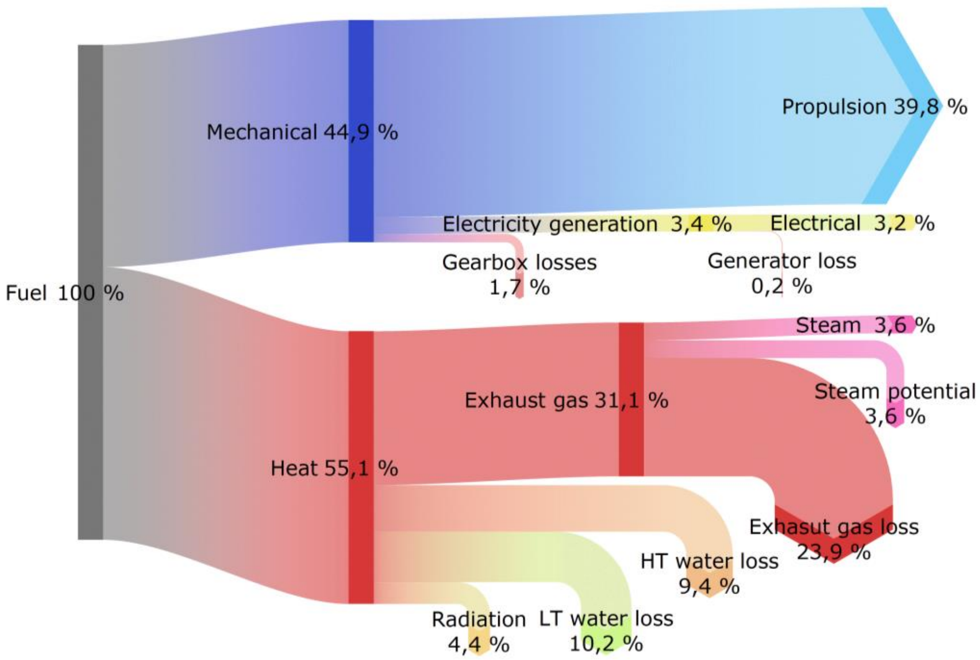

Figure 3.16 Sankey diagram of the case study ship's winter navigation energy analysis

3.4 Waste heat recovery retrofit analysis

3.4.1 Steam Rankine cycle

The steam turbine generator could use the exhaust gas economizers' heat recovery potential to produce steam to convert otherwise lost heat into electrical energy. The technical solution under consideration is a screw expander designed for steam wet expansion, as mentioned under item 2.5.4. The screw expander is connected to the ship's main steam line. The steam flow to the expander is regulated by a control valve. Steam expands, driving the rotor coupled with an electric generator. After the expander, steam is directed into the condenser. The condenser cooling media is seawater.

The Rankine cycle energy rate balance described in formula (3.5) is the fundamental equation used in the calculation model developed for the waste heat recovery retrofit analysis. The right side of formula (3.5) represents the cycle's net power. The calculation model was developed in MATLAB using the CoolProp library to obtain working fluid properties [77, 78].

$$\dot{Q}_{in} - \dot{Q}_{out} = \dot{W}_t - \dot{W}_p - \dot{W}_{cp}, \tag{3.5}$$

where $\dot{Q}_{in}$ – is the heat rate into the cycle, J/s,

$\dot{Q}_{out}$ – is the heat rate from the cycle, J/s,

$\dot{W}_t$ – is the turbine power, J/s,

$\dot{W}_p$ – is the pump power, J/s,

$\dot{W}_{cp}$ – is the condenser pump power, J/s.

In the real world, according to the Second Law of Thermodynamics, processes are irreversible. In the real Rankine cycle, principal irreversibility is associated with turbine and pump isentropic efficiencies [64]. Formula (3.5) terms accounting for the irreversibility of the process are turbine power $\dot{W}_t$ (3.6), feed pump power $\dot{W}_p$ (3.7), and condenser pump power $\dot{W}_{cp}$ (3.8). The formulas include the isentropic efficiency of the respective components. Table 3.4 presents isentropic efficiency values of the components used in the steam Rankine cycle calculation. The electric generator efficiency used in the calculation is 90 %.

$$\dot{W}_t = \dot{m}\,(h_{in} - h_{out})\eta_t\,, \tag{3.6}$$

where $\dot{m}$ – is the working fluid mass flow rate, kg/s,

h_{in} – is the working fluid specific enthalpy at turbine inlet, J/kg,

h_{out} – is the working fluid specific enthalpy at turbine outlet, J/kg,

η_t – is the turbine isentropic efficiency.

$$\dot{W}_p = \frac{\dot{m}\,(h_{out} - h_{in})}{\eta_p}, \tag{3.7}$$

where $\dot{m}$ – is the working fluid mass flow rate, kg/s,

h_{in} – is the working fluid specific enthalpy at pump inlet, J/kg,

h_{out} – is the working fluid specific enthalpy at pump outlet, J/kg,

η_t – is the pump isentropic efficiency.

$$\dot{W}_{cp} = \frac{\dot{m}\,(h_{out} - h_{in})}{\eta_{cp}}, \tag{3.8}$$

where $\dot{m}$ – is the condenser cooling fluid mass flow rate, kg/s,

h_{in} – is the condenser cooling fluid specific enthalpy at pump inlet, J/kg,

h_{out} – is the condenser cooling fluid specific enthalpy at pump outlet, J/kg,

η_t – is the condenser cooling pump isentropic efficiency.

