THESIS FOR THE DEGREE OF DOCTOR OF ENGINEERING

Modelling, analysis and optimisation of ship energy systems

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CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2016

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ISBN: 978-91-7597-359-3
Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr 4040
ISSN 0346-718X.

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Cover:
Emergence in ship modelling. Photo by Francesco Baldi, Edited by Sandro Baldi

Printed by Chalmers Reproservice
Gothenburg, Sweden 2016

Abstract

Shipping is the backbone of today's economy, as 90% of global trade volumes is transported by sea. Much of our lifestyle today is only made possible by the existence of shipping as a cheap and reliable mean of transportation across the globe.

However, the shipping industry has been challenged in the latest years by, among others, fluctuating fuel prices and stricter environmental regulations. Its contribution to global warming, although today relatively small, has been set under scrutiny: for shipping to be part of a sustainable economy, it will need to reduce its emissions of greenhouse gases.

Increasing ship energy efficiency allows reducing fuel consumption and, hence, carbon dioxide emissions. The latest years have witnessed a multiplication of the efforts in research and development for increasing ship energy efficiency, ranging from improvements of existing components to the development of new solutions. This has also contributed to ship energy systems to become more complex. The optimisation of the design and operation of complex systems is a challenging process and the risks for sub-optimisation are high.

This thesis aims at contributing to the broader field of energy efficiency in shipping by adopting a systems perspective, which puts a special focus on system requirements and on interactions within the system. In this thesis, the energy systems of two case study ships were analysed using energy and exergy analysis to identify energy flows and inefficiencies. Then, solutions for improving the energy efficiency of the existing systems were proposed and evaluated accounting for the ship's observed operating range and for how added elements influenced the existing systems and their performance.

The results of this thesis show the importance of modelling the interactions between different parts of the energy systems. This allows not only a more accurate estimation of the benefits from the installation of new technologies, but also the identification of potential for additional energy savings. This is particularly important when the broad range of ship operations is included in the analysis, rather than focusing on the performance of the system in design conditions. In addition, the results of this thesis also show that there is potential for further improving ship energy efficiency by putting additional focus on heat losses from the engines and on how to efficiently recover them.

List of publications

Appended papers

This thesis represents the combination of the research presented in the following appended papers:

Paper I : Baldi F. , Johnson H. , Gabrielii C. , & Anderssson, K. (2015). Energy and exergy analysis of ship energy systems: the case study of a chemical tanker. *International Journal of Thermodynamics*, **18**(2), 82-93.
The author of this thesis is the main contributor to ideas, planning, data collection, calculations, and writing.

Paper II : Baldi F. , Ahlgren F. , Nguyen T.V. , Gabrielii C. & Anderssson K. (2015). Energy and exergy analysis of a cruise ship. *Proceedings of the 28th International Conference on Efficiency, Cost, Optimisation, Simulation and Environmental Impact of Energy Systems (ECOS)* June 2015 Pau, France.
The author of this thesis is the main contributor to ideas, planning, calculations, and writing.

Paper III : Baldi F. , Theotokatos G. & Anderssson K. (2015) Development of a combined mean value-zero dimensional model and application for a large marine four-stroke Diesel engine simulation. *Applied Energy* **154**, 402-415.
The author of this thesis participated to ideas, planning, data collection, calculations, and writing.

Paper IV : Baldi F. & Gabrielii, C. (2015). A feasibility analysis of waste heat recovery systems for marine applications. *Energy* **80**, 654-665.
The author of this thesis is the main contributor to ideas, planning, data collection, calculations, and writing.

Paper V : Baldi F. , Larsen U. & Gabrielii C. (2015). Comparison of different procedures for the optimisation of a combined Diesel engine and organic Rankine cycle system based on ship operational profile. *Ocean Engineering* **110**, 85-93.
The author of this thesis is the main contributor to ideas, planning,

data collection, and writing, while Ulrik Larsen performed the main part of the simulations and optimisation process.

Paper VI : Baldi F. , Ahlgren F. , Melino F. , Gabrielii C. & Anderssson, K. (2016). Optimal load allocation of complex ship power plants. Submitted to *Energy Conversion and Management on 2016-03-30.* *The author of this thesis is the main contributor to ideas, planning, calculations, and writing.*

Other publications

Baldi F. , Gabrielii C. , Anderssson K. & Petersen B.-O. (2012). From Energy Flows to Monetary Flows-An Innovative Way of Assessing Ship Performances Through Thermo-Economic Analysis. *Proceedings of the International Association of Maritime Economists Conference (IAME)* June 2012 Taipei, Taiwan.

Baldi F. , Bengtsson S. & Anderssson K. (2013). The influence of propulsion system design on the carbon footprint of different marine fuels. *Proceedings of the Low Carbon Shipping Conference* September 2013 London, United Kingdom.

Baldi F. , Larsen U. , Gabrielii C. & Anderssson K. (2015). Analysis of the influence of the engine, propeller and auxiliary generation interaction on the energy efficiency of controllable pitch propeller ships. *Proceedings of the International Conference of Maritime Technology* July 2014 Glasgow, United Kingdom.

Larsen U. , Pierobon L. , Baldi F. , Haglind F. & Ivarsson A. (2015). Development of a model for the prediction of the fuel consumption and nitrogen oxides emission trade-off for large ships. *Energy* **80** 545-555.

Baldi F. , Gabrielii C. , Melino F. , & Bianchi M. (2015). A preliminary study on the application of thermal storage to merchant ships. *Proceedings of the 7th International Conference on Applied Energy* March 2015 Abu Dhabi, United Arab Emirates.

Coraddu A. , Oneto L. , Baldi F. & Anguita D. (2015). Ship efficiency forecast based on sensors data collection: Improving numerical models through data analytics. *Proceedings of the OCEANS 2015* May 2015 Genoa, Italy.

Baldi F. , Lacour S. , Danel Q. , & Larsen U. (2015). Dynamic modelling and analysis of the potential for waste heat recovery on Diesel engine driven applications with a cyclical operational profile. *Proceedings of the 28th International Conference on Efficiency, Cost, Optimisation, Simulation and Environmental Impact of Energy Systems (ECOS)* June 2015 Pau, France.

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Symbols and abbreviations

Roman Symbols

B	Exergy $[kW]$
g	Gravitation acceleration on earth $[\frac{m}{s^2}]$
H	Enthalpy $[kJ]$
I	Irreversibility $[kJ]$
J	Advance ratio
K_Q	Adimentional propeller torque
K_T	Adimentional propeller thrust
m	Mass $[kg]$
P	Power $[kW]$
P/D	Propeller pitch over diameter ratio
Q	Heat $[kJ]$
S	Entropy $[\frac{kJ}{K}]$
T	Temperature $[K]$
v	Speed $[kn]$
W	Work $[kJ]$
z	Altitude above the sea level $[m]$

Greek Symbols

δ	Efficiency loss ratio
ϵ_t	Total exergy efficiency
ϵ_u	Task (exergy) efficiency
η	Energy efficiency
γ	Relative irreversibility
λ	Load
ω	Speed $[rpm]$

Subscripts

0	Reference ambient conditions
en	Energy
ex	Exergy
in	Inlet
out	Outlet
ph	Physical

Abbreviations

0DEM	Zero-dimensional
AE	Auxiliary engine
BSFC	Brake specific fuel consumption $[\frac{g}{kWh}]$
CAC	Charge air cooler
CO_2	Carbon dioxide
CPP	Controllable pitch propeller
DLS	Data logging system
EC	European Commission
ECA	Emission controlled area
EEDI	Energy Efficiency Design Index
EU	European Union
FC	Frequency Converter
FPP	Fixed pitch propeller
GB	Gearbox
GHG	Greenhouse gas
HFO	Heavy fuel oil
HHV	Higher heating value
HRSG	Heat recovery steam generator
HT	High temperature
HVAC	Heat, ventilation, and air conditioning
IFO	Intermediate fuel oil
IMO	International Maritime Organisation
JW	Jacket water
LHV	Lower heating value

SYMBOLS AND ABBREVIATIONS

LNG Liquified Natural Gas

LO Lubricating oil

LT Low temperature

MCR Maximum continuous rate [kW]

MDO Marine diesel oil

ME Main engine

MGO Marine gas oil

MVEM Mean value engine model

NO$_X$ Nitrogen oxides

ORC Organic Rankine cycle

PM Particulate matter

S/G Shaft generator

SCR Selective catalytic reactor

SEEMP Ship Energy Efficiency Management Plan

SO$_X$ Sulphur oxides

USD United States dollars

VGT Variable geometry turbine

WHR Waste heat recovery

Chapter 1

Introduction

Low freight rates, fluctuating fuel prices, stricter environmental regulations, and expectations to reduce greenhouse gas (GHG) emissions make the current situation particularly challenging for the shipping industry. In this context, the interest in solutions for reducing ship fuel consumption has increased in the latest years, together with the technological improvements in ship energy efficiency. This thesis aims at contributing to the knowledge required for the reduction of fuel consumption from shipping. This is done by focusing on the potential for improvement coming from the application of energy systems engineering to ship on board energy systems.

1.1 Rationale

The rationale behind this thesis is related to both environmental and economic aspects.

From an **environmental perspective**, the main connection between energy efficiency and the environment relates to GHG emissions, which are today the main responsible of global warming today (IPCC, 2014). In spite of the fact that in 2012 carbon dioxide (CO_2) emissions from shipping amounted to only 2.5% of the total global anthropogenic emissions, they are expected to increase in the future by between 50% and 250% as a consequence of growing trade volumes (Smith *et al.*, 2014).

From an **economical perspective**, despite today's low fuel prices, there are reasons to advocate for improved fuel efficiency in shipping. Fuel prices have shown to be volatile in history, and there is no guarantee that they will not rise again in the future. In addition, environmental regulations are becoming stricter all over the world, and compliance often relates to higher fuel expenses. This is particularly true in the aforementioned case of CO_2, as market based measures are being discussed at different levels for incentivising a faster transition to low-carbon shipping.

The improvement of energy efficiency in shipping constitutes a relatively broad field of studies, from logistics and social studies to engineering. Narrowing the perspective to the latter, the latest research and development efforts have resulted in a large number of potential solutions, ranging from improvements of existing components (e.g. propellers and Diesel engines), applications of land-based technologies to shipping (e.g. waste heat recovery, fuel cells, batteries) to completely new solutions (e.g. hull air lubrication, Flettner rotors).

These technical innovations make ship energy systems to become increasingly complex, being composed of a large number of components interacting with each other. Solely focusing on individual parts of the system, thereby neglecting or over-simplifying the interactions between the components, can lead to misleading results and sub-optimisation. In spite of this observation, research in the application of systems science and engineering, that focuses expressively on complex systems, is limited to a handful of examples. This constitutes the main rationale of this thesis, which focuses on **looking at ship energy systems from a systems perspective**.

1.2 Aim and research questions

The aim of this thesis is to analyse the benefits of employing an energy systems engineering approach in the quest for improving energy efficiency in shipping.

This analysis is structured in two main objectives, each of them further represented by a number of research questions.

The first objective is to apply a systematic procedure for **analysing the performance** of ship on board energy systems. This can be related to two main research questions:

RQ 1.1 What type of information about the performance of the ship on board energy systems can be gathered based on the data/documentation typically available from on board monitoring systems?

RQ 1.2 What useful insight of the system can be gained by applying energy and exergy analysis to ship on board energy systems?

The improved understanding that results from an in-depth analysis of the system leads to the identification of opportunities for its improvement. Hence, the second objective of this thesis is the **synthesis of potential solutions for improving the performance** of ship on board energy systems towards a reduction of its fuel consumption. This is done according to principles of systems engineering, hence leading to the following additional research questions:

RQ 2.1 What can be gained by looking at interactions within the system rather than focusing on the performance of individual components?

RQ 2.2 What can be gained by looking at a broader range of expected ship operations rather than at one specific design point?

RQ 2.3 Based on the above principles, what is the potential for reducing fuel consumption by improving ship on board energy systems?

1.3 Delimitations

Energy focus : While the discipline of systems engineering is interdisciplinary in its original definition, this thesis focuses on the ship as an energy system and on the minimisation of the energy input for a given energy output. Economical aspects are briefly touched upon, but do not constitute the main focus of this thesis. Environmental, human factors, and other technical aspects (such as maintenance) lie outside of the main scope of this work.

System boundaries : In this thesis, the ship power plant constitutes the main system of interest. This includes the main components on board that are involved in the process of energy conversion to its final use. The different final energy users, such as the propeller, the heating systems and electric components, are not part of the main system of interest.

Case studies : Although the methods and principles presented and discussed in this thesis are general in their purpose, they are here applied specifically to two case study vessels.

Commercial vessels : This thesis focuses on large commercial vessels. Smaller ship types, such as inland ferries and leisure crafts are not directly covered by the results of this study.

Mathematical modelling : The work presented in this thesis focuses on the use of computational models for the analysis and evaluation of ship on board energy systems. This excludes, for instance, direct experimentation and the realisation of prototypes.

1.4 Thesis outline

Chapter 2 provides a brief introduction to the shipping sector (Sec. 2.1) and to the main drivers for research in the field of energy efficiency (Sec. 2.2). The main features of ship energy systems are described in Sec. 2.3, while a review of some of the most promising technical measures for energy efficiency is presented in Sec. 2.4.

Energy systems engineering represents the methodological basis of this thesis. Chapter 3 provides the reader with an introduction to its main principles (Sec. 3.1), and a description of the tools used in this study: energy and exergy analysis (Sec. 3.2) and mathematical models (3.3).

Chapter 4 describes how energy systems engineering principles were applied in this thesis. This includes an introduction to the general methodological approach (Sec. 4.1) and a description of the two case studies (Sec. 4.2) and of the data available for each of them (Sec. 4.3). The chapter also summarises the main assumptions employed in each of the studies that build up this thesis (Sec. 4.4).

Chapter 5 reports the main results of this thesis, subdivided between systems analysis (Sec. 5.1, related to Papers I and II) and synthesis (Sec. 5.2, related to Papers III to VI).

Chapter 6 then discusses how these results provide evidence of the benefits of an energy systems engineering approach, both in the analysis (Sec. 6.1) and in the synthesis process (Sec. 6.2). The chapter further develops by discussing how the findings presented in this thesis can be used to advocate for an increased focus on solutions for more efficient on board energy systems (Sec. 6.3). As this thesis focuses on the analysis of two case studies, the generalisability of the findings is also discussed (Sec. 6.4).

Proposals for future research in the field and suggestions to stakeholders are presented in Sec 7.1 and 7.2, while the conclusions are finally summarised in the last chapter (Chapter 8).

Chapter 2

Background

Shipping and energy efficiency

Chapter 2 represents an introduction to the domain of shipping. In Section 2.1 the main characteristics of shipping with particular focus on energy efficiency matters are presented; Section 2.2 describes the details of the rationale for working on energy efficiency, summarised into the economic and environmental standpoints. The ship as an energy system is described from a technical perspective in Section 2.3. Section 2.4 finally provides a survey of the current efforts for improving ship energy efficiency for the two technologies that are mostly dealt with in this thesis: waste heat recovery systems and hybrid propulsion systems.

2.1 An introduction to shipping

Throughout the course of the history of mankind, the development of society has gone hand in hand with trade. In spite of the importance of local and international land trade routes, shipping has always been the main mean of transportation for goods and people over long distances.

Merchant shipping has been growing continuously over the past years, hand in hand with global trade. The volume of world seaborne trade increased from 2.6 to 9.8 billion tons of cargo from 1970 to 2014, and today anything from iron ore, coal, oil and gas to cars, grains and containerized cargo is transported by sea, making shipping the backbone of global economy (UNCTAD, 2015). Today, shipping contributes to an estimated 80-90% of the global trade[1] (Maritime Knowledge Centre, 2012; UNCTAD, 2015).

As any other sector, shipping has some business-specific features, some of which influence the processes of designing and operating ships for reduced fuel consumption[2]:

[1]in ton km, i.e. based on the amounts of goods transported and the distance covered

[2]For a broader picture concerning energy efficiency in shipping from an organisational perspective,

- The fact that the owner of the cargo, the owner of the ship and the operator of the ship are often different actors generates **split incentives**. In particular, as the shipowner does not pay for the fuel, he/she does not have any incentive in building or buying a more energy efficient ship. On the other hand, when not even the ship operator pays for the fuel either (the cargo owner can pay for it, depending on the charter party), he/she does not have any incentive for saving fuel on an operative basis, for instance by sailing at a lower speed. This situation often hinders efforts in efficient ship operations and slows down the uptake of energy efficient technologies (Faber *et al.*, 2011; Jafarzadeh & Utne, 2014; Agnolucci *et al.*, 2014).

- Differently from e.g. planes and cars, ships are built on **individual or small-series basis**, which discourages research and development as they become too expensive if performed on an individual ship basis. This is not true for most ship components, such as engines and propellers, which partly explains why most technical developments for energy efficiency are seen in component development more than in ship design. In addition, when order books are full, shipyards tend to only accept orders for very "standard" designs which require little effort and allow maximizing the revenues (Devanney, 2011; Faber *et al.*, 2011).

- The **operational life** of a vessel can range from 15 to more than 30 years (Stopford, 2009). Ships built according to non-optimal standards for energy efficiency will therefore have an impact for a long time.

- Ships are sometimes used as **mere assets** by investors, who look more at the value of the sales and purchase market rather than at the energy efficiency of the vessels. As a consequence, efficient vessels are not always associated to a higher value on the second-hand ship market (Jafarzadeh & Utne, 2014).

2.2 The need for energy efficiency in shipping

2.2.1 The environmental standpoint: cutting GHG emissions

The question of reducing fuel consumption from shipping is related to one of the most important challenges of today's society: global warming.

CO_2 **emissions** are known to be the main cause of the anthropogenic contribution to global warming. While shipping-related emissions contribute today to 2.5% of the total of anthropogenic emissions[1] (Smith *et al.*, 2014), these emissions are expected to increase in the future by up to 250% as a consequence of growing trade volumes (see Figure 2.1), at the same time as emissions from other sectors are expected to decrease[2]

the reader is suggested to check the Hannes Johnson (2016) PhD thesis.

[1]Note that this number refers to CO_2 emissions, while the contribution to the total GHG emissions is lower.

[2]The predictions from IMO 3^{rd} GHG study propose 16 alternative scenarios, of which only one predicts lower emissions in 2050 compared to 2012 levels (Smith *et al.*, 2014).

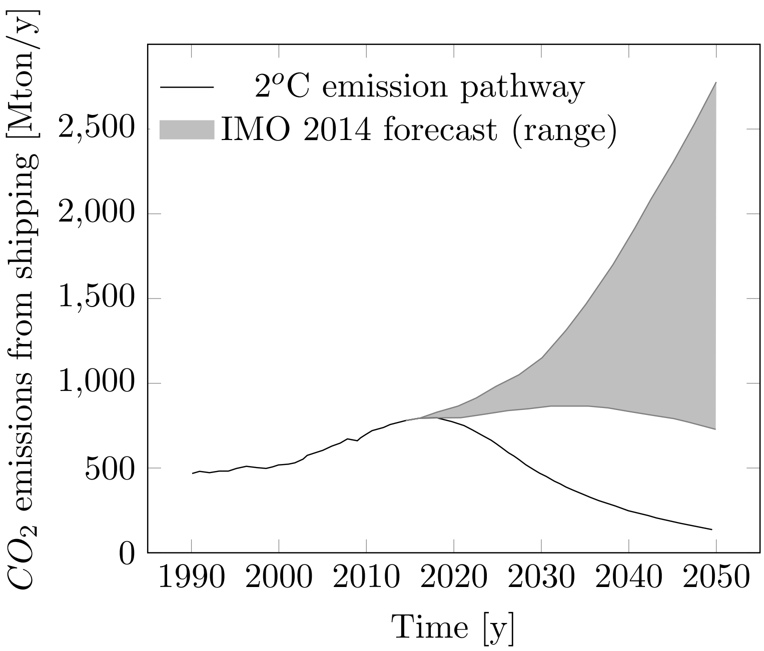


Figure 2.1: Comparison between forecast GHG emissions from shipping and viable pathways for achieving the 2 degrees climate goal. Adapted from (Anderson & Bows, 2012)

(Smith *et al.*, 2014).

However, even in the most optimistic scenario presented by IMO reports, emissions from shipping will reach much higher levels compared to what required for keeping global climate from warming beyond acceptable limits (see Figure 2.1). When more pessimistic scenarios are taken into account the picture becomes even gloomier Anderson & Bows (2012).

In 2013 the International Maritime Organisation (IMO) issued two main regulations connected to the reduction of shipping contribution to global CO_2 emissions (MEPC, 2011):

Energy Efficiency Design Index (EEDI) : A technical indicator of the ship's design energy efficiency. It is measured in tons of CO_2 emitted per ton of cargo transported and per km travelled. The EEDI is calculated based on the ship's performance when it is delivered and compared to a baseline value.

Ship Energy Efficiency Management Plan (SEEMP) : A document that has to be kept on board of every vessel where the ship operator must show that he/she has addressed the improvement of ship energy efficiency and that there is a plan for action for the future.

Although these measures represent a step forward for a reduction of CO_2 emissions from shipping, their effectiveness has been put under question for being inaccurate and not sufficiently ambitious (Johnson *et al.*, 2012; Bazari & Longva, 2011; Smith *et al.*, 2014).

2.2.2 The economic standpoint: much more than fuel prices

Shipping is primarily a business, and regardless all environmental concerns its main purpose is to generate a profit.

The most direct economic incentive to reduce fuel consumption is related to **fuel costs**. Research have shown that there is a large number of measures that could increase energy efficiency at a negative cost (Eide *et al.*, 2011). These considerations, however, heavily depend on the current fuel price.

Box 2.1: Marine fuels

As a consequence of the generally low requirements from an environmental standpoint and of the flexibility of marine engines, the shipping industry has been able to choose among a wide variety of different fuels:

Residual fuels : residual oils are mainly made of the heavy fraction remaining after the oil refinement process. Because of the high viscosity, these fuels need to be heated to up to 150^{o}C to achieve proper atomisation properties before injection. Normally, residual fuels have a relatively high sulphur content (up to 3.5% is today allowed), although low-sulphur residual fuels are available on the market. The two main variants of residual fuels are **heavy fuel oil** (HFO), made almost entirely of residual oils, and **intermediate fuel oil** (IFO), where HFO is partly blended with distillate fuels.

Distillate fuels : distillate fuels are made of lighter fractions of the oil refining process. The "lightest" of the distillate fuels is **Marine gas oil** (MGO), which is equivalent to Diesel fuels used in the automotive sector, while **Marine Diesel oil** (MDO) is a light blend of MGO and residual oil.

Other fuels : Mostly as a consequence of stricter environmental regulations, new fuels are being tested for use in the marine sector. This includes, among others, natural gas (generally in its liquefied form, LNG), ehtanol, and methanol.

In fact, fuel prices today are far from the peak achieved in 2012 (see Figure 2.2). According to observations of the past years, HFO prices tend to oscillate between 71% and 76% of the crude oil price (Ship&Bunker, 2015). Today's forecasts for crude oil prices suggest that they will range between 30 and 100 USD per barrel until 2020, which would suggest bunker fuel prices ranging between 226 and 753 USD per metric ton, while most likely remaining somewhere around 400 USD/ton (Ship&Bunker, 2015).

However, looking at the forecasts for bunker fuel prices issued in 2010, before the recent drop in crude oil prices (Figure 2.2), it appears that the reliability of these

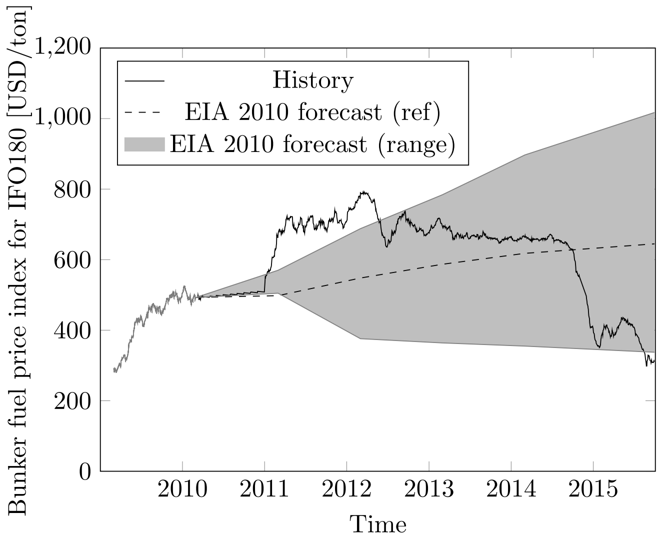


Figure 2.2: Historical IFO180 bunker prices evolution since 2009 and comparison with 2010 EIA forecast

forecasts can be questioned[1]. Although fuel prices are low today, they might rise again in the future.

2.2.3 Shipping and the environment: an economic matter

Fuel prices are not the only element influencing fuel-related costs. In recent years environmental concerns have become significantly stricter, adding to various types of operational costs on board and, particularly, on fuel related costs.

Sulphur oxides (SO_X) are emitted as a consequence of the sulphur in the fuel, which entirely oxides to SO_2 and SO_3 during combustion. SO_X emissions cause several harmful effects on the environment, such as acid rain and ocean acidification, and are precursors to the formation of particulate matter (PM) which is also harmful both to the environment and to human health. Today's global limit for the sulphur content is 3.5% on a weight basis, to be reduced to 0.5% in 2020[2] (IMO, 2013), while the global average was estimated to lie around 2.8% in 2012 (Mestl *et al.*, 2013). In emission controlled areas (ECAs), the limit was reduced to 0.1% since 2015.[3] Low-sulphur

[1]Dan Sten Olsson, manager at Stena Lines, recently declared in an interview *"When we designed the HSS-ships in 1992 oil prices were around 20 USD per barrel and further sank down to 12 USD/barrel. The ships were designed to be able to withstand a fuel price increase of up to 60%, although we never really considered an increase of more than 50% to be possible. To be able to be competitive up to 40, 100 USD/barrel was simply unthinkable" (Davidsson, 2015)*

[2]This decision will be subject to a review in relation to the availability of distillate fuels and systems for compliance, and might be postponed to 2025

[3]In spite of the recent reductions, these limits are still much higher compared to those valid for land-based transportation: fuel for trucks and Diesel trains can contain a maximum of 0.001% sulphur, 100 times less than what allowed for shipping in ports and ECAs today (EEA, 2013).

fuels are more expensive (the premium for distillate fuels normally ranges between 200 and 300 USD/ton), while scrubbers are costly to install and require energy during operations. Therefore, stricter regulations of SO_X emissions will provoke an increase of fuel costs.

Nitrogen oxides (NO_X) are emitted as a consequence of the high temperatures in the Diesel engines during combustion, which causes nitrogen and oxygen in the combustion air to react. Nitrogen oxides contribute to the processes of water eutrophication and acidification, are precursors to toxic chemicals (ground level ozone, secondary particulate matter) and can damage plant growth (Magnusson, 2014). Today NO_X emissions are regulated from the perspective of engine design (IMO, 2013). The global limit (Tier II) can be met by using today's engine technology stand-alone. Tier III limits (today valid only in US coastal waters, but under discussion in other areas of the world), on the other hand, can only be met via the installation of a selective catalytic reactor (SCR) or the use of alternative fuels (such as LNG and methanol).

Carbon dioxide (CO_2) is, as previously mentioned, the main driving force, from an environmental perspective, for improving ship energy efficiency This is generating political efforts to push shipping companies towards energy efficiency. Apart from the aforementioned IMO measures (EEDI and SEEMP), the European Union (EU) has recently decided to actively address the matter of including emissions from shipping in its GHG reduction policies (EC, 2013a), that will include, as a first step, the implementation of a monitoring, reporting and verification scheme for ships from 2018 (EC, 2013b). This will be followed by the definition of reduction targets and by the application of market based measures (EC, 2013a). Although the reduction targets for shipping have not been set yet, they are expected to be in the range of 40% to 50% by 2050, compared to 2009 levels inside the EU (EC, 2013a). Compared to current expectations of future development of CO_2 emissions from shipping (Smith *et al.*, 2014), this is an ambitious objective that will require a strong commitment.

2.3 The ship as an energy system

A ship needs fuel for operations. In the most general case, fuel is converted on board to energy in the form required for its final use: mechanical power for propulsion, electric power for on board auxiliaries and thermal power for heating purposes.

2.3.1 Energy demand

A ship is built and operated for a specific reason, normally referred to as mission, that varies from ship to ship (e.g. transporting cargo, transporting passengers, bringing fighting power at sea, etc.). In order to achieve this mission, a ship needs to be able to perform a certain amount of functions in addition to propulsion. These may range from providing a safe support for on board activities to ensuring hotel facilities for the

Box 2.2: Ship energy systems: definitions

In this thesis, different terms are used to refer to the ensemble of component and subsystems that are installed on board and that contribute to the behaviour of the ship from an energy perspective

Ship energy systems : the entirety of the ship systems that can be considered to be relevant from an energy perspective. Therefore, also hull and propeller are included.

Ship on board energy systems : the part of the ship energy systems located inside the hull. From an energy perspective, the propeller shaft constitute the main boundary of the system.

Ship power plant : the part of the on board ship energy system that is responsible for energy conversion. It therefore includes engines, generators and boilers, but not users (e.g. pumps, compressors, heaters, etc.). The ship's power plant is the main focus of this thesis.

Propulsion system : the part of the ship energy system devoted to propulsion. It generally includes the main engine(s) and the propeller(s).

crew[1].

On board energy demand is generally subdivided in three main categories (see also Fig. 2.3) (Woud & Stapersma, 2003):

Propulsion power : Ship movement generates a resistance from the water and, to a minor extent, from the air. This resistance depends primarily on a ship's speed and on the specifics of the hull (e.g., the shape, state, and wetted surface)[2]. External factors, such as the growth of various marine organisms on the hull and adverse weather conditions, also have an influence on the demand for propulsion power (Woud & Stapersma, 2003).

Auxiliary electric power : Many components on board require electric power during ship operations. Some of them are present on all ships and are related to basic support functions, such as the navigation equipment, cooling and lubricating pumps, compressors in air conditioning (HVAC) system, fans, ballast water pumps, and lights[3]. Specific ship types might require the operation of energy in-

[1]The focus of this thesis lies on the energy aspect of the ship systems. The analysis therefore focuses on the parts of the ship that have a significant influence on the ship's fuel consumption. As an example, the radar is a crucial part of the ship's navigational system, but it is not particularly interesting from an energy perspective since it requires little power to be operated.

[2]The following equation is broadly accepted as a simple approximation of the dependence of ship resistance on speed: $R_{ship} = Cv_{ship}^2$

[3]This base load can be roughly estimated as a function of the installed engine power: $P_{el}[kW] =$

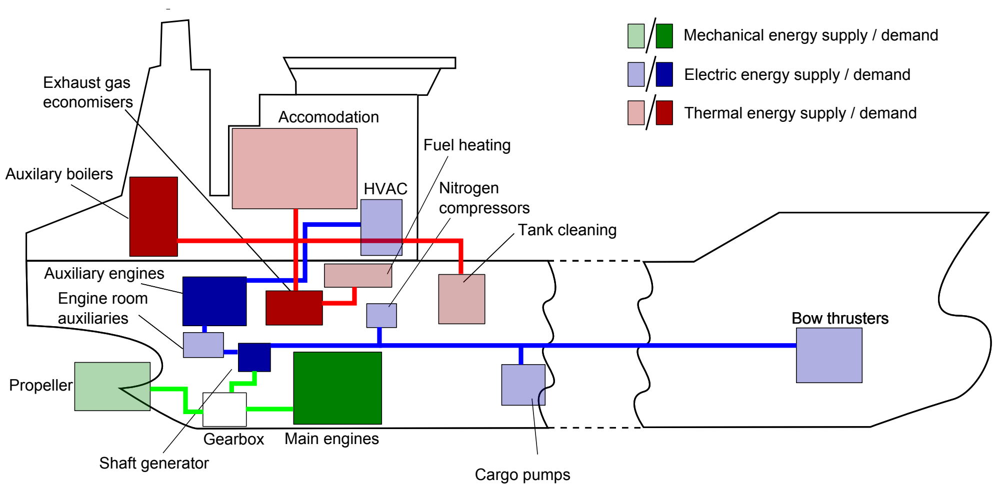


Figure 2.3: Schematic representation of the ship energy systems of a chemical tanker

tensive mission-related equipment, such as inert gas compressors and cargo pumps on tankers, refrigerated containers on containerships, etc.

Auxiliary thermal power : Heating is generally required for three main uses on board: accommodation, fuel heating, and fresh water generation. Similarly to auxiliary electric power demand, special ship types have additional requirements for heating, such as in the case of product tankers (for heating low-viscous cargo) and cruise ships (for accommodation)

2.3.2 Prime movers and energy converters

In order to provide energy in the required form to the different demands, the energy system of a ship is equipped with a number of devices for energy conversion.

Propulsors

The propeller is the most widespread solution for converting mechanical power from the engine shaft into a thrust force. Thrust bearings connect the shaft to the ship, thus allowing the further conversion of the thrust force into ship motion.

Fixed pitch propellers (FPP) represent the most common and basic propeller type and are characterized by having blades whose angle relative to the axis of the shaft (pitch) is fixed. FPPs are the most widespread solution for ship propulsion, and are particularly common among container ships, tankers, and bulk carriers (Carlton, 2012).

Controllable pitch propellers (CPP) allow the variation of the propeller pitch. This ability provides the CPP with an extra degree of freedom in addition to its rotational speed. As a consequence, CPPs are installed for increasing ship manoeuvrability,

$100 + 0.55(MCR_{ME})^{0.7}$ (Woud & Stapersma, 2003).

for improving the ability of adapting load to drive characteristic, and for giving the possibility to generate constant-frequency electric power with a generator coupled to the main engines (Woud & Stapersma, 2003). CPPs are generally more expensive and delicate than FPPs. They are most favoured on passenger ships, ferries, general cargo ships, tugs, and fishing vessels (Carlton, 2012), and represent today roughly 35% of the propeller market.

Other types of propulsors are used only in very specific applications. Waterjets are generally installed when propellers cannot be used, particularly for very high speed vessels; cycloidal propellers (Kirsten-Boeing and Voith-Schneider) are generally employed when very high manoeuvrability or station-keeping are required (Molland *et al.*, 2011).

Internal combustion engines

Diesel engines are the most widespread solution for the conversion of chemical to mechanical energy, representing 96% of installed power on board of merchant vessels larger than 100 gross tons (Eyring *et al.*, 2010). The main marine Diesel engines features are (see also Table 2.1)[1]:

Efficiency : Diesel engines can reach up to more than 50% brake efficiency (Woud & Stapersma, 2003).

Load flexibility : Diesel engines allow low-load operations (down to 10% of the maximum continuous rating (MCR) (Laerke, 2012)) with a rather flat efficiency curve.

Fuel flexibility : Low and medium speed Diesel engines allow operations on both residual (HFO and IFO) and distillate fuels (MDO and MGO)) (Woud & Stapersma, 2003). Recent efforts from the main engine manufacturers also allowed operations on alternative fuels, such as natural gas and methanol (Aesoy *et al.*, 2011).

Maintenance : Compared to other prime movers, such as gas turbines, Diesel engines offer more possibilities to be repaired by the crew on board.

Diesel engines can be used both for providing propulsion (in which case they are normally referred to as main engines, ME) and auxiliary power (auxiliary engines, AE). Two stroke engines are generally used only for propulsion, while other engine types are used for different scopes depending on the application.

Gas turbines are today the only alternative to Diesel engines for ship power plants. Despite being less efficient (30-40%), and less flexible with load and fuel quality compared to Diesel engines (Woud & Stapersma, 2003), their main advantage lies in their higher power density. This makes them suitable for applications where high power and low weight are required, as in the case of fast ferries or naval vessels.

[1] For a more detailed description the reader is invited to refer to the extensive literature on the subject, such as the writings of Heywood (1988); Stone (1999); Woud & Stapersma (2003)

Table 2.1: Performance parameters of Diesel engines, state of art 2001 (Woud & Stapersma, 2003)

	Diesel Engines		
	Low-speed	Medium-speed	High-speed
Process	2-stroke	4-stroke	4-stroke
Construction	Crosshead	Trunk piston	Trunk piston
Output power range [kW]	8000 - 80000	500 - 35000	500 - 9000
Output speed range [rpm]	80 - 300	300 - 1000	1000 - 3500
Fuel type	HFO/MDO	HFO/MDO	MDO
SFOC [g/kWh]	160 - 180	170 - 210	200 - 220
Specific mass [kg/kW]	60 - 17	20 - 5	6 - 2.3

2.4 Selected technologies for energy efficiency in shipping

The potential for improving ship energy efficiency in shipping based on technologies available today was estimated to lie between 25% and 75% (Buhaug *et al.*, 2009), even when only cost-effective measures are considered (Eide *et al.*, 2011; Faber *et al.*, 2011).

Reviews such as those presented by Buhaug *et al.* (2009) and Faber *et al.* (2011) generally refer to all type of measures that can potentially reduce fuel consumption: from logistics to improved hull and propeller design. While a complete review of these technologies would be out of the scope of this thesis, the following section focuses on research related to two specific solutions that will be further investigated in this thesis: waste heat recovery (WHR) systems, and hybrid propulsion systems.