Table 3.4 The steam Rankine cycle components isentropic efficiency

Component	**Isentropic efficiency / %**	**Source**
Turbine (screw expander)	60 %	[65]
Pump	70 %	[79]
Condenser pump	70 %	[79]

The calculated steam turbine electric power generation during the case study ship roundtrip is presented in Figure 3.17.

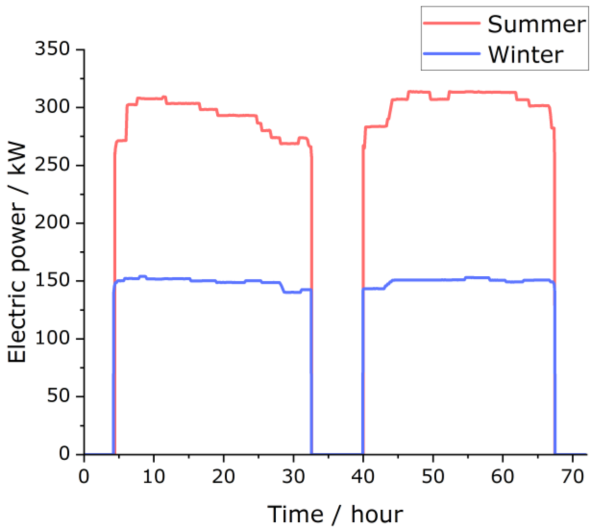

Figure 3.17 The steam turbine's electric power production during summer and winter roundtrips

In the summer, higher steam availability allows the steam turbine to achieve electricity generation power of 300 kW. In the winter, the steam turbine's electric power generation is at the level of 150 kW because of the ship's higher steam demands. The steam turbine effect on the ship energy system is demonstrated in Figure 3.18. During summer, the steam turbine waste heat recovery can achieve 0,6 % energy saving. During winter, the energy saving is 0,3 %. Still, 6,7 % of energy in steam (temperature over 100 °C) after the steam turbine will be lost in the seawater condenser.

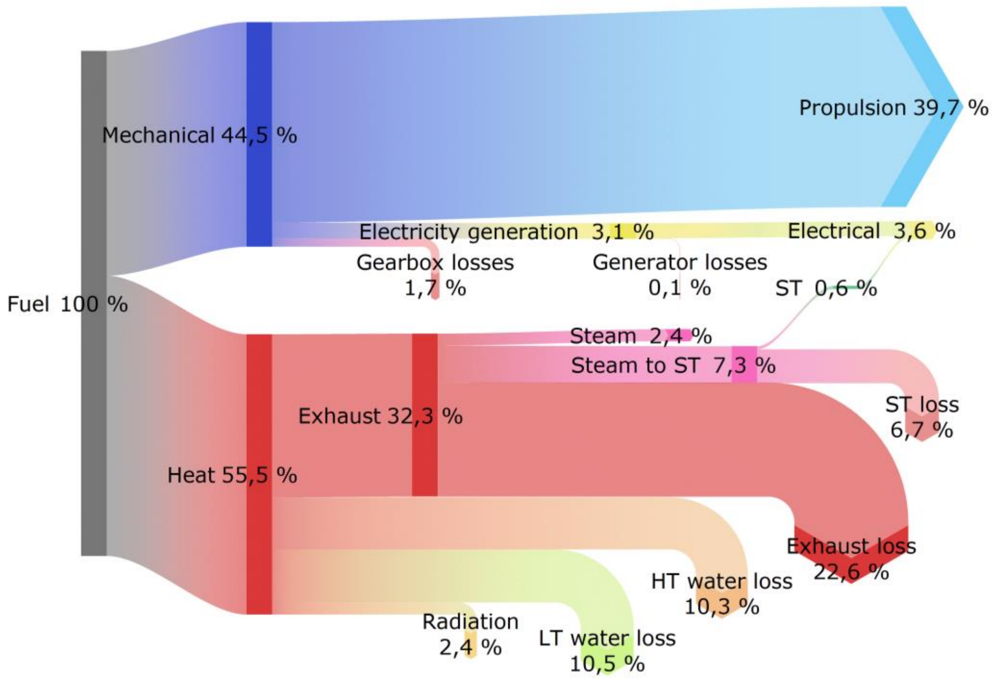

Figure 3.18 Sankey diagram of the ship energy system with the steam turbine during summer

3.4.2 Organic Rankine cycle

The main engine's high-temperature (HT) cooling water heat could be used in the organic Rankine cycle power generator to convert otherwise lost heat into electrical energy. The technical solution under consideration is an ORC system designed for ship engine HT cooling water heat recovery mentioned under item 2.5.5 Organic Rankine cycle. The ORC evaporator is connected to the engine cooling water system. In the evaporator pressurized working fluid is heated to change its phase to vapor. The working fluid vapor expands in the turbine, driving its shaft. The turbine shaft is coupled with the electric generator. The expanded vapor is cooled in the seawater condenser to change the phase to liquid and repeat the cycle.

Reducing the working fluid condensing temperature could improve the ORC efficiency. The seawater temperature variations from the refence ship measurements during summer and winter roundtrips between the northeast and southwest part of the Baltic Sea are presented in Figure 3.19. The seawater temperature during summer is close to 20 °C, and during winter, it is assumed to be 5 °C.

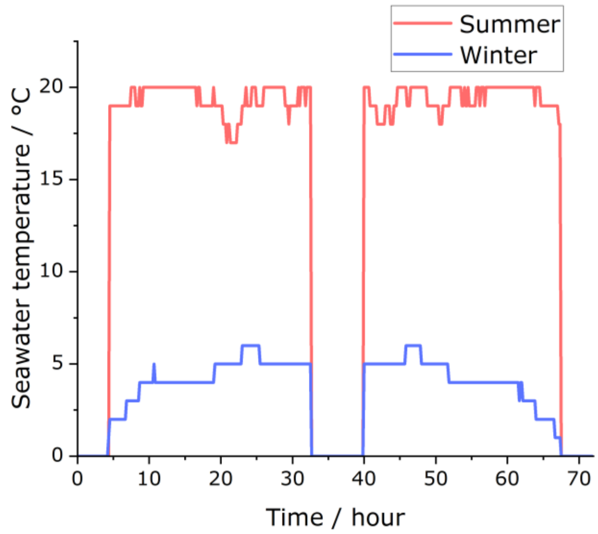

Figure 3.19 The seawater temperature summer and winter variations during the ship roundtrip

The organic Rankine cycle calculation model was developed using the same principles as the steam Rankine cycle one. The working fluid selected for the ORC ship retrofit is R-245fa as explained under item 2.5.5 Organic Rankine cycle. The turbine isentropic efficiency η_t used in the calculation is 80 % [66]. The pinch point temperatures of the evaporator and condenser used in the calculation are 5 °C. Figure 3.20 presents the calculated electric power generation from the ORC during the case study ship roundtrip.

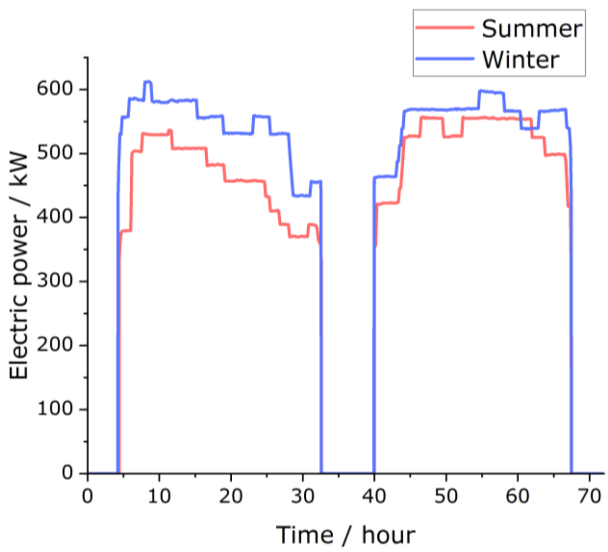

Figure 3.20 The ORC's electric power production during summer and winter roundtrips

During winter, lower seawater temperature, and hence, working fluid's lower condensing pressure, allows the ORC to achieve electricity generation power of more than 500 kW. During summer, ORC-generated electric power varies between 400 and 500 kW. The ORC electric power generator effect on the ship energy system is demonstrated in Figure 3.21. During winter, the ORC waste heat recovery can achieve 1 % energy saving. During summer, the energy saving is 0,9 %. The 175 °C steam potential could not be applied directly to the ORC circuit since it will overheat the working fluid. An intermediate heat transfer circuit would be needed to prevent overheating and use the steam potential. Another option to increase the ORC production would be a combination with the steam turbine outlet steam with a temperature of around 100 °C.