2.4.1 Waste heat recovery systems

Waste heat recovery (WHR) systems refer to technical devices designed to make use of the thermal energy that would otherwise be wasted to the environment, a solution which is widely used in various industrial sectors.

A Diesel engine presents four main sources of waste heat (see Table 2.2). The **exhaust gas** are simply released to the atmosphere through the funnel, while waste heat from the **lubricating oil**, **charge air** and **engine walls** needs to be cooled on board.

On most ships, two cooling systems are installed: the **high-temperature** (HT) cooling system, with temperatures ranging between 70 and 90^{o}C, is responsible for cooling the cylinder walls (jacket water cooler, JWC) and part of the charge air flow (charge air cooler (CAC), HT section); the **low-temperature** (LT) cooling system, with temperatures normally ranging between 30 and 50^{o}C, is responsible for cooling the lubricating oil (lubricating oil cooler, LOC) and the remaining part of the charge air flow. LT cooling systems are also responsible for cooling the remaining systems on

Table 2.2: Waste heat from Diesel engines

Source	Temperature [^oC]	Energy share [%]
Exhaust gas	380	25.2
Jacket water cooling	85[a]	5.2
Charge air cooling	210 (85[a], 40[b])	13.7
Lubricating oil cooling	80(40[b])	6.3

Values refer to a four-stroke engine (Wärtsilä, 2007) at 100% load. The share changes at lower load, particularly in the case of the charge air cooling heat losses that decrease more with decreasing load then the rest.
[a] Available temperature at the HT cooling systems
[b] Available temperature at the LT cooling systems

board, such as the gearbox, propeller bearings, etc (Grimmelius *et al.*, 2010).

Heat-to-heat recovery

The recovery of waste heat from the main engines for fulfilling on board heat demand is today common practice. This is generally done by making use of the thermal energy content of the exhaust gas from the main engines, using an **heat recovery steam generator** (HRSG)[1] to generate steam which is then distributed to different users on board, such as HVAC and fuel heating (McCarthy *et al.*, 1990; Bidini *et al.*, 2005). The use of heat as means for ballast water treatment has also been proposed (Balaji *et al.*, 2015).

Heat from the engine cooling water is also often used for fulfilling on board energy demand. On many ships, this is used for freshwater generation using low-pressure evaporators (McCarthy *et al.*, 1990; Marty, 2014). When heat demand is higher, such as in the case of cruise ships, waste heat from the cooling systems can also be used for HVAC systems (Baldi *et al.*, 2015).

Heat-to-power recovery

The amount of waste heat available from the prime movers often exceeds the on board demand for heat, thereby driving engineers and researchers to investigate further opportunities for WHR[2].

[1] HRSG is a term most used in the land-based industry. In shipping it is often frequent to refer to these heat exchangers as exhaust gas economisers, or exhaust gas boilers.

[2] In principle, the expression "waste heat recovery" and the acronym WHR refer to any type of technology used for recovering waste heat. In current scientific literature, however, it is common to use this term to refer particularly to heat-to-power systems. This convention is also applied in this thesis.

One of the most interesting solutions concerns the conversion of waste heat to mechanical power. Although different technologies are available (Shu *et al.*, 2013), **Rankine cycles** have been particularly successful because of their well-known technology, safety, and relatively high efficiency (Tchanche *et al.*, 2011; DNV, 2012). Standard Rankine cycles are based on the generation of high-pressure steam and its subsequent expansion in a turbine, which generates mechanical power.

Steam-based Rankine cycles have been proposed for the application to many ship types: containerships (Dimopoulos *et al.*, 2011, 2012; Yang Min-Hsiung, 2014), ferries (Livanos *et al.*, 2014) and bulk carriers (Theotokatos & Livanos, 2013), referring to the use of both simple and dual-pressure cycles. Single-pressure steam-based Rankine cycles are installed, for instance, on E-class and on Triple-E class Maersk vessels (Maersk, 2014), and ready technical solutions are offered by several engine manufacturers (Mest *et al.*, 2013). The estimated fuel savings vary between different ship types and WHR technologies, ranging between 1% (Theotokatos & Livanos, 2013) and 10% (Dimopoulos *et al.*, 2012).

In some cases the use of steam as a working medium for Rankine cycles is not the most convenient choice. This is mainly due to the fact that:

- At low temperatures of the heat source it is not possible to maintain a sufficiently high evaporating pressure while ensuring the required minimum level of superheating (Invernizzi, 2013).

- The expansion turbine for a steam cycle is normally too expensive for low-power applications. This is due to the high enthalpy drop and low volumetric flow, which makes the design of the turbine particularly challenging (Invernizzi, 2013).

Organic Rankine cycles (ORC) are often used when only low-temperature waste heat (i.e. approximately below 250^oC) is available (Invernizzi, 2013), which makes the more suitable in the case of two-stroke engine; their working process is analogous to that of a steam-driven Rankine cycle, but they make use of different working fluids with more suitable thermodynamic properties.

The need of choosing the working fluid among many potential candidates implies an additional degree of freedom and, therefore, higher expected performance but also a more challenging optimisation process. This made ORCs to become the subject of many studies in scientific literature, with applications to containerships (Larsen *et al.*, 2013; Choi & Kim, 2013), LNG carriers (Soffiato *et al.*, 2014), handy-size tankers (Burel *et al.*, 2013) and passenger vessels (Ahlgren *et al.*, 2015). Grljušić *et al.* (2015) also proposed the application to oil tankers by attempting to integrate the ORC system with on board heat requirements.

The fuel savings related to the installation of ORCs are slightly higher then what estimated for steam-based WHR cycles, especially in the case of two-stroke engines where the temperatures of the available heat sources are lower. For instance, Larsen *et al.* (2015) showed that 10% fuel savings can be achieved on a marine two-stroke engine if an ORC is installed, at design load.

Rankine cycles are not the only way proposed for recovering waste heat on board. Power turbines, driven by the exhaust gas at high engine load, are efficient and have low capital investment, although they are generally connected to lower fuel savings (Dimopoulos *et al.*, 2011; Matsui *et al.*, 2010).

Other WHR technologies

Absorption refrigeration allows the use of heat for chilling purposes (Shu *et al.*, 2013). Although not common, it is sometimes employed on cruise vessels (R718.com, 2012). Finally, thermoelectric generation refers to processes based on the Seedback effect for the direct generation of electricity from a temperature difference without the need of any thermodynamic cycle (Shu *et al.*, 2013; Georgopoulou *et al.*, 2016).

2.4.2 Hybrid propulsion

Although propulsion arrangements based on a hybridisation of mechanical and electric propulsion have been historically commonly installed on some specific ship types, such as naval ships and supply vessels (Woud & Stapersma, 2003), these systems are today also being studied for other vessel types.

The main engines are generally designed for the large propulsion power demand of sailing conditions at design speed. When sailing at low speed or manoeuvring, however, the demand for propulsion power decreases. In a conventional, direct-drive propulsion system (see Figure 2.4a) engines are operated at low load and, consequently, low efficiency.

Hybrid propulsion systems (Figure 2.4c) can be a solution to this issue. By allowing the main engines to be used to generate auxiliary power and the auxiliary engines to contribute to propulsion,s they allow additional flexibility in how the system deals with the generation of both propulsion and auxiliary power and proved to allow savings of 1-2% (Sciberras *et al.*, 2013).

Diesel-Electric systems (Figure 2.4d) can be even more attractive when higher flexibility is required. In Diesel-Electric systems there are no main and auxiliary engines: all the power generated by the prime movers is converted to electricity and further redirected to the different users, including the electrical motors driving the propeller shafts. These systems require however additional effort both in the design phase (Solem *et al.*, 2015) and in the definition of the control strategy (Vučetić *et al.*, 2011; Kanellos *et al.*, 2012).

Finally, the installation of batteries for energy storage has also gained ground as a consequence of the recent improvements in battery technology, showing a potential for savings of up to 28% (Grimmelius & de Vos, 2011; Dedes *et al.*, 2012; Sciberras *et al.*, 2013; Zahedi *et al.*, 2014).

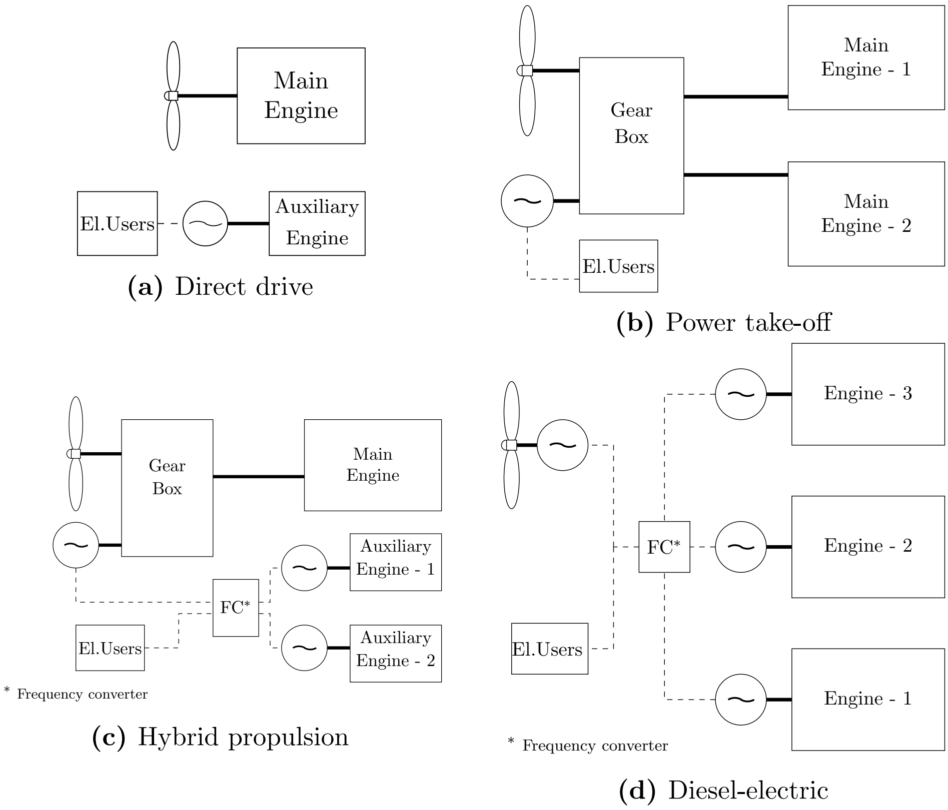


Figure 2.4: Schematic representation of alternative power system configurations

Chapter 3

Theory

Energy systems engineering

Chapter 3 introduces the main principles and tools of energy systems engineering. First, the fundamentals of systems engineering are described (Sec. 3.1). Then, the main tools for energy systems analysis are presented: energy and exergy analysis (Sec. 3.2), and energy systems modelling (Sec. 3.3).

3.1 The energy systems engineering approach

The central focus of this thesis lies on the premise that ships' design and operation, with regards to energy efficiency, can be improved if the subject is approached by considering the ship as a system rather than by concentrating on its individual components.

This type of approach, normally referred to as systems approach, requires however additional effort and resources, while often reducing the focus on each individual part of the system. Its use should therefore be motivated: a systems approach is all about dealing with complexity (Flood & Carson, 1993).

3.1.1 Complexity in ship energy systems

According to Yates (1978), complexity arises when one or more of the following attributes are found:

Significant interactions : The different parts of the entity under study influence each other's behaviour.

High number of parts : The higher number of parts, the more possibilities for the different parts of the system to interact.

Non-linearity : The behaviour of the parts and their interactions cannot be represented by linear mathematical relationships. The influence of non-linearity can be seen intuitively, but is particularly relevant when dealing with models and, in particular, with optimisation (Chang, 2010).

Emergence : The interactions within the different parts are directed towards a common goal; simpler entities exhibit properties and capabilities that the simple entities themselves are not capable of. *Instead of being merely an aggregation of shaped materials, an airplane can fly. Instead of being a blob of cells, we can walk and talk.* (Flood & Carson, 1993).

Asymmetry : The interactions among the parts are not symmetrical.

Nonholonomic constraints : Some of the parts can go, temporarily, outside central control, generating localised, transient anarchy.

It is easy to observe that the energy system of a ship shows at least four of the six features mentioned above. As presented in Chapter 2, a ship is made of a large number of parts interacting with each other (hull, propeller, main engine(s), auxiliary engine(s), auxiliary electric equipment, boilers, etc.); these parts show a non-linear behaviour (e.g. the efficiency of the engine as a function of its power requirement) and operate towards a common goal. Although the degree of complexity varies between ship types, ship energy systems can be classified as complex according to the definition above.

When complexity arises "*a major contributory factor* [to erroneous predictions of systems behavior] *has been the unwitting adoption of piecemeal thinking, which sees only parts and neglects to deal with the whole* "(Flood & Carson, 1993). Inefficient design is often connected to erroneous predictions of system behaviour, which are normally originated by counter-intuitive behaviour. However, referring again to (Flood & Carson, 1993),

> *this* [counter-intuitive behavior] *is not an intrinsic property of phenomena; rather, it is largely caused by our neglect of, or lack or respect being paid to, the nature and complexity that we are trying to represent. That is one reason why we need systems thinking, methodologies, and models. We argue that without this formal thinking we see only parts, the extremes, the simple explanations or solutions.*

3.1.2 From systems to systems engineering

The discipline approaching the engineering design process from a system perspective is normally referred to as systems engineering. Four main traits can be found and are emphasised in most of the available definitions (Blanchard & Fabrycky, 2006):

- The use of a an approach that views the **system as a whole** and that focuses on **interactions** within the system rather than on its individual components.
- A long-sighted approach that puts significant emphasis on **systems operations** and not only on the design.

- A detailed description of the **requirements** from the system.
- An **interdisciplinary** approach.

In this thesis, only the first three aspects of systems engineering are retained. The focus being on the energy part of the system, the approach employed in this work can be referred to as **energy systems engineering** (Vanek *et al.*, 2012).

3.1.3 Ship energy efficiency from a systems perspective

This work aims at contributing to the field of energy efficiency in shipping by applying a systems perspective. Although not as widely as in other fields, and often not explicitely in relation to systems engineering, other authors have published on this subject in the past. This is particularly true for ship energy and exergy analysis, and for studies that broadened the perspective of ship design by enlarging the boundaries of the system of interest and by taking a broader range of operational conditions into account.

Ship energy analysis

As introduced in Section 3.2, the work published to date concerning ship energy and exergy analysis can be broadly divided in two main category: studies based on a data-driven approach, and employing a model-based one.

The former approach is employed in two main studies: Thomas *et al.* (2010) and Basurko *et al.* (2013), both proposing the energy audit of fishing vessels. The results suggest that, for the selected case studies, propulsion represents a major part of the total on board energy consumption (76% in the case analysed by Thomas *et al.* (2010), 84% to 88% in the cases presented by Basurko *et al.* (2013)). In the case presented by Thomas *et al.* (2010), however, fishing equipment (14%) and lighting (6%) also showed to be relevant for the overall energy budget. None of the two aforementioned studies, however, touches the subject of thermal energy demand.

Marty *et al.* (2012); Marty (2014) proposed instead the application of model-based energy and exergy analysis. The results of his work confirmed that cruise ships a more varied energy demand compared to other ship types. Although the energy demand shares depend on each individual case, Marty (2014) estimated a share of approximately 40%-30%-30% for propulsion, auxiliary electric power and auxiliary heat for a cruise ship during sailing.

Interactions within the system

Although not common, more than one author accounted for interactions between different part of the systems in their analysis. The most notable examples come from two fields: WHR systems and hybrid propulsion.

In the case of WHR, the characteristics of the prime mover can be subject to modifications aiming at improving the performance of the whole system. Modifications to the turbocharger can influence the efficiency of the full power plant (in the case proposed by Dimopoulos *et al.* (2012) this allowed reducing the estimated payback time from 8 to 4 years). Similarly, the fine-tuning of engine injection and valve timing to optimise the efficiency of the combined engine-WHR system showed that up to 1.0% improvements in the overall efficiency can be achieved compared to optimising the components individually Larsen *et al.* (2015).

More in general, the larger the boundaries of the system of interest, the higher the expected improvement. This is mostly true for particularly complex systems, such as combined cycles (Dimopoulos & Frangopoulos, 2008) and Diesel-electric propulsion systems (Solem *et al.*, 2015; Zahedi *et al.*, 2014; Dedes *et al.*, 2012).

An appropriate understanding of system interactions is of utmost importance when the field of control systems is involved. In the case of hybrid and Diesel-electric propulsion systems, the issue of system control is not trivial and requires an additional effort in understanding how to operate all components for optimal efficiency (Grimmelius & de Vos, 2011; Dedes *et al.*, 2012; Sciberras *et al.*, 2013; Zahedi *et al.*, 2014; Vučetić *et al.*, 2011; Kanellos *et al.*, 2012).

Design for operational conditions

When a new solution for energy efficiency is proposed or optimised, a reference case is generally proposed as an example of the behaviour of the specific application, or to showcase the proposed method. Many times, however, the system under study is only evaluated at one operational condition, which most often only partly represents ship operations.

Some authors have taken into account a reference voyage, rather than a single operational point (Dedes *et al.*, 2012; Choi & Kim, 2013). Although constituting an improvement with respect to design-point evaluations, this approach misses to take into account the variability of the voyage pattern of a vessel in terms of speed, draft, weather encountered, time spent in port, etc. More in general, a correct evaluation of a proposed design should be performed on an operational profile representative of real ship operations (Ahlgren *et al.*, 2015), as these are generally substantially different from design conditions (Coraddu *et al.*, 2014).

In a design process, a correct accounting of the expected range and distribution of system operations can make the difference between a success and a failure (Gaspar *et al.*, 2010; Motley *et al.*, 2012). Kalikatzarakis & Frangopoulos (2014) showed that depending on the assumed operational profile, the net present value of the proposed WHR system after 20 years could vary by as much as 50%.

3.2 Energy and exergy analysis

The correct understanding of the requirements of a system constitutes one of the main building blocks of the systems engineering approach. In the case of energy systems, this demands for a detailed, systematic analysis of the system's energy performance. Apart from standard data analysis tools that can be used for dealing with typical marine engineering variables of interest, two additional tools used in this thess is: energy and exergy analysis.

3.2.1 Energy analysis

Energy analysis is based on the 1^{st} law of thermodynamics, which can be read as *Energy cannot be created nor destroyed.* The energy balance of a given component can be written as follows:

$$\frac{dU}{dt} = \dot{Q} - \dot{W} + \sum_i \dot{m}_{in,i} \left(h_{in,i} + \frac{1}{2} v_{in,i}^2 + g z_{in,i} \right) - \sum_j \dot{m}_{out,j} \left(h_{out,j} + \frac{1}{2} v_{out,j}^2 + g z_{out,j} \right) \tag{3.1}$$

where U, Q, W, m, h, v, g and z represent internal energy, heat, work, mass, specific enthalpy, fluid velocity, gravitational acceleration and altitude, respectively.

From an energy analysis perspective, the *energy efficiency* of a component is broadly defined as (Patterson, 1996):

$$\eta = \frac{\Delta H_{out}}{\Delta H_{in}} \tag{3.2}$$

where ΔH_{out} and ΔH_{in} represent the totality of the **useful** energy output and of the energy input to the system, respectively. Examples of the useful output of a system are the mechanical power (in the case of a Diesel engine) or the enthalpy content of a steam flow (for a boiler).

Energy analysis is generally done on either a data-driven or a model-based approach. According to a **data-driven** approach, the performance of a system is evaluated starting from measurements of relevant quantities on board. On the other hand, in **model-based** the majority of the data required in the energy analysis is generated using mathematical models of the investigated system.

3.2.2 Exergy analysis

Exergy is a thermodynamic quantity which allows combining considerations of energy quantity and quality, and is defined as "*the maximum shaft work that can be done by the composite of the system and a specified reference environment* " (Dincer & Rosen, 2013). For this reason exergy analysis is often integrated with energy analysis to get a better understanding of the system, and in particular for (Dincer & Rosen, 2013):

- Combining and applying the conservation of mass and energy and the second law of thermodynamics.

- Revealing whether or not and by how much it is possible to design more efficient systems by reducing the inefficiencies in existing systems.

Box 3.1: The quality of thermal energy

Energy analysis is based on the assessment of energy quantities, where all forms of energy are treated at the same level. This assumption is valid for most of energy forms. Given a certain amount of electric energy, this can be converted with almost 100% efficiency to any other form: using an electric motor (conversion to mechanical energy), or a resistance (to thermal energy), etc.
Thermal energy is different from other energy forms. This is a consequence of the fact that, in contrast to mechanical and electrical energy, thermal energy results from a disorganised motion of particles (Atkins, 1994).
The conversion from disorganised to organised movement does not happen "for free". As stated in the 2^{nd} law of thermodynamics, a given amount of thermal energy cannot be converted to an equal amount of mechanical energy. The efficiency of the conversion depends on several variables, where the temperature at which the thermal engine receives the heat, and that at which the heat is rejected, are the most important.
These observations have a number of practical consequences:

- Waste heat cannot be entirely converted into work. In fact, only a relatively small portion of the heat released by an engine to the environment can be converted to mechanical or electric power, even when assuming ideal conversion machines.

- Not all sources of waste heat on board of a ship are of equal importance. The energy in the exhaust gas, which (depending on the engine type) is released at between 200 and 400^{o}C is of higher quality than that contained in the cylinder cooling water (90^{o}C) or in the charge air (up to 200^{o}C at full engine load).

- The recovery of waste heat on board can be a particularly challenging process if the objective is to harvest it in the most efficient way. Using high-temperature exhaust gas to generate 8 bar steam corresponds to an inefficient use of the original energy flow and to a loss of energy quality, as the same result could have been achieved with a heat source at lower temperature. The same process occurs when 8 bar steam is used to heat fuel oil to 70^{o}C in the storage tanks.

- Analysing ship energy efficiency based solely on energy quantity can be misleading. A ship might recover all of its waste energy for heating purposes, which would appear efficient from an energy perspective. However, full recovering all available waste heat does not necessarily imply that this is done efficiently. This is the domain where exergy analysis demonstrates the greatest potential for identifying the inefficiencies of thermomechanical systems.

Electric, kinetic and potential exergy quantities coincide with their energy counterparts. The **physical exergy** content of a flow instead can be calculated as follows:

$$\dot{B}_{ph} = \dot{m}[(h - h_0) + T_0(s - s_0)] \tag{3.3}$$

where $\dot{B}$, h, and s respectively stand for exergy flow, specific enthalpy, and specific entropy, while the subscript 0 refers to the conditions of the reference environment.

Similarly, the exergy counterpart of a heat flow at a given temperature can be calculated as:

$$\dot{B}_{heat} = \dot{Q}[1 - \frac{T_0}{T}] \tag{3.4}$$

where T represents the temperature at which the heat is transferred.

Differently from energy, exergy is not conserved. Any non-reversible process involves a loss of exergy. This contribution to the exergy balance, generally known as **irreversibility rate**, is calculated as:

$$\dot{I} = T_0\dot{S}_{gen} \tag{3.5}$$

where $\dot{S}_{gen}$ stands for the entropy generation rate in the component.

The fact that exergy is not conserved leads to the fact that a large amount of alternative performance indicators can be defined, and to date there is not a complete agreement in the scientific community concerning which ones should be used when performing an exergy analysis (Lior & Zhang, 2007). A list of the performance indicators used in this thesis is provided in Table 3.1[1].

Table 3.1: Summary of the exergy-based performance indicators employed in this work

Name	Defining equation	Function
Total exergy efficiency (ϵ_t)	$\frac{\sum \dot{B}_{out,i}}{\sum \dot{B}_{in,i}}$	Measures what fraction of the exergy input to the component is not destroyed
Task efficiency (ϵ_u)	$\frac{\sum \dot{W}_{u,i} - \sum \dot{W}_{p,i} + \sum \dot{B}_{h,u,i} + \sum \dot{B}_{c,u,i}}{\sum \dot{B}_{h,p,i} + \sum \dot{B}_{c,p,i} + \sum \dot{B}_{ch,p,i}}$	Measures the ability of the component to generate useful output
Efficiency loss ratio (δ)	$\frac{\dot{I}}{\sum \dot{B}_{in,i}}$	Measures what fraction of the exergy input to the component is destroyed
Relative irreversibility (γ)	$\frac{\dot{I}}{\sum \dot{I}_j}$	Measures the contribution of the component to the total exergy destruction of the system

[1] A detailed review of exergy-based performance indicators can be found in dedicated literature (Kotas, 1980; Lior & Zhang, 2007).

3.3 Energy systems modelling

When applying the principles of systems engineering, tools are required for being able to correctly estimate how the engineering system will perform given different operational conditions, and on how these conditions will influence the internal processes. The process of modelling refers to the act of constructing a tool for reproducing or imitating the behaviour of a real system, which is easier to study than the system itself (Kramer & de Smit, 1977).

3.3.1 Introduction to mathematical modelling

The act of modelling can refer to many different types of actions, from verbal modelling (describing the behaviour of a system in words) to physical modelling (building a physical reproduction of the system, generally in smaller scale, to perform tests). This work focuses on **mathematical models**, where the relationships between entities in the model are represented in mathematical terms (Kramer & de Smit, 1977), and in particular on models with a **predictive** purpose, i.e. that are meant to be able to simulate the behaviour of the system under varying conditions (Flood & Carson, 1993).

Mathematical models can be further subdivided in different categories depending on their defining aspects[1].

Mechanistic (often referred to also as **white-box**) models attempt to describe the physical phenomena that characterise a system by making use of physical laws (e.g. conservation of mass and energy) or semi-empirical equations (e.g. heat transfer correlations) (Duarte *et al.*, 2004). In contrast, **empirical** (also known as **black-box**) models are trained on observed data to predict the output of a system given the input (Duarte *et al.*, 2004).

Empirical models do not require any knowledge of the underlying system's physics, and are often more accurate compared to mechanistic models. However, not only they require large datasets for model training, but they also generally perform poorly when extrapolating outside of the training dataset (Duarte *et al.*, 2004).

An additional categorisation is based on how the model treats time as an internal variable. Depending on whether the time domain is included among the modelling independent variables or not, a model is called **steady-state** or **dynamic**. Steady-state models are generally easier to solve and are preferred when there is no interest in the dynamic component of the system.

Finally, a model that, given a certain input, generates one and only one possible output is called **deterministic**. **Stochastic** models instead can deal with uncertainty and are normally used in processes, such as robust optimisation, where the focus lies not only in finding one optimal solution, but also in limiting the effect of uncontrollable variations to the system's inputs and its behaviour (Sahinidis, 2004).

[1] This categorisation is a personal adaptation based on Grimmelius (2003)

3.3.2 Energy systems modelling in shipping

Computational models are extensively used for application to ship energy systems, and propulsion systems in particular, as already exemplified in early work in the field (DeTolla & Fleming, 1984; Neilson & Tarbet, 1997; Depuis & Neilson, 1997)[1].

Models of ship energy systems are generally used for three main purpose: for the control of existing systems, for the evaluation of new designs or retrofitting options, and for optimisation. Although each model is different depending on the individual study, models used in the framework to which this thesis aims to contribute are generally **mechanistic** and **deterministic**.

System control

Models used for **control** purposes are subjected by the intrinsic requirement of being dynamic. Most models proposed in academic literature in this field relate to the control of relatively complex systems, where the task of optimising the control strategy is more challenging. This is the case for instance of Diesel-electric power plants, where the total electric load needs to be allocated to different prime movers (Kanellos *et al.*, 2012), and to systems equipped with batteries (Grimmelius & de Vos, 2011; Han *et al.*, 2014), where the optimal strategy for battery charge and discharge needs to be defined. Finally, Grimmelius & Stapersma (2001) also provide an example of the use of computational models for determining the impact of the control of the propulsion plant on the thermal loading of the engine.

Prediction for system design

Mathematical models have been extensively applied to the prediction of the performance of a given design (or retrofitting) and, therefore, to its evaluation.

Many of the proposed are used to predict the performance of the system in terms of energy efficiency and fuel consumption. In these regards, it is often assumed that for many ship types the influence of ship dynamics on fuel consumption is marginal and, therefore, focus on the steady-state performance of the system[2].

Some authors presented different modelling strategies without focusing on specific uses. While Shi & Grimmelius (2010) and Theotokatos & Tzelepis (2015) focused on the ship's propulsion system, other authors leaned towards a more holistic perspective. Calleya *et al.* (2015), Cichowicz *et al.* (2015) and Tillig *et al.* (2015) proposed general, holistic modelling framework for the simulation of the performance of the ship in different operational conditions and for evaluation of different energy saving technologies; these models focused on the hydrodynamic part of the ship, while Zou *et al.* (2013)

[1]For other examples of reviews in the literature of energy systems modelling the reader is referred to the works of Tillig *et al.* (2015); Ginnetti (2014).

[2]It should be noted that, although the models presented in these papers are mostly used for predicting the performance of the system in steady-state conditions, they are often dynamic models. Most models are based on intrinsically dynamic modelling platforms, such as Simulink, Simscape and Modelica.

and Lepistö *et al.* (2016) put the emphasis on thermal energy flows on board. Pedersen & Pedersen (2012) proposed the use of bond-graph modelling for ship energy systems, and particularly for the application to Diesel-electric systems.

Other authors proposed the use of mathematical models for the evaluation of specific design solutions. Viola *et al.* (2015) focused on the design of wind-assisted propulsion; Zahedi *et al.* (2014) proposed the use of DC hybrid power systems for Diesel-electric ships, and evaluated their performance against more standard AC systems; Livanos *et al.* (2014) evaluated various propulsion systems for LNG-powered ferries, also included WHR systems in the picture, while Burel *et al.* (2013) focused on handymax tankers; Dedes *et al.* (2012) and Sciberras *et al.* (2013) attempted to asses the potential for fuel savings of hybrid propulsion systems.

Dealing with the propulsion system, dynamic models are often used for the prediction of ship performance during manoeuvring or, in general, to simulate the behaviour of the ship systems during transients (acceleration, crush-stop, turns) (Campora & Figari, 2003; Benvenuto & Figari, 2011; Theotokatos, 2008; Schulten, 2012).

Box 3.2: Black-box and stochastic modelling in shipping

Although the focus of this thesis lies on mechanistic and deterministic models, examples of the use of alternative modelling strategies can be found in academic literature.

In the latest years, the use of **black-box models** has been increasing as a consequence of the growing availability of measured data from ship operations. In particular, artificial neural networks (Petersen *et al.*, 2012a; Shi & Grimmelius, 2010), Gaussian processes (Petersen *et al.*, 2012b), regularised least squares, Lasso regression, and random forest methods (Coraddu *et al.*, 2015) have been tested, and compared to white box models. In presence of sufficiently extensive measurements of ship operations, black-box models are more reliable than white-box models in the accuracy of the predictions (Leifsson *et al.*, 2008).

The use of hybrid (gray-box) models allows achieving an accuracy comparable to that of a black-box model while requiring a lower amount of measurements and improving the performance of the model for extrapolation (Coraddu *et al.*, 2015; Leifsson *et al.*, 2008).

Although most models presented so far are deterministic, there are few examples of including **uncertainty** in the discussion. Kalikatzarakis & Frangopoulos (2014); Coraddu *et al.* (2014), for instance, proposed a sensitivity analysis, where the influence of varying operational parameter on the efficiency of the design was evaluated. Vrijdag *et al.* (2007) proposed instead an uncertainty analysis, mostly accounting for the uncertainty in model parameters and inputs. Stochastic optimisation in ship design has only been introduced in relation to ship hydrodynamics, and in particular on the choice of the ship's main dimensions Hannapel & Vlahopoulos (2010); Diez & Peri (2010).

Optimisation

The models presented in the previous section are used for aiding the designer in evaluating a pre-determined design. Models can however also be used at a even higher level of the design process: in the field of design optimisation, parts of the design choices are delegated to an optimisation procedure that helps the designer in the identification of the set of parameters or system configuration that, according to the output of the model, shows the most optimal performance.

Optimisation in ship design has been applied extensively to the choice of the ship main dimensions (among others, Ölçer (2008)), to the configuration of the power plant (Dimopoulos & Frangopoulos, 2008; Dimopoulos *et al.*, 2008; Solem *et al.*, 2015) and to the design of retrofitting options, particularly for WHR systems (Dimopoulos *et al.*, 2011; Larsen *et al.*, 2013).

Optimisation generally requires the system to be simulated a large number of times, which leads to models used for this purpose being less computational intensive. Models used for system optimisation are steady-state; the use of linear models, although not common, has also been proposed (Solem *et al.*, 2015).

3.3.3 Modelling of individual components

The choice of the modelling detail goes hand in hand with considerations related to modelling accuracy and computational time based on the requirements of the problem to be solved. In this section, the available choices for modelling the main parts of the ship energy systems are reviewed.

Propellers

Mechanistic modelling of propeller performance can be performed in three, main ways (Molland *et al.*, 2011):

Performance maps : Performance maps are generally provided by the propeller manufacturer and provide a graphical relation between the main variables of the propeller (e.g. adimensional thrust and torque, and efficiency), valid for one specific propeller model.

Standard series : Propeller series have been systematically analysed in order to derive relatively simple models for the prediction of propeller performance. The Wageningen series propellers are largely the most known and employed in scientific literature (Oosterveld & Van Oossanen, 1975), although models of several other series have been developed (Molland *et al.*, 2011).

Theory-based models : Different theories have been developed over the years for modelling propellers and their interaction with the water flow. These types of models are generally rather computationally expensive and rarely used in energy systems models.

When available, performance maps are preferred as they are easy to use and provide accurate predictions. When a performance map is not available, standard series, and in particular the Wageningen series, are by far the most employed in academic literature about modelling of ship propulsion systems (see Table 3.2).

Diesel engines

Modelling the Diesel engine can require different effort depending on the specific problem under investigation:

Empirical models represent the relationship between engine main operative variables (typical outputs are efficiency, exhaust temperature and mass flow, waste heat to cooling systems) using empirical input-output relations. In the simplest case, these are defined as polynomial functions of the engine load alone (e.g. in Kanellos *et al.* (2012); Calleya *et al.* (2015)) or of load and speed (Marty, 2014). These functions can be based on the engine's technical documentation or on experimental data. Performance maps, such as those described in the case of propellers, can also be provided by engine manufacturers. More complex models, such as those based on artificial neural networks, have also been employed (Grimmelius *et al.*, 2007).

Mean value engine models (MVEM) are based on the assumption that engine processes can be approximated as a continuous flow through the engine, and hence average engine performance over the whole operating cycle (Theotokatos, 2008; Dimopoulos *et al.*, 2011).

Zero-dimensional engine models (0DEM) models operate per crank-angle basis by solving the mass and energy conservation equations, along with the gas state equation, in their differential form. Combustion is modelled by using phenomenological models of either one or multi zones, where the latter are favoured when a more detailed representation of the combustion process and the prediction of exhaust gas emissions are needed (Scappin *et al.*, 2012).

CFD engine models are based on principles of fluid dynamics and feature the inherent ability of providing detailed geometric information on in-cylinder mass and energy flows by solving the governing flow equations.

As shown in Table 3.2, different authors have employed different types of models for simulating engine behaviour in ship energy system models. Empirical models, MVEMs and 0DEMs are all employed, while CFD models are more common for research in specific combustion-related topics and when accurate predictions of pollutant emissions (particularly NO_x and PM) are required.

Electric machinery

The modelling choices related to the electric machinery on board varies depending on the type of energy system analysed and on the scope of the work.

Table 3.2: A review of the modelling choices in scientific literature on ship propulsion systems modelling

	Type	Propeller (K_T, K_Q)	Main engines ($\dot{m}_{fuel}$)
Benvenuto & Figari (2011)	Dyn	Map ($J, P/D$)	0DEM
Campora & Figari (2003)	Dyn	Map ($J, P/D$)	0DEM
Pedersen & Pedersen (2012)	Dyn	StSe	EM ($\dot{W}_{ME}$)
Schulten (2012)	Dyn	Map ($J, P/D$)	MVEM
Theotokatos (2008)	Dyn	StSe	MVEM
Grimmelius *et al.* (2010)	Con	StSe	EM ($\dot{W}_{ME}$)
Larroudé *et al.* (2013)	Con	$P_2(J)$	EM ($\dot{W}_{ME}$)
Kanellos *et al.* (2012)	Con	-	EM ($\dot{W}_{ME}$)
Shi & Grimmelius (2010)	Mod	StSe	EM ($\dot{W}_{ME}, \omega_{ME}$)
Theotokatos & Tzelepis (2015)	Mod	StSe	MVEM
Cichowicz *et al.* (2015)	Mod	StSe	MVEM
Coraddu *et al.* (2014)	Mod	TB	EM ($\dot{W}_{ME}, \omega_{ME}$)
Calleya *et al.* (2015)	Des	StSe	EM ($\dot{W}_{ME}$)
Liu & Fan (2010)	Opt	StSe	EM ($\dot{W}_{ME}$)

Abbreviation	Model type
Dyn	Dynamic
Con	Control
Mod	General models
Des	Design evaluation
Opt	Optimisation
Map	Performance map
StSe	Standard series (e.g. Wageningen)
TB	Theory-based methods
EM	Empirical model
0DEM	Zero-dimensional model
MVEM	Mean value engine model

When dealing with "traditional" propulsion systems, where power demand for propulsion and for electric auxiliaries are provided by different systems, auxiliary generators are often neglected (Theotokatos & Livanos, 2013).