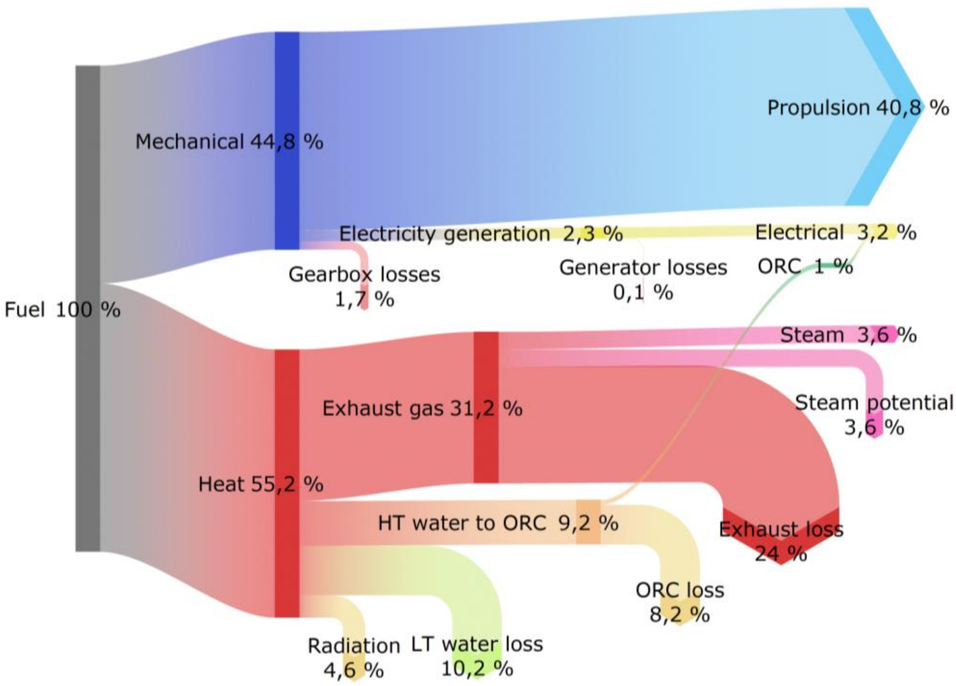

Figure 3.21 Sankey diagram of the ship energy system with the ORC during winter

3.4.3 Combination

The combination of steam turbine and ORC power generation using the heat cascade principle could be used to increase the ORC power output and the ship's energy efficiency. First, the HT cooling water preheats the ORC working fluid. Then, the expanded steam after the turbine is used to evaporate the ORC working fluid. It would increase the mass of the ORC working fluid and the pump's power consumption, but the ORC power generation should reach the highest levels. Figure 3.22 presents the total calculated electric power generation of the steam turbine and ORC during the case study ship roundtrip.

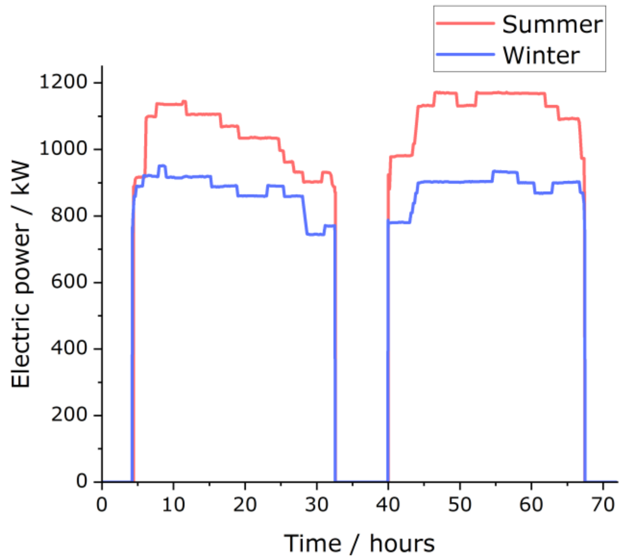

Figure 3.22 The ST and ORC total electric power production during summer and winter roundtrips

Summer operation of the combined steam turbine and ORC crosses the 1000 kW electric power generation level and yields over 1100 kW. Most of the time, the ORC-generated electric power during summer is above 700 kW. On the return part of the ship's roundtrip, the ORC electric power is over 800 kW. During winter, the ORC electric power is around 700 kW. The combined steam turbine and ORC electric power generation effect on the ship energy system is demonstrated in Figure 3.23. During summer, the combined system waste heat recovery can achieve 2,1 % of energy saving. During winter, the energy saving is 1,7 %.

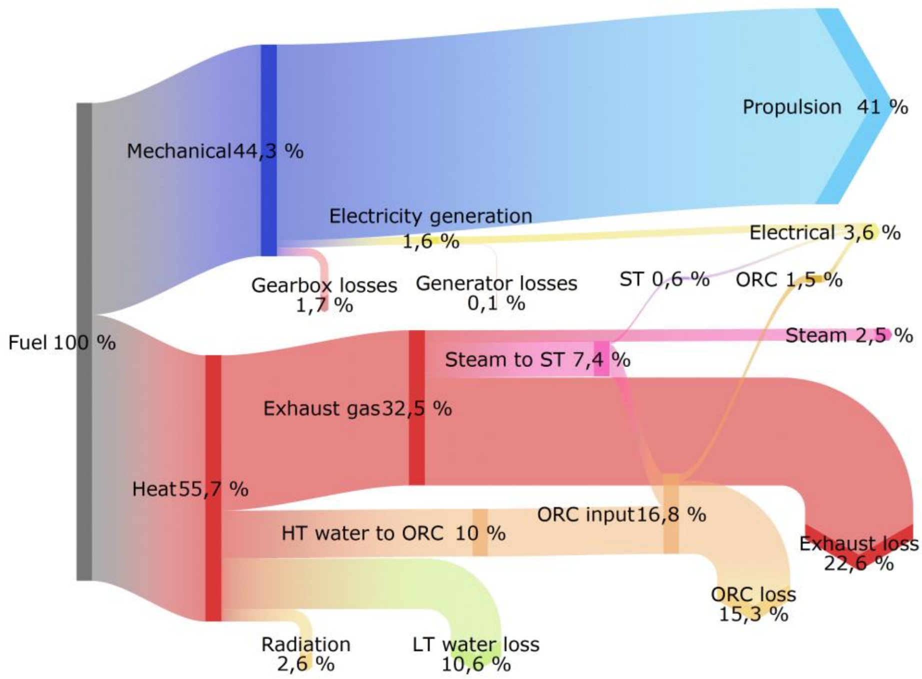

Figure 3.23 Sankey diagram of the ship energy system with the combined ST and ORC system during summer

4 EVALUATION RESULTS AND CONCLUSIONS

4.1 Operational

The results presented are obtained from the calculations made using the case study ship energy model and the waste heat recovery Rankine cycle models developed as part of the case study. The evaluation of energy efficiency and emission reduction potential of the waste heat recovery solutions considered for the case study ship is made on an annual operating basis. The case study ship's annual operating time is assumed to be 50 weeks and 1 day, allowing her to carry out 117 roundtrip iterations according to the operational profile. It is assumed that the studied summer and winter weather effects on the ship's annual operations are equal.

Table 4.1 presents the annual operational results of the waste heat recovery solutions applied to the case study ship navigating at sea. The conversion factor between fuel consumption and CO_2 emission for heavy fuel oil used for the case study ship navigation is 3,114 per IMO guidelines [80]. The case study ship's energy efficiency before evaluation was 45,8 %.

Table 4.1 The evaluation operational results

Waste Heat Recovery solution	Energy efficiency / %	Fuel saved / t	CO_2 emissions prevented / t
Steam turbine ST	46,2 (+0,4)	284	884
ORC	46,8 (+1,0)	658	2049
Combination ST and ORC	47,7 (+1,9)	1239	3858

The case study ship's steam turbine retrofit could achieve an additional 0,4 % of the ship's energy efficiency. The calculated fuel saving potential is 284 t of heavy fuel oil per year. The amount of fuel saved prevents 884 t of CO_2 emissions per year. The steam turbine retrofit is considered to have the lowest risk and safety impact on ship operations. The turbine is a steam consumer connected to the ship's steam system. The components of the steam Rankine cycle, such as boilers, feed water pumps, and condensers already exist and operate in the ship steam system. In an emergency, the steam flow to the turbine has to be shut down automatically, ship steam production is reduced, and excess steam left in the system is condensed. The system pressurized part has to meet the ship classification society rules on pressure equipment and piping systems. The steam turbine generator electrical part has to be integrated to the ship low voltage electric distribution system according to the electrical installations rules. The steam turbine electric generator has to be certified for installation on board ships.

The case study ship ORC retrofit could achieve an additional 1,0 % of the ship's energy efficiency. The calculated fuel saving potential is 658 t of heavy fuel oil per year. The amount of fuel saved prevents 2049 t of CO_2 emissions per year. The ORC retrofit would have a higher risk and safety impact on the ship operations since it is connected to the ship main engines' cooling water system. The ORC system, including the turbine, pumps, evaporator, and condenser, has to be properly designed and installed. In an emergency, e.g., working fluid leakage or pump malfunction, the ship engine cooling water system has to remain functional to keep the engines running and prevent potential engine damage. The ship ORC solutions available on the market have a marine certificate issued by classification societies that approves the ORC ship installation.

The case study ship's steam turbine and ORC retrofit could achieve an additional 1,9 % of the ship's energy efficiency. The calculated fuel saving potential is 1239 t of heavy fuel oil per year. The amount of fuel saved prevents 3858 t of CO_2 emissions per year. The steam turbine and ORC combination would have the highest risk and safety impact on the ship operations since it interconnects the ship's steam heating and main engine cooling systems. Additional equipment and piping, such as the steam condenser – organic working fluid evaporator, need to be added to the system. It will increase the system's complexity and its associated risks. The combined system's electric power generation above 1000 kW levels requires special attention to the system integration into the ship power management system. With the increase of the proposed waste heat recovery solution, energy efficiency and emission reduction potential increase the system complexity, risks, and safety measures.

4.2 Regulations

The shipping industry regulatory bodies encourage the application of energy efficiency and emission reduction technologies on ships. As discussed under item 2.2.1, the International Maritime Organization enforced the mandatory ship energy efficiency technical measures such as EEDI and EEXI. The technical measures require energy-efficient ships now and even more efficient ships in the future. The effect of the technical measures is also reflected in the ship's mandatory operational measure, CII, enforced at the same time as the EEXI. Both EEXI and CII compliance is mandatory for the case study ship. The effect of waste heat recovery technologies on the case study ship attained EEXI is calculated according to formula (2.1) and the IMO guidelines [81]. The case study ship attained CII value is calculated using formula (2.3).

Table 4.2 presents the regulatory results of the waste heat recovery solutions applied to the case study ship navigating at sea. The solution's design electric power is estimated based on the case study ship waste heat recovery calculation model results to use in the attained EEXI calculation.

Table 4.2 The evaluation regulatory results

Waste Heat Recovery solution	**Design electric power / kW**	**Attained EEXI reduction / g CO_2/t·nm**	**Attained CII reduction / g CO_2/t·nm**
Steam turbine ST	250	0,68 (3 %)	0,12 (0,9 %)
ORC	500	1,33 (6 %)	0,29 (2,1 %)
Combination ST and ORC	1000	2,67 (12 %)	0,54 (4,0 %)

The case study ship steam turbine retrofit could achieve attained EEXI reduction of 0,68 g CO_2 emissions per ton·nautical mile, which is 3 % lower than before ST retrofit. The attained CII reduction would be 0,12 g CO_2/t·nm, which is 0,9 % lower than before. The ORC retrofit could double the attained EEXI and CII reduction. The ORC attained EEXI could be reduced by 1,33 g CO_2/t·nm or 6 % lower than before the ORC retrofit. The attained CII reduction would be 0,29 g CO_2/t·nm, which is 2,1 % lower than before. Finally, the combination of steam turbine and ORC retrofit to the case study ship could achieve attained EEXI reduction of 2,67 g CO_2/t·nm or 12 % lower than before the ST and ORC retrofit. The attained CII reduction would be 0,54 g CO_2/t·nm, which is 4,0 % lower than before.