The modelling of hybrid or Diesel electric systems does not allow neglecting the influence of electric machinery, as this would lead to overestimating the performance of the system. In order to take this aspect into account it can be sufficient to model the electric components with **constant efficiencies**, as done, among others, by Dedes *et al.* (2012). Although electric machines generally have flat efficiency curves, their efficiency drops at very low load; this can be taken into account using **empirical correlations** (see McCarthy *et al.* (1990)).

Many authors, however, favour a more detailed modelling of the electric machinery, both for including the influence of these components in terms of system control (Kanellos *et al.*, 2012) and for improving the accuracy of the prediction of energy losses (Zahedi *et al.*, 2014). The use of the standard d-q (direct and quadrature axes) equations (Sciberras *et al.*, 2013) is a typical example of a more advanced modelling of on board electric machinery.

Waste heat recovery systems

As most of the work published in the literature related to the application of waste heat recovery systems (and, particularly, of Rankine cycles) to ships is focused on the estimation of the performance of the system in different conditions and on its optimisation, WHR systems are always modelled based on a component-by-component principle.

Some of the presented work, in fact, focuses on the **working cycle** without a specific modelling of the individual components. In these cases the standard principle lies in fixing a value for the pressure of the working fluid and of the minimum temperature difference in the heat exchangers (pinch point), which define the main features of the thermodynamic cycle (Larsen *et al.*, 2013; Livanos *et al.*, 2014). Once the thermodynamic cycle has been identified, the features of the heat exchangers (UA value) can be determined, while the performance of the expansion turbine and of the pump are normally determined using their isoentropic (Choi & Kim, 2013) or politropic (Larsen *et al.*, 2013) efficiencies.

The requirements in terms of model assumptions become more complex once the design parameters are identified, and the off-design performance of the system is to be evaluated. Larsen *et al.* (2015) and Dimopoulos & Kakalis (2010) provide some examples of how to determine the part-load performance of heat exchangers and expanders[1].

[1] It should be noted that the available literature on WHR systems based on Rankine cycles is significantly wider than what published in the field of shipping. For the interested reader, the work of Quoilin (2011) provides very good guidance in these regards.

Chapter 4

Methodology

Case studies, data collection, and modelling choices

Chapter 4 presents the methodology employed in this work. It includes a summary of the methodological approach (Sec. 4.1), a description of the case studies (Sec. 4.2) and information on the availability and quality of the data that could be gathered for the two case study vessels (Sec. 4.3). Finally, the main assumptions employed in each of the studies that build up this thesis are summarised (Sec. 4.4).

4.1 Methodological approach

The central focus of this thesis is to **apply principles of energy systems engineering to the analysis and improvement of ship on board energy systems**. This general aim is subdivided into two, main objectives:

- To systematically **analyse** the performance of on board ship energy systems.
- To propose the **synthesis** of solutions for improving ship energy efficiency and to evaluate their potential energy savings.

In this thesis, the proposed themes were addressed by focusing on two case studies. In both cases, operational measurements and technical documentation were used to analyse the performance of the system. Based on the results of this initial analysis, potential improvements to the systems were proposed and evaluated. In both phases, computational models were used to improve the understanding of the system and to predict its behaviour.

4.1.1 Analysis

The first objective of this thesis relates to the analysis of the existing systems.

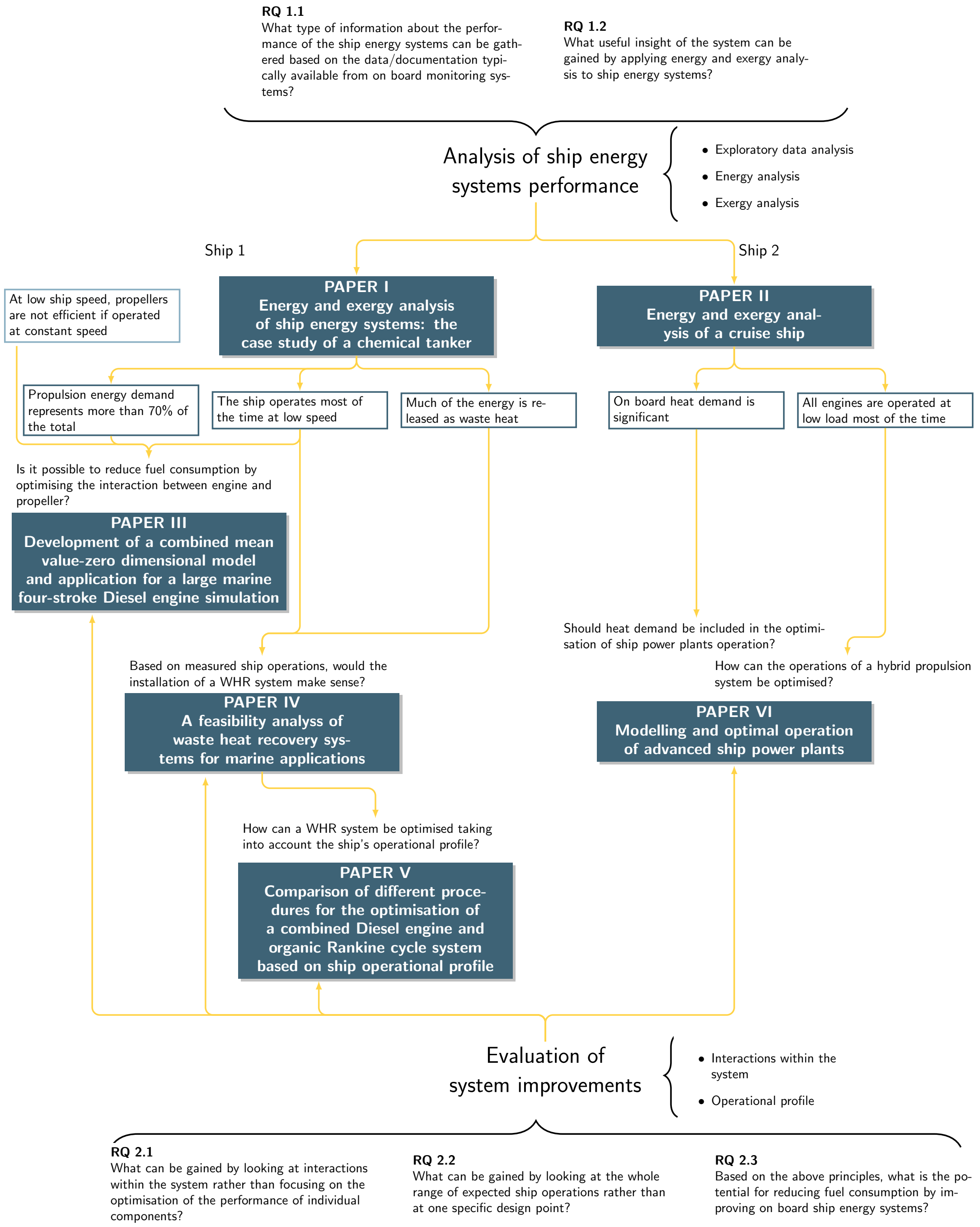


Figure 4.1: Overview of the methodology (1)

In order to approach this subject, the work of this thesis started from analysing the information available for the two case study vessels (both from monitoring systems and from technical documentation, as detailed in Section 4.3), and using it to gain an insight about the related energy systems.

The analysis of the available data was divided in two main parts:

Preliminary analysis (also referred to as *exploratory data analysis*), with the aim of getting a broad view of what type of data are available, and what can be understood about the operations of the vessel by a simple, structured observation of the data (Tukey, 1977). This phase included, for instance, understanding the typical operational profile of the ship in terms of speed, engine loads, power demands, etc.

Energy and Exergy analysis , with the aim of applying a more structured and systematic analysis of the ships' systems with the focus on their energy performance. This phase included the estimation of, among others, energy and exergy flows and efficiencies for the different parts of the ship.

The work related to this part of the thesis is the main focus of Paper I (in relation to Ship-1) and Paper II (Ship-2).

4.1.2 Synthesis

Starting from the insight gained in the previous part, the second objective of this thesis moves from the analysis of the existing systems to the synthesis and evaluation of ways to improve the energy efficiency of these systems. More specifically, this led to three applications:

- Engine/propeller interaction (Paper III)
- Waste heat recovery (Paper IV and Paper V)
- Ship power plant operational optimisation (Paper VI)

4.1.3 System boundaries and modelling

As a general principle, this thesis focuses on the **ship's power plant** as the main system of interest. This puts an ideal boundary of the system on the propeller shaft, on the switchboard, and on the steam pipes. The parts of the ship that are excluded from the main system of interest (propeller and hull, individual electric and thermal power consumers) are considered as power demands to the ship power plant. The choice of excluding the propeller from the main system of interest was challenged in Paper III, where the focus lies on the interaction between the engine and the propeller.

The models employed in this thesis depend on the specific aim of each of the Papers, and are further described in Section 4.4. As a general principle, the model employed in the first two Papers of this thesis are **descriptive**, as they are used for processing

Table 4.1: Summary of the level of detail in the modelling for Papers III to VI

	III	IV	V	VI
Propulsion	Op.Prof.	Op.Prof.	Op.Prof.	Op.Prof.
Aux. electric	Const.	Op.Prof.	Op.Prof.	Op.Prof.
Aux. heat	Const.	Op.Prof.	Not Incl.	Op.Prof.
Main engines	NonLin(M)	NonLin(E)	NonLin(E)	NonLin(M)
Auxiliary engines	Lin	Lin	Lin	NonLin(M)
Propeller	NonLin(E)	Not Incl.	Not Incl.	Not Incl.
Auxiliary boilers	Not Incl.	Lin	Not Incl.	NonLin(E)
WHR system	Not Incl.	Lin	NonLin(M)	NotIncl.

Op.Prof.: Operational profile
Const.: Constant demand
Not Incl.: Not included
Lin: Linear modelling (i.e. constant efficiency)
NonLin(E): Non-linear modelling , empirical
NonLin(M): Non-linear modelling , mechanistic

the measurements from ship operations, while the models used in Papers III to VI are **predictive**, as they are used to estimate the behaviour of the system given a set of operational conditions.

Furthermore, all models in this thesis are **steady-state**, and it was assumed that dynamic effects do not significantly affect the results of this work. All models are also **deterministic**, i.e. uncertainty in both model accuracy and inputs is not taken into account. Finally, the thesis makes use of a mixture of both **mechanistic** and **empirical** models, depending on the required accuracy, on the computational demands and on the available information on the system.

Table 4.1 summarises the main choices in terms of system boundaries and modelling detail for each of the parts of this thesis. The modelling choices and assumptions are then presented more in detail in the following sections, and in the respective papers.

4.2 Case studies

In this thesis the research questions were approached by looking at two case study vessels: a chemical tanker and a passenger vessel. These two vessels were selected mainly based on the availability of measured data and of technical documentation. In the case of Ship-2, the additional complexity of a system with high requirement of both mechanical, electric and thermal energy constituted a rationale for the choice of the vessel as case study.

4.2.1 Ship-1 (M/T Tambourin): A chemical/product tanker

The first case study (from now on referred to as Ship-1) is a handy-max tanker used for the transportation of different types of liquid bulk cargo, such as oil products (kerosene, gasoline, etc.), molasses, vegetable oils, etc. The ship is 183 m long and 32.2 m wide, with a maximum draft of 12.7 m, for a total cargo capacity of 53000 m^3.

The power plant of Ship-1 consists of two four-stroke main engines connected to a common gearbox (GB), which provides power to both the propeller and a shaft generator (S/G). Auxiliary power is also provided by two auxiliary engines, while heat demand is fulfilled by two exhaust boilers recovering energy from the exhaust gas of the main engines, and two auxiliary, oil fired boilers (see Table 4.2 and Figure 4.2)

For both electric power and heat, most auxiliary consumers are the same that can typically be found on most merchant ships. Special systems connected to the ship mission are the following:

Inert gas production and compression: Nitrogen needs to be produced on board and pumped into cargo tanks when flammable liquids are transported. Nitrogen compressors have a high power demand (4 compressors rated 285 kW each) but are only operated intermittently.

Cargo pumping: When unloading the vessel, cargo pumps are required (high pressure in the shore-based tanks is normally sufficient for cargo loading). They can require a large amount of power when operated simultaneously (11 pumps for a total rated power of 1310 kW).

Tank cleaning: After one cargo has been unloaded, tank cleaning is generally necessary in order to prepare the cargo tanks for the following shipment. This operation is performed either directly in port or during ballast trips, and requires a large amount of heat for a short time.

Cargo heating: Some specific liquids are characterized by very high viscosity at ambient temperature, which makes them unsuitable for handling. For this reason, cargo heating can be ensured by means of process steam. This operation is, however, very seldom required.

4.2.2 Ship-2 (M/S Birka Stockholm): A passenger ship

The second case study ship (Ship-2) is a passenger vessel that operates daily tours in the Baltic Sea between Stockholm and Mariehamn on the Åland islands. The ship is 176.9 m long and 28.6 m wide and can accommodate up to 1800 passengers and entertain them with restaurants, night clubs and bars, as well as saunas and pools. Worth of mention, Ship-2 was built to fulfil the Det Norske Veritas' "Clean Design" rule relating to environmentally friendly design solutions (DNV, 2004).

According to its daily schedule, the ship leaves at around 6 PM from Stockholm and sails at reduced speed in the Stockholm archipelago until it reaches the open sea,

Table 4.2: Main components number and sizes of the two case studies

Ship 1			Ship 2		
Component	N	Size [kW]	Component	N	Size [kW]
Main engine	2	3840	Main engine	4	5850
Auxiliary engine	2	682	Auxiliary engine	4	2760
Shaft generator	1	3200	HRSG (ME)	2	1500
HRSG	2	390	HRSG (AE)	4	700
Auxiliary boiler	2	7600	Auxiliary boiler	2	4700

where it stops for the night; early in the morning, the ship starts sailing again and arrives in Mariehamn at around 7 AM. The ship then leaves Mariehamn at around 9 AM and arrives back to Stockholm at around 4 PM (see Figure 4.3).

The propulsion system consists of two propulsion lines composed of two main engines, a gearbox, and a propeller each (see Table 4.2 and Fig. 4.4). The MEs are four Wärtsilä 4-stroke Diesel engines rated 5850 kW each.

On board electrical power demand is fulfilled by the four Wärtsilä AEs, rated 2760 kW each. Electrical power is needed on board for a number of alternative functions, from pumps in the engine room to lights, restaurants, ventilation and entertainment for the passengers.

All AEs and one ME for each propulsion line (i.e. six engines in total) are equipped with HRSGs, which allow covering a large part of on board thermal power demand; in addition, the HT cooling systems of all engines are connected to a heat recovery system based on pressurised water which allows using the waste heat for the pre- and re-heater in the air treatment unit of the HVAC system and for water heating; finally, when thermal power demand is higher than the recoverable waste heat, two auxiliary boilers are used.

All engines are equipped with SCRs for NO_X emissions abatement. Although the Baltic Sea is only subject to TierII limits on NO_X emissions, the ship enjoys up to a 10% reduced harbour fees in Stockholm if these emissions are reduced below a certain level.

4.3 Data collection

4.3.1 Data sources

In this work, data collected from on board measurements and from available technical documentation were used for the analysis. The work included the collection of already existing datasets and other types of useful information, and did not involve additional measurements performed in situ. This part of the study therefore falls under the category of *observational studies*, i.e. *conducted on existing data that typically had been*

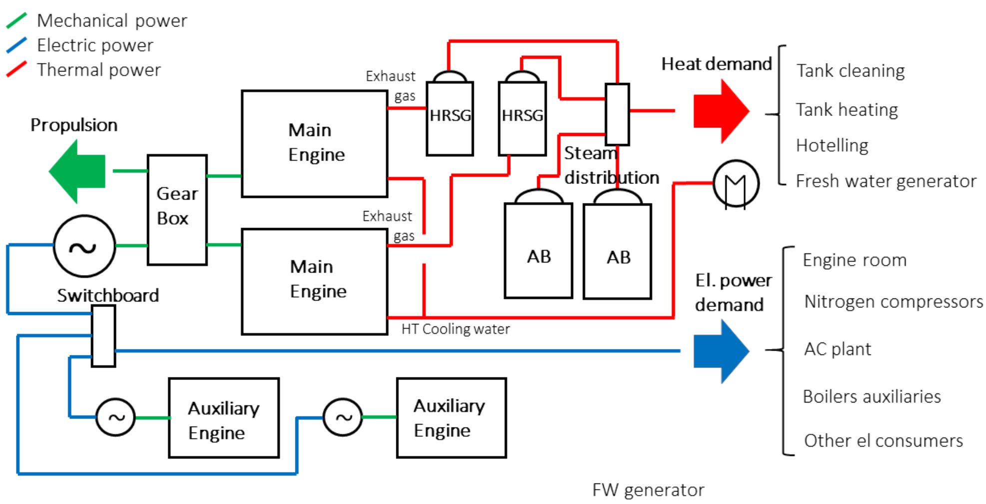


Figure 4.2: Conceptual representation of energy systems and flows of Ship-1

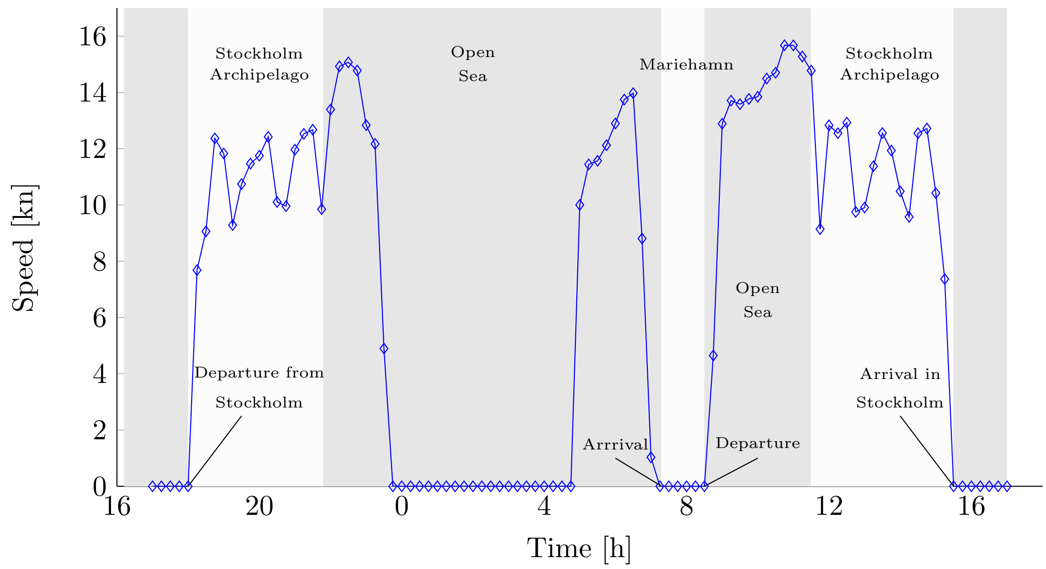


Figure 4.3: Typical operational profile of Ship-2

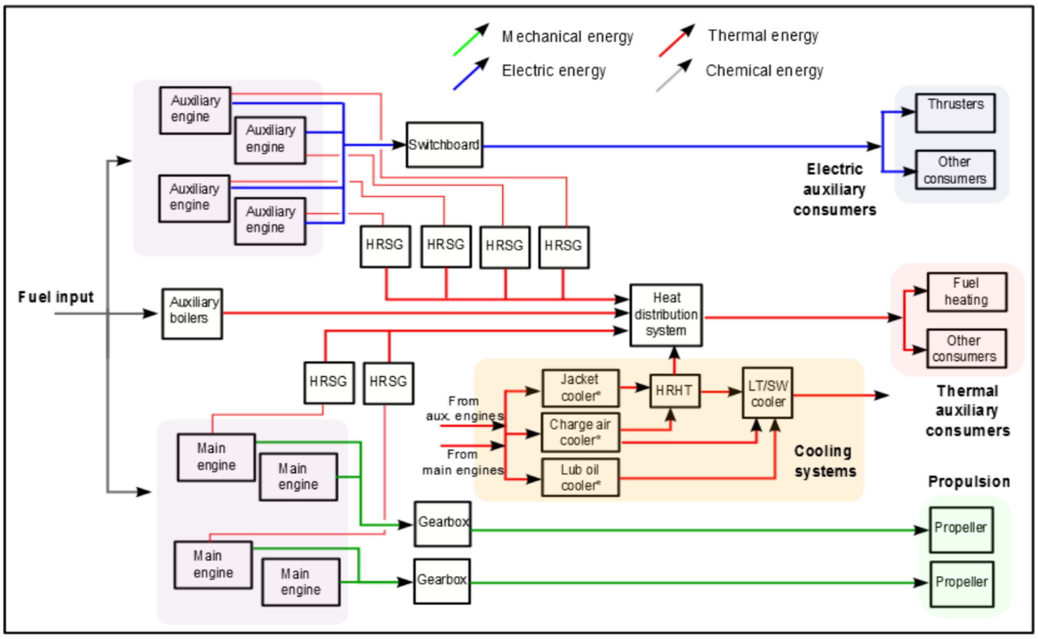


Figure 4.4: Conceptual representation of energy systems and flows of Ship-2

obtained for purposes other than to conduct (statistical) data analysis (Doganaksoy & Hahn, 2012).

Hereafter the available documentation for the two case studies analysed in this work is summarised.

Data logging system

Both Ship-1 and Ship-2 are equipped with a data logging system (DLS) which logs on board measurements on a dedicated server. In both cases, data were gathered for 1 year of ship operations. A list of the variables available from the DLS of Ship-1 and Ship-2 is presented in Table 4.3

Other sources

Not all variables of interest for this work were available from the data logging system on board. Quite extensive technical documentation was made available by the partner companies, and was used to gather additional information related to the ship systems performance.

These data relate to the nominal performance of the system and of some of its sub-systems and do not provide operational information. This documentation was therefore used for modelling the system, both in the phase of data processing and in the evaluation of possible improvements to existing systems.

Table 4.3: Summary of the available measurements from the data logging systems for the two case studies

Ship 1
Ship general
Speed over ground
Speed through water
Draft (fore, aft, starboard, port)
GPS heading
Power plant
Propeller torque
Propeller speed
ME fuel consumption
AE power
AE fuel consumption
SG power
Environment
Wind speed
Wind direction
Sea water temperature

Ship 2
Ship general
Speed over ground
Main engines
Fuel rack position
Exhaust gas temperature (before EGB)
Exhaust gas temperature (after EGB)
Charge air temperature
Charge air pressure
Auxiliary engines
Fuel rack position
Exhaust gas temperature (before EGB)
Exhaust gas temperature (after EGB)
Charge air temperature
Charge air pressure

Hereafter a short description of the different documents used in this work is provided, while Table 4.4 summarises what documents were available for the two case studies

Engines project guides contain information directly provided by the engine manufacturer and publicly available online. The data here provided comply with ISO 3046/1 and 15550 standards. Information connected to engine performance, inlet and outlet flows, and thermal losses to the environment are used in the study.

Engine shop tests contain experimental data provided by test performed by a classification society and measured under well-defined conditions. Information on engine performance for different loads, including efficiency and exhaust temperature, is available from this type of technical document.

Ship sea trials are performed when the construction of the ship is completed to verify that the actual vessel performance conforms to the requirements set by the customer. These documents provide propulsion and auxiliary power demand in conditions of clean hull, calm seas for different ship speeds and are therefore often used for benchmarking.

Propeller curves are represented as a diagram provided by the propeller manufacturer and generated through numerical codes. They provide information on propeller performance for different values of the propeller pitch, speed and power and for different ship speeds.

Combinator diagrams map the characteristics of the control system installed on board for engine-propeller interaction. The combinator diagram is used when the ship is run at variable propeller speed, and is needed for engine protection versus too high torque at low speed, which would result in excessive thermal loading for the engine.

Ship electric balance is provided by the shipyard and summarises the expected power consumption of different auxiliary components depending on ship operational mode based on which the power plant was designed.

Ship heat balance is supplied by the shipyard and provides details on the different parameters used in the calculations for the design of the boilers and steam distribution systems, such as heat exchange areas and heat transfer coefficients.

Noon reports and their aggregates are manual measurements collected daily by the crew and logged in paper and electronic format. Although the accuracy and reliability of these data is often questioned (Aldous *et al.*, 2015), they constitute an additional source of information and are used in this thesis when none of the previously mentioned sources could provide the required information.

Table 4.4: Summary of the technical documentation available for the two case studies. Documents marked with ✓are available, those with × are not.

Document	Ship-1	Ship-2
Engine project guide	ME,AE	ME,AE
Engine shop test	ME	ME,AE
Ship sea trials	✓	×
Propeller curves	✓	×
Combinator diagram	✓	×
Electric balance	✓	✓
Heat balance	✓	×
Noon reports	✓	✓

4.3.2 Considerations about data quality

The quality of the data retrieved from the DLS is high in terms of sampling frequency, but low in terms of measurement accuracy. As measured values come from on board sensors, this does not allow an appropriate control of measurement accuracy and reliability. This is a situation that often occurs in observational studies and that is generally connected to limitations in data quality (Hahn & Doganaksoy, 2008).

The original data frequency measured by the monitoring system is of 1 point every 15 seconds on both Ship-1 and Ship-2. However, in both cases the amount of data points to be handled would become too large if the original sampling frequency was used for one year of ship operations.

For this reason, an averaging of the data was performed. In the case of Ship-1 the averaging was automatically performed by the energy management system provider, while in the case of Ship-2 the averaging was performed by the data logging system on board. In both cases, although it is most likely that the output of the averaging was generated using an arithmetic mean, it was not possible to get access to the computation algorithm.

Neither in the case of Ship-1 nor in that of Ship-2 it has been possible to perform an appropriate test and calibration of the sampling probes. However, general considerations concerning the accuracy of the meters installed on board are hereafter reported:

Ship speed (LOG) : The speed of the ship through the water (LOG speed) is generally measured using a small impeller or paddle wheel attached to the bottom of the hull. This type of measurement device is known to be often unreliable as a consequence of the fact that the flow through the measurement device can be disturbed by the interaction with the hull or by other environmental conditions (Insel, 2008).

Ship speed (GPS) : The speed of the ship compared to a fix reference (speed over ground, or GPS speed) is measured by on board GPS sensors. GPS speed measurements are rather reliable; however, the GPS speed does not account for the influence of currents, which can be as strong as 2-3 knots depending on time and location, and is therefore of lower interest compared to the LOG speed.

Fuel consumption : In the case of Ship-1, fuel consumption is measured using a mass flow meter based on the Coriolis effect. This type of meter allows reducing measurement uncertainty when compared to volumetric flow meters (more commonly installed on board ships), as the latter are sensitive to errors in the calculation of fuel density.

Fuel energy content (LHV) : Measurements of fuel lower heating value (LHV) are rarely available, thereby introducing an additional element of uncertainty in the analysis. Fuel LHV is mostly influenced by its sulphur content, water content, and carbon/hydrogen ratio for variations that could reach up to $\pm 5\%$. In this thesis, a constant value of 40.4 MJ/kg is used, following the fact that no measurement of fuel LHV was available (Bengtsson *et al.*).

Propeller torque : is calculated based on optical measurements of the shaft's elastic deformation. The estimated accuracy is $\pm 1\%$ based on information provided by the shipyard.

Propeller speed : Propeller speed is measured optically on the propeller shaft, with an accuracy estimated to $\pm 0.1\%$ based on information provided by the shipyard.

Electric power : The electric power demand is calculated starting from the power delivered by the electric generators (shaft generators, auxiliary generators) based on measurements of electric current and voltage. Although detailed information was not available for the specific instruments installed on both ships, electrical measurements are generally accurate and reliable (Blackburn, 2001).

Flow temperatures : Temperature measurements available from data logging systems are measured with thermocouples, which are widely used industrially due to their reasonable accuracy and reliability and low cost (Kutz, 2013). Nominal accuracy ranges from $\pm 1K$ for T type thermocouples, normally used for temperatures up to 540 K, and $\pm 2.2K$ for K type thermocouples, for up to 1530 K. In practical applications, however, the accuracy is generally lower due to decalibration over time and to perturbations in the electric signal (Kutz, 2013).

4.3.3 Data cleaning

Data cleaning refers to the process of detecting and correcting (or excluding from the analysis) corrupt or inaccurate values from a dataset (Doganaksoy & Hahn, 2012).

The detection of faulty measurements is a particularly challenging task:

> *While it may be obvious that a value is missing from a record, it is often less obvious that a value is in error. The presence of errors can (sometimes) be proven, but the absence of errors cannot. There is no guarantee that a data set that looks perfect will not contain mistakes. Some of these mistakes may be intrinsically undetectable: they might be values that are well within the range of the data and could easily have occurred. Moreover, since errors can occur in an unlimited number of ways, there is no end to the list of possible tests for detecting errors. (de Veaux & Hand, 2005)*

In this work an automatic, rule-based data cleaning process was applied to the original dataset. This process led to the elimination of specific data points which did not pass checks of consistency and of belonging to a specific range.

In the case of Ship-1, the following selection rules were used:

Total fuel consumption Data points for $\dot{m}_{fuel} > 1500 kg/h$, which would correspond to fuel flow above the maximum permitted value, were excluded.

Main engines power Data points for $P_{prop} + P_{S/G} > 8000$, which would correspond to $P_{ME} > MCR_{ME}$, were excluded.

Main engines efficiency The main engines' break specific fuel consumption (BSFC) was calculated based on measurements of the engine power and of the fuel consumption: $BSFC_{ME} = \frac{\dot{m}_{ME}\left[\frac{kg}{h}\right]}{10^3 P_{ME}[kW]}$. According to the engine project guide, the engine maximum efficiency in ISO conditions is estimated at $178 \frac{g}{kWh}$. Consequently, all points for which $BSFC_{ME} < 178 \frac{g}{kWh}$ were considered invalid. For these values, the error was assumed to originate from faulty measurements of the fuel consumption, which is more fault-prone than propeller or S/G power. These values were hence corrected by providing a new calculated value for the engine $BSFC = P_2(\lambda_{ME})$, where $P_2(\lambda_{ME})$ is a 2^{nd} degree based on a polynomial regression based on the entire dataset.

In the case of Ship-2, the following selection rules were used:

Seawater temperature For some of the points in the dataset, the measurement of the seawater temperature was missing. In this cases the measured air temperature was used as a reasonable estimation of seawater temperature.

Auxiliary engines, exhaust gas temperature All values for which $T_{eg,turbine,in} < 0K$ and/or $T_{eg,turbine,out} < 0K$ were substituted by $T_{eg,turbine,in} = 650K$ and/or $T_{eg,turbine,out} = 550K$ respectively. This allowed not to eliminate these data points, while maintaining a conservative approach to the estimation of the waste energy flows.

Auxiliary engines on/off For data points with $\lambda_{AE} < 0.05$ the auxiliary engines were assumed not to be running, and therefore all inputs and outputs were set to 0.

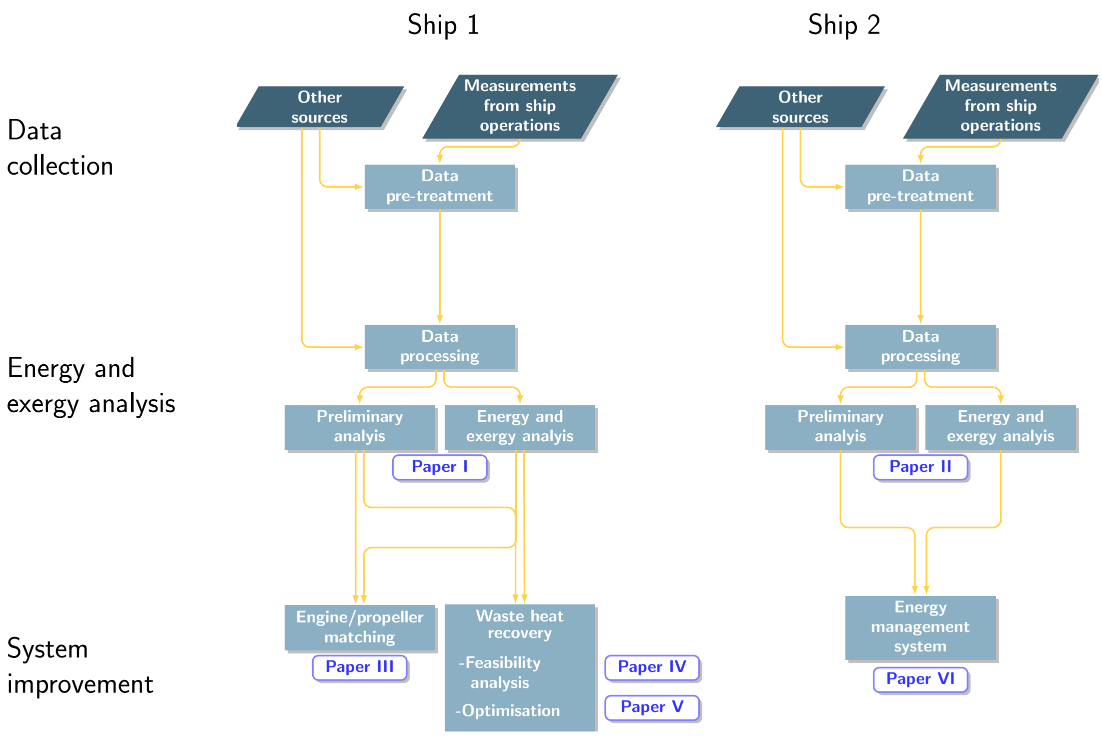


Figure 4.5: Overview of the methodology (2)

4.4 Summary of the approach of the appended papers

In Section 4.1 the general approach of energy systems engineering that was applied in this thesis was presented. This section introduces how the different papers presented in this thesis relate to the central theme of this thesis (see also Figures 4.1 and 4.5).

Each of the papers is presented by describing its main aim and the methods specifically employed. In addition, the novel element of each paper compared to the existing literature is highlighted, together with how the paper contributes to the main subject of the thesis.

4.4.1 Data processing for energy and exergy analysis (Paper I and II)

Aim : To investigate the energy flows of the case study ships (Ship-1 and Ship-2) over one year of operation and, hence, to improve the understanding of these systems.

Method : The energy and exergy flows for each time step of the datasets are calculated by elaborating available measurements. This elaboration is performed using models based on a combination of white- and black-box approaches.

Novelty : Existing literature aiming at the estimation of ship energy flows mostly focuses on energy flows (Thomas *et al.*, 2010; Basurko *et al.*, 2013). Only Marty (2014) included exergy in the analysis. Paper I and II constitute additional case studies for the application of energy and exergy analysis to ship energy systems and, therefore, towards an improved understanding of these systems.

Red thread : From a systems engineering perspective, the use of energy and exergy analysis for analysing the behaviour of a system based on operational measurements represents the systems analysis phase, in which the existing system is investigated to identify possibilities for improvement.

4.4.2 Propeller/engine matching (Paper III)

Ship-1 can operate in two alternative operational modes:

Fixed speed : The engine and propeller are operated at fixed speed. The auxiliary power is fulfilled by the shaft generator.

Combinator mode : The propeller speed is left free to vary adapting to the best conditions for propeller efficiency. The auxiliary power demand is fulfilled by the auxiliary engines.

In the first case, auxiliary power is generated at a higher efficiency, since the main engines are more efficient than the auxiliary engines. In addition, the main engines are operated at higher load and therefore, in principle, more efficiently. However, in the second case the propeller can operate at variable speed and closer to its optimal point.

Aim : To investigate the trade-off between these two opposites contributions and to compare the two modes of operations based on the expected difference in fuel consumption.

Method : The propulsion system is modelled and simulated for a range of ship speeds (10 to 15 kn). The engine was modelled using a combined 0D-MVEM model which enabled to make predictions of the influence of the speed of the engine on its energy efficiency, while the propeller was modelled based on the Wageningen B-series polynomials.

Novelty : Although many authors before have modelled the entirety of the propulsion system (Benvenuto & Figari, 2011; Theotokatos & Livanos, 2013), there is no documented effort of explicitly analysing the consequences of the interaction between the engine and the propeller when comparing operations at fixed speed versus in combinator mode. Furthermore, the requirements of the problem led to the development of an innovative combined 0D-MVEM engine model suitable for use in ship energy system models.

Red thread : The work presented in Paper III is intended to show how the identification of optimal ship operations in different sailing conditions can be improved when interactions within the system are studied more in detail.

Table 4.5: Details of the conditions in the WHR cases investigated in Paper IV

Case	Waste heat source	Final use
A1	Exhaust gas	Electric power
A2	Exhaust gas	Electric and propulsion power
B1	Exhaust gas and HT cooling	Electric power
B1	Exhaust gas and HT cooling	Electric and propulsion power
C1	All primary waste heat sources	Electric power
C2	All primary waste heat sources	Electric and propulsion power

4.4.3 Waste heat recovery systems (Paper IV and V)

The potential for waste heat recovery for Ship-1 was evaluated in two different studies: Paper IV and Paper V.