The considered waste heat recovery solution retrofits have potential to aid the case study ship comply with the ship energy efficiency EEXI and CII regulations. Certainly, the evaluated solutions improve the ship’s energy efficiency, however, their impact and role in the regulation context is mostly supportive, preventing the ship EEXI from exceeding the required reference line or dropping the CII rating from C to D.

The evaluation financial results include the effect of the considered solutions retrofit on the European Emission Trading System regulation. The ship’s operational CO_2 emissions in European waters are subject to taxation under the EU ETS. The ship CO_2 emissions prevented by means of the considered waste heat recovery solutions could increase the solutions’ profitability and make the retrofit option financially more attractive.

4.3 Financial

The ship waste heat recovery retrofit financial aspect, also known as the business case, is the decisive factor for a shipowner whose goal is to benefit from their assets and activities. The shipping industry's energy efficiency and emissions reduction are also important goals, but the industry's main stakeholders, including regulatory bodies, technical solutions suppliers, cargo owners, and shipowners, have to negotiate and agree on the appropriate leverage between environmental and economic aspects.

Table 4.3Table 4.1 presents the annual financial results of the waste heat recovery solutions applied to the case study ship navigating at sea. The heavy fuel oil price used in the calculations is taken as 500 €/t [82]. The CO_2 emissions EU ETS tax used in the calculations on intra-EU voyages is taken as 70 €/t [83]. The emissions percentage included in the EU ETS calculation is 100 %.

Table 4.3 The evaluation financial results

Waste Heat Recovery solution	**Cost of fuel saved / €**	**Cost of CO_2 emissions prevented / €**	**Total savings / €**
Steam turbine ST	142 000	61 880	203 880
ORC	329 000	143 430	472 430
Combination ST and ORC	619 500	270 060	889 560

The case study ship's steam turbine retrofit could achieve an annual saving of 203 880 €. The ORC retrofit annual savings would increase up to 472 430 €. The highest annual savings could be achieved by combining the steam turbine and ORC, reaching 889 560 €. The share of the prevented CO_2 emissions cost is around 30 % of total annual savings. The other 70 % is the saved fuel cost.

The project's material and labor costs were estimated to get an idea of the investment required for the waste heat recovery retrofit project. The case study ship waste heat recovery retrofit project costs are presented in Table 4.4. The labor cost includes the installation of the materials listed, as well as the project feasibility study, design and engineering, and project management. The estimated materials cost include:

- Waste heat recovery equipment cost;
- Auxiliary equipment (pumps, heat exchangers) cost;
- Structural modifications, foundations steel cost;
- Piping integration materials cost;
- Electrical integration materials cost;
- Automation integration materials cost.

Table 4.4 The case study ship waste heat recovery retrofit project cost

Waste Heat Recovery solution	Material cost / €	Labor cost / €	Total invetment cost / €
Steam turbine ST	710 000	540 000	1 250 000
ORC	1 400 000	900 000	2 300 000
Combination ST and ORC	2 950 000	2 300 000	5 250 000

The estimated project cost of the case study ship's steam turbine retrofit is 1 250 000 €. It considers the integration of the screw turbine designed for wet steam expansion with an electric power output of around 250 kW. The annual operating cost of the steam turbine is estimated to be 12 500 €. One of the ship voids must be converted into a machinery room to install the turbine. The steam turbine integration could be executed in operation.

The ORC retrofit estimated cost is 2 300 000 €. It considers the integration of the ORC system designed specifically for the heat available on board with an electric power output of around 500 kW. The annual operating cost of the ORC is estimated to be 50 000 €. The ORC installation also requires void conversion and an additional seawater pump for the condenser cooling. The ORC seawater pump capacity may require its sea chest and overboard, which require ship dry docking.

The estimated project cost of the steam turbine and ORC combination retrofit is 5 250 000 €. It considers the integration of the two pairs of steam turbine and ORC, one pair on the port side and the other pair on the starboard side. The pair means a combination of standard products available on the market, a steam screw turbine with electric power output around 150 kW and an ORC with electric power output around 350 kW. The annual operating cost of the combined system with two pairs of standard steam turbine and ORC is estimated to be 70 000 €.

The operating time required for a project to pay for itself is payback time. It is calculated using formula (4.1) and shows how many years of operation are needed for the project to start making profit and realize its business case. Table 4.5 presents the case study ship’s waste heat recovery retrofit projects payback time.

$$Payback\ time = \frac{Total\ investment\ cost}{Annual\ savings - Annual\ operating\ cost} \tag{4.1}$$

Table 4.5 The case study ship's waste heat recovery retrofit project payback time

Waste Heat Recovery solution	Payback time / year
Steam turbine ST	6,5
ORC	5,5
Combination ST and ORC	6,4

The case study ship steam turbine retrofit estimated payback time is 6,5 years, which nominally is the longest. The steam turbine retrofit could offer a simpler system compared to the ORC. The ORC system is more complex and involves higher risks but could offer a payback time of 5,5 years. To mitigate the risks, the considered ORC solution design could be reconsidered to combine low-power modular standard ORC solutions to achieve the power of one large ORC unit. The modular approach is used in the steam turbine and ORC combination retrofit, with an estimated payback time of 6,4 years. The benefit of the combined system, which uses most of the case study ship's waste heat recovery potential, is significant profit after the project pays off.

As a conclusion on the Baltic Sea ro-ro passenger ship waste heat recovery potential evaluation, the optimal technical solution is selected, which is the organic Rankine cycle system, which recovers heat from the main engine high temperature cooling water to produce electrical energy. The ORC solution is selected considering the evaluation's following findings:

- Significant annual operational profit of around 400 000 €;
- Moderate energy efficiency and emissions reduction;
- Short payback time 5,5 years;
- Manageable risks and safety requirements;
- Marine-certified ORC standard solutions market availability;
- Cycle efficiency improvement through cold Baltic Sea water exploitation.

SUMMARY

The thesis solves the problem of waste heat recovery on the Baltic Sea ro-ro passenger ship to improve energy efficiency and reduce emissions. The case study ship's energy system operation was analysed using data from a real reference ship. The ship waste heat recovery market was explored to find the most suitable technical solutions. The solutions considered were steam turbines and organic Rankine cycle systems based on the Rankine cycle principle. The steam turbine, ORC, and combination of both were applied to the ship's energy system calculation model to estimate their performance. The calculation model was developed and validated using data and measurements from the reference ship. The optimal technical solution for waste heat recovery on the case study ship was identified.

The shipping regulatory bodies' levers, such as IMO EEXI, CII, and EU ETS regulations, to control the sector's greenhouse gas emissions were studied to apply in the evaluation. It was found that the studied waste heat recovery solutions' role in the mandatory EEXI and CII energy efficiency measures is rather to support the ships to stay in compliance than to improve the index significantly. The EU ETS emission taxation system has demonstrated its efficiency in stimulation for waste heat recovery retrofits. The ETS emission tax share in the waste heat recovery system's annual savings was estimated at 30 %.

During winter, the Baltic Sea water temperature of 5 °C allowed the ORC solution power output to increase by more than 10 %. Ro-ro passenger ships have a significant footprint in the Baltic Sea region, creating more than a quarter of shipping emissions. Despite the slow steaming practice, the ro-ro passenger ships' speed is still relatively high, increasing energy consumption.

The Rankine cycle ship waste heat recovery previous studies and installations review attempted to find promising configurations and solutions. However, the challenge was determining the previous studies' results in one equivalent, the saved fuel energy equivalent. Nevertheless, the ship steam turbine and ORC waste heat recovery systems demonstrated fuel energy recovery from 2 % to 4 % on fast ships such as container ship, ro-ro passenger and cruise ships.

Two reference ship visits were organized to collect design and operational data to develop the case study ship energy model. Ship energy flow field measurements were conducted to validate the calculation model and improve its accuracy. The case study

ship's calculated fuel consumption error was ±2 % compared to the reference ship data. The steam turbine and ORC performance calculation model was combined with the case study ship's energy model for waste heat recovery potential evaluation.

The operational results revealed that the steam turbine retrofit could add 0,4 % to the case study ship's energy efficiency and reduce 884 t of CO_2. The ORC retrofit would add 1,0 % to the energy efficiency and reduce 2049 t of CO_2. The combination of both systems could add 1,9 % to the case study ship energy efficiency and reduce 3858 t of CO_2.

The financial results revealed that the steam turbine retrofit's estimated payback time is 6,5 years, and annual operational profit is around 190 000 €. The ORC retrofit's estimated payback time could be 5,5 years, achieving annual operational profit around 400 000 €. The complex combination of both systems would achieve a payback time of 6,4 years and annual operational profit around 800 000 €.

Based on a comprehensive analysis of risk and safety, cost estimation, and retrofit complexity, the study concluded that the optimal technical solution for the Baltic Sea ro-ro passenger ship waste heat recovery retrofit is the ORC system. The ORC system offers manageable risk and safety requirements, with marine certified solutions on the market. In the Baltic Sea context, the ORC system could achieve higher thermodynamic cycle efficiency through the use of the cold sea water, thereby enhancing its performance. The considerable annual operational profit, moderate energy efficiency and emission reduction, and the shortest payback time further support this conclusion.

The conducted study extension could attempt to improve energy efficiency further by exploitation of low-temperature engine cooling water for ORC preheating and/or evaluating the use of reversed ORC as a heat pump to generate heat using shore power electricity when the ship is in port.