WHR feasibility analysus

Aim : To present and test a method for evaluating the potential for WHR on board of a ship starting from measurements of ship operations without designing the recovery system. The method is tested on Ship-1.

Method : The potential of the installation of a WHR system is calculated starting from the exergy flows of Paper I. The energy generated by the WHR system is presented as a function of the the WHR's exergy efficiency, which is treated as an independent variable. According to this approach, the exergy efficiency of a system is used as an indicator of the technological level of the system (e.g. the quality of its components, the complexity of the thermodynamic cycle, the size of the heat exchangers). The evaluation was performed for different scenarios, depending on the final use of the recovered energy and on the waste energy sources used for recovery (see Table 4.5).

Novelty : Differently from other literature on the subject, the paper puts its focus on the estimation of the feasibility of the WHR system rather than on the optimal design of the system itself.

Red thread : The work presented in Paper IV is intended to show the importance of accounting for how the ship is operated in the systems engineering process, and in particular in the process of designing a WHR system.

Modelling and optimisation of an ORC system

Aim : To propose and optimise the design of a WHR system for Ship-1 based on the knowledge of its operational profile.

Method : A WHR system based on a Rankine cycle was modelled on a component-by-component basis (see Figure 4.6)[1]. The design and operational parameters of a WHR system have to be defined in the design phase, requiring an optimisation process. In this paper four different optimisation procedures (see Table 4.6) are compared based on:

- The extent to which part-load operations were accounted in the definition of the objective function. In the "simplest" optimisation procedure the system was optimised only based on its performance at design load. In the "most advanced" procedure, the objective function was calculated as a weighted average of the performance of the WHR system at different engine loads, where the weights were assigned based on how often the ship was found to operate at that specific load.
- The parameters included in the optimisation. In the simplest case, only typical cycle parameters (design pressure, fluid) were included. In the most advanced case, also the switching load between one- and two-engines operations and the maximum operational range for the WHR system were included as optimisation parameters.

The engine outputs (efficiency and energy flow in the exhaust gas) were modelled using polynomial interpolations as functions of engine load based on the model presented in Paper III. In addition, it was assumed that the entire waste heat available in the main engines' exhaust gas could be used for conversion to electric power. This implies that the on board heat demand was assumed to be fulfilled using the energy in the cooling water.

Novelty : Differently from other literature on the subject, design parameters of the WHR system are optimised based on the ship's operational profile rather than on one operating point. Furthermore, some engine operational parameters are also allowed to be part of the optimisation process instead of only focusing on the WHR system.

Red thread : The work presented in Paper V is intended to show the benefits that can be achieved, when designing ship energy systems (a WHR system in this specific case), by optimising the system based on its operational profile and by broadening the boundaries of the system of interest (in this case, from the WHR system alone to including the main engines).

4.4.4 Ship power plant operational optimisation (Paper VI)

Aim : To propose an on board energy management system capable of allocating the energy demand to different prime movers (namely: main engines, auxiliary engines, and boilers) while minimising the fuel consumption.

[1]It should be noted that the paper stems from a collaboration with Ulrik Larsen, who provided the most significant contribution to the modelling of the Rankine cycle.

Table 4.6: Details of the WHR optimisation procedures investigated in Paper V

Case	Description
DP	WHR system optimised at the propulsion system's design point
DP$_+$	As in DP, but the system is also evaluated at 50% of the propulsion system's design point. If the system cannot work in these conditions, the design is discarded
OP	The WHR system is optimised on the measured operational profile of the ship
OP$_+$	As in OP, but some engine operational parameters are also included in the optimisation procedure

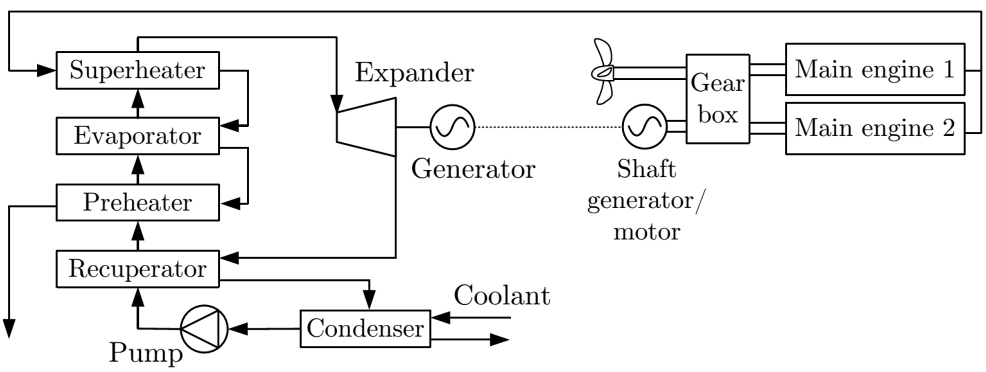


Figure 4.6: Layout of the waste heat recovery systems proposed for Ship-1

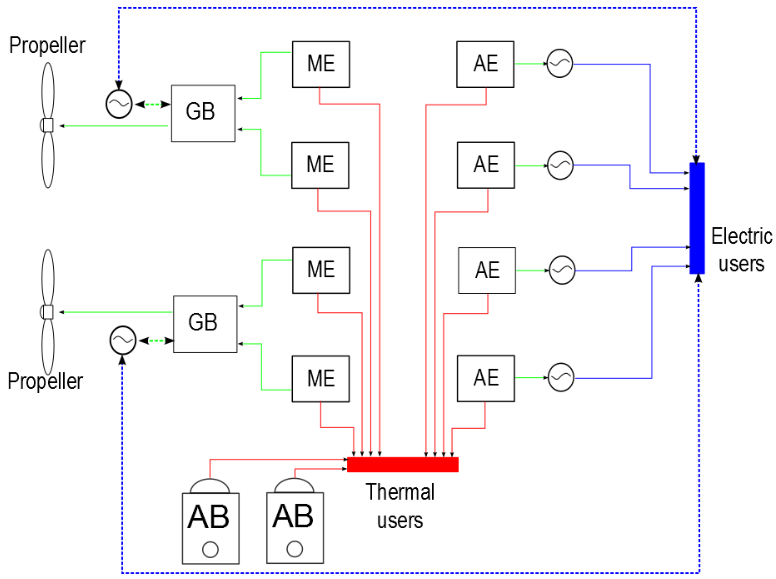


Figure 4.7: Layout of hybrid propulsion system proposed for Ship-2. The dashed connections represent the additions compared to the existing system

Method : The proposed energy management system is applied to both the existing power plant installed on Ship-2, and as means to evaluate the potential for a proposed hybrid propulsion system which includes the installation of a shaft motor/generator on each of the propulsion lines (see Figure 4.7). The main engines are modelled using a combination of white- and black box modelling approaches, while all other components on board are modelled using empirical correlations. The optimisation of the load-allocation is performed by stating the problem as a mixed integer and nonlinear programming (MINLP) problem, which is solved using a SQP algorithm (for the NLP part) and a brach-and-bound method (for the integer part).

Novelty : Compared to existing literature on the subject (e.g. Solem *et al.* (2015)), the proposed method also includes the fulfilment of heat demand (and, therefore, fuel consumption from the boilers).

Red thread : The work presented in Paper V is intended to show the benefits of system modelling and optimisation in the evaluation of ship power plants where the load allocation problem is not trivial. Furthermore, the work shows how considering additional interactions within the system (i.e. heat demand and boilers) allows achieving further fuel savings.

Chapter 5

Results

Analysis and synthesis of ship energy systems

Chapter 5 presents the results for the application of the principles of energy systems engineering to the two case studies considered in this thesis, with a particular focus on underlining how the proposed approach represents an improvement compared to standard non-systemic practices. Section 5.1 focuses on the analysis of the existing systems, laying the ground for the synthesis and evaluation of possible improvements presented in Sec. 5.2.

5.1 Energy system analysis: Improving the understanding of the system

The work presented in the first part of this thesis aims at improving the understanding of the ships selected as case studies from an energy perspective. The content of this section is a summary of what presented in Paper I and Paper II.

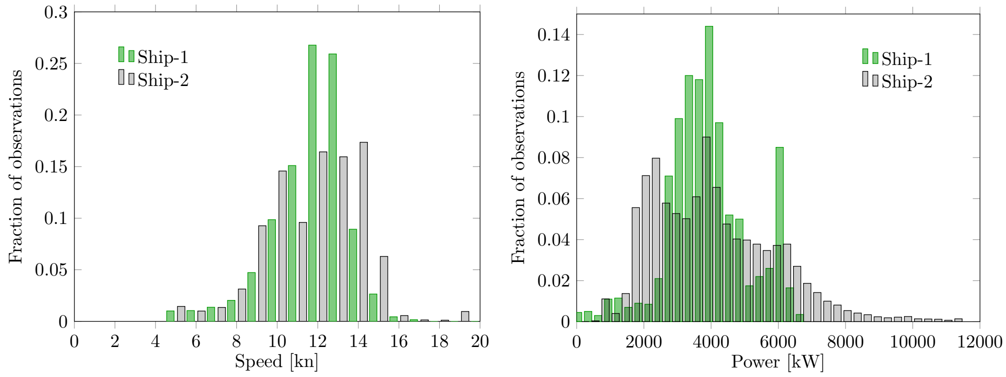
5.1.1 Energy analysis

Both Ship-1 and Ship-2 show large variations in their power demand, particularly for propulsion (Fig. 5.1) but also for heat and electric power (Fig. 5.2). This observation is particularly of interest as it highlights the importance of accounting for this variability in the design process, which will be further discussed in the following section.

In addition, the results suggest that, although propulsion demand appears predominant in both case studies, auxiliary heat and electric power demand also represent a significant share of the total energy demand (20% and 12% for Ship-1, 33% and 25% for Ship-2 respectively, see also Figure 5.4a and 5.4b).

This situation is related to two main observations:

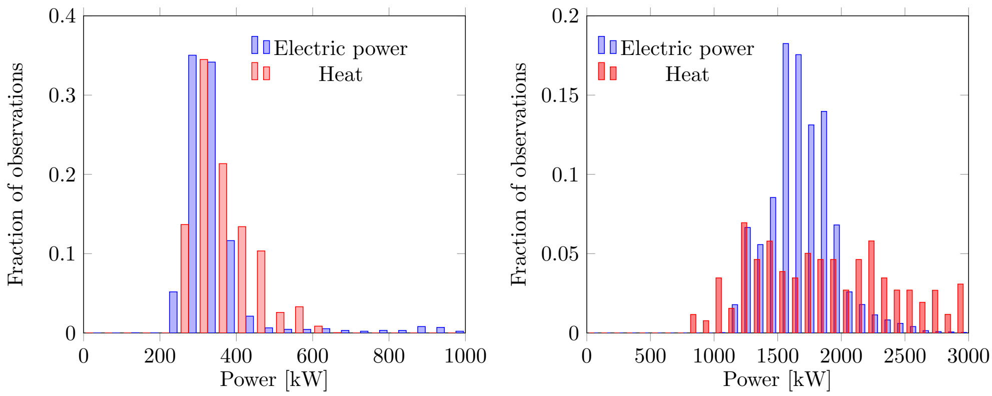
- Both ships spend a large amount of time in port (see Fig. 5.3), where there is no propulsive power demand.



(a) Speed

(b) Propulsion power

Figure 5.1: Case studies operational analysis: Speed and propulsion power distribution



(a) Ship-1

(b) Ship-2

Figure 5.2: Case studies operational analysis: Auxiliary power distribution

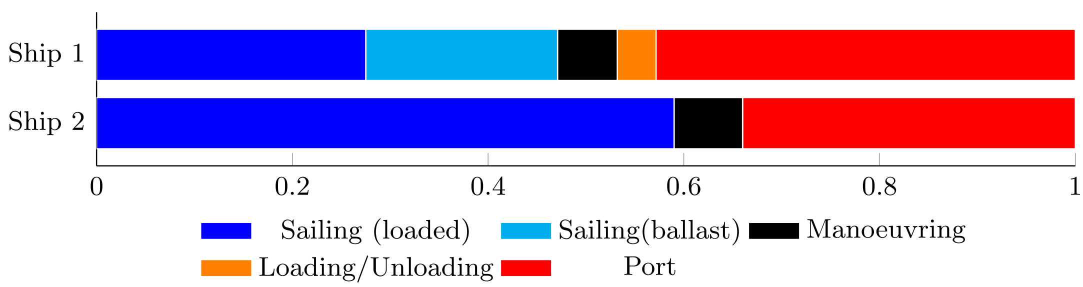


Figure 5.3: Operational share, time-based

- Both ships, for different reasons, generally operate far from the design speed of the vessel. When the ship operates at low speed, the power demand for propulsion is reduced, while auxiliary heat and electric demand tend to remain approximately constant.

An additional observation resulting from the energy analysis relates to the availability of waste heat. In both cases, in spite of the installed HRSGs, the exhaust gas contains a significant amount of energy that could be recovered for other purposes. Most of the heat demand on board is already satisfied without the need of the use of oil-fired boilers (64% and 63% for Ship-1 and Ship-2 respectively), whose fuel consumption represent only a minor part of the total (resp. 4.1% and 5.2%).

The fact that there is waste heat available and, at the same time, that the oil-fired boilers are necessary for satisfying heat demand can be explained by the large amount of time spent in port, when the main engines are not running. During sea voyages, the amount of waste heat available for recovery often exceeds the heat demand, and it could be used for generating electric power. In these regards, however, the different sources of waste heat from the engines have different potential, in relation to their different temperatures.

5.1.2 Exergy analysis

Exergy analysis, by taking into account both energy quantity and quality, allows a more realistic estimation of the potential for waste heat recovery.

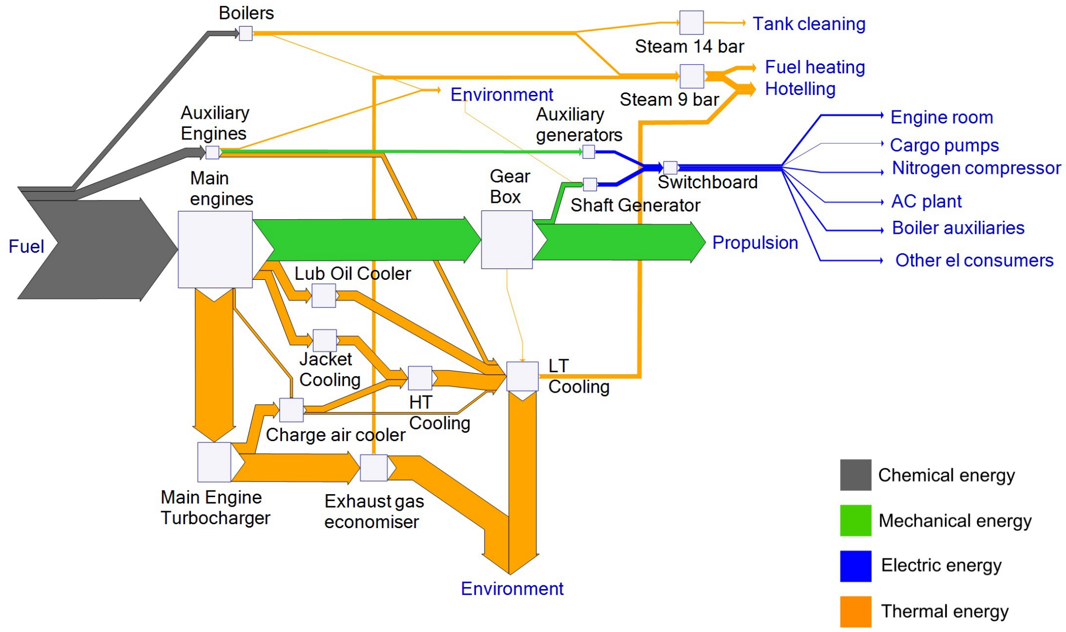
In the case of Ship-1, the exergy loss to the environment through the exhaust gas of the main engines (after the HRSG) equals to 14 TJ/year, to be compared to a total exergy output for propulsion of 68 TJ/year. In the case of Ship-2, the same flows accounted for 20 TJ/year and 75.2 TJ/year respectively, with 8.7 TJ/year more from the auxiliary engines.

This results in only 11% of the waste energy (or 10% of the waste exergy) being recovered on board in the case of Ship-1. These numbers are higher in the case of Ship-2 (23% and 25%), showing that the energy system of Ship-2 makes a more efficient use of the energy on board.

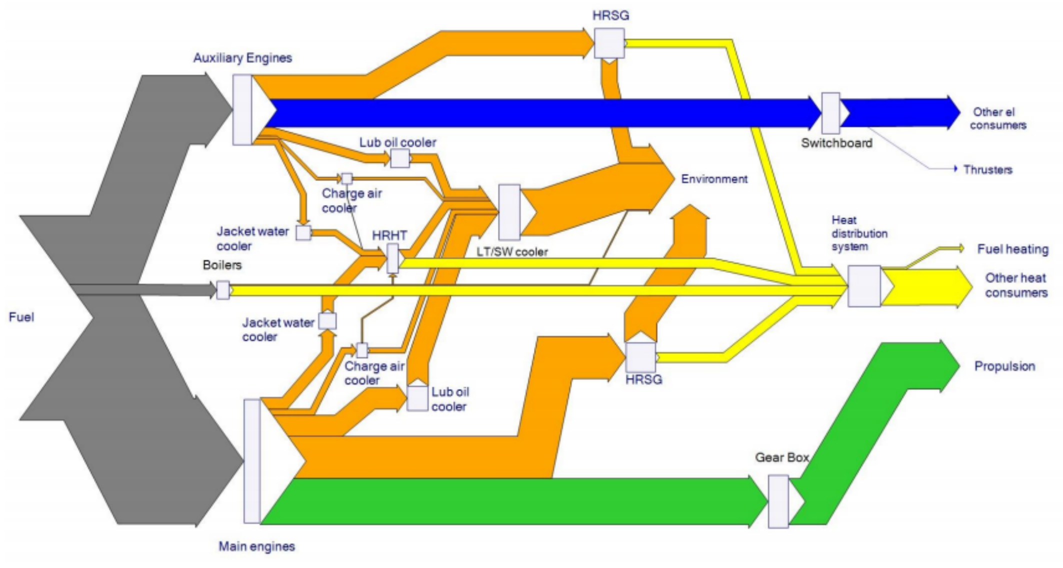
5.1.3 About on board measurements

The process of gathering and analysing data obtained from ship operations, and in particular the process of energy and exergy analysis, allows a reflection on the relative importance of different measurements. In particular, the fact that the two case study vessels did not have the same amount and type of measurements allowed the comparison of the two experiences:

Propulsion power can be obtained from measurements of speed and torque on the propeller shaft. Having accurate data related to this variable is of utmost importance for the estimation of propulsion power demand, engine efficiency, fouling effects on hull and propeller.



(a) Ship-1



(b) Ship-2

Figure 5.4: Sankey diagram for ship energy systems. Note that the scale is not the same for the two diagrams, so flow sizes can be compared within each diagram, but not between them

Electric power is normally measured at the output of the generators, providing a reliable estimation of the total electric power demand. However, component-by-component measurements would allow more detail in the analysis, and especially in sight of optimising the energy usage of individual consumers, such as pumps, HVAC compressors, and fans.

Thermal power is hardly measured at all. On Ship-2 it was possible to estimate part of the contribution based on measurements of the temperature of the exhaust gas before and after the HRSG. On Ship-1, instead, all information was based on the technical documentation provided by the shipyard. If thermal systems are to be included in the process of improvement of the system, more accurate information is required, both on the demands and on the waste heat flows from the engines. More specifically, these should include **temperature** and **flow measurements** on:

- Steam distribution network
- Air and exhaust gas flows to and from the engines.
- Cooling water systems (both HT and LT).

5.2 Synthesis: Proposing solutions for system improvement

The second, core part of an energy systems engineering approach consists in the synthesis and evaluation of possible solutions for improving the systems from an energy efficiency perspective.

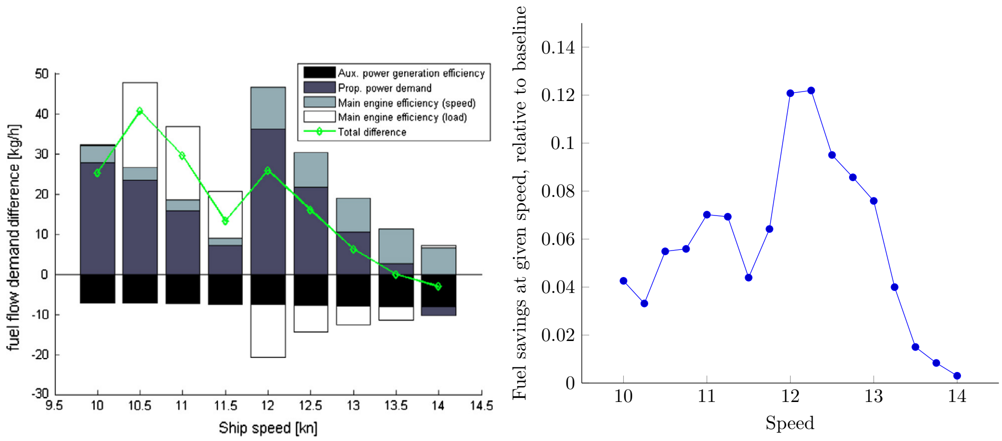
5.2.1 Potential for energy efficiency

Based on the results of the initial phase, different alternative solutions were proposed and evaluated for improving the performance of the studied systems from an energy perspective.

Engine-propeller interaction

In Paper III, two alternative operational modes for Ship-1 were compared, based on whether the propeller was operated at fixed or variable speed.

The results show that operating the propulsion system at variable propeller speed can lead to lower fuel consumption in the 10-13 kn range. The estimated improvements range from a minimum of $0kg/h$ at $13.5kn$ to a maximum of $41kg/h$ at $10.5kn$ (see Fig. 5.5a). As a consequence of the ship's operational profile, the fuel savings are concentrated in the range between 12-13 kn (see Fig. 5.5b and amount approximately to 1.9% of the yearly fuel consumption.



(a) Speed-based difference **(b)** Yearly operations

Figure 5.5: Engine-propeller interaction, comparison between fixed- and variable-speed operations.

The reduction of fuel consumption comes as a combination of different contributions, as shown in Figure 5.5a[1]. In particular, it can be noted that:

- When operating at variable speed, the positive effect on propeller efficiency largely overcomes the negative effect on the efficiency of the generation of auxiliary power.

- At low speed, the effect of the main engines' load is positive (i.e. it contributes to reduce fuel consumption, compared to the baseline case). At speeds above 12 kn, as soon as engine operations switch from one- to two-engines running, the effect of main engines' load becomes negative instead.

- Operating the engine at lower speed leads to a small, yet positive impact on the engine's efficiency

Waste heat recovery

As presented in Section 5.1, there is a significant amount of heat wasted from the existing systems, in both case studies. This thesis, focused on the evaluation of the possibility of taking advantage of this potential in the case of Ship-1[2].

The results of the initial feasibility analysis (see Fig. 5.6) confirmed the expectations on the existence of a potential for heat recovery on board, as recovering heat from the exhaust gas alone (A) can generate fuel savings between 4% and 7%. This choice would constitute the simplest and least costly retrofit, also in view of the fact that there would

[1] Note: positive values refer to higher fuel consumption in the fixed engine speed case

[2] For an evaluation of potential WHR systems for Ship 2, the reader can refer to Ahlgren *et al.* (2015).

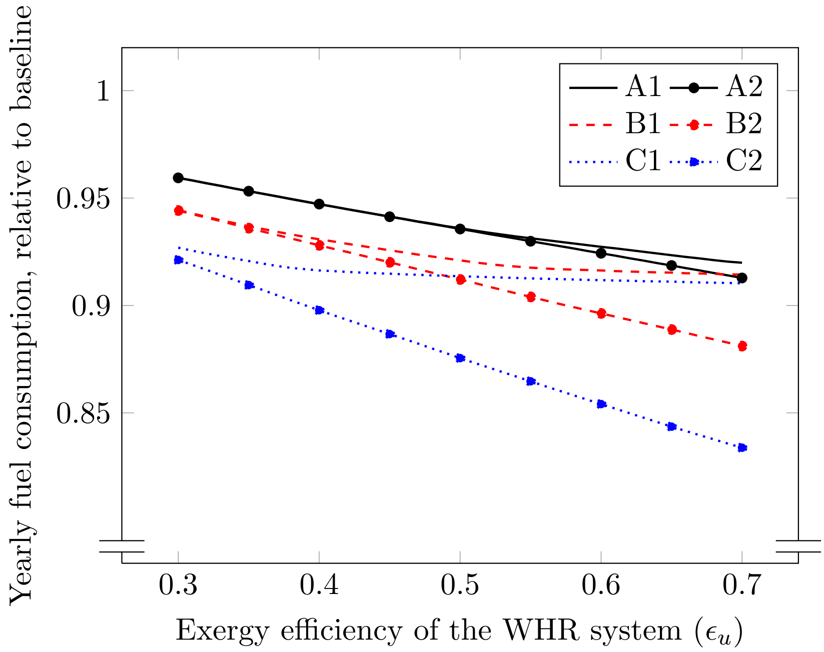


Figure 5.6: Calculated yearly fuel consumption with the installation of a WHR system on Ship-1, compared to baseline

be no sufficient extra power to be used for propulsion and, therefore, no need to install an electric motor on the propeller shaft.

Adding the HT cooling to the recovered sources (B) could improve the results; however, such improvement would be limited to approximately 1% unless i. the WHR system had a high performance ($\epsilon_u > 0.5$) and ii. the energy generated by the WHR system was also used for propulsion (B2).

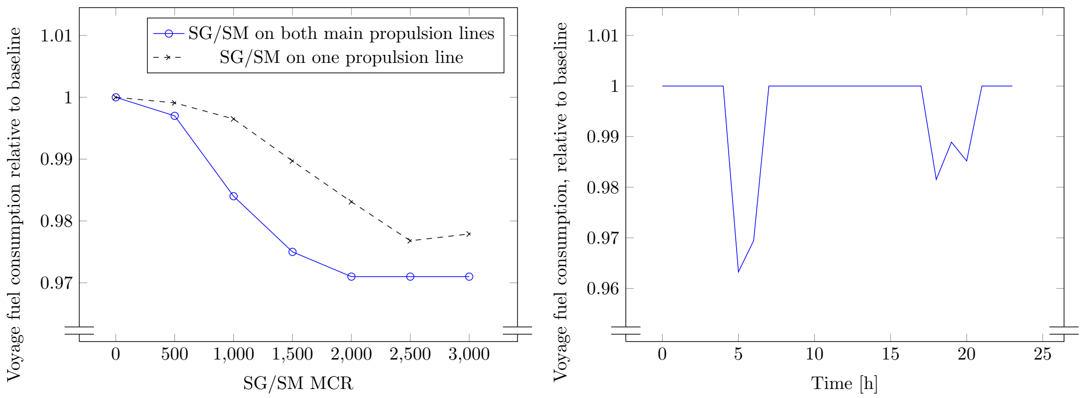
Finally, more for a matter of comparison than foreseeing a real installation, the potential of WHR when accounting for all waste heat sources on board (C) was calculated. In this case hypothetical savings could sum up to over 15%.

In Paper V, the possibility of installation of a WHR system, particularly based on a Rankine cycle, was studied in further detail, showing that yearly savings of up to 10.8% could be achieved based on the installation of an ORC-based WHR system on the engine exhaust gas line.

Hybrid propulsion systems

In Paper VI, the performance of the existing power plant of Ship-2 was compared to a power plant retrofitted for allowing more flexibility in the generation of both propulsion and electric power.

The results, as shown in Figure 5.7a, show that the hybrid propulsion system would allow fuel savings of up to 3% for the reference voyage. Lower savings, but with a lower



(a) Relative fuel consumption versus shaft generator/motor design power

(b) Relative fuel consumption, effect of including heat demand in the optimisation[1]

Figure 5.7: Ship-2: Estimated savings from the hybridisation of the propulsion system

capital cost, can be achieved if only one of the two shaft lines is equipped with a shaft motor/generator.

The savings achieved through the hybridisation of the propulsion system relate to the possibility of operating the engines closer to their design load and, hence, at a higher efficiency. On the other hand, the additional conversion steps through generators, motors and frequency converters imply higher transmission losses, thereby reducing the benefits in fuel consumption.

5.2.2 Operational profile

The main driver for the research presented in Paper III relates to the realisation that the ship operates most of the time far from its design conditions. As showed in Figure 5.5b the benefits from operating at variable propeller speed can only be observed at low ship speeds, with the break-point located at around 14 kn. As the ship was designed for operating at 15 kn, the choice of operating at constant propeller speed appears reasonable, if only design conditions are taken into account.

In the latest year, however, Ship-1 has been operating most of the time at speeds between 11 and 13 kn (see Figure 5.1a), which is where the variable speed drive provides the largest efficiency improvement. Including the yearly operational profile into the picture allows a more accurate estimation of the expected benefits, as shown in Figure 5.5b.

However, the clearest contribution to showing the importance of the operational profile presented in this thesis relates to the work included in Paper V, where an

[1]The figure shows the ratio between the fuel consumption when the thermal part of the energy demand and the fuel consumption are included in the optimisation procedure, over the reference case where only the fuel consumption of the Diesel engines is optimised.

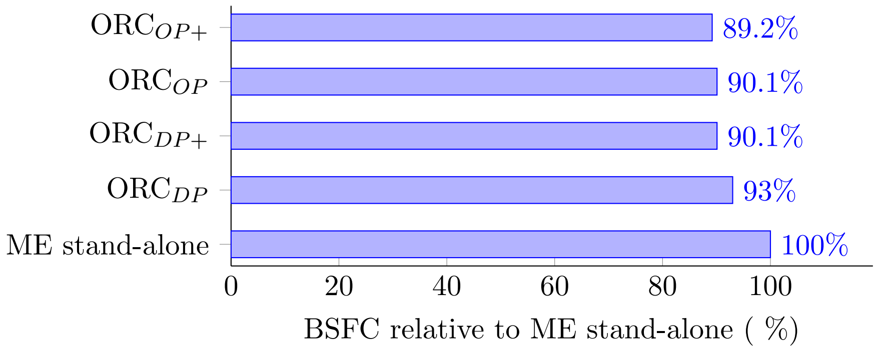


Figure 5.8: Comparison between alternative procedures for WHR systems optimisation: yearly fuel consumption compared to the baseline case

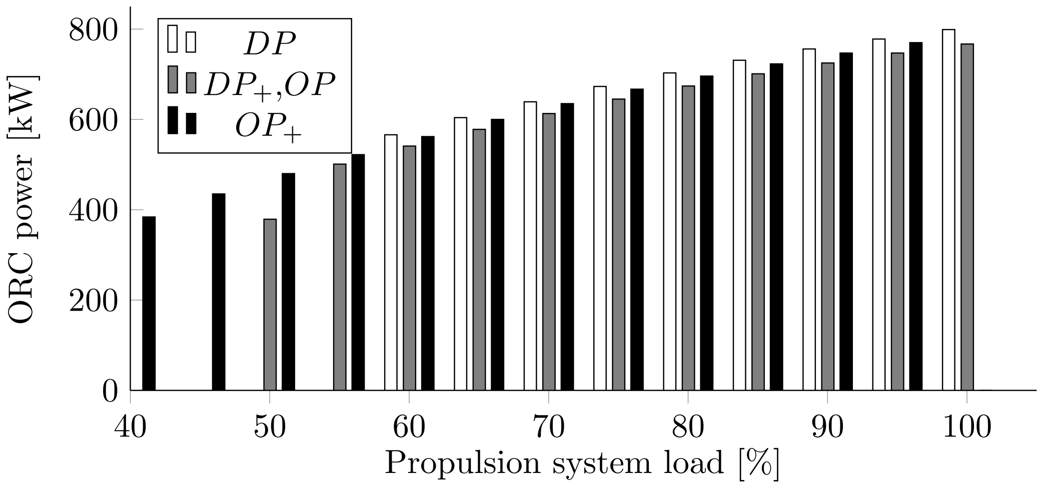


Figure 5.9: Comparison between alternative procedures for WHR systems optimisation: WHR power production at different loads

optimisation procedure based on the evaluation of the system's performance only at the design point was compared to one where the whole operational profile of the ship is accounted for.

Looking at the results of Paper V, it can be observed that the system optimised according to the DP procedure shows the largest fuel savings (10.4%) when evaluated at the design point of the propulsion system (i.e. both engines operated at 90% of their MCR). However, when the part-load performance of the system is included in the analysis and the performance of the DP design is evaluated against the full operational profile of Ship-1, the calculated fuel savings are reduced to 7.0% (see ORC$_{DP}$ in Figure 5.8).

When the whole operational profile is instead included in the optimisation (i.e. the WHR system performance is calculated, for each evaluation of the objective function, at different values of the load of the propulsion system), the results are different. The power produced by the cycle at design conditions is slightly lower (767 kW instead of 799 kW) but the yearly savings are increased to 9.9% of the yearly fuel consumption of the original propulsion system (see ORC_{OP} in Figure 5.8), mostly because the system can operate at lower load (see Fig. 5.9).

However, the results of the application of the optimisation procedure DP_+ suggest that it might not be needed to simulate the WHR design over the whole operational profile for the optimisation to converge to the optimal design. In fact, the DP_+ procedure reaches the same conclusion of the OP procedure, while only requiring two simulations: one at the design point of the system, and one at the minimum load at which the system is expected to be required to operate.

In Paper VI, similarly to Paper III, although the full operational profile is not included in the optimisation procedure, the subject under study stems in itself from the observation of a variable operational profile and from the fact that this requires an improved flexibility of the power plant.

5.2.3 Interactions

The results presented in this thesis suggest that the wider the system boundaries included in the modelling and in the evaluation, the larger the benefits to the systems engineering process. Expanding the boundaries of the system of interest directly implies including more components and, hence, a larger number of significant interactions into the analysis.

The work presented in Paper III is the most prominent example in this thesis of the importance of systems interaction. The results indicated that including the whole propulsion system in the analysis allows not only a more complete estimation of the advantages and disadvantages of the two options, but also an improved understanding of what are the effects that play a role in the overall behaviour of the system.

Although the work presented in Paper V focuses on the importance of the operational profile, it also includes aspects related to the interaction between the main engines and the WHR system. In particular, the OP_+ optimisation procedure also includes one engine operational parameter in the optimisation of the system.

Compared to an optimisation procedure based on the WHR system alone, the expected yearly savings increased from 9.9% to 10.8%. This improvement is mainly due to the fact that the WHR system can also operate at lower loads (40%-50%, see Fig. 5.9). This is achieved without requiring any additional capital expense compared to the "non-systemic" optimised system.

In Paper VI, the heat demand and boiler fuel consumption were included in the objective function of the optimisation, compared to the standard practice of optimising the operations of the system only based on propulsion and electric power demand. As shown in Figure 5.7b, depending on the instantaneous demand, this can lead to up to 4% fuel savings. In practice, this means that it can be sometimes more efficient to operate the engines at a load which does not maximise their mechanical efficiency, but that allows to recover more waste heat therefore operating the whole power plant more energy efficiently.

Chapter 6

Discussion

Chapter 6 elaborates on the results of the thesis in three different ways. First, the results are discussed as part of a broader perspective, and their contribution to the field is highlighted. Secondly, the methods and assumptions used in the thesis are discussed and put under scrutiny, based on the experience gained at the end of the work. These aspects are discussed separately for the two main parts of this thesis: systems analysis (Sec. 6.1) and synthesis (6.2). The results are also discussed in relation to the potential for energy efficiency of the technologies evaluated in this thesis (6.3), i.e engine-propeller interaction, waste heat recovery, and hybrid power plants. The chapter is concluded with a reflection on the generalisability of the results (6.4), i.e. on the extent to which the findings of this thesis can be considered to be representative of the shipping sector as a whole.

6.1 A systematic procedure for analysing ship on board energy systems

In this thesis, the use of energy and exergy analysis as systematic tools for improving the understanding of ship on board energy systems was proposed.

6.1.1 Significance and contribution to the field

Energy and exergy analysis are widely employed tools for land-based energy systems. In shipping, however, only three papers could be found in scientific literature that explicitly aim to analyse ship energy flows (Thomas *et al.*, 2010; Basurko *et al.*, 2013; Marty *et al.*, 2012). Compared to these publications, the work presented in this thesis presents a combination of different aspects:

Yearly operations : Most of the work related to the analysis or design of ship on board energy systems focuses either on a limited amount of operating points or voyages (e.g. Marty (2014)). The work presented in this thesis, similarly to what proposed by Thomas *et al.* (2010) and Basurko *et al.* (2013), bases the analysis on the ship operations over an extended period of time and therefore provides a more accurate picture of the importance of different energy demands.

Heat demand : Of the work available in the scientific literature, only Marty (2014) takes the heat demand into account. Although this contribution is often limited, the work presented in Paper I suggests that the heat demand can constitute a non-negligible contribution to the yearly energy demand also for cargo vessels.

Waste heat : Many authors who presented work on WHR systems included an evaluation of the available waste heat from the main engines' exhaust gas. Including waste heat from the charge air cooling is rare (Dimopoulos *et al.*, 2011)), while even fewer also include evaluations of other cooling-related waste heat flows (e.g. Marty (2014); Grimmelius *et al.* (2010)). The work presented in this thesis represents a new case of the estimation of the available waste heat in the cooling systems accounting for ship operations.

Exergy : The use of exergy analysis is substantially new in the field of ship on board energy systems. Dimopoulos *et al.* (2012) applied the concept of exergy as an aid in the process of optimising a marine WHR system, while Zhao & Zhaofeng (2010) analysed a combined marine power plant from an exergetic perspective. However, similarly to the point previously discussed, an estimation of the availability of waste heat from the ship on board energy system which included all sources of waste heat over the ship's operational profile had not been presented before.

Concerning its practical application, the proposed method has two main advantages:

First, it represents a **systematic and effective tool** for the analysis of ship on board energy systems and, consequently, for the process of determining how energy efficiency should be addressed on a vessel.

Secondly, the ensemble of actions required for successfully performing all the steps of the process (gathering of on board measurements, assessment of data quality, data processing) allows getting an **improved insight of the energy system of a vessel**. Consequently, even when the numerical results of the energy and exergy analysis do not provide clear suggestions for improvement, the designers will be able to propose solutions based on their improved knowledge of the ship's energy systems.

6.1.2 Validity: Methodological choices and assumptions

The data processing phase required for the energy and exergy analysis, given the absence of many relevant measurements, proved particularly challenging both in the case of Ship-1 and Ship-2. Hereafter, the most "sensitive" assumptions are summarised:

- On **Ship-1**, a number of assumptions were made in the attempt of **subdividing the electrical energy demand among different groups of consumers**. In the case of **Ship-2**, this was done only for the case of **bow thrusters** given the large amount of electrical consumers on board.

- On both **Ship-1** and **Ship-2**, the modelling of the main engines had to be substantially simplified, especially for what concerns **heat losses**[1]. This concern becomes even larger concerning some assumptions related to the **cooling water mass flows**, which needed to be estimated for calculating exergy flows but for which there was no information available aside of the pumps design flows.

- On **Ship-2** there were no measurements available for the amount of **heat recovered from the HT cooling systems**. This contribution had to be estimated based on assumptions[2]. The estimation of heat demand and its subdivision among different consumers was also challenging in the case of **Ship-1**.

These uncertainties could have been reduced by either improving the detail of the modelling (e.g. by modelling the details of the cooling systems in terms of pumps, valves and heat exchangers, as proposed by Marty (2014)) or by excluding the uncertain elements from the analysis.

In general, the approach used in this thesis was an attempt to achieve a good compromise between providing as much information as possible based on the available data without requiring a too extensive modelling effort. This choice related to the intention of proposing a method that could be used in conditions of limited time and resources.

6.2 The benefits of an energy systems engineering approach

In this thesis, the matter of increasing ship energy efficiency was addressed by employing an energy systems engineering approach, which involved a specific focus on interactions within the system and on the impact of the ship's operational profile on its performance.

6.2.1 Significance and contribution to the field

Modelling the interactions between different parts of the system of a ship has been done many times before in the field of marine engineering. This is particularly true for the interaction between engine and propeller, whose role is of utmost importance in the determination of the behaviour of the system during manoeuvres (see the work of Benvenuto & Figari (2011); Coraddu *et al.* (2014); Shi (2013)).

Similarly, accounting for the operational profile in the optimisation of ship energy systems is not uncommon in available literature in the field (see Motley *et al.* (2012) for the application to propeller design and Dimopoulos *et al.* (2011); Choi & Kim (2013) to WHR system design).

[1] When the engineers on board of Ship-2 looked at the result of our work, they were very puzzled by the amount of waste heat going to the lubricating oil cooling systems.

[2] See Paper II. These assumptions were strongly questioned by the reviewers.

The novelty of the work presented in this thesis lies in the combination of the two aspects, which can be observed in all of the papers presented in this thesis related to system synthesis (Paper III to Paper VI[1]).

The work proposed in this thesis aimed at providing evidence for the need of extending the boundaries of ship energy systems modelling, and of accounting with additional detail for how the energy system will be operated in the foreseeable future. Although none of the models presented in this work claimed to be holistic, it was showed that more accurate results and higher potential for energy savings were found every time the system boundaries were enlarged and ship operations were included with additional detail.

- Paper III showed the improvements related to both the engine and the propeller operating more efficiently at variable speed, which offsets the lower efficiency of the auxiliary engines. The yearly savings, estimated to a total of 1.9% of the yearly fuel consumption, are estimated based on the ship's real operational profile. Looking at the engine and the propeller separately would have led to an inaccurate estimation of the potential savings; similarly, looking at the ship's performance only at its design point would not have allowed to identify any saving at all.

- In Paper IV the aim was to provide tools for choosing whether to consider the installation of a WHR system or not and, in case, what yearly savings could be expected based on the expected efficiency of the WHR system, on the sources of waste heat recovered and on the final use of the generated power. The combination of these aspects, evaluated over the whole operational profile provides a simple, yet reliable tool for supporting decisions in relation to WHR systems in shipping. The challenge of the optimal design of the recovery system, which requires additional time, resources and competences, is postponed to after the evaluation of the convenience of the investment.

- In Paper V a WHR system was optimised for its application on Ship-1. Optimising the system for performance over the whole ship operational range **and** modelling the interaction between the operations of the engines and of the WHR system was estimated to allow yearly fuel savings of 10.8%. In the same study, the performance of a WHR system optimised only at its design point and with no modifications to the engine management strategy was tested. When this WHR system was evaluated over one year of ship operations, it allowed "only" 7.0% savings. These findings are in line with what previously proposed by Larsen *et al.* (2015), where it was pointed out that when a WHR system is added to a Diesel engine, the system's most efficient operating point does not coincide with the engine's most efficient load.

[1] In the case of Paper VI the analysis was performed on a "reference voyage" rather than on the whole measured operations. It should be noted, however, that the ship operates on a fixed route and, therefore, the variations of power demand are less sensitive compared to ships operating on the spot market.

- In Paper VI an on board energy management system for the optimal allocation of the load to the different parts of the power plant of Ship-2 based on a given demand was proposed. Although there are examples in literature of similar tools for optimal load-allocation (e.g. Zahedi *et al.* (2014); Kanellos *et al.* (2012)), none of these included heat demand as part of the modelling, nor was the the fuel consumption of the boilers included in the optimisation. As shown in Paper VI, this can lead to operating the system in sub-optimal conditions and to up to 4% higher fuel consumption, according to the investigated scenario.

From measurements to predictions, what will the operational profile look like in the future?

In this thesis, the systems were both optimised and/or tested on the measured operational profile in the previous year of ship operations. This implies the assumption that ship operations in the future will be equal, or at least similar, to what observed in the previous year of operation. However, the work presented by Banks *et al.* (2013) suggests that in correspondence with fast changes in fuel prices and freight rates ship speed distributions change remarkably over the years.

In the case of Ship-1, given the extension of the available database, it was possible to provide a comparative analysis of some operational years. Figure 6.1 shows how the operational speed of the ship evolved in the 2012-2014 period. It appears that any improvement based on the operational profile as measured in 2012 would have overestimated the amount of time spent at high speed, compared to what happened in 2013 and 2014.

More in general, the savings estimated in an optimisation study such as that presented in Paper V represent an ideal maximum based on a system that is tested on the same operational profile it was optimised for. If real operations after the installation of the optimised system changed compared to the dataset used for optimisation, fuel savings would most likely be lower.

As a consequence, the optimisation procedure applied in Paper V is most advised in those cases where the operational profile is little dependent on external conditions, such as market forces or environmental conditions. This is typically the case of e.g ferries and cruise ships. For other ship types, for which the operating speed is more fluctuating, the results of the application of the proposed method should be taken with additional care.

6.2.2 Validity: Methodological choices and assumptions

In spite of dealing with a systems approach, none of the work presented in the attached Papers included the modelling of the full system. In addition, not all components were modelled with the same level of detail.

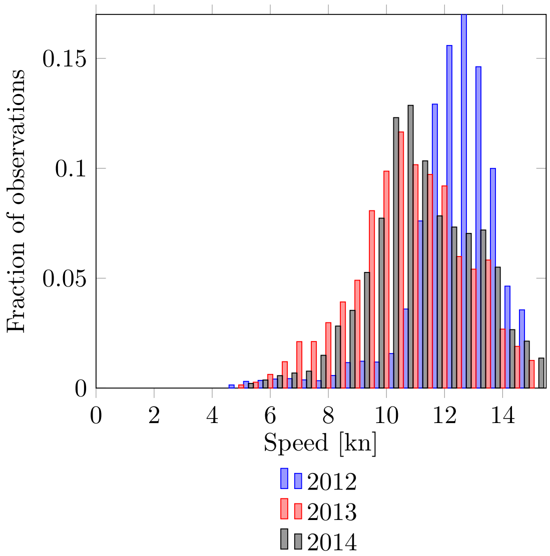


Figure 6.1: Comparison of the yearly operational profile of Ship-1 from 2012 to 2014

System boundaries

In this thesis, the power plant on board was selected as the main system of interest. This includes all the components on board that are responsible for the conversion of chemical energy (fuel) to energy in the form required for the use by other subsystems on board (mechanical energy for the propeller, electric energy for the auxiliary systems, and heat for accommodation and fuel heating).

This choice implied that many relevant subsystems were not included in the analysis. Based on the aforementioned principle that every extension in the system boundaries improves the quality of the assessment, excluding components limits the scope and reliability of the study.

The choice of excluding the propeller from the main system of interest in all but one of the studies (see Paper III) represents the most notable of the choices. As the work presented in Paper III showed, there is a significant interaction between the engine and the propeller, suggesting that future studies in connection to the ship power system should not overlook this contribution. In Paper V, for instance, the optimisation of the WHR system did not include the possibility of operating the propeller at variable speed, which would influence the engine operational point and, therefore, the quantity and quality of the heat available to the WHR system. Similarly, the energy management problem addressed in Paper VI would have been even more complex to solve, had the speed of each of the two propellers been added to the variables to be optimised.

In practice, from the perspective of the ship's hydrodynamics, the boundaries could have been extended even further by including the hull in the model through the estimation of ship resistance and of the effects of the interaction between hull and propeller. In Paper III, although the propeller was included in the model, it was decided to exclude the effects of the ship's added resistance in wind and waves, as well as the effects of biofouling on the hull. These aspects are known to have an influence not only on the power required for sailing at a given speed, but also on the matching between the engine and the propeller.

Similarly, with reference to auxiliary components on board, it was chosen not to model any of the parts of the system which contribute to the on board auxiliary energy demand (i.e. pumps, fuel heaters, HVAC systems, etc.). This choice was made based on the limited amount of information about the system and especially for the validation of the models. The addition of these components to the system model would have brought additional depth to the analysis and, most likely, allowed the identification of further potential for energy savings. Including the details of each of the components of the heat demand in Paper VI could have allowed, for instance, including the heat recovery from the LT systems, or proposing solutions for adapting the energy demand for optimal operations of the full system (demand-side management).

Extending the boundaries of the system comes, however, at a cost. The higher the amount of components, the more complex the model, and the higher the computational burden. But what makes the difference is the time required to the modeller for gathering sufficient information and data for achieving a satisfactory level of fidelity in the models. A compromise is required. In this thesis, it was decided to focus on the power plant on board, and to leave the task to broaden the boundaries of the system even further to future research.

Component modelling

Enlarging the boundaries of the system comes at the cost of additional computational effort. Assuming that the available computational power is constant and that the employed algorithms cannot be improved, the only choice is to decrease the computational requirement for each (or, at least, part of) the models that build up the full system.

The level of detail required on each model depends on the required amount of inputs and outputs that need to be handled. The case of the Diesel engine will be used in the following text as an illustrative example.

The Diesel engine is part of the on board energy system of both the ships included in this study, and was therefore modelled in all the four "synthesis" papers. In Paper III, the engine model had one, main requirement: it should be able to predict the influence of both the required torque and speed on the engine's energy efficiency.

As no measured data was available in connection to the influence of engine speed on its efficiency, a more detailed modelling effort was required. The engine speed influences the amount of air entering the cylinder, and therefore the combustion process. The model should therefore be able to capture these phenomena, which led to the choice of the hybrid 0D-MVEM.

The use of these models in energy systems modelling is, however, rare, and not always advised given the high computational time required for the model to converge. A common practice, which was in this thesis applied both in Paper III and Paper V, is that of using the output of the model to create a performance map to be used via interpolation in the energy system model.

As shown in Table 3.2, this approach is the most common in existing literature in ship energy system modelling. Exceptions to this principle can be identified in the work of Benvenuto & Figari (2011); Campora & Figari (2003); Schulten (2012); Theotokatos (2008). Their choice was, however, justified by the need of accurately simulating engine dynamics. In this thesis, and particularly in Paper III, the direct use of the 0D-MVEM was also proposed in order to evaluate the influence of the use of a variable geometry turbine (VGT) for the selected engine. In this case, given the intrinsic influence of the VGT on the engine operations, it was not possible to use a simplified version of the model.

Similar considerations could be added for the models used for, e.g., the electric machinery and the boilers. In this thesis, these components were modelled using as constant efficiencies, or as simple regressions as suggested in dedicated literature (McCarthy *et al.*, 1990). This approach was deemed sufficient for the scope of the thesis and it represents an improvement when compared to other studies (e.g. (Dedes *et al.*, 2012)). However, other researchers in the field have adopted more complex models, particularly for electric machines (e.g. Zahedi *et al.* (2014)).

6.3 Advanced marine power plants

Albeit the main focus of this thesis consists in the evaluation of the benefits of a methodological approach (energy systems engineering) to a specific area of engineering (marine engineering), the thesis included results concerning the applications of specific technologies, which should be seen as a contribution to the respective fields.

6.3.1 Propulsion systems versus power plants

After the beginning of the *slow-steaming era*, the attention on off-design performance of ship systems, and in particular of the propulsion system, has increased significantly. Therefore, for practitioners in the field, concluding that *at low speed it is better to operate at variable rather than fixed propeller speed* does not come as a surprise.

The main point of this part of the work lies in the more general consideration that the interaction between the engine and the propeller is of utmost importance for a proper and efficient functioning of the the propulsion system and they should therefore not be considered separately in the phase of ship design.

The choice of the engine, in particular, requires further attention. The engines installed on Ship-1 are efficient, both at design point and at low load. However, their operational envelope is narrow, posing very restrictive limits on reducing the engine speed, which is the typical case of heavily turbocharged Diesel engines. In retrospective, the choice of installing a less efficient, but more speed-flexible engine could have been a better choice with reference to the overall efficiency of the system.

Similarly, it was shown in this work (see Paper V), as well as in previous literature

(Larsen *et al.*, 2015), that the installation of a WHR system impacts many choices in relation to the remaining part of the propulsion system, and in particular of the engine. If a WHR system is used, it could be more efficient to install a less efficient engine with higher exhaust gas temperature.

6.3.2 Waste heat recovery systems

The results of Paper IV showed that, for the specific case of Ship-1, the focus should lie on the installation of a medium-performance system which only recovers energy from the engine exhaust gas. This solution would be the most cost effective, as it would allow minimizing installation costs while providing significant fuel savings (estimated to 4-7% on a yearly basis).

These results were confirmed once an ORC was optimised for this application, based on the ship's operational profile. As it was assumed that on board heat requirements could be fulfilled using heat from the cooling systems, the heat recovery potential was even higher, and a 10.8% improvement was calculated.

This also resulted in a low payback time for the system, which ranged from less than 2 to 5.5 years depending on the fuel price and on the assumptions made for the installation cost of the WHR system. This is in agreement with previous results presented in the scientific literature: Dimopoulos *et al.* (2011) and Theotokatos & Livanos (2013) calculated a payback time of around 8.1 and 2.4 years for a medium-sized containership and for a large bulk carrier, respectively.

These results suggest that, in theory, WHR systems should be very common in shipping. Given that merchant vessels normally have a long operative life, ranging from 10 years for tankers to more than 30 years for, e.g., ferries, a payback time of 6 years appears more than reasonable and leaves extensive possibilities for WHR to be a profitable choice.

As a matter of fact, however, the payback time allowed for such investments in the shipping business is normally 2 years, rarely going up to 5 (DNV, 2012). As a consequence, although research in WHR technology can still lead to improvements in system performance, it can be argued that the focus should shift to understanding how to allow for companies to broaden their time perspective for this type of investments[1].

6.4 Generalisability of the results

The work presented in this thesis is based on two case studies: a chemical tanker and a passenger vessel. Although the methods proposed in this work are applicable to any ship type, the question is whether the benefits obtained are specific of the two case studies, or could be expected to be observed on any other ship.

[1]Note that this reasoning does not only apply to WHR systems, but to energy-savings technologies in general.

6.4.1 Complex systems

Although employing a systems approach improves the understanding and the accuracy of the analysis for any type of system, it is generally the case that the higher the complexity of the system, the larger the benefits.

In this sense, the two case studies presented in this thesis can be used as reference for a rather broad set of ship types. Ship-2 is an example of a system characterised by a high system complexity, with a large number of elements in the power plant (four main engines, four auxiliary engines, two boilers) and an energy demand which varies in time over ship operations. In relation to the findings presented in Paper VI, for instance, it is likely that the effect of including the heat demand in the optimisation would not be as high had the energy system of a containership been taken into account instead. The operational pattern and the energy demand of Ship-1, on the other hand, are similar to those of many cargo vessels, such as tankers, bulk carriers and containerships.

6.4.2 Data availability and quality

Garbage in - garbage out. When modelling or analysing a system, the access to relevant information is of utmost importance. For the two case studies, thanks to the competence and professionalism of the two shipping companies involved[1], access to extensive datasets from on board measurements and technical documentation was available.

The models and methods employed in both the analysis and synthesis part of this thesis are flexible to different levels of information available. It is clear, however, that in the absence of on board measurements and of technical documentation related to the installed machinery on board, the work presented in this thesis would have been different[2].

6.4.3 Engine/propeller matching

That the optimisation of the interaction between engine and propeller is not an easy question is nothing new (Woodward, 1972). This is however particularly true for the case of vessels powered by controllable pitch propellers, where the additional degree of freedom given by the possibility to change the propeller pitch poses additional challenges to the optimal design and control.

Although FPPs are more common, there are today more than 18000 ships in the world powered by CPPs and four-stroke engines, as in the case proposed in Paper III. In particular, almost 3500 vessels have specifically an MaK engine of the same series (M32C) as the ones installed on Ship-1; the propulsion system of these vessels is

[1] And to the hard work of my colleague Fredrik Ahlgren from Linnaeus University, who went on board Ship-2 to download logged data from the on board alarm system, to whom go my warmest thanks

[2] During my PhD experience, I have had the chance to supervise two very smart Master Students, Alexander and Kari. Their work on hybrid propulsion system was made much harder by the fact that, in their case, on board measurements were scarce to say the least, and technical documentation of the machinery on board (engine, propeller) had been mysteriously lost on the way.

therefore expected to behave in a very similar way compared to the one presented in Paper III.

6.4.4 Waste heat recovery systems

The work proposed in Paper IV and V relates to the installation of a WHR system on board Ship-1. The results presented in this thesis suggest that there is a lot to gain from the installation of WHR systems on ships.

Although this conclusion is supported by an extensive literature on the subject, and by an increasing use of such system on board, it should be noted that the results presented in this thesis refer to the application of WHR to one specific case.

In particular, it should be noted that, as it is widely accepted, four-stroke engines have higher exhaust gas temperatures and, therefore, take more advantage from the installation of WHR systems when compared to two-stroke engines. As previously pointed out in literature this leads to WHR systems performing better in the former case (see for instance Theotokatos & Livanos (2013), who showed that the achievable efficiency increase in the case of two-stroke and four-stroke engines was in the range of 0.4%-1.4% and 3.0%-3.3% respectively).

Engine size is also an important factor, although this generally does not appear from the simulations. Steam turbines become inefficient at low power levels ($<$ 1 MW Invernizzi (2013)) and, in general, the performance of every component decreases with size. In this sense, WHR applications are generally more convenient for larger vessels.

Chapter 7

Outlook

Future research and recommendations to stakeholders

7.1 Suggestions for future research

Research is just as much about finding new questions to ask as it is about replying to known questions.

- The extent of methods for energy systems analysis departing from the 2^{nd} law of thermodynamics goes beyond what proposed in this work. More advanced methods for exergy analysis, such as those looking at **endogenous** and **exogenous**, **avoidable** and **unavoidable exergy losses**, as well as **exergoeconomic analysis**, could be applied for further improving the insight of the ship energy systems.

- The work presented in Paper III suggests that the operational envelope of the main engines installed on board limits the possibilities for operating the whole propulsion system in optimal conditions. Future research should investigate alternative solutions for broadening the range of engine operations, such as **variable geometry turbine (VGT)** and **sequential turbocharging**, and their effect on the efficiency of the whole propulsion system.

- This work focused on the steady-state performance of ship energy systems. Although most ships operate in constant conditions for long periods of time, they still require a **control system**. Complex energy systems such as those presented in this paper are challenging from a control perspective, and future research should look further into optimal control strategies for hybrid power plants and waste heat recovery systems.

- This work focused on three main energy demands: propulsion, auxiliary electric power, and auxiliary heat. In many applications, the demand for **refrigeration** is also relevant. The existence of systems such as absorption coolers provides additional challenges to the integrations of such systems with the rest of the on board energy systems for optimal efficiency.

- The existence of a number of heat sources and heat sinks suggests that benefits could be achieved through the use of **process integration**. Process integration is a collection of methods aimed at finding the network of heat exchangers that minimises the need for external heating and cooling given a set of heat and cooling demands to be fulfilled. Process integration could prove particularly useful in those cases where there is a large and diversified heat demand on board.

- In this work, every study involved the generation of "ad-hoc" algorithms and models. This approach was considered to be suitable given the specific conditions of this work. However, with a long-time perspective in mind, the approach to energy systems modelling should become more systematic. In particular, the development of a **standard, flexible modelling platform** to be used for the implementation of different sub-models and for the simulation of different conditions is considered as a necessity if a research group aims at strategically invest in this field.

- More research should be performed in the future to improve the understanding of **ship auxiliary energy demand**, both electric and heat. This would allow including these parts of the energy systems in the retrofitting process by improving their efficiency and their integration into the system. This step is seen as a requirement for improving the potential for optimising the full energy system, rather than keeping the focus on the propulsion system.

- In this work, different solutions where proposed for improving the efficiency of ship energy systems. The optimisation of engine-propeller interaction proposed in Paper III led to estimated savings of approx. 2%. The WHR system proposed in Paper V was expected to provide up to almost 11% savings. However, in the future, the demand for reduction of ship fuel consumption will achieve another level of magnitude, as ships will be expected to consume 50-90% less fuel as they do today. Research should therefore also focus on more **radical ship designs**, such as wind propulsion, utilisation of fuel cells, or improvements in the logistic chain to allow for slower sailing speeds.

- The process of modelling of ship energy systems, as any modelling effort, involves many uncertainties, both in relation to the system's inputs and to the behaviour of individual parts of the system. Therefore, it is here suggested that for making the process of design, evaluation and optimisation of ship energy systems more accurate and complete, it should also involve **stochastic modelling**[1].

[1] This aspect was briefly investigated during the thesis, leading to a poster publication (Baldi, 2015).

7.2 Recommendations to stakeholders

- The presented methodology for energy and exergy analysis allows achieving an increased insight of the ship energy system, as a consequence of both the analysis of the results and of the process of generating them. It could therefore be applied as part of a routine for improving ship energy efficiency, and particularly as a milestone of a **SEEMP**.

- Future ships will be designed by naval architects, and in this sense it will be important that technologies for sustainability will be given a much higher focus in their **education**. The same should apply for skills related to data analysis for future ship operators.

- Knowledge is power, and **data analysis** is a good way to achieve knowledge. Although a promising trend can today be observed in the shipping industry, shipping companies should make sure that they invest enough resources in taking relevant measurements on board and in their analysis to keep control on ship performance.

- Whenever considering retrofitting options on their vessels, shipowners should make sure that the **influence of the new component on the rest of the ship energy system** is correctly investigated, as well as their behaviour in **all the expected operational conditions** of the ship. This will require a deep understanding of the energy system, and the development of holistic ship models will be a useful tool in this direction.

- **Policies and decisions** based only on the performance of a vessel in its design point (i.e. the EEDI) should be questioned in their validity and possibly improved, in order to better account for **how a ship is operated in reality** and, therefore, provide a more accurate evaluation of a ship's performance.

Chapter 8

Conclusion

The aim of this thesis was to contribute to the subject of improving ship energy efficiency by answering the question "what is there to gain by looking at this matter from an energy systems engineering perspective?".

The study was based on two case studies, a chemical tanker and a passenger vessel, and was divided in two main parts. First, it included an in-depth analysis of the energy systems of two existing vessels based on the available information in terms of on board measurements and technical documentation, leading to improve the understanding of the system. The process included the use of energy and exergy analysis as structured, systematic methods to investigate the energy flows on board.

In a second part, improvements to energy efficiency were proposed and evaluated: variable propeller speed operations (Ship-1), waste heat recovery (Ship-1), and hybrid propulsion (Ship-2). The evaluation was based on accounting also for the performance at off-design conditions, and on focusing on interactions between different parts of the system. This was achieved by building ad-hoc mathematical models for each study, and by using the models to simulate the performance of the system in different conditions.

The results of this thesis confirmed the initial hypothesis, that looking at the energy system of the ship with a systems perspective leads to an increased understanding of the system, to a more accurate estimation of the benefits deriving from the installation of additional components and to the achievement of higher energy savings:

Energy and exergy analysis are a good complement to existing methods and practices, and constitute a structured and systematic way to gather information concerning the ship's energy systems, thus allowing improving the understanding of these systems. This comes as a consequence of the results of the analysis, in terms of energy and exergy flows and efficiencies, but also of the process itself of gathering and processing data and information concerning the ship under study.

Accurate and reliable measurements on board are a crucial requirement for providing an accurate and, hence, useful analysis of the system, which can in turn be used for its improvement. From the experience gathered from the two case studies it can be concluded that there is need for more focus on measuring thermal energy demand on board, and on adding details to the limited knowledge of the electric energy demand.

Thermal energy is an important part of the analysis, and including this element in the system analysis and synthesis process could lead to remarkable fuel savings. This is particularly true for ships like passenger vessels, where this energy demand represent a large share of the total. In addition, the results of this thesis showed that **waste heat** is available on both of the two vessels investigated in this thesis. The potential for improving ship energy efficiency through the application of WHR systems was estimated to be in the range of approximately 5-10% when taking advantage of the wasted heat that is the easiest to recover, but that savings of up to 15% could be foreseen in the case higher levels of heat integration were achieved.

Interactions among the different parts and the **operational profile** must be taken into account when dealing with the analysis of ship energy systems. This allows improving the accuracy of the evaluation of design or retrofitting options: if only individual parts of the systems are considered, or if the system is only evaluated at one operating condition, there is the risk of sub-optimisation and of providing an inaccurate estimation of the expected savings. The work presented in this thesis reinforced this view by providing examples of situations where the systems approach brings a clear advantage: the interaction between the propeller and the engines (estimated savings by improved practice: 1.9%), the installation of a WHR system (from 9.0% to 10.8%), and the optimal energy management of a hybrid propulsion system.

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Postface

Whenever I read other people's theses, this is how they appear to me: rational; logically structured; straight forward in their approach, from problem identification, to background and method, through the results and discussion and finally to the conclusion. They look like they were carefully planned in detail from the beginning. Like someone had sat down on day one and drew the plan for how all the work would be done in the coming five years.

I also wrote my thesis according to the same principle, as this is what is normally required by the academic community. If I succeeded, after reading my thesis you will think that I had a plan. That I had a clear picture of the problem to be solved, of how to solve it, and then I simply started building up my models and analysing my data, which of course was carefully gathered according to the initial plan.

Well, that's not how it went.

The whole process has been messy, to say the least. I started by thinking I could easily model alone the whole ship, and that I would start from the Diesel engine as it is, clearly, the most important part. That's how I ended up spending a good part of the first two years of my PhD, stubbornly polishing my model to the finest detail, keeping repeating myself everyday "OK, this is the last day I work on this, tomorrow I will start with something else". And if you look at the contribution of this part of the work to the final thesis, you probably will feel like "well, I actually missed that". And it's not your fault, because it is hidden in one of the papers, outside of the main scope of this thesis.

The data came thanks to Hannes Johnson, who had a good collaboration with Laurin Maritime, and to the fact that they had just decided to install an on board monitoring system when I started my PhD. This came very handy, but it was not planned. Also when it comes to the second dataset: it might look like it was all well prepared, but hadn't I met Fredrik Ahlgren right after my Licentiate, and hadn't we found out that he had a lot of data that we could use together for something interesting, that whole half of my PhD thesis would not be there.

So, if you are a PhD student and, reading these words, will think "oh, really? Because in my case, everything worked smoothly according to the plan", then I can tell you that you are lucky, because that makes things much easier. But if you are a student that, reading my thesis felt ”oh, damn, this looks so logical and consequential, my research instead is a mess”, than my message is: don’t worry, it is normal. That’s how research works most of the times (would it be really research if you already knew from the beginning what to expect?), especially for PhD students’ research. We are STUDENTS, so we are supposed to learn, and make mistakes in the process.

One more thing. Many people say that the PhD thesis is the final result of five years of a PhD student’s work. That all your work as a PhD student is included, summarised there, in that thick bunch of text, tables and figures.

Well, that’s wrong.

The five years of my PhD are way, way more than what you can read in my thesis. And I am not only talking about the ”other publications”, each only briefly mentioned in the beginning of this thesis, each requiring months of work and effort. I have been to conferences and met people, present and future researchers to collaborate with. I have been on board of real ships, talked to the crew, learned about their experiences and lives. I have talked with other PhD students in the department, learning about all sorts of things such as VTS, biofouling, effects of oil spills on meiofauna, social aspects of implementing energy efficiency, and much more. I have planned the structure of a whole MSc course on marine propulsion systems, something I knew nothing about only 5 years ago. I have supervised students on a variety of subjects, ranging from hybrid propulsion systems to cost-benefit analysis of shore connection. I have applied for many different scholarships for doing anything from going to conferences to financing my networking. I have learned a new language, and have become part of new communities (both the shipping and the Swedish ones). I participated to the organization of two conferences and to the redaction of a book. I have made three posters. I have taken courses on design of experiments, on leadership, on project management, on programming and on data analysis. Most importantly, I have (hopefully) learned about what it means to be a researcher, about how to channel my inner curiosity, how to critically assess information and knowledge, how to proceed to transform a simple question to something that will contribute to human knowledge.

So, if you are a PhD student and you are reading this postface, here’s my advice. Remember, always, that the final result of your PhD is not your thesis. It is not your papers either.

It is you.

Therefore go out, don’t be afraid to make mistakes; try, experience, learn, knowing that even if doing this might not contribute to writing a better thesis, it will probably help in making you a better researcher.

Acknowledgements

The journey to becoming a doctor of engineering is a long and, often, lonely one. And yet, when looking back at these past five years, it feels that this journey would not have been the same fun, or even possible, hadn't it been for those many, amazing people that I have had the honour to share this journey with.

First and foremost, I must thank Karin and Cecilia who entrusted me with this project. The journey through a PhD can lead to frustrating moments, but now that I find myself at the end of it, with a thesis written and printed, and with a green tick on almost all the items of the researcher check-list, I realise that this cannot be a coincidence. So thank you Karin, thank you Cecilia, for being able to understand me and to guide me the way I needed.

Big thanks also go to Gerasimos "Makis" Theotokatos, for all the time you spent with me discussing about Diesel engine models, but also for the beers we have been drinking together in the rainy Glasgow. The success of my Scottish expedition depended a lot on your contribution. I must also thank all the welcoming and professional people I have met in my collaboration with Laurin Maritime and Marorka, and particularly Bengt-Olof Petersen, Pär Brandholm, Stefán Gunnsteinsson, Kristinn Aspelund and Jon Agust Thorsteinsson.

It is much more fun to do research and write papers in good company, and I must thank my co-authors for making this whole process more interesting and motivating. Thank you T.V., Andrea, Hannes, Fredrik and Ulrik, because when I met you I realised that I had not only found skilled researchers to work with, but also great friends with whom it was just fun to spend time with. A big thank also goes to Francesco, whose enthusiasm for new opportunities gave me the chance to "go back to the origins".

Although you "betrayed" me by moving to Malmö, it would still be unfair not to have a special mention for you, Gesa. Even though sometimes your straight comments and criticism were hard to digest, they have been crucial for me to get where I am. Of all the people I have met since I moved to Sweden, you are the one who contributed the most, both professionally and personally, to my experience. So thank you, from the bottom of my heart.

Great thanks also to all the people who accompanied me at work during these five years. How to name them all, without making anyone upset for having been forgotten? I'll give it a try, in as random order as possible: Carolina, Hanna, Hannes, Elma, Hiba, Florian, Luis, Anna, Maria, Henrik, Ulrik, Johannes, Hedy, Nicole, Steven, Lesly, Mathias, Philip, Selma, Erik, ... All of you have been, in a way or another, crucial for my experience as a PhD student, sharing ideas, fears, jokes, thoughts, frustration and all sorts of emotions that fill a PhD student's mind and life. The same, big thanks also go to all the crew at Strathclyde University, and particularly to Dennj, Charlotte and Ruhan.

How couldn't I thank all my friends, all those who made my life outside the walls of the university something great to live for? Whether you are now sitting in Gothenburg, Bologna, Glasgow, Paris, Bern, or whatever other part of the world, I would like to thank you all. You did not write papers with me. You did not do any exergy analysis with me, and you probably do not even know what it is. And yet, I would not be here without you. Thank you all!

Infine, ultimi come sempre accade per le cose piú importanti, un grazie grande grande a papá e mamma, perché in cinque anni, no, anzi, in trenta anni di vita in cui tutto cambia giorno dopo giorno, voi siete stati il mio faro nella tempesta, quelle persone su cui, al di sopra di ogni altre ho sempre potuto contare. E per questo, come per tutto il resto, non posso che dirvi grazie.

Appended papers

Paper I

International Journal of Thermodynamics (IJoT)
ISSN 1301-9724 / e-ISSN 2146-1511
www.ijoticat.com

Vol. 18 (No. 2), pp. 82-93, 2015
doi: 10.5541/ijot.70299
Published online: June 1, 2015

Energy and Exergy Analysis of Ship Energy Systems – The Case study of a Chemical Tanker

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Received 25 September 2014, Accepted 17 April 2015

Abstract

Shipping contributes today to 2.1% of global anthropogenic greenhouse gas emissions and its share is expected to grow together with global trade in the coming years. At the same time, bunker prices are increasing and companies start to feel the pressure of growing fuel bills in their balance sheet.

In order to address both challenges, it is important to improve the understanding of the energy consumption trends on ships through a detailed analysis of their energy systems. In this paper, energy and exergy analysis are applied to the energy system of a chemical tanker, for which both measurements and technic knowledge of ship systems were available. The application of energy analysis to the case-study vessel allowed for the comparison of different energy flows and therefore identifying system components and interactions critical for ship energy consumption. Exergy analysis allowed instead identifying main inefficiencies and evaluating waste flows.

Results showed that propulsion is the main contributor to ship energy consumption (70%), but that also auxiliary heat (16.5%) and power (13.5%) needs are relevant sources of energy consumption. The potential for recovering waste heat is relevant, especially from the exhaust gases, as their exergetic value represents 18% of the engine power output.

Keywords: *Energy analysis; exergy analysis; shipping; energy efficiency.*

1. Introduction

As shipping is facing a number of challenges related to increased fuel costs and stronger focus on environmental impact energy efficiency is more and more a subject of study. In this condition, however, detailed studies on energy generation, use and losses on board, together with similar evaluations related to exergy, are lacking in existing scientific literature.

1.1 Background

International trade is the core of today’s economy and lifestyle. Its size, compared to 1950, is today more than 100 times larger in terms of volume and value of goods transported [1]. In this picture shipping, which is responsible for between 80% and 90% of the overall global trade [2] has a crucial role in global economy and, more in general, in all human activities.

However, shipping is now subject to a large number of important challenges. Bunker fuel prices are today three times higher than they were in the 80's [3], and fuel costs are estimated to account for between 43% and 67% of total operating costs depending on vessel type [4]. Moreover, upcoming environmental regulations on sulfur oxides, nitrogen oxides and greenhouse gases (shipping is estimated to contribute to 2.1% of global anthropogenic GHG emissions [5]) will exert an additional leverage on fuel costs [6]. This phenomenon will be more pronounced in emission controlled areas, i.e. USA coastal waters, the Baltic Sea, and the North Sea, where regulations will be stricter.

Various fuel saving solutions for shipping are available and currently implemented. Operational measures include improvements in voyage execution, engine monitoring, reduction of auxiliary power consumption, trim/draft optimization, weather routing, hull/propeller polishing, slow-steaming. Design related measures can relate to the use of more efficient engines and propellers, improved hull design, air cavity lubrication, wind propulsion, fuel cells for auxiliary power generation, waste heat recovery, liquefied natural gas as fuel, pump frequency converters, cold ironing [7]. Several scientific studies have been conducted on these technologies, and a more detailed investigation would be out of the scope of this work.

Even if efforts have been put in order to evaluate the benefits associated with the use of each of these solutions and of their combined effect [7], [8], it has also been acknowledged that the world fleet is heterogeneous; from the perspective of a ship owner or operator, measures need to be evaluated on a ship-to-ship basis [9]. In this process, a deeper understanding of energy use on board of the specific ship is vital.

1.2 Previous Work

Some studies presenting the analysis of ship energy systems can be found in literature. Thomas et al. [9] and Basurko et al. [10] worked on energy auditing fishing vessels; Shi et al. [11], [12] proposed models for predicting

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ship fuel consumption in design and off-design conditions; Balaji and Yaakob [13] analyzed ship heat availability for use in ballast water treatment technologies. However, a more thorough, holistic thermodynamic analysis of a ship, such as that proposed by Nguyen et al. [14] for oil platforms, is, to the best of our knowledge, lacking in scientific literature. The work proposed by Zaili and Zhaofeng [15], though looking in the right direction, still does not represent the required level of detail as they only focus on the main engines and propose an analysis based on design values rather than on measured data.

Analyses based on the First law of thermodynamics lack insight of the irreversibilities of the systems, as well as of the different quality of heat flows, since they do not account for the additional knowledge provided by the Second law of thermodynamics [16]. Exergy analysis, which is based on both the First and the Second laws of thermodynamics, can help addressing this shortcoming. Widely used in other industrial sectors, exergy analysis in not commonly employed in maritime technology studies, and is mostly related to waste heat recovery systems [17], [18] and refrigeration plants [19], [20].

1.3 Aim

The aim of this paper is to provide a better understanding of how energy is used on board of a case study vessel and where the largest potential for improvement is located by performing an energy and an exergy analysis of a the ship's energy systems. Compared to what can be found in the scientific literature, the present research presents elements of novelty, because it:

- Is based on a combination of measurements and design information.
- Embraces all ship energy systems.
- Analyses energy input, output, and internal energy flows.
- Focuses on both energy and exergy analysis, hence including considerations about energy quality.

2. Methodology

The methodology employed in this work consists in the analysis of measured operational data with the aid of technical knowledge of the system and theoretical principles whenever measured data are not available or the quantity of interest is not directly measureable.

2.1 Exergy Analysis

When dealing with energy flows of different nature, energy analysis alone can lead to misleading results, as it does not account for energy quality. This problem can be partially overcome by the use of exergy analysis. Exergy is defined as the maximum shaft work that can be done by the a system in a specified reference environment [16]. The exergy content of a flow depends on the quality of the energy content. Additionally, differently from energy, exergy is not conserved and can be destroyed, representing the deterioration of energy quality.

The exergy content of a material flow is generally divided in four parts: physical, chemical, kinetic and potential. Potential and kinetic exergy flows coincide with their energy counterparts. In the case of chemical exergy, substantial differences can be found when analyzing systems involving a more advanced chemistry; in this case combustion is the only chemical reaction taken into account, and it is assumed that the specific chemical exergy content of the fuel can be calculated as suggested by [21] based on its LHV and its H/C ratio. Finally, the physical component of an exergy flow is defined as showed in Eq. (1).

$$\dot{B}_{ph} = \dot{m}[(h-h_0)+T_0(s-s_0)] \tag{1}$$

where $\dot{B}$, h, and s respectively stand for exergy flow, specific enthalpy, and specific entropy, while the subscript 0 refers to reference conditions, which in this work coincide with measurements of seawater temperature.

Energy flows that are not associated to material stream flows are also associated to a corresponding exergy flow. In the case of work and electricity the exergy exchanged coincides with the correspondent amount of energy; in the case of heat, the exergy exchanged depends on the temperature at which the exchange takes place, according to Eq. (2):

$$\dot{B}_{heat} = \dot{Q}\left(1-\frac{T_0}{T}\right) \tag{2}$$

With reference to an open system, the exergy balance of the system can be expressed in accordance with Eq. (3):

$$\dot{B}_{in} = \dot{B}_{out} + \dot{I} \tag{3}$$

where $\dot{B}_{in}$ and $\dot{B}_{out}$ represent the flow of exergy entering and leaving the component, respectively. The term $\dot{I}$ is known as irreversibility rate (or exergy destruction) and can be calculated, in its general form, as:

$$\dot{I} = T_0\dot{S}_{gen} \tag{4}$$

where $\dot{S}_{gen}$ represents the entropy generation rate in the component.

Accounting for the second law of thermodynamics allows for a large number of possible definitions of efficiency, and there is limited agreement in the scientific community concerning what exergy-based efficiencies are to be used in these analyses. In this study, four different quantities measuring efficiency according to exergy analysis will be used based on the work of Kotas [16] and Lior and Zhang [22]:

- The *total exergy efficiency* (ε_t) is used in this study as defined by [22] according to Eq. (5)

$$\varepsilon_t = \frac{\sum \dot{B}_{out}}{\sum \dot{B}_{in}} \tag{5}$$

where the subscripts *out* and *in* respectively refer to outputs and inputs. As suggested by Kotas [16] and originally proposed by Bruges [23], in the case of heat exchangers Eq. (5) can be interpreted as presented in Eq. (6) by assuming the reduction in exergy of the hot stream as the input to the system and the increase in exergy of the cold stream as the desired output:

$$\varepsilon_t = \frac{\dot{B}_{c,out} - \dot{B}_{c,in}}{\dot{B}_{h,out} - \dot{B}_{h,in}} \tag{6}$$

- The *task efficiency* (ε_u) is used in this study as defined by Lior and Zhang [22] according to Eq. (7).

$$\varepsilon_u = \frac{\sum \dot{W}_u - \sum \dot{W}_p + \sum \dot{B}_{h,u} + \sum \dot{B}_{c,u}}{\sum \dot{B}_{h,p} + \sum \dot{B}_{c,p} + \sum \dot{B}_{ch,p}} \tag{7}$$

where the subscripts *u*, *p*, *h* and *c* represent the "useful" output of the system, the "paid" input to the system, heating and cooling flows. In this study, the equation originally proposed by Lior and Zhang [22] was adapted by also including fuel exergy inputs to the denominator of the fraction. The task efficiency is not used for heat exchangers, in this study, as depending on whether it is applied to a heater or a cooler the result would be $\varepsilon_u = \varepsilon_t$ or $\varepsilon_u = \varepsilon_t^{-1}$, none of which would add significant contribution to the analysis.

- The *efficiency loss ratio* (δ) is used according to the definition proposed by Kotas [24] and represents the proportion of the exergy input to a component that is lost due to irreversibilities:

$$\delta = \frac{\dot{I}}{\sum \dot{B}_{in}} \tag{8}$$

In the case of heat exchangers, the difference $\dot{B}_{h,out} - \dot{B}_{h,in}$ is used as denominator to the equation instead in order to be consistent with the definition of total exergy efficiency.

- The *relative irreversibility* (γ) is defined as the ratio between the exergy destroyed in the component "i" and the total rate of exergy destruction in the whole system:

$$\gamma = \frac{\dot{I}_i}{\sum \dot{I}_i} \tag{9}$$

2.2 Ship Description

The ship under study is a Panamax chemical / product tanker. Relevant ship features are provided in Table 1, while Figure 1 conceptually represents the ship energy systems. Figure 2 gives a more detailed representation of the main engine systems, including the cooling systems. The ship is propelled by two 4-stroke Diesel engines (ME) rated 3,840 kW each. The two engine shafts are connected to a common gearbox (GB). One of the gears reduces the rotational speed from 600 rpm to 105.7 rpm, the design speed for the controllable pitch propeller.

Another shaft from the gearbox connects it to the electric generator (SG) which provides 60 Hz current to the ship. Additionally, two auxiliary engines (AE) rated 682 kW each can provide electric power when the MEs are not in operation, or whenever there is a failure in the SG. Auxiliary heat needs are fulfilled by the exhaust gas economizers (EGE) or by auxiliary boilers (AB) when the MEs are not running or heat demand is higher than what provided by the EGEs.

Table 1. Main Ship Features.

Dimension	Value
Deadweight	47,000 tons
Installed power (Main Engines)	7,700 kW
Installed power (Auxiliary Engines)	1,400 kW
Shaft generator design power	3,200 kW
Exhaust boilers design steam gen.	1,400 kg/h
Auxiliary boilers design steam gen.	28,000 kg/h

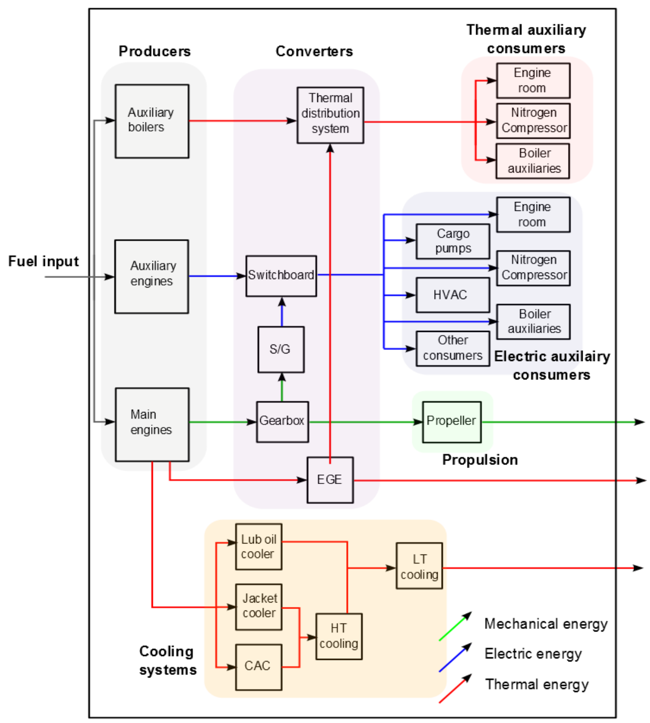

Figure 1. Conceptual representation of ship energy systems.

2.3 Data Gathering and Processing

The main source of measured data for the analysis is a continuous monitoring system (CMS) installed on board. Measurements are logged on board with a frequency of 1 to 15 s depending on the measured quantity. The raw data are sent to the energy management system provider, where they are elaborated and made available online to the company as 15 min averages. The 15 min averaged dataset was used for the analysis in this work.

These data were filtered in order to eliminate entries that showed to be clearly inconsistent (e.g. negative fuel flows). Unfortunately, as a consequence of not having access to the raw measurements, it was not possible to derive information in relation to measurement accuracy in addition to what provided by the shipyard (±0.1% for propeller speed, ±2% for propeller power, ±3% for main engines fuel flow). The analysis was therefore performed under the assumption that no relevant bias was present in the original data as a consequence of measurement inaccuracies.

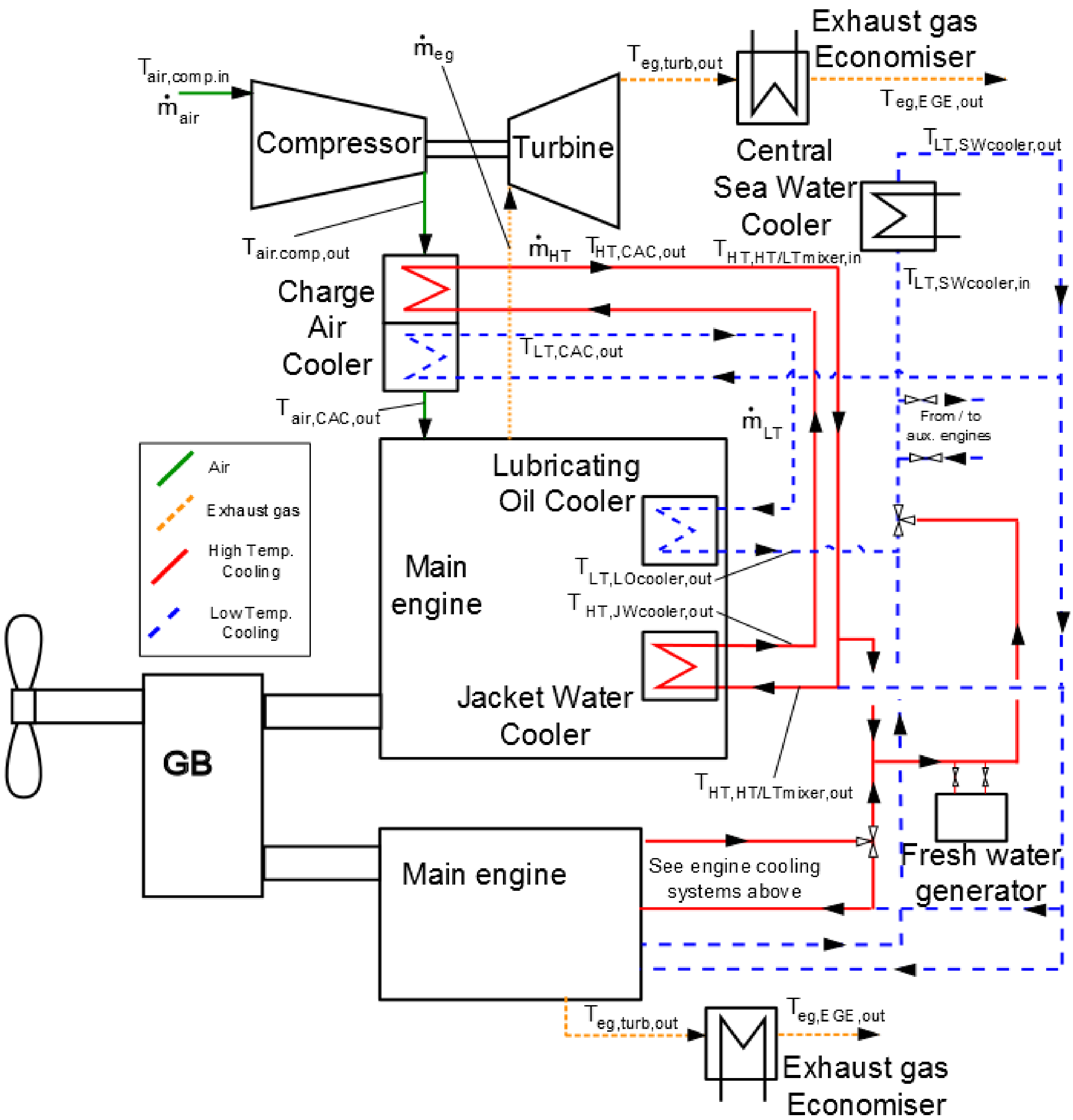

Figure 2. Conceptual representation of main engine systems.

Values available from the CMS were:

- Propeller torque
- Propeller speed
- Propeller power
- Engine fuel consumption
- Auxiliary generator power output
- Auxiliary engines fuel consumption
- Main engines fuel consumption
- Shaft generator power output
- Ship speed
- Sea water temperature
- Ambient temperature
- Ambient pressure

In addition to the aforementioned approximations, it should be noted the measurements in moments of highly dynamic behavior (i.e. maneuvering) were filtered out from the averaged dataset. This was done as a consequence of clear inconsistence in the calculated engine efficiency, which is apparently generated by the averaging process. The amount of data points filtered out of the database sum up to a negligible amount of the total (0.8%) and does therefore not influence the reliability of the final results.

In addition to logged measurements, technical documentation was available for on board machinery and was used as input for numerical regressions: heat and electric balance of the ship were provided by the shipyard; ship sea trials performed by the shipyard when the ship was first sailed and direct communication with on board and onshore personnel were also available.

Engine properties are based on measurements of power, speed and fuel mass flow and on empirical polynomial regressions based on information provided by the engine manufacturer. A detailed accounting of all relationships and assumptions employed in this study in order to process the raw measured data are shown in Tables A1 to A3 in Appendix A. Table 2 shows the values taken by the main engine parameters given specific measured inputs of power and fuel flow rate; exergy flows from the engine are similarly shown in Figure 3.

*Table 2: Calculated Engine Temperatures and Flows for Different Total Main Engines Power. Values Marked with * Are Calculated in the Table, But Measured in the Application of the Model to the Case Study.*

Power [kW]	1500	2500	3500	4500	5500	6500	7500
# Engines running	1	1	2	2	2	2	2
Engine load	0.39	0.65	0.46	0.59	0.72	0.85	0.98
Engine bsfc [g/kWh]	224*	206*	218*	209*	204*	203*	207*
$\dot{m}_{air}\left[\frac{kg}{s}\right]$	2.8*	4.6*	6.5*	8.3*	10.2*	12.1*	13.9*
$T_{air,Comp,in}\left[K\right]$	308	308	308	308	308	308	308
$T_{air,Comp,out}\left[K\right]$	376	441	397	429	452	473	494
$T_{air,CAC,\mathrm{out}}\left[K\right]$	328	328	328	328	328	328	328
$\dot{m}_{eg}\left[\frac{kg}{s}\right]$	2.9	4.8	6.7	8.6	10.5	12.4	14.3
$T_{\mathrm{eg},Turb,in}\left[K\right]$	749	736	745	738	737	747	770
$T_{\mathrm{eg},Turb,\mathrm{out}}\left[K\right]$	687	614	664	627	605	595	600
$T_{\mathrm{eg},EGE,\mathrm{out}}\left[K\right]$	573	546	615	590	574	569	577
$T_{LO,\mathrm{LOcooler},in}\left[K\right]$	337	337	337	337	337	337	337
$T_{LO,\mathrm{LOcooler},out}\left[K\right]$	352	355	353	354	356	358	361
$T_{\mathrm{HT},JWcooler,in}\left[K\right]$	351	345	350	347	343	340	335
$T_{\mathrm{HT},JWcooler,out}\left[K\right]$	356	351	355	353	350	347	344
$T_{\mathrm{HT},CAC,\mathrm{out}}\left[K\right]$	358	358	358	358	358	358	358

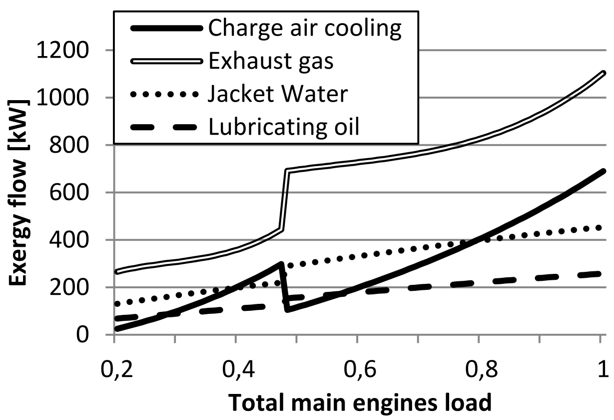

Figure 3: Calculated exergy flows for different values of total main engines power.

Auxiliary power consumption measurements are available from the CMS. These measurements, however, do not include details about the individual consumers. In order to give an estimation of the power needed by different consumers, information from the electric balance was used. Since the measured consumption is different from design figures, this operation required a number of assumptions:

- For seagoing mode (loaded), it is assumed that the power consumption is subdivided according to the electric balance. Therefore, proportions between different consumers are maintained. For all points where auxiliary load is larger than 500 kW nitrogen compressors are assumed to account for the additional consumption. Nitrogen compressors are needed for keeping an inert atmosphere into the cargo tanks when inflammable liquids are transported.
- For seagoing mode (ballast) the same repartition is assumed as for seagoing mode (loaded) if auxiliary power is lower than 500 kW. If power consumption is higher the difference is assumed to be connected to the operations of nitrogen compressors and boilers auxiliaries (in connection to tank cleaning), which are subdivided according to their respective design power.
- For maneuvering the same assumptions as for seagoing mode (loaded) are employed.
- For cargo loading and unloading all consumption going over 500 kW is allocated to nitrogen compressors and cargo pumps, with repartition according to maximum installed power. It should be noted that cargo loading operations normally do not require the use of cargo pumps, as port storage facilities can provide the needed overpressure for loading the cargo.
- For waiting time the same proportions as reported in the ship electric balance are used, with the exception of engine room consumption, which is halved, since when waiting in port only auxiliary engines are used.

Fuel heating is needed because of high fuel viscosity, and is computed starting from the design heat balance and using sea water temperature and outer air temperature measurements. Hotel facilities needs are calculated assuming a linear correlation between calculated values given in the heat balance, assumed at an outer temperature of 2°C, depending on outer air temperature. Heat consumption for fresh water generation is calculated including service water for machinery and cooling systems and consumption for the crew according to common practice [25]. Since the generation of fresh water is connected to the (HT) cooling systems, the value of heat of vaporization for water was taken at 50°C and equal to 2382 kJ/kg.

During ballast legs, saturated steam at 14 bar is needed for tank cleaning, which requires the operation of the auxiliary boilers. Energy use for tank cleaning is derived from the aggregated boiler fuel consumption, under the assumption of 90% boiler efficiency accounting for combustion losses and heat flow in the exhaust gas, limited at 200°C to prevent sulfuric acid condensation in the funnel. Auxiliary boilers are also used when the main engines are not in operation. In this condition, as boilers are operated at very low load, a reduced efficiency of 80% was assumed instead.

3. Results

3.1 Energy Analysis

Figure 4 shows the Sankey diagram of ship energy systems. Summaries of cumulated input and output energy flows over one year of ship operations are shown in Tables 3 and 4, while Table 5 presents an overview of all the ship flows analyzed in this study.

Propulsion represents the main source of energy consumption, as it accounts for 68% of the yearly ship energy demand. This also translates in the main engines consuming the largest share of the overall energy input of the system (87.9%). Hence, efforts directed towards the reduction of propulsive power are highly justified for the ship under study.

Both auxiliary engines and auxiliary boilers (respectively representing 8.0% and 4.1% of ship energy input) on one side, and auxiliary power and heat consumers (12% and 20% of ship energy demand respectively) on the other, should be given significant attention.

Boiler auxiliary electric demand should also be taken into account as it also represents a significant share of the total demand (2.7%).

Table 3: Summary of Input Energy Flows.

Input flow	Flow type	$\dot{E}\left[\frac{TJ}{year}\right]$	$\dot{E}\left[\%_{in,tot}\right]$
Fuel to MEs	Chemical	187.6	87.9%
Fuel to AEs	Chemical	17.0	8.0%
Fuel to boilers	Chemical	8.7	4.1%

Table 4: Summary of Output Energy Flows.

Output flow	Flow type	$\dot{E}\left[\frac{TJ}{year}\right]$	$\dot{E}\left[\%_{out,tot}\right]$
Propulsion	Work	67.7	31.7%
Tank cleaning	Heat	3.1	1.5%
Fuel heating	Heat	7.7	3.6%
Hotel facilities	Heat	9.6	5.4%
Nitrogen compressors	Electricity	2.1	1.0%
Cargo pumps	Electricity	0.8	0.4%
HVAC	Electricity	1.8	0.8%
Engine room	Electricity	1.5	0.7%
Boiler auxiliaries	Electricity	2.7	1.3%
Miscellaneous	Electricity	2.6	1.2%
Exhaust gas (ME)	Waste heat	45.9	21.5%
Exhaust gas (AE)	Waste heat	4.4	2.1%
Exhaust gas (AB)	Waste heat	1.4	0.7%
Radiated heat (ME)	Waste heat	6.2	2.9%
Sea water cooling	Waste heat	52.1	24.4%
Shaft losses	Waste heat	0.7	0.3%
SG losses	Waste heat	1.0	0.5%

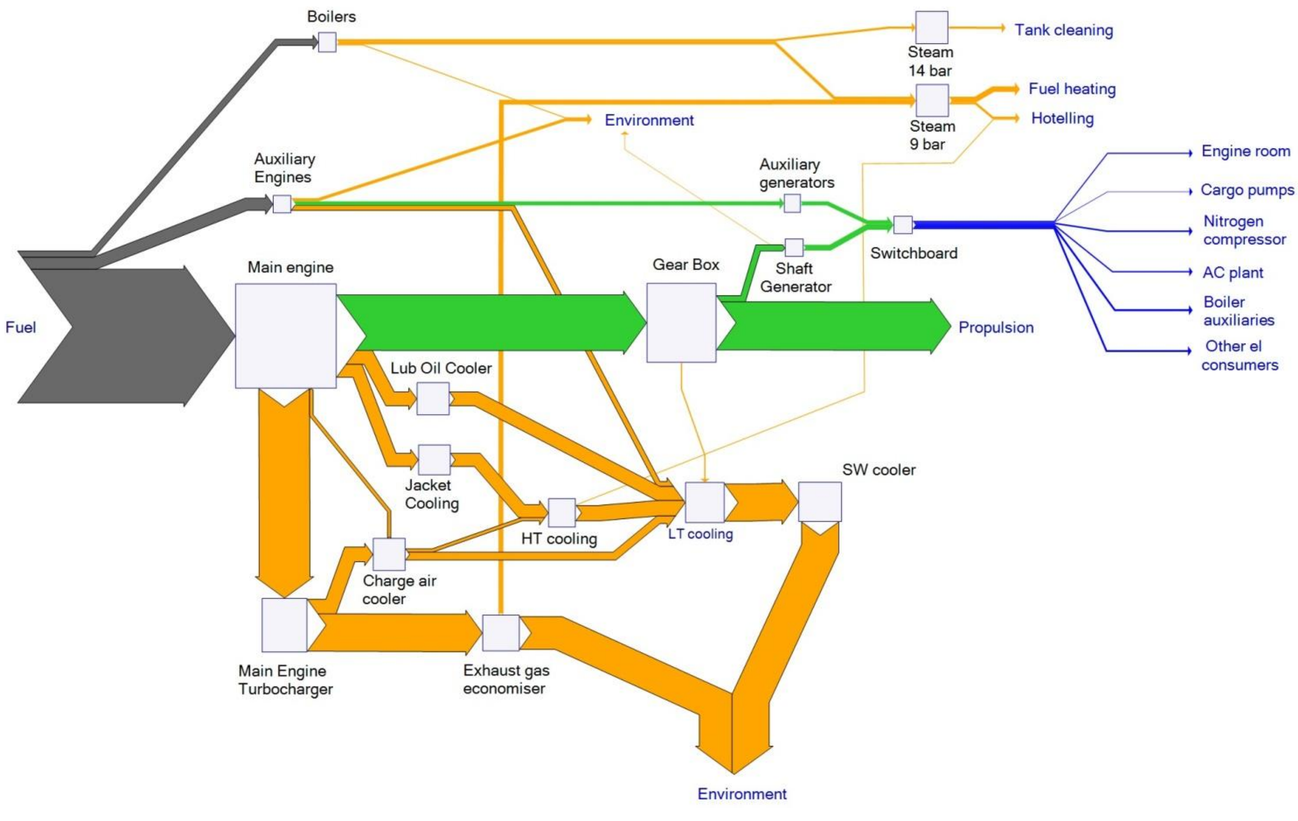

Figure 4. Sankey diagram of ship energy systems.

Table 5: Yearly Energy Flows for the Selected Case Study Vessel, in TJ/year.

cComponent	$\dot{E}_{ch,in}$	$\dot{E}_{ph,c,in}$	$\dot{E}_{ph,c,out}$	$\dot{E}_{ph,h,in}$	$\dot{E}_{ph,h,out}$	$\dot{E}_{w,in}$	$\dot{E}_{w,out}$	$\dot{E}_{q,in}$	$\dot{E}_{q,out}$
Cylinders (ME)	187.6	5.5	71.3	0.0	0.0	0.0	78.0	0.0	43.9
Turbocharger (ME)	0.0	1.8	20.4	71.3	52.6	0.0	0.0	0.0	0.0
Lub oil cooler (ME)	0.0	44.8	64.3	61.7	42.3	0.0	0.0	0.0	0.0
Jacket water cooler (ME)	0.0	148.0	166.2	0.0	0.0	0.0	0.0	18.1	0.0
CAC, HT stage (ME)	0.0	166.2	170.9	20.4	15.7	0.0	0.0	0.0	0.0
CAC, LT stage (ME)	0.0	33.8	44.8	15.7	4.7	0.0	0.0	0.0	0.0
LT/HT mixer	0.0	64.3	85.9	169.7	148.0	0.0	0.0	0.0	0.0
SW cooler	0.0	0.0	52.1	85.9	33.8	0.0	0.0	0.0	0.0
Exhaust Gas Economizer	0.0	2.5	9.3	52.6	45.9	0.0	0.0	0.0	0.0
Gearbox	0.0	0.0	0.0	0.0	0.0	78.0	76.2	0.0	1.8
Shaft generator	0.0	0.0	0.0	0.0	0.0	8.6	7.8	0.0	0.8
Switchboard	0.0	0.0	0.0	0.0	0.0	13.5	13.4	0.0	0.1
Boiler	8.7	2.9	10.2	0.0	1.4	0.0	0.0	0.0	0.0
Tank cleaning	0.0	0.0	0.0	4.5	1.3	0.0	0.0	0.0	3.1
Fuel heating	0.0	0.0	0.0	10.6	2.8	0.0	0.0	0.0	7.7
Hotel facilities	0.0	0.0	0.0	4.4	1.2	0.0	0.0	6.4	9.6
Auxiliary engines	17.0	0.0	4.4	0.0	0.0	0.0	5.7	0.0	7.0

Auxiliary boilers are run at low load most of the time, leading to low efficiency. Fuel heating also represents a surprisingly high share of the overall ship energy consumption (7.8%). This high influence of auxiliary needs is partly connected to the ship spending large amount of time in port, when there is no propulsion power demand.

Finally, a large amount of energy is wasted to the environment through the exhaust gas (21.5% of total ship energy output), and the SW cooler (24.4%). This suggests that there is potential for the recovery of these waste flows. The amount of energy available in the cooling systems can however be evaluated more consistently using exergy analysis so to also account for the different energy quality of the available cooling flows.

3.2 Exergy Analysis

The results from the exergy analysis are presented graphically in Figure 5; a summary of exergy based efficiencies is presented in Table 6; Tables 7 and 8 present input and output exergy flows; Table 9 finally shows the detail of the exergy flows between components onboard.

The analysis of exergy flows shows a different picture from the energy analysis. Heat demand accounts for only 3.0% of the total onboard exergy demand, while propulsion (83%) and auxiliary power (14%) represent a higher relative share of the total demand.

Looking at waste flows, the results suggest that the main engine exhaust are by large the main source of exergy loss onboard (14.1% of total ship exergy output). Exergy losses from sea water cooling are negligible.

Exergy efficiency helps understanding which components make the best use of the quality of their energy input. It can be seen, for example, that according to this definition, boilers (ε_t =36.3%) are much less efficient than both main (ε_t =59.2%) and auxiliary engines (ε_t =53.0%). This holds true when looking at task efficiency (ε_u), although the difference is smaller.

A further analysis of the cooling systems allows the identification of where the largest amount of exergy is destroyed. All the different coolers present a significant contribution of onboard exergy destruction, which sums up to 10.1% of the total. These irreversibilities could potentially be reduced thus providing an additional source of heat for energy recovery. When calculated at the engine output, the total amount of exergy available for recovery accounts for 10 TJ/year, which is comparable to the amount available from the exhaust gas (13.8 TJ/year).

These results suggest that three is a significant potential for improving the efficiency of the energy system by enhancing the recovery of waste heat. Waste heat recovery (WHR) systems for heat-to-power conversion are often proposed for enhancing marine propulsion systems efficiency [18], [26]–[28] . In this context exergy analysis, compared to energy analysis, provides a more accurate estimate of the amount of power that could be generated through a WHR system.

The analysis of the total exergy efficiency (ε_t) allows identifying where the aforementioned potential for improvement is larger. The LT stage of the CAC (ε_t =25.5%) appears to be the one where the highest potential for improvement is located, followed by the HT/LT mixer (49%). Other coolers have efficiencies included between 52% and 55.5% (see Table 6).

In practice, however, these improvements would require larger heat exchangers, at the cost of an increased capital investment. This work focuses on a thermodynamic analysis of ship energy systems; methods for thermoeconomic analysis and optimization have been proposed in literature and should be employed in further developments of this work (e.g. by Szargut and Sama [29]).

The relatively high total exergyu efficiency of the EGE (67%) was somewhat unexpected, since it generates relatively low pressure steam (9 bar, 448 K saturation temperature) at the expense of heat at higher temperatrure in the exhaust gas (between 650 and 550 K, see Table 2). It should be noted, however, that among all the heat exchanger analysed in this work, the EGE is the only one that has a heating (rather than cooling) function. This suggests that it should not be directly compared with other exchangers meant for different purposes.

Heat demands for tank cleaning and fuel heating also involve a high rate of exergy destruction. In the first case, 14 bar steam generated by the auxiliary boilers is used to warm up water from 50 to 85ºC, which represents a clearly inefficient exchange; in the same way, the use of 9 bar steam for fuel heating, which mostly happen at temperatures comprised between 50 and 90ºC, is clearly identified by the exergy analysis as a potential source for improvement. In the case of hotel facilities, the use of HT water for freshwater generation increases the overall efficiency significantly. This could be done, for example, by using a different heat transfer fluid or, in alternative, steam at a lower pressure. Fuel handling and hoteling, for instance, only require temperatures as low as 70-80°C (a part from fuel heaters before the engine, which warm HFO up to around 90-100°C), which could be provided at much lower temperature than by 9 bar steam.

Table 6: Exergy-based Efficiencies of Different Ship Components (%).

Component	ε_t	ε_u	δ	γ
Cylinders (ME)	59.2	41.5	40.8	65.8
Turbocharger (ME)	35.6	-	64.4	5.8
Lub oil cooler (ME)	52.0	-	48.0	1.2
Jacket water cooler (ME)	53.7	-	46.3	2.2
CAC, HT stage (ME)	55.5	-	44.5	0.6
CAC, LT stage (ME)	25.5	-	74.5	1.3
LT/HT mixer	49.0	-	51.0	1.9
SW cooler	2.5	-	97.5	3.5
Exhaust Gas Economizer	67.0	-	33.0	1.0
Gearbox	98.3	97.7	1.7	1.1
Shaft generator	93.2	90.7	6.8	0.5
Switchboard	99.3	99.0	0.7	0.1
Boiler	36.3	28.0	63.7	5.1
Tank cleaning	25.3	-	74.7	0.7
Fuel heating	26.2	-	73.8	1.7
Hotel facilities	51.1	-	48.9	0.7
Auxiliary engines	53.0	33.5	47.0	6.9

Table 7: Summary of Input Exergy Flows.

Input flow	Flow type	$\dot{B}\left[\frac{TJ}{year}\right]$	$\dot{B}\left[\%_{in,tot}\right]$
Fuel to MEs	Chemical	199.6	87.9
Fuel to AEs	Chemical	18.1	8.0
Fuel to Boilers	Chemical	9.3	4.1

Table 8: Summary of Output Exergy Flows.

Output flow	Flow type	$\dot{B}\left[\frac{TJ}{year}\right]$	$\dot{B}\left[\%_{in,tot}\right]$
Propulsion	Work	67.6	69.0
Tank cleaning	Heat	0.9	0.9
Fuel heating	Heat	0.7	0.7
Hotel facilities	Heat	0.9	0.9
Nitrogen compressors	Electricity	2.1	2.1
Cargo pumps	Electricity	0.8	0.8
HVAC	Electricity	1.8	1.8
Engine room	Electricity	1.5	1.5
Boiler auxiliaries	Electricity	2.7	2.8
Miscellaneous	Electricity	2.6	2.7
Exhaust gas (ME)	Waste heat	13.8	14.1
Exhaust gas (AE)	Waste heat	1.9	1.9
Exhaust gas (AB)	Waste heat	0.2	0.2
Radiated heat (ME)	Waste heat	0.0	0.0
Sea water cooling	Waste heat	0.1	0.1
Shaft losses	Waste heat	0.2	0.2
SG losses	Waste heat	0.2	0.2

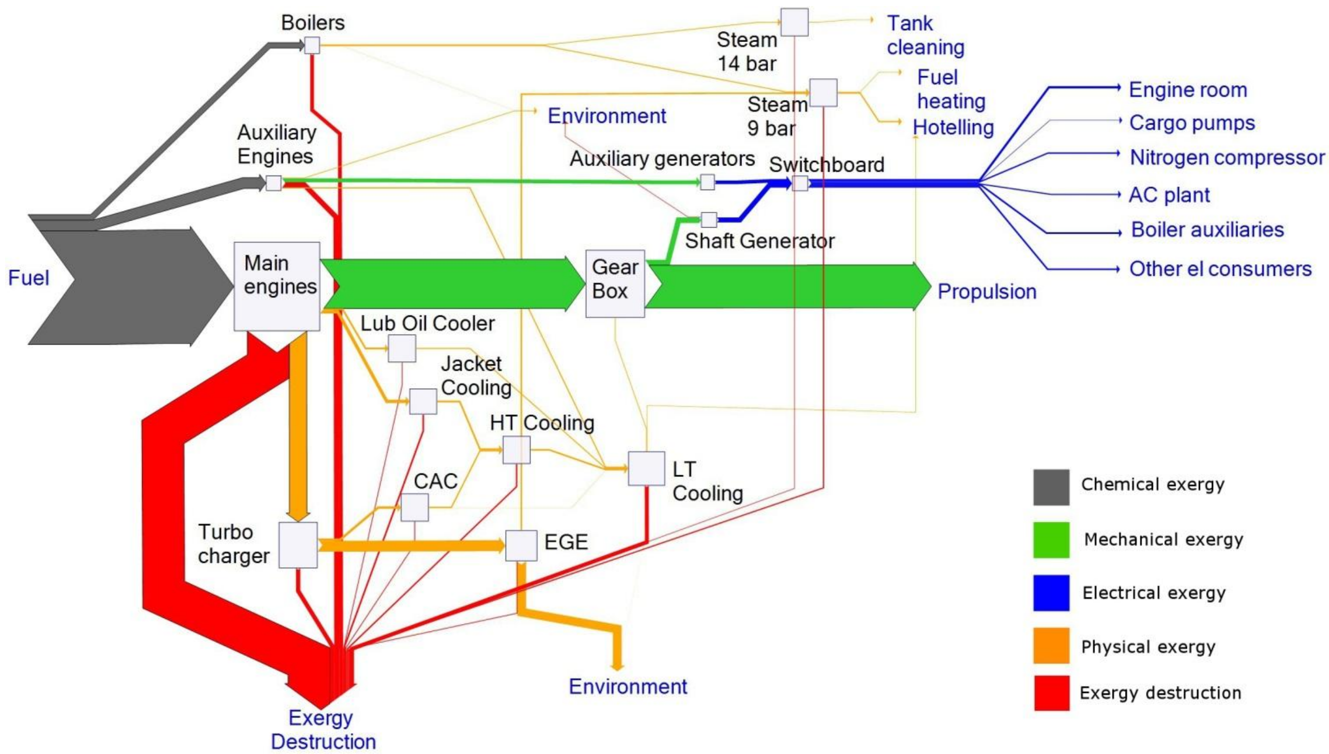

Figure 5. Grassmann diagram of ship energy systems.

Table 9: Yearly Exergy Flows for the Selected Case Study Vessel, in TJ/year.

Component	$\dot{B}_{ch,in}$	$\dot{B}_{ph,c,in}$	$\dot{B}_{ph,c,out}$	$\dot{B}_{ph,h,in}$	$\dot{B}_{ph,h,out}$	$\dot{B}_{w,in}$	$\dot{B}_{w,out}$	$\dot{B}_{q,in}$	$\dot{B}_{q,out}$	$\dot{I}$
Cylinders (ME)	199.6	0.4	27.8	0.0	0.0	0.0	78.0	0.0	5.5	76.7
Turbocharger (ME)	0.0	0.0	3.8	27.8	17.3	0.0	0.0	0.0	0.0	6.8
Lubricating oil cooler (ME)	0.0	1.4	2.9	5.7	2.8	0.0	0.0	0.0	0.0	1.4
Jacket water cooler (ME)	0.0	12.3	15.3	0.0	0.0	0.0	0.0	5.5	0.0	2.5
CAC, HT stage (ME)	0.0	15.3	16.1	3.8	2.3	0.0	0.0	0.0	0.0	0.6
CAC, LT stage (ME)	0.0	0.9	1.4	2.3	0.3	0.0	0.0	0.0	0.0	1.6
LT/HT mixer	0.0	2.9	5.0	16.7	12.3	0.0	0.0	0.0	0.0	2.2
SW cooler	0.0	0.0	0.1	5.0	0.9	0.0	0.0	0.0	0.0	4.0
Exhaust Gas Economizer	0.0	0.4	2.8	17.3	13.8	0.0	0.0	0.0	0.0	1.2
Gearbox	0.0	0.0	0.0	0.0	0.0	78.0	76.2	0.0	0.5	1.3
Shaft generator	0.0	0.0	0.0	0.0	0.0	8.6	7.8	0.0	0.2	0.6
Switchboard	0.0	0.0	0.0	0.0	0.0	13.5	13.4	0.0	0.0	0.1
Boiler	9.3	0.5	3.1	0.0	0.2	0.0	0.0	0.0	0.0	5.9
Tank clearing	0.0	0.0	0.0	1.4	0.3	0.0	0.0	0.0	0.3	0.9
Fuel heating	0.0	0.0	0.0	3.2	0.5	0.0	0.0	0.0	0.7	2.0
Hotel facilities	0.0	0.0	0.0	1.3	0.2	0.0	0.0	0.5	0.9	0.8
Auxiliary engines	17.0	0.0	1.9	0.0	0.0	0.0	5.7	0.0	1.4	8.0

4. Discussion

The implications of the hypotheses made in this study will be here further discussed, together with the generalizability of the results.

4.1 Generalization of the Results

The numerical results presented in the energy and exergy analysis are expected to be representative of the selected vessel and its sister ships: as aggregated data over one year of operation were used, any voyage-specific feature (weather influence on propulsive power, sea water temperature, etc.) is supposed to be levelled when accounting for longer periods of time.

It should be noted, however, that some phenomena can be observable only under longer time perspectives. In particular, today's low markets and high fuel prices have pushed down the operative speed of the vessel, and it is reasonable to expect that the share of propulsive power would be larger (together with recoverable energy) if the vessel were to operate at higher speed.

The variability of ship operational speed is the most important limit to the generalization of the results for future operations of the same vessel, as changes in market conditions could easily lead to an increase in the average operational speed. Were the engines to be operated at higher average load, it would be possible to see a number of changes, such as:

- Increase of propulsion share of total energy consumption
- Increase of the share of the HT stage in the heat balance of the CAC.
- Larger waste flows, both in exergy and energy terms.

The large influence of vessel speed on ship energy systems performance makes the design and retrofitting on these systems a challenge.

There are a number of conditions for the extension of the results presented in this study to other vessels.

The vessel should not present any major ship-specific auxiliary power or heat demand. In the case of chemical tankers, this reduces to the operations of tank cleaning and nitrogen compressors, which only account for a minor share of the total energy demand. Ships like passenger ships or reefers have a remarkably different energy demand and are therefore not represented by the vessel studied in this work.

The propulsion system of this ship is based on four-stroke engines. Although the difference in efficiency compared to two-stroke engines of similar size is limited, it could still be seen in the analysis. In addition, exhaust temperatures are significantly lower in the case of two-stroke engines, making results related to the waste heat availability in the exhaust gas obtained in this study not applicable to two-stroke engine powered vessels.

Finally, the study presented in this paper does not account for dynamic ship behavior. This approximation is justified in the case of merchant, ocean going vessels, but not in the case of small ferries, tugs, or in general other ships were the dynamic component of the energy consumption cannot be neglected.

We therefore call for more case studies related to energy and exergy analysis of ship energy systems, particularly in relation to other vessel types. The extension of the results of this work to other ship categories would improve the understanding of ship energy systems and reinforce the need for the utilization of these methods in efforts for improving ship design, retrofitting, and operations.

4.2 Input Data

One strength of the procedure employed lies in the variety of input data that can be used in order to elaborate the structure of on board energy flows. Input data for calculations were obtained from the CMS, manufacturers' technical documentation, shipyard technical documentation, and reported measurements from the crew. This mixture of different data sources made it possible to use all available information, with the drawback of reduced consistency in data sources and accuracy.

Some variables were not measured and needed to be either assumed or calculated. This was particularly limiting in the case of exhaust gas and air properties (flow and temperature), which were calculated based on the regression of manufacturer's data. In reality many parameters, such as engine and turbocharger wear and fuel type, will influence engine performance.

Heat flows to jacket water and lubricating oil also had to be estimated based on the assumption that the engine behaves according to manufacturer's information.

Regressions also required extrapolation outside of the original domain whenever the engine load was measured to be below 50% of the engine MCR. Apart from air and exhaust temperatures, unfortunately, all other aforementioned variables are very seldom measured on board of existing vessels and it is therefore expectable that approximations will be required also for future similar studies. The estimation of heat was also based on a large number of assumptions and is should therefore be treated with care. The same can be said for the repartition of auxiliary power demand among individual consumers.

The availability of measurements of total heat demand, as well as of individual heat and power consumers, would provide the possibility to discuss savings related to consumers, and not only to converters. Heat demand for hotel facilities, for instance, is largely influenced by the assumptions employed in the calculation of the required amount of freshwater to be generated onboard, which is determined according to common practice and is therefore subject to large variability.

Given the absence of available measurements, it was not possible to validate the assumptions employed in this study.

5. Conclusions

The paper presented the energy and exergy analysis of a chemical / product tanker, based on a mixed top-down and bottom-up approach applied to one year of ship operation. The exergy analysis was used as a basis for evaluating the potential for waste heat recovery on the vessel.

The application of the proposed method to the case study ship led to an improved understanding of onboard energy use and of inefficiencies in the system, obtained through the estimation of energy and exergy flows. Energy analysis allows estimating the main consumers, producers, and hence allows understanding where most of the energy goes and were losses are located. Exergy analysis, on the other hand, improves the understanding of the potential for WHR, and helps in the identification of inefficiencies in the handling of waste heat.

The analysis showed, as expected, that propulsion power is the major energy consumption (68%), while also demonstrating that auxiliary demands of both electric power (12%) and heat (20%) are not negligible. A large amount of energy is wasted to the environment through the engine cooling and the exhaust gas. Using exergy analysis, the potential for WHR from these losses was estimated. Large amounts of exergy are destroyed in the cooling systems, as exchanges are not optimized for conserving energy quality.

The availability of such amounts of waste heat would suggest further investigating the possibility of installing WHR systems; future work can be directed towards the design and optimization of WHR cycles for the generation of auxiliary power, such as steam-based and Organic Rankine cycles, which have been extensively treated in literature (e.g. Larsen et al. [28]). In addition to the aforementioned technologies, complementary uses for waste heat from Diesel engines for shipping application have been extensively reviewed by Shu et al. [30]

Acknowledgments

The work presented in this paper is part of a larger project entitled “energy systems modelling in shipping” financed by the Swedish Energy Agency and part of the Lighthouse competence center. The authors would also like to thank Laurin Maritime and all its employees for the supportive and collaborative attitude that made it possible to achieve the results of this work.

Appendix A

Table A1. Defining Equations and Assumptions for on Board Material Flows.

Flow	Equation
Air	$T_{air,Comp,in} = 35^o C$ $T_{air,Comp,out} = T_{in}\beta_{comp}^{\frac{k-1}{k\eta_{pol,comp}}}$ $T_{air,CAC,out} = 55^o C$ $m_{air} = \sum \rho_{in,i}\frac{n_{ME,i}}{120}V_{cyl,\max}N_{cyl}$
Exhaust gas	$T_{eg,turb,out} = P_2(\lambda_{ME})$ $T_{eg,\mathrm{EGE},out} = T_{eg,turb,out} - \frac{\dot{Q}_{heat}}{\dot{m}_{eg}c_{p,eg}}$ $\dot{m}_{eg} = \dot{m}_{air} + \dot{m}_{fuel}$
Lub oil	$T_{LO,\mathrm{LOcooler,out}} = 60^o C$ $T_{LO,LOcooler,in} = T_{LO,LOcooler,out} + \frac{\dot{Q}_{LO}}{c_{LO}\dot{m}_{LO}}$ $\dot{V}_{LO} = 65\frac{m^3}{h}$
HT cooling	$T_{HT,\mathrm{HT/LTmixer,in}} = 90^o C$ $T_{HT,HT/LTmixer,out} = T_{HT,HT/LTmixer,in} - \frac{\dot{Q}_{HT}}{c_W\dot{m}_{HT}}$ $\dot{V}_{HT} = 70\frac{m^3}{h}$
LT cooling	$T_{LT,SW\,cooler,out} = 34^o C$ $T_{LT,SWcooler,in} = T_{LT,SWcooler,out} + \frac{\dot{Q}_{LT}}{c_W\dot{m}_{LT}}$ $\dot{V}_{HT} = 80\frac{m^3}{h}$

Table A2. Defining Equations and Assumptions for on Board Energy Flows.

Energy flow	Equation
Exhaust gas	$\dot{Q}_{eg} = \dot{m}_{eg}c_{p,eg}(T_{eg,turb,out} - T_0)$
Charge air cooler	$\dot{Q}_{CAC} = \dot{m}_{air}c_{p,air}(T_{air,comp,out} - T_{air,comp,in})$
Jacket water cooling	$\dot{Q}_{JW} = 0.414(\dot{Q}_{fuel} - \dot{W} - \dot{Q}_{eg} - \dot{Q}_{CAC})$
Lub oil cooling	$\dot{Q}_{LO} = 0.444(\dot{Q}_{fuel} - \dot{W} - \dot{Q}_{eg} - \dot{Q}_{CAC})$
HT cooling	$\dot{Q}_{HT} = \dot{Q}_{JW} + P_2(\lambda_{ME})\dot{Q}_{CAC}$
LT cooling	$\dot{Q}_{LT} = \dot{Q}_{LO} + \dot{Q}_{CAC} + \dot{Q}_{JW} - \dot{Q}_{FWgen}$
Main engine power	$P_{ME} = \frac{\frac{P_{prop}}{\eta_{shaft}} + \frac{P_{SG}}{\eta_{SG}}}{\eta_{GB}}$
Auxiliary engine power	$P_{AE} = \frac{P_{AG}}{\eta_{AG}}$

Table A3. Defining Equations and Assumptions for Selected Components.

Component	Equation
Compressor	$\beta_{comp} = P_2(\lambda_{ME})$
Compressor	$\eta_{pol,comp} = P_2(\lambda_{ME})$
Shaft generator	$\eta_{SG} = 0.95 P_{2.}(\lambda_{SG})$ [25]
Gearbox	$\eta_{GB} = 0.983$
Shaft	$\eta_{shaft} = 0.99$ [12]

Nomenclature

b	specific exergy, J/kg
B	exergy, J
$\dot{B}$	exergy flow, W
bsfc	break specific fuel consumption, g/kWh
c	specific heat, J/kg K
E	energy, J
$\dot{E}$	energy flow, W
h	specific enthalpy, J/kg
$\dot{I}$	irreversibility rate, W
k	specific heat ratio
m	mass, kg
$\dot{m}$	mass flow, kg/s
n	rotational speed, rpm
N_{cyl}	number of cylinders
p	pressure
P_n	polynomial of order n
$\dot{Q}$	heat flow, W
s	specific entropy, J/(kg K)
$\dot{S}_{gen}$	entropy generation rate, W/K
T	Temperature, K or ^{o}C
V	Volume, m^3
$\dot{V}$	Volume flow, m^3/s

Acronymes

AE	auxiliary engine
AG	auxiliary generator
CAC	charge air cooler
CMS	continuous monitoring system
EGE	exhaust gas economizer
HT	high temperature
JW	Jacket water
LO	lubricating oil
LT	low temperature
ME	main engine
SG	shaft generator
SW	sea water
WHR	waste heat recovery

Greek letters

β	compression ratio
λ	engine load
δ	irreversibility share
ε_t	total exergy efficiency
ε_u	task efficiency
γ	irreversibility ratio
η	energy efficiency
ρ	density, kg/m^3
Δ	finite difference

Subscripts

c	cold
comp	compressor
eg	exhaust gas
h	hot
i	component
in	inlet flow
out	output flow
p	paid
pol	politropic
prop	propeller
tot	total
u	useful
0	reference state

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Paper II

Energy and exergy analysis of a cruise ship

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Abstract:

The shipping sector is today facing numerous challenges. Fuel prices are expected to increase in the medium-long term, and a sharp turn in environmental regulations will require several companies to switch to more expensive distillate fuels. In this context, passenger ships represent a small but increasing share of the industry. The complexity of the energy system of a ship where the energy required by propulsion is no longer the trivial main contributor to the whole energy consumption thus makes this kind of ship of particular interest for the analysis of how energy is converted from its original form to its final use on board.
To illustrate this, we performed an analysis of the energy and exergy flows of a cruise ship sailing in the Baltic Sea based on a combination of available measurements from ship operations and of mechanistic knowledge of the system. The energy analysis allows identifying propulsion as the main energy consumer (41% of the total) followed by heat (34%) and electric power (25%) generation; the exergy analysis allowed instead identifying the main inefficiencies of the system: exergy is primarily destroyed in all processes involving combustion (88% of the exergy destruction is generated in the Diesel engines and in the oil-fired boilers) and in the sea water cooler (5.4%); the main exergy losses happen instead in the exhaust gas, mostly from the main engines (67% of total losses) and particularly from those not equipped with heat recovery devices.
The improved understanding which derives from the results of the energy and exergy analysis can be used as a guidance to identify where improvements of the systems should be directed.

Keywords:

Energy analysis; exergy analysis; low carbon shipping

1. Introduction

1.1 Background

According to the third IMO GHG Study 2013, in 2012 shipping contributed to global anthropogenic CO_2 emissions with a total of 949 million tonnes, which represents roughly the 2.7% of the total [1]. Although such contribution appears relatively low, the trend is that shipping will play an even greater role in the CO_2 emissions in a near future due to the increased transport demand according to all IMO future scenarios. As an example, global transport demand has increased by 3.8 % in 2013, compared to a global GBP growth of 2.3 % the same year, which shows how shipping tends to rise even faster than global economy [2].

International Energy Agency data from marine bunker show that the OECD countries in fact have reduced the CO_2 impact from shipping, but a larger amount has been moved to the non-OECD countries [3]. The fact that shipping needs to even further reduce its CO_2 emissions in the near future is essential for being able to achieve the goals of maintaining the climate below a 2-degree level in 2050 [4]. Finally, in the Baltic Sea an emission control area is enforced by the International Maritime Organisation since January 2015 which stipulates that the fuel used must not contain more than 0.1 % sulphur, therefore requiring the use of more expensive distillate fuels.

Altogether, these conditions present a challenge to shipping companies, which are attempting to reduce their fuel consumption in an attempt to reduce both environmental impact and operative costs. A wide range of fuel saving solutions for shipping are available and partially implemented in the existing fleet, both from the design and operational perspective; several specific studies have been conducted on these technologies, and a more detailed treatise would be out of the scope of this work. In this context, it has been acknowledged that the world fleet is heterogeneous, and measures need to be evaluated on a ship-to-ship basis [5,6]. In this process, a deeper understanding of energy use on board of the specific ship is vital.

1.2 Previous work

A number of studies concerning ship energy systems can be found in literature. Thomas et al. [7] and Basurko et al. [8] worked on energy auditing fishing vessels; Shi et al. [11, 12] proposed models for predicting ship fuel consumption for some specific vessel types; Balaji and Yaakob [9] analysed ship heat availability for use in ballast water treatment technologies. These studies have been of particular interest in their relative fields, but a more comprehensive approach of the totality of the ship energy system is missing. In addition, an analysis purely based on the First law of thermodynamics does not account for the irreversibilities of the systems and for the different quality of heat flows [16]. Exergy analysis, which is based on both the First and the Second laws of thermodynamics, can help addressing this shortcoming. Widely used in other industrial sectors, exergy analysis in not commonly employed in maritime technology studies, and is mostly related to waste heat recovery systems [17, 18] and refrigeration plants [19, 20]. The application of exergy analysis in shipping is still limited; Zaili and Zhaofeng [10] proposed the energy and exergy analysis of the propulsion system of an existing vessel showing that there is potential in improving ship power plant efficiency by recovering the exergy in the exhaust gas and by improving operations of the main engines.

In a previous study of the energy and exergy analysis of a product tanker [11] the dominance of propulsion as main consumer on board was highlighted, together with the substantial availability of waste heat for recover. On cruise vessels, the number of different uses of energy is larger and a complex system of different energy carriers (chemical, thermal, electrical or mechanical) is present in order to fulfil the needs for transport combined with passenger services and comfort, such as cooking and cooling in restaurants, air conditioning, and passenger entertainment facilities.

The complexity of the energy system of a ship where the energy required by propulsion is no longer the trivial main contributor to the whole energy consumption thus makes this kind of system of particular interest for the analysis of how energy is produced, transformed, and used on board. The complexity of such systems was modelled and investigated previously by Marty et al. [12,13], but to the best of our knowledge there is no study in literature describing cruise ships' energy and exergy analyses based on actual measurements.

1.3. Aim

The aim of this paper is to provide a better understanding of how energy is used on board of a cruise ship and where the largest potential for improvement is located by applying energy and exergy analysis to the a case study. The combination of a method rarely applied in the shipping sector to a ship type featuring a complex energy system is considered as the main contribution of this work to the existing literature in the field.

2. Methodology

This paper proposes the application of energy and exergy analysis (further described in Section 2.1) as a mean for improving understanding of energy conversion on board of a cruise ship. This application is shown for a specific case study vessel (see Section 2.2) for which extensive measurements from on board logging systems were available (see Section 2.3 for details on data gathering and processing). The results from the energy and exergy analyses are then discussed in order to propose possible improvements for ship operations and design.

2.1 Energy and exergy analysis

Energy and exergy analysis were performed on the system of study, taking the ship energy system, as presented above, as control volume. The energy flows are calculated by assuming that the chemical energy of the fuel flows is equal to its lower heating value, and the physical energy is taken as its relative enthalpy. Such an analysis allows for tracking all the energy streams flowing through the ship and depicts the main heat and power users.

Energy may be transformed from one form to another, but it can neither be created nor destroyed; this results in the fact that a conventional energy analysis provides limited information on the system inefficiencies. *Exergy* is defined as the `*maximum theoretical useful work as the system is brought into complete thermodynamic equilibrium with the thermodynamic environment while the system interacts with it only*'. At the difference of energy, exergy is *not* conserved in real processes, and the exergy destroyed, or irreversibility rate, quantifies the system irreversibilities. The general exergy balance can be written as:

$$\dot{I} = \sum \dot{EX}_{in} - \sum \dot{EX}_{out} \ , \tag{4}$$

where:

- $\dot{I}$ denotes the irreversibility rate, also called exergy destruction, which can also be calculated from the Gouy-Stodola theorem;

$$\dot{I} = T_0 \dot{S}_{gen} \tag{5}$$

- $\dot{EX}_{in}$ is the exergy flow entering the component/system under investigation;
- $\dot{EX}_{out}$ represents the corresponding exergy outflow. This term is normally further subdivided in two parts: products($\dot{EX}_{prod}$) and losses ($\dot{EX}_{loss}$)

The exergy of a material flow is divisible into its physical, chemical, kinetic and potential components, in the absence of nuclear and magnetic interactions. The physical exergy represents the maximum amount of work obtainable from bringing the material stream from its initial state to the environmental state, defined by p_0 and T_0, taken here as the ambient pressure and the seawater temperature. The chemical exergy represents the maximum amount of work obtainable as the stream under consideration is brought to the dead state, by chemical reaction and transfer processes. The fuel chemical exergy is assumed equal to its higher heating value, which is derived based on the fuel H/C ratio according to the equation proposed by Szargut et al. [14]. The potential and kinetic exergies are neglected. The exergy transferred with power has the same value as its energy, while the exergy transferred with heat is lower and its value depends on the temperature at which heat transfer takes place. For more details, the reader is referred to the reference books of Szargut et al. [14], Kotas [15] and Moran [16].

The system performance is measured using several performance indicators:

- the *exergy efficiency (η_{ex})*, defined as the ratio between the product $\dot{E}_p$ and fuel $\dot{E}_f$ exergies

$$\eta_{ex} = \frac{\dot{EX}_p}{\dot{EX}_{in}} \tag{6}$$

where the product exergy represents the desired output of the component or system, and the fuel exergy denotes the resources required to drive this process.

- the *irreversibility ratio (λ)*, proposed in [15], which illustrates how much of the exergy input to the system $\dot{E}_{in}$ is actually lost through irreversibilities $\dot{I}$;

$$\lambda = \frac{\dot{I}}{\dot{EX}_{in}} \tag{7}$$

- the *irreversibility share (δ)*, proposed in the works of Kotas [15] and Tsatsaronis [17], which is defined as the ratio between the exergy destroyed in the *i*-th component $\dot{I}_i$ in relation to the total system irreversibilities $\dot{I}_{tot}$.

$$\delta_i = \frac{\dot{I}_i}{\dot{I}_{tot}} \tag{8}$$

2.2 Case study vessel

The ship under study is a cruise ship operating on a daily basis in the Baltic Sea between Stockholm and the island of Åland. The ship is 176.9 m long and has a beam of 28.6 m, and has a design speed of 21 knots. The ship was built in Aker Finnyards, Raumo Finland in 2004. The ship has a capacity of 1800 passengers and has several restaurants, night clubs and bars, as well as saunas and pools. This means that the energy system regarding the heat and electricity demand is more complex than a regular cargo vessel in the same size. Typical ship operations, although they can vary slightly between different days, are represented in Figure 1. It should be noted that the ship stops and drifts in open sea during night hours before mooring at its destination in the morning, if allowed by weather conditions.

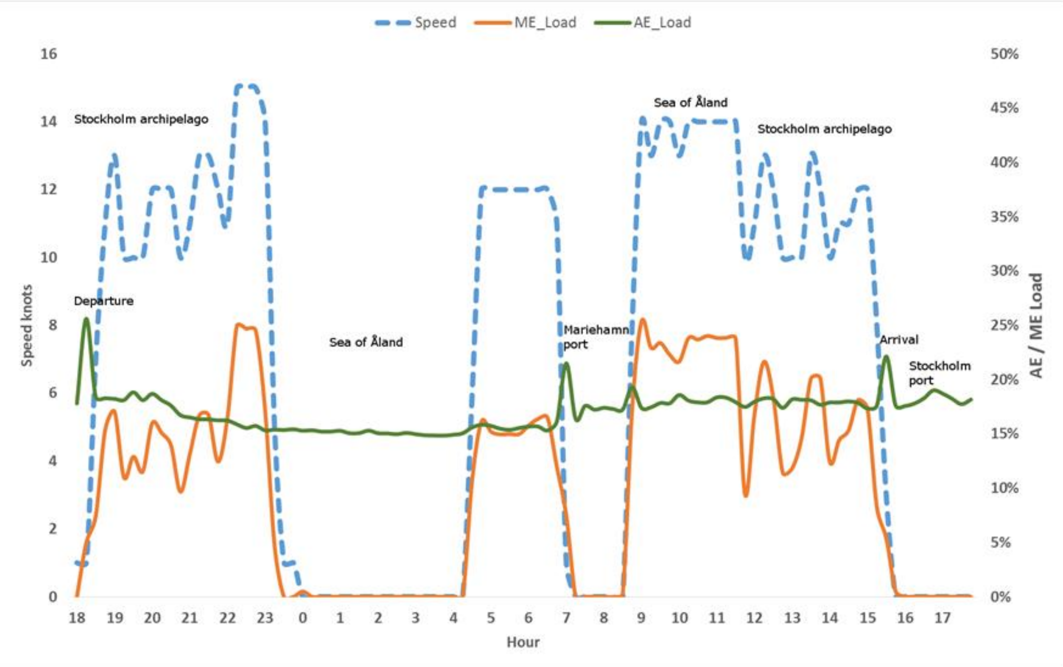


Fig. 1. Typical operational profile (ship speed, main engines load and auxiliary engines load) for the selected ship.

The ship systems are summarized in Figure 2. The propulsion system is composed of two equal propulsion lines, each made of two engines, a gearbox, and a propeller. The main engines are four Wärtsilä 4-stroke Diesel engines (ME) rated 5850 kW each All engines are equipped with selective catalytic reactors (SCR) for NO_X emissions abatement. Propulsion power is needed whenever the ship is sailing; however, it should be noted that the ship rarely sails at full speed, and most of the time it only needs one or two engines operated simultaneously.

Auxiliary power is provided by four auxiliary engines (AE) rated 2760 kW each. Auxiliary power is needed on board for a number of alternative functions, from pumps in the engine room to lights, restaurants, ventilation and entertainment for the passengers.

Auxiliary heat needs are fulfilled by the exhaust gas steam generators (HRSG) located on all four AEs and on two of the four MEs or by oil-fired auxiliary boilers (mainly when in port, or during winter), by the heat recovery on the HT cooling water systems (HRHT), and by the auxiliary, oil-fired boilers (AB). The heat is needed for passenger and crew accommodation, as well as for the heating of the highly viscous heavy fuel oil used for engines and boilers. This last part, however, is drastically reduced since the 1st of January 2015, as new regulations entering into force require the use of low-sulphur fuels, which require a much more limited heating.

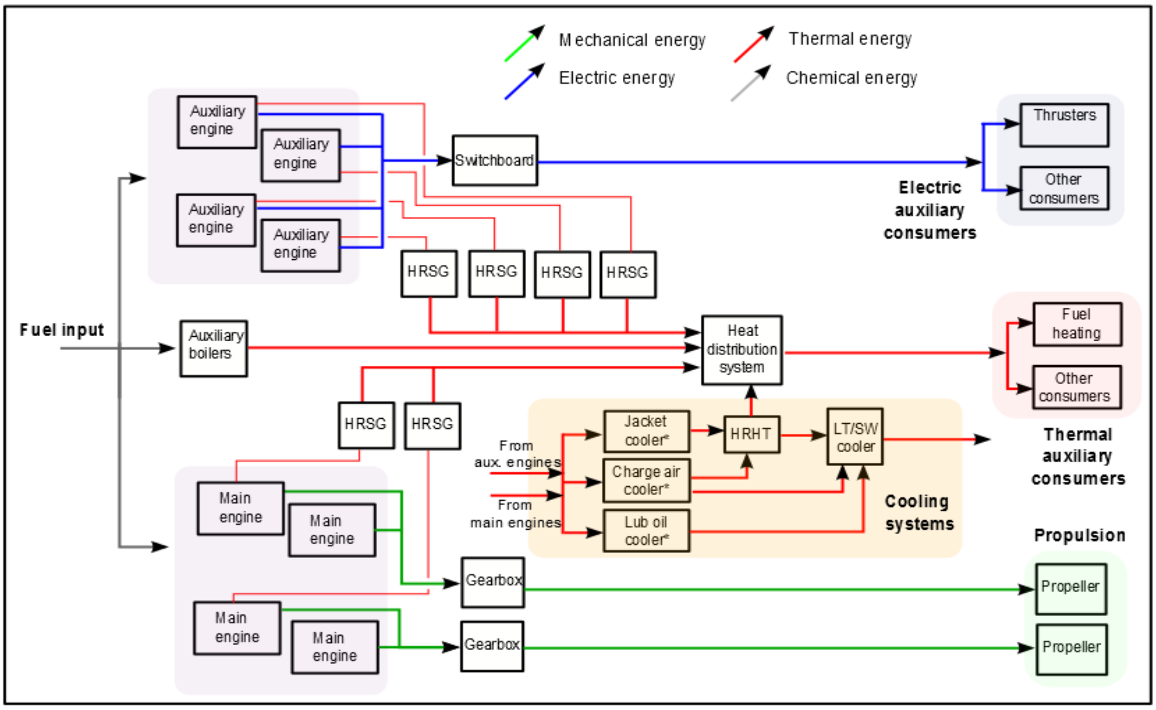


Fig. 2. Schematic representation of ship energy systems

2.3 Data gathering and processing

The ship under study is equipped with an extensive system for measuring and logging of operational variables, which logs the data with a 60 second interval. For this study an averaged 15 minute interval was chosen in order to cover a total of approximately one year of ship operations under the constraints related to the maximum number of data points in the database export tool.

A detailed accounting of all relationships and assumptions employed in this study in order to process the raw measured data are shown in Table A1 in Appendix A. Hereafter only the most relevant assumptions are discussed.

Main engines power and fuel mass flow ($\dot{m}_{fuel,ME}$) were not directly measured. In this study it was assumed that measures of the normalized fuel rack position (frp_{norm}) can be used as a predictor for the amount of fuel injected per cycle. The fuel flow to the main engines is consequently calculated according to the following equation:

$$\dot{m}_{fuel,ME} = \dot{m}_{fuel,ME,des}\left(a_0 + a_1 frp_{norm}\right)\left(\frac{n_{ME}}{n_{ME,des}}\right), \qquad (1)$$

where the subscript *des* refers to design conditions of the engine at 100% of the maximum continuous rating (MCR). The regression coefficients a_0 and a_1 where determined based on the engine shop trial tests documents. Estimations calculated using (1) were validated against fuel flow measurements obtained from a recently installed mass flow meter.

The engine specific fuel oil consumption (*bsfc*) for both main and auxiliary engines is calculated as a 2nd degree polynomial function of the engine load. In the case of the auxiliary engines, since measurements of engine power were available, no further assumption was required.

The heat demand was not directly measured, and therefore needed to be estimated. As previously mentioned, on board heat demand is fulfilled by three different systems: the HRSGs, the HRHT and the ABs. The heat recovered in the HRSGs was estimated based on calculated engine exhaust flow and measured temperatures before and after the HRSGs; no information was available regarding the heat recovered in the HRHT; finally, the AB daily fuel consumption was available from a second logging system. In order to provide a reasonable assumption for the contribution of each of the above mentioned systems to the total amount of heat generated on board, it was assumed that heat demand is constant during each day. In addition, discussions with the crew allowed making the assumption that the ABs are only used when the main engines are not running. Based on these considerations, the following approximation was employed in this study:

$$\dot{Q}_{tot}(t) = \frac{m_{fuel,AB}}{\Delta t_{port}} LHV_{fuel} + \frac{\sum_i \dot{Q}_{HRSG,port}(t_i)\Delta t_i}{\Delta t_{port}} + 500 \text{ [kW]}, \quad (2)$$

where the first term represents the contribution from the oil fired boilers, the second term that of the HRSGs operated in port, and the third that of the HT cooling systems based on design calculations provided by the shipyard. The heat demand calculated according to (2) is considered to be constant during the day. When the ship is sailing/manoeuvring the ABs are turned off, and the required heat on board is generated by the HRSGs and the HRHT:

$$\dot{Q}_{rec,HT}(t) = \dot{Q}_{tot}(t) - \dot{Q}_{HRSG}(t) \quad (3)$$

3. Results

3.1 Operational profile

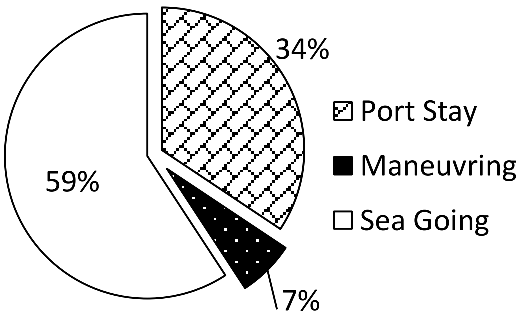
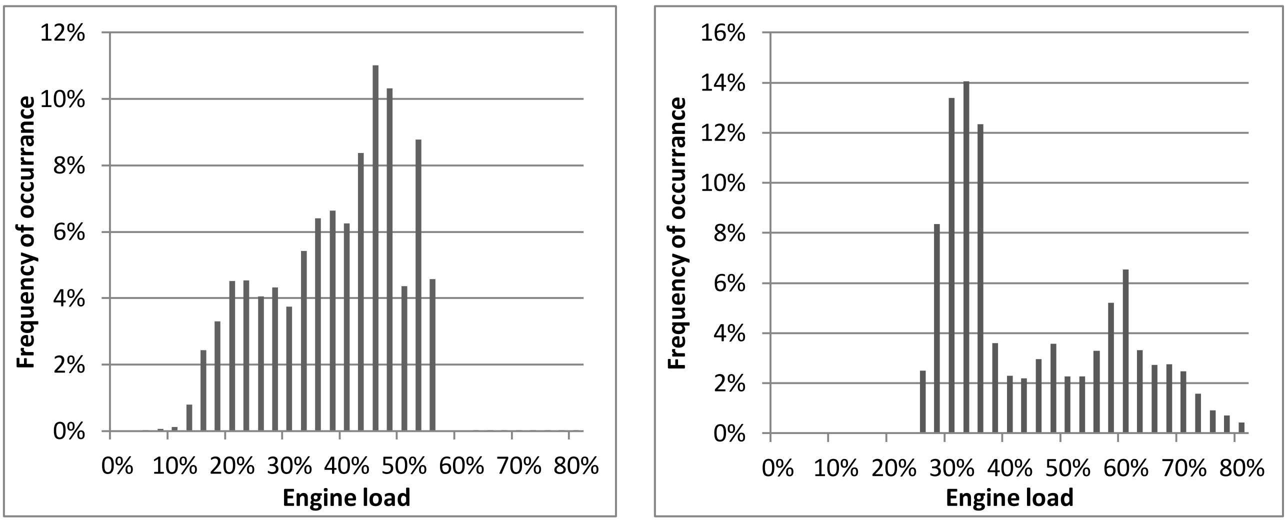


Fig. 3. Time spent by the ship in different operational modes



(a) **(b)**

Fig. 4. Load distribution for a) main engines and b) auxiliary engines

As shown in Figure 3 the ship spends most of the time sailing, while a significant amount of time is also spent in port. This is not surprising, in relation with the typical operations of this type of ship where loading and unloading of passengers is an operation that require a significant amount of time.

Figure 4 shows the load distribution for the main engines (a) and auxiliary engines (b), respectively. As it is observable from the figure, the main engines are most often operated at very low load, which leads to sub-optimal conditions in terms of efficiency and wear. This is a result of two concurring factors, as discussions with the crew revealed:

- The ship is operated most of the time at a speed which is much lower than the design value. This leads consequently to a strong reduction in propulsion power demand

The engines are divided in two groups, each driving one propeller. This means that, even at very low load, it is not possible to operate on only one engine at medium-high load.

3.2 Energy analysis

Figure 4 shows the Sankey diagram for ship energy producers, converters, and consumers, where values are presented numerically in Table A2 in Appendix A. From the consumers' side, it can be seen that the energy demand for auxiliary power is comparable in size to that for propulsion. This is situation is expectable in the case of cruise ships /passenger ferries, but not common in other ship types. Thrusters, although they represent a high punctual consumption, do not significantly contribute to the overall energy consumption. Auxiliary heat demand is also particularly large, but is mostly fulfilled by heat recovery boilers. It should be noted, however, that the yearly fuel consumption from the auxiliary boilers in the case under study is significant (7 %).

The contribution from the HT cooling water systems is also significant, comparable to that of the boilers and the HRSGs. Although this value was not directly measured and is therefore subject to a larger uncertainty, this observation suggests that heat integration has been carefully and successfully taken into account in the design of this particular vessel.

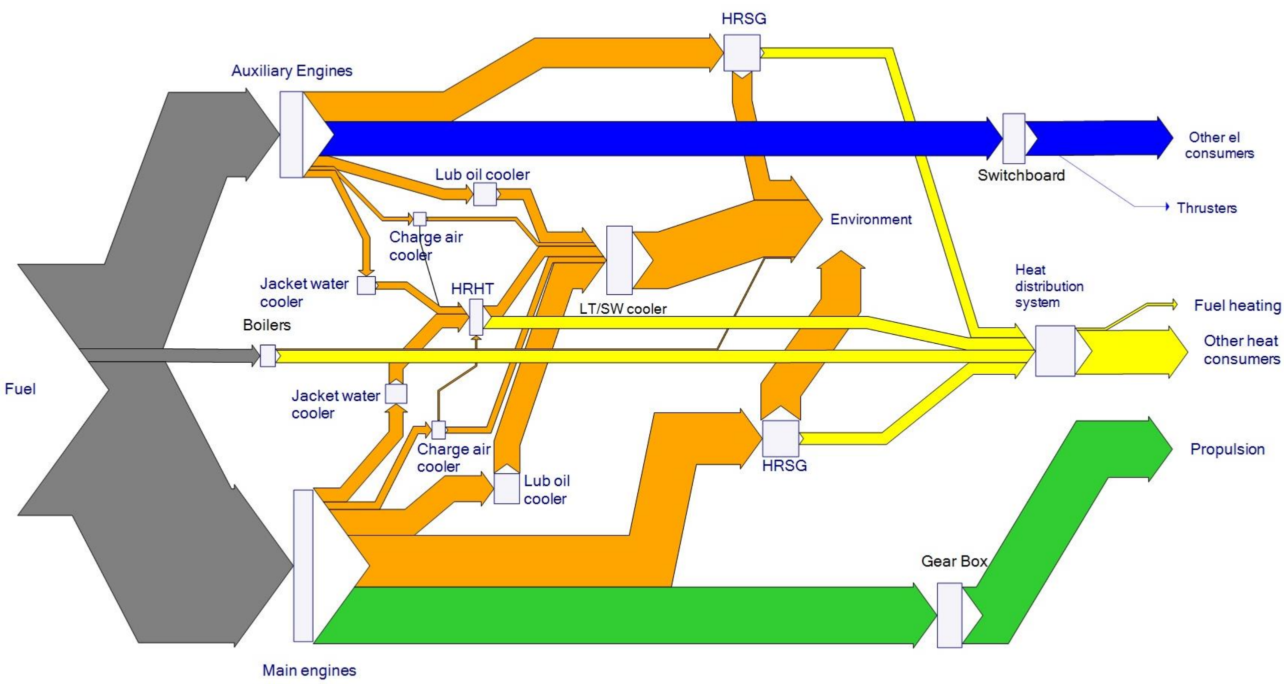


Fig. 4. Sankey diagram for ship energy systems. Values represent the aggregated consumption over one year of operations. All main engines and auxiliary engines are grouped together.

The energy analysis also shows that a large amount of heat is rejected to the environment, mainly with the exhaust gases exiting the heat recovery steam generators installed after the main engines, and heat from the low-temperature and seawater cooling systems. The amount of energy dispersed to the environment is in the same order of magnitude as the heat consumption of the whole ship energy system. This situation suggests that additional heat could be harvested for other uses on-board, e.g. for use in heat recovery and heat-to-power systems, which would result in a smaller fuel consumption of the boilers or/and of the engines. This aspect will be however further investigated using the exergy analysis, which gives a better picture of energy quality and a better estimation of the amount of energy that could be actually recovered and converted into electricity. Large amount of energy is also dispersed via the LO cooling. This waste heat flow is by no means recovered on ship systems, differently from the heat to the HT systems.

The absence of any dedicated measurement made it impossible to identify the individual consumers; however, the use of bow thrusters during manoeuvring constitute a clear spike in the total auxiliary power consumption and are therefore possible to separate from the total.

Figure 5 shows the repartition of the energy production among different generators and for the different operational modes. As expectable, propulsion represents the main part of energy use, but is only present when at sea or manoeuvring. Electric energy consumption is instead rather constant over time and therefore it scales proportionally to the time in each phase.

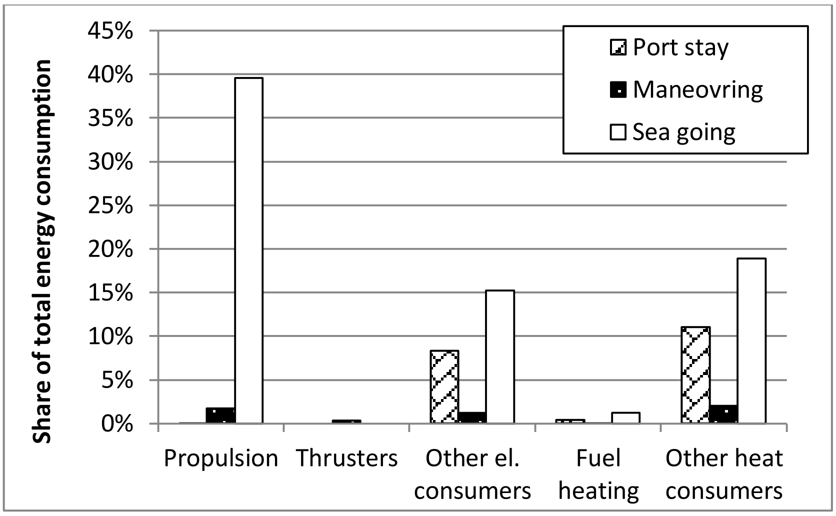


Fig. 5. Yearly energy demand for different consumers, separated per operational mode. Note that the label "port stays" also includes the time spent drifting in open sea with the main engines off

Table 1: Yearly shares for the different energy consumers on board, divided by operational mode

Consumer	Port stay	Manoeuvring	Sea going	**All modes**
Propulsion	0,0%	1,7%	39,6%	**41,3%**
Thrusters	0,0%	0,3%	0,0%	**0,3%**
Other el. consumers	8,3%	1,2%	15,3%	**24,8%**
Fuel heating	0,4%	0,1%	1,2%	**1,7%**
Other heat consumers	11,0%	2,0%	18,9%	**31,9%**
All consumers	**19,7%**	**5,3%**	**75,0%**	**100,0%**

3.3 Exergy analysis

The observation of the Grassmann diagram (Figure 6) allows for identifying, locating and quantifying the main sources of exergy losses and destruction. Most exergy destruction takes place in the main and auxiliary engines (respectively 53.1% and 27.8% of the total) followed by the boilers (6.9%), where the high rate of exergy destruction is strongly connected to the process of conversion of chemical to thermal energy, as well as to mixing and friction phenomena and heat transfer. A significant part of the exergy destruction takes place in the cooling systems (9.1%), which highlights potential for improvement in the design of the heat exchanger network. Finally is the exergy flow rate lost to the environment in the exhaust gas after the HRSGs (8.0% of the total exergy input, 20.3% if compared to the total exergy output of the system) also represents a significant potential for improvement. It should be noted, as shown in Table 3, that most exergy losses take place during the seagoing phase, when on board heat demand is fulfilled through the use of waste heat available on board. The recoverable exergy theoretically available during port stays constitutes however more than half (52.5%) of the total exergy flow produced by the boilers.

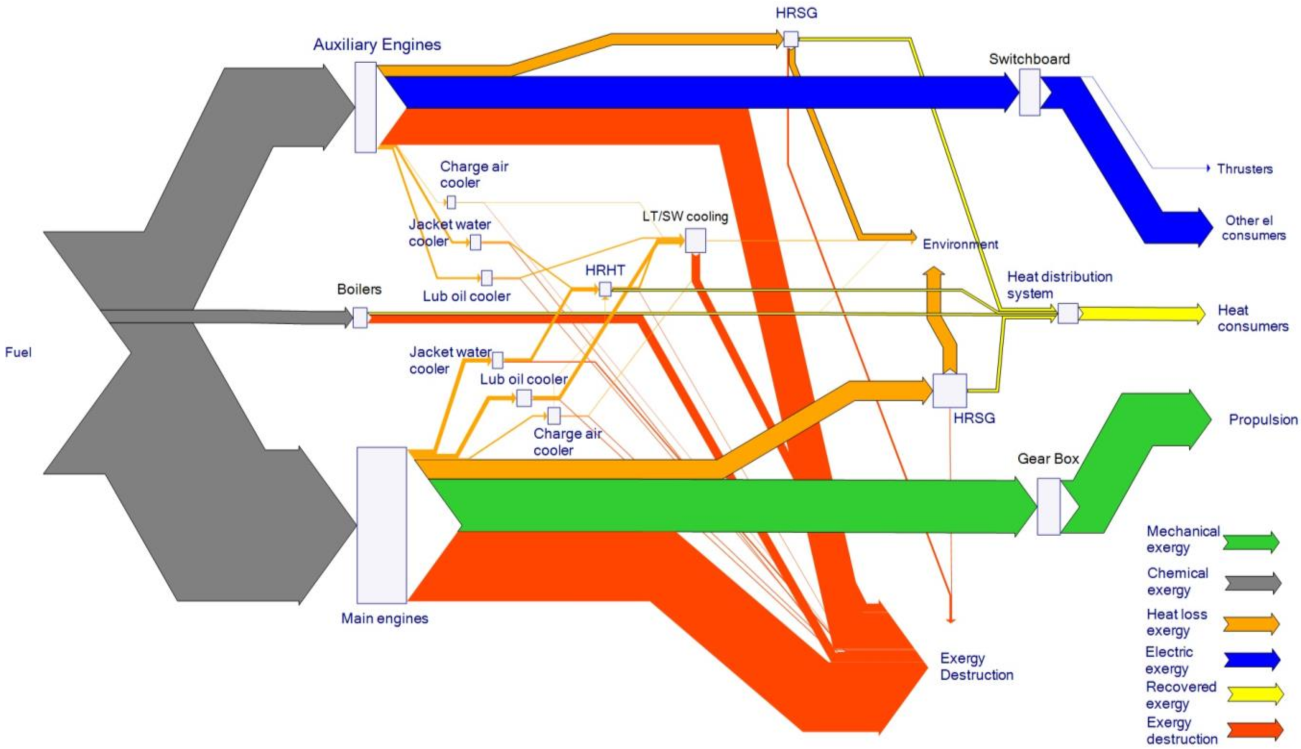


Fig. 6: Grassmann diagram for ship energy systems. Values represent the aggregated consumption over one year of operations. All main engines and auxiliary engines are grouped together.

Table 2: Exergy efficiency, irreversibility ratio and irreversibility share for ship energy system thermal components

Component	Exergy efficiency	Irreversibility ratio	Irreversibility share
Main engines	34,7%	46,2%	53,1%
Auxiliary engines	38,3%	44,0%	27,8%
Boilers	29,2%	70,8%	6,9%
Charge air cooler (ME)	-	4,7%	0,5%
Charge air cooler (AE)	-	3,2%	0,2%
Lub oil cooler (ME)	-	6,6%	0,6%
Lub oil cooler (AE)	-	10,5%	0,5%
Jacket water cooler (ME)	-	7,8%	1,0%
Jacket water cooler (AE)	-	7,4%	0,5%
HT - LT mixer	-	1,7%	0,4%
LT - SW cooler	-	65,7%	5,4%
HRSG (ME)	37,0%	18,8%	1,5%
HRSG (AE)	28,5%	15,9%	1,4%

Table 3: Yearly shares for the different exergy losses on board, divided by operational mode

Consumer	Port stay	Manoeuvring	Sea going	**All modes**
Exhaust, AE	9,1%	1,7%	19,2%	30,0%
Exhaust, ME	0,0%	2,6%	64,5%	67,1%
LT - SW cooler	0,6%	0,2%	2,2%	2,9%
All exergy losses	**9,7%**	**4,5%**	**85,9%**	**100,0%**

The analysis of exergy performance indicators further highlights a number of observations about the system. Main engines show a lower efficiency compared to the auxiliary engines (34.7% compared to 38.3%) despite their higher efficiency at design conditions, which provide further evidence to the fact that the main engines are often operated in non-optimal conditions. The high irreversibility ratio of the boilers (70.8%) suggests that reducing their use should be a priority in view of the exergy optimisation of the system. Among the heat exchangers, the highest potential for improvements appears to lie in the lubricating oil coolers and in the jacket water coolers, where the high temperature difference between hot and cold flows suggests that the heat exchange process could be improved.

4. Discussion

The energy and exergy analysis of the selected ship reveal a rather well thought design, where there has been a significant attempt into the reduction of energy consumption, especially from the heat demand perspective. The amount of energy recovered from the exhaust gas and the engine cooling systems amounts to the most significant fraction of the overall head demand on board, and the boiler is used only in those situations when on board generated waste heat would not be sufficient to fulfil the totality of the heat demand. However, the system shows possibilities of improvement.

4.1 Suggested improvements

From an operational point of view, the main engines are often operated at very limited load, which significantly reduces the efficiency of the energy conversion. This situation is mostly due to high installed power (the vessel was designed for 21 knots but is normally operated at a maximum of 16 knots and, most often, at even lower speeds). The limitations to one-engine operations imposed by the regulations prevents operating on only one engine, which would in turn improve propulsion efficiency by operating the engine at higher load; it should be noted, however, that the achieved higher engine efficiency might be compensated by higher hull rudder resistance.

From a retrofitting/design perspective, even if engine substitution might not be possible due to the high related investment cost, engine de-rating through cylinder disconnection and substitution of the turbocharger could be viable options. An alternative worth investigating could also be that of an hybridisation of the whole system, through the installation of shaft motors/generators, which would allow both main and auxiliary engines to contribute to both propulsive and auxiliary electric power demand and, therefore, increase the flexibility of the system.

Efforts for improving the performance of a ship energy system should however not only focus on the main engines but on avoidable irreversibilities in the rest of the system, such as those caused in the HRSGs and cooling systems. These may be reduced by decreasing the temperature differences between the heat source (e.g. exhaust gases, lubricating oil) and the receiver streams.

From a thermal perspective, the existence of energy and exergy flows potentially available for recovery suggests that there is potential for improving the system's efficiency and therefore reducing fuel consumption. However, the fact that most of the waste heat is available during sea passages, when on board heat demand is already fully fulfilled by the use of waste heat from the main and auxiliary engines, suggests that improvements would require more complex technical arrangements.

The utilisation of heat-to-power technologies represents one possible solution for making use of the waste heat available during sea passages. This possibility was explored by Ahlgren et al. [18] and showed significant potential for improving vessel performance. The use of WHR systems on board for heat-to-power conversion could also justify efforts in the improvement of the heat exchanger network in order to minimize exergy destruction and, therefore, allowing additional exergy to be recovered to useful power for on board use. This additional effort would be particularly justified in the case of the lubricating oil cooler and the jacket water cooler, where the exergy destruction happens at a higher rate therefore suggesting that the highest potential for improvement is located.

The use of thermal energy storage devices could constitute an alternative solution for reducing fuel consumption by providing a buffer between the excess energy available during sea passages and the unfulfilled demand during port stays. A dedicated study, as proposed by the authors in the case of a product tanker [19] is required for providing an estimate of the potential for recovery and of the required thermal storage capacity.

4.2 Limitations and further work

The limited amount of data, both from measurement and design perspective, limits parts of the analysis and therefore prevents to dig further into certain parts of the ship energy systems. The absence of measurements of the temperature levels of the heat demand in different parts of the ship prevents further considerations on heat integration. It is likely that a number of users on board require low-grade it, as in the case of HVAC pre-heaters and re-heaters, which could be provided by recovering heat from low-grade heat sources such as the lubricating oil cooler.

A similar discussion can be presented for on board electrical consumers. The absence of measurements makes it impossible to draw conclusion on possible design and operational savings related to a minimized consumption. This influenced the possibility to analyse the operative efficiency of a number of systems and components, particularly HVAC, cooling systems and engine room ventilation, which not only are expected to contribute extensively to on board energy consumption, but that are also often related to important improvement potential. Most of the cooling pumps on board are in fact equipped with frequency controllers, whose efficiency in the reduction of pump power demand was however impossible to determine.

5. Conclusion

The results of this study showed that the system under analysis was designed with significant efforts for improving energy efficiency; however, many parts of the system could be improved in order to reduce fuel consumption.

- The main potential proved to come from the main engines, which are most often operated at low load and, therefore, at low efficiency. This situation could be improved by engine de-rating or by a hybridization of the system.
- Exergy losses, mostly in the exhaust gas, also provide potential for improvement. Waste heat recovery through heat-to-power technologies are a viable option for the system under study, an option which would also benefit from re-designing the heat exchanger network in order to reduce exergy destruction in critical points such as lubricating oil coolers and jacket water coolers.
- Alternatively, the unbalance between heat availability and demand during sea passages and port stays could be solved through the use of a thermal energy storage system, which would lead to a reduction in the amount of fuel needed by the auxiliary boilers.

Acknowledgments

The authors would like to thank the Swedish Energy Agency and the Swedish Maritime Administration for financing the main authors of this work. The authors would also like to thank Rederiaktiebolaget Eckerö for contributing with data and information regarding the case study vessel and the crew of M/S Birka Stockholm for their help and support during the visits for data gathering.

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Appendix A

Table A1: Summary of the assumptions employed in the processing of measured values for ship energy and exergy systems analysis. Note that λ corresponds to the engine load.

$T_{air,comp,in}$	*Measured*
$T_{air,Comp,out}$	$T_{air,comp,in} + \frac{T_{air,comp,in}(\beta^{\left(\frac{k-1}{k}\right)} - 1)}{\eta_{TC,is}}$
β_{comp}	*Measured*
$\eta_{TC,is}$	$P_2(\lambda)$
$\dot{m}_{air}$	$\eta_{vol}\frac{V_{cyl,\max}\rho_{air} n N_{cyl}}{60*2}$
$\dot{m}_{fuel,ME}$	$\frac{fr_{ME}}{fr_{ME,des}}\frac{n_{ME}}{n_{ME,des}}\dot{m}_{fuel,ME,des}$
$bsfc$	$P_2(\lambda)$
$\dot{W}_{ME}$	$\dot{m}_{fuel,ME} * bsfc$
$\dot{m}_{eg}$	$\dot{m}_{air} + \dot{m}_{fuel}$
η_{vol}	$\eta_{vol} = \frac{r_c}{r_c - 1}\frac{T_{air,in}[\mathrm{K}]}{313 + \frac{5}{6}T_{air,in}[^{o}C]}$ [20]
ϱ_{air}	$\frac{p_{air,CAC,out}}{R_{air}T_{air,\mathrm{CAC,out}}}$
$\dot{m}_{eg,TC}$	$\dot{m}_{air}\frac{c_{p,air}(T_{air,comp,out} - T_{air,comp,in})}{\eta_{\mathrm{mech},TC}c_{p,eg}(T_{eg,turb,in} - T_{eg,turb,out})}$
$\dot{m}_{eg,bypass}$	$\dot{m}_{eg} - \dot{m}_{eg,TC}$
$\dot{Q}_{cooling}$	$\dot{Q}_{fuel} + \dot{Q}_{air,in} - \dot{Q}_{eg} - \dot{W}_{out}$
$\dot{Q}_{LT}$	$\frac{P_{2,LT}(\lambda)}{P_{2,LT}(\lambda) + P_{2,LT}(\lambda)}\dot{Q}_{cooling}$
$\dot{Q}_{HT}$	$\dot{Q}_{cooling} - \dot{Q}_{LT}$
$\dot{m}_{w.HT}$	$\dot{m}_{w,HT,des}\lambda$
$\dot{m}_{w.LT}$	$\dot{m}_{w,\mathrm{L}T,des}\lambda$
$\dot{m}_{LO}$	$\dot{m}_{w,LO,des}\lambda$
$\dot{m}_{w.LT}$	$\dot{Q}_{CAC}P_2(\lambda)$
$\dot{Q}_{CAC,LT}$	$\dot{Q}_{CAC} - \dot{Q}_{CAC,HT}$
$\dot{Q}_{JW}$	$\dot{Q}_{HT} - \dot{Q}_{CAC,HT}$
$\dot{Q}_{LO}$	$\dot{Q}_{LT} - \dot{Q}_{CAC,LT}$

Table A2. Summary of the energy and exergy flows represented in the Sankey and Grassmann diagrams, referred to 11 months of ship operations. Values are provided in TJ

Type	From	To	Energy flow	Exergy flow
CH	Fuel	Main engines	203.5	216.6
CH	Fuel	Auxiliary engines	111,8	119,2
CH	Fuel	Auxiliary boilers	17,3	18,4
M	Main engines	Gearbox	75,2	75,2
H	Main engines	Charge air cooler (ME)	10,1	1,1
H	Main engines	HRSG (ME)	68,7	27,9
H	Main engines	Lubricating oil cooler (ME)	34,4	6,3
H	Main engines	Jacket water cooler (ME)	18,5	6,1
H	Charge air cooler (ME)	HT cooling systems	2,0	0,4
H	Charge air cooler (ME)	LT cooling systems	8,2	0,7
H	Jacket water cooler (ME)	HT cooling systems	18,5	4,2
H	Lubricating oil cooler (ME)	LT cooling systems	34,4	5,2
H	HRSG (ME)	Heat distribution system	15,8	5,4
H	HRSG (ME)	Environment	52,9	19,6
EL	Auxiliary engines	Gearbox	45,7	45,7
H	Auxiliary engines	Charge air cooler (AE)	5,0	1
H	Auxiliary engines	HRSG (AE)	39,5	16,0
H	Auxiliary engines	Lubricating oil cooler (AE)	14,2	2,7
H	Auxiliary engines	Jacket water cooler (AE)	9,5	3,1
H	Charge air cooler (AE)	HT cooling systems	0,3	0,1
H	Charge air cooler (AE)	LT cooling systems	4,7	0,5
H	Jacket water cooler (AE)	HT cooling systems	9,5	2,1
H	Lubricating oil cooler (AE)	LT cooling systems	14,2	1,9
H	HRSG (AE)	Heat distribution system	13,4	7,3
H	HRSG (AE)	Environment	26,1	8,7
M	Gearbox	Propeller	75,2	75,2
EL	Switchboard	Thrusters	0,6	0,6
EL	Switchboard	Other el. consumers	45,1	45,1
H	Heat distribution system	Fuel heating	3,2	1
H	Heat distribution system	Other heat consumers	58	17,6
H	HT cooling systems	LT cooling systems	14,0	2.8
H	HT cooling systems	Heat distribution system	16,3	3.3
H	LT cooling systems	Environment	75.6	0.9

Paper III

Applied Energy 154 (2015) 402–415

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Development of a combined mean value–zero dimensional model and application for a large marine four-stroke Diesel engine simulation

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HIGHLIGHTS

- Development of a combined mean value–zero dimensional engine model.
- Application for simulating a large marine Diesel four-stroke engine.
- Results comparable to the respective ones of the mean value model.
- Enhancement of mean value models predictive ability with adequate accuracy.
- Appropriate where the mean value approach exceeds its limitations.

ARTICLE INFO

Article history:
Received 3 December 2014
Received in revised form 3 May 2015
Accepted 9 May 2015

Keywords:
Combined mean value–zero dimensional model
Marine engine modelling
Engine parameters mapping
Variable turbine geometry
Propulsion plant operating modes

ABSTRACT

In this article, a combined mean value–zero dimensional model is developed using a modular approach in the computational environment of Matlab/Simulink. According to that, only the closed cycle of one engine cylinder is modelled by following the zero-dimensional approach, whereas the cylinder open cycle as well as the other engine components are modelled according to the mean value concept. The proposed model combines the advantages of the mean value and zero-dimensional models allowing for the calculation of engine performance parameters including the in-cylinder ones in relatively short execution time and therefore, it can be used in cases where the mean value model exceeds its limitations. A large marine four-stroke Diesel engine steady state operation at constant speed was simulated and the results were validated against the engine shop trials data. The model provided results comparable to the respective ones obtained by using a mean value model. Then, a number of simulation runs were performed, so that the mapping of the brake specific fuel consumption for the whole operating envelope was derived. In addition, runs with varying turbocharger turbine geometric area were carried out and the influence of variable turbine geometry on the engine performance was evaluated. Finally, the developed model was used to investigated the propulsion system behaviour of a handymax size product carrier for constant and variable engine speed operation. The results are presented and discussed enlightening the most efficient strategies for the ship operation and quantifying the expected fuel savings.

1. Introduction

The shipping industry has been facing a number of challenges due to the unprecedented rise of fuel prices [1–3], the increasing international concern and released regulations for limiting ship emissions and their impact on the environment [4] as well as the reduction of charter rates [5]. This combination of conditions has brought the subject of energy efficiency to the agenda of the maritime industry and of the corresponding academic research.

Improvements in energy efficiency can be obtained in several areas of ship operations and design [6,7]. Among the different components responsible for energy losses on-board a ship, however, it has been widely shown that the main engine(s), and in a less extent the auxiliary engines, occupy a crucial role, as they are responsible for the conversion of the fuel chemical energy to mechanical, electrical or thermal energy for covering the respective ship demands [8]. In this respect, engine manufacturers have developed a number of measures for improving engine efficiency and reducing pollutant emissions. In electronically controlled

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http://dx.doi.org/10.1016/j.apenergy.2015.05.024
0306-2619/

Nomenclature

Symbols

A	area (m^2)
$BMEP$	brake mean effective pressure (bar)
$bsfc$	brake specific fuel consumption (g/kW h)
c_d	discharge coefficient
c_V	specific heat at constant volume (J/kg K)
d	Cylinder bore (m)
h	specific enthalpy (J/kg); heat transfer coefficient (W/m^2 K)
HR	Heat release rate (J/°CA)
$\dot{H}$	Energy flow (W)
I	Polar moment of Inertia (kg m^2)
LHV	fuel power heating value (J/kg)
k	Coefficients; revolutions per cycle
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
N	rotational speed (r/min)
p	pressure (Pa)
pr	pressure ratio
P	power (W)
Q	heat transfer (J)
$\dot{Q}$	heat transfer rate (W)
R	gas constant (J/kg K)
r_c	compression ratio
t	time (s)
T	temperature (K)
u	specific internal energy (J/kg)
V	Volume
V_D	Engine displacement volume (m^3)
w	Velocity (m/s);weight factors (–)
W	Work (J)
z_{cyl}	number of engine cylinders

Greek symbols

γ	ratio of specific heats
Δ	difference
$\Delta\phi$	Crank angle difference (°)
Δ_{cy}	engine cycle duration (°)
ε	Air cooler effectiveness
η	Efficiency
λ	Air–fuel equivalence ratio (–)
ρ	density (kg/m^3)
ϕ	crank angle (°)
τ	torque (Nm)

Subscripts

a	air
amb	ambient
AC	air cooler
AE	Auxiliary engines
AF	air filter
$comb$	combustion
cor	corrected
cy	cycle
cyl	cylinder
C	compressor
d	downstream
E	engine
e	exhaust gas
el	electrical
ep	exhaust pipe
eq	equivalent
ER	exhaust receiver
EV	exhaust valve
EVO	Exhaust valve open
f	fuel
GB	gearbox
ht	Heat transfer
id	ignition delay
in	inlet
IR	inlet receiver
IV	inlet valve
IVC	inlet valve closing
ME	Main engines
out	outlet
$pump$	pumping
P	propeller
ref	reference
$scav$	scavenging
SG	shaft generator
Sh	shafting system
SOC	start of combustion
T	turbine
TC	turbocharger
tot	total
u	upstream
vol	volumetric
w	wall
W	Cooling water

engines [9,10], timings for injection and exhaust valve opening/closing are managed by computer-controlled high-pressure hydraulic systems instead of being operated directly by the camshaft; waste heat recovery systems [11–14] are now used for recovering part of the energy rejected by the engines to produce thermal and/or electrical power; with the aim of improving the propulsion engines low loads performance, retrofitting packages for turbocharger units isolation, exhaust gas bypass and turbochargers with variable geometry turbines have been presented [15–18].

Design, experimentation and prototyping are expensive processes in manufacturing industries, and in particular in the case of marine engines. As a solution to this issue, computer modelling of engines and their systems/components has been extensively used as a mean of testing alternative options and possible improvements during the engine design phase by employing a limited amount of resources. Engine models of a varying range of accuracy and computational time can be employed depending on the required application [19,20]. Cycle mean value engine models (MVEM) [21–29] and zero-dimensional models (0-D) [30–36] are extensively used both for the evaluation of engine steady-state performance and transient response, in cases where the requirements for predicting details of the combustion phase are limited. The former are simpler and faster and provide adequate accuracy in the prediction of most engine output variables [25,29]; the latter include more detailed modelling of the engine physical processes and therefore, more realistic representation of the physical processes as well as higher accuracy can be obtained at the expense of additional computational time.

MVEMs are based on the assumption that engine processes can be approximated as a continuous flow through the engine, and hence average engine performance over the whole operating cycle. As a consequence, the in-cycle variation (per crank-angle degree) of internal parameters such as pressure and temperature cannot be estimated [27,37]. MVEMs have been extensively described in the scientific literature [38–40] and were employed for modelling of marine Diesel engines, both two-stroke [25–29] and four-stroke [21–23].

Zero dimensional (0-D) models operate per crank-angle basis by using the mass and energy conservation equations, along with the gas state equation, which are solved in their differential form, so that the parameters of the gas within the engine cylinders and manifolds, such as pressure, temperature and gas composition can be calculated. Combustion is modelled by using phenomenological models of either one zone, which are an adequate compromise of process representation and accuracy, or multi zones, which offer more detailed representation of the combustion process and prediction of exhaust gas emissions.

Both MVEMs and 0-D models offer specific trade-offs in terms of accuracy, computational time, and required input/provided output. However, when the focus lies in the energy performance and analysis of the system, it is widely recognised that the single-zone 0-D models provide the best trade-off between computational effort and performance, whilst more advanced modelling is generally needed for obtaining additional details on pollutant formation processes [19].

MVEMs have been used to simulate marine engines operation and predict the engine performance parameters, but they cannot predict the in-cylinder parameters variation as well as the specific fuel consumption for the cases where inlet receiver pressure varies comparing with a baseline value e.g. in electronically controlled versions of marine engines or in engines using turbochargers with variable geometry turbine. On the other hand, 0-D models can handle such cases with the drawback of considerable execution time. An attempt to combine MVEMs with 0-D models were presented in Livanos et al. [41], where mapping of the cylinders using a 0-D tool was first performed and subsequently the maps were linked with a mean value model. The derived model was used to design the control system and test alternative control schemes for an ice-class tanker performing manoeuvres in iced sea water. In Ding et al. [42], a Seiliger cycle approach was used in conjunction with a MVEM for representing the in-cylinder process. In both cases, a considerable pre-processing phase is required; in the first case to set up and run the 0-D model as well as for elaborating the results and create the required maps; in the latter case for calibrating the Seiliger model constants based on available experimental data.

The objective of this work is to propose a modelling approach that combines the computational time of a mean value approach with the required contribution from a 0-D model for calculating in-cylinder parameters that cannot be available if a pure mean value approach is employed. This combined MV-0D modelling approach uses a 0-D model for representing the cylinder closed cycle (IVC to EVO in the case of a four-stroke engine), whereas it employs a faster mean value approach for simulating the open part of the cycle (EVO to IVC) as well as for the other engine components. In this respect, the in-cylinder parameters variation as well as the engine performance can be adequately predicted in the whole engine operating envelope, thus surpassing the limitations of the mean value engine models.

2. Engine model description

The engine model is implemented in MATLAB/Simulink environment according to a modular approach. The utilised blocks and connections are shown in Fig. 1.

The engine inlet and exhaust receivers are modelled as control volumes, whereas the turbocharger compressor and turbine are modelled as flow elements. For the engine cylinders a hybrid flow element- control volume approach is used as explained below. An engine governor controls the fuel flow using a proportional-integral (PI) controller law with torque and scavenging pressure limiters, whilst the engine load and the ordered speed are considered input variables to the model. The working fluid (air and exhaust gas) is considered ideal gas and therefore, the fluid properties depend on gas composition and temperature. For the calculation of the exhaust gas composition, the following species were taken into account: N_2, O_2, H_2O and CO_2.

2.1. Shafts dynamics

The engine crankshaft and turbocharger shaft rotational speeds are calculated according to the following equations, which represent the conservation of the angular momentum in the respective shaft lines:

$$\frac{dN_E}{dt} = \frac{30(\eta_{Sh}\tau_E - \tau_P)}{\pi I_{Sh}} \tag{1}$$

$$\frac{dN_{TC}}{dt} = \frac{30(\tau_T - \tau_C)}{\pi\, I_{TC}} \tag{2}$$

where I_{Sh} represents the total inertia of the engine-propeller shafting system including the engine crankshaft, gearbox, shafting system, propeller and entrained water inertia, τ represents the torque and N the shaft speed, whilst subscripts E, P, T, C and TC represent the engine, propeller, turbine, compressor and turbocharger

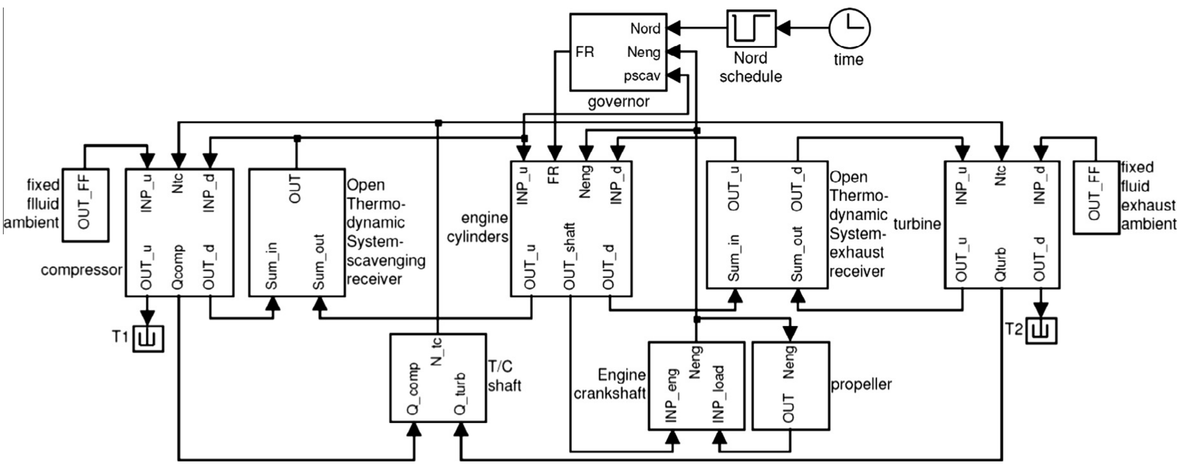

Fig. 1. Matlab/Simulink implementation of marine diesel engine model.

elements, respectively. The shafting system efficiency is considered a function of the engine load as described in [43].

2.2. Turbocharger components

The compressor is modelled using its steady state performance map, which provides the interrelations between the compressor performance variables, in specific: corrected flow rate, pressure ratio, corrected speed and efficiency. Turbocharger speed and pressure ratio are considered input to the model, which allows the computation of the corrected flow rate and efficiency through interpolation [27]. The turbocharger shaft speed is calculated in the turbocharger shaft block, whilst the compressor pressure ratio is calculated according to the following equation, which accounts for pressure losses in the air cooler and filter:

$$pr_C = \frac{p_{IR} + \Delta p_{AC}}{p_{amb} - \Delta p_{AF}} \tag{3}$$

where pr_C, represents the compressor pressure ratio and the subscripts *IR*, *AC*, *AF* and *amb* represent the inlet receiver, air cooler, air filter and ambient conditions, respectively. The pressure in the inlet receiver and the ambient pressure are taken from the inlet receiver and fixed fluid elements, connected downstream and upstream of the compressor element, respectively. The air filter and air cooler losses are considered to be proportional to the square of the compressor air mass flow rate.

The temperature of the air exiting the compressor is calculated according to the following equation, which was derived by using the compressor efficiency definition equation [44]:

$$T_{C,d} = T_{C,u}\left(1 + (pr_C^{(\gamma_a-1)/\gamma_\alpha} - 1)/\eta_C\right) \tag{4}$$

where $T_{C,d}$, $T_{C,u}$, γ_a and η_c represent the compressor outlet and inlet temperature, air heat capacities ratio and compressor efficiency, respectively. The compressor absorbed torque can subsequently be calculated according to the following equation:

$$\tau_C = 30\dot{m}_C(h_{C,d} - h_{C,u})/(\pi N_{TC}) \tag{5}$$

The specific enthalpy of the air exiting the compressor is calculated by using the respective temperature calculated from Eq. (4), whereas the specific enthalpy of the air entering the compressor is taken from the fixed fluid element connected upstream.

The temperature of the air exiting the air cooler is calculated based on the air cooler effectiveness definition equation