KOKKUVÕTE

Lõputöö lahendab heitsoojuse taaskasutamise probleemi Läänemere ro-ro reisilaeval, et parandada energiatõhusust ja vähendada heitkoguseid. Juhtumiuuringu laeva energiasüsteemi tööd analüüsiti tõelise võrdluslaeva andmete põhjal. Sobivaimate tehniliste lahenduste leidmiseks uuriti laevade heitsoojuse taaskasutamise lahenduste turgu. Kaalutud lahendused olid auruturbiinid ja orgaaniline Rankine'i tsüklisüsteemid, mis töötavad Rankine'i tsükli põhimõttel. Auruturbiini, ORC, ja mõlema kombinatsiooni rakendati laeva energiasüsteemi arvutusmudelis, et hinnata nende jõudlust. Arvutusmudel töötati välja ja valideeriti võrdluslaeva andmete ja mõõtmiste abil. Selgitati välja optimaalne tehniline lahendus heitsoojuse taaskasutamiseks juhtumiuuringu laeval.

Hindamisel uuriti laevandust reguleerivate asutuste hoobasid, nagu IMO EEXI, CII ja EU ETS määrused, et kontrollida sektori kasvuhoonegaaside heitkoguseid. Leiti, et uuritud heitsoojuse taaskasutuslahenduste roll kohustuslike EEXI ja CII energiatõhususe meetmete puhul on pigem laevade vastavuse toetamine kui indeksi oluline parandamine. ELi heitkoguste maksustamise süsteem on näidanud oma tõhusust heitsoojuse taaskasutamise moderniseerimise stimuleerimisel. ETS heitmemaksu osakaal heitsoojuse taaskasutussüsteemi aastases säästus oli hinnanguliselt 30%.

Talvel Läänemere veetemperatuur 5 °C võimaldas ORC lahenduse võimsust suurendada rohkem kui 10%. Ro-ro reisilaevadel on Läänemere piirkonnas märkimisväärne jalajälg, mis tekitab enam kui veerandi laevanduse heitkogustest. Vaatamata *slow steaming* praktikale on ro-ro reisilaevade kiirus endiselt suhteliselt suur, mis suurendab energiatarbimist.

Rankine'i tsükli laeva heitsoojuse taaskasutamise varasemad uuringud ja paigaldiste ülevaated püüdsid leida paljutõotavaid konfiguratsioone ja lahendusi. Väljakutseks oli aga varasemate uuringute tulemuste määramine ühes ekvivalendis, säästetud kütuseenergia ekvivalendis. Sellegipoolest näitasid laeva auruturbiin ja ORC heitsoojuse taaskasutussüsteemid kiiretel laevadel, nagu konteinerlaevad, ro-ro reisilaevad ja kruiisilaevad, kütuseenergia taaskasutamist 2–4%.

Korraldati kaks võrdluslaeva külastust, et koguda projekteerimis- ja tööandmeid juhtumiuuringu laevade energiamudeli väljatöötamiseks. Arvutusmudeli kinnitamiseks ja selle täpsuse parandamiseks viidi läbi laeva energiavoolu väljamõõtmised. Juhtumiuuringu laeva arvestuslik kütusekulu viga oli ±2 % võrreldes laeva võrdluslaeva andmetega. Auruturbiini ja ORC jõudluse arvutusmudel kombineeriti juhtumiuuringu laeva energiamudeliga heitsoojuse taaskasutamise potentsiaali hindamiseks.

Töötulemustest selgus, et auruturbiini moderniseerimine võib suurendada juhtumiuuringu laeva energiatõhusust 0,4% ja vähendada 884 t CO_2. ORC moderniseerimine suurendaks energiatõhusust 1,0% ja vähendaks 2049 t CO_2. Mõlema süsteemi kombinatsioon võib suurendada juhtumiuuringu laeva energiatõhusust 1,9% ja vähendada 3858 t CO_2.

Majandustulemustest selgus, et auruturbiini moderniseerimise eeldatav tasuvusaeg on 6,5 aastat ja aastane tegevuskasum ligikaudu 190 000 €. ORC moderniseerimise hinnanguline tasuvusaeg võib olla 5,5 aastat, saavutades aastase tegevuskasumi umbes 400 000 €. Mõlema süsteemi kompleksne kombinatsioon saavutaks tasuvusaja 6,4 aastat ja aastase tegevuskasumi ligikaudu 800 000 €.

Tuginedes põhjalikule riskide ja ohutuse analüüsile, kuluhinnangule ja moderniseerimise keerukusele, jõuti uuringus järeldusele, et Läänemere ro-ro reisilaevade heitsoojuse taaskasutamise moderniseerimise optimaalne tehniline lahendus on ORC-süsteem. ORC-süsteem pakub juhitavaid riski- ja ohutusnõudeid ning turul on meresõidusertifikaadiga lahendusi. Läänemere kontekstis võib ORC-süsteem saavutada suurema termodünaamilise tsükli efektiivsuse külma merevee kasutamise kaudu, parandades seeläbi selle toimivust. Märkimisväärne aastane tegevuskasum, mõõdukas energiatõhusus ja heitkoguste vähendamine ning lühim tasuvusaeg kinnitavad seda järeldust veelgi.

Läbiviidud uuringulaiendus võib püüda energiatõhusust veelgi parandada, kasutades madala temperatuuriga mootori jahutusvett ORC eelsoojenduseks ja/või hinnates ümberpööratud ORC kasutamist soojuspumbana soojuse tootmiseks kaldalt elektrienergiat kasutades, kui laev on sadamas.

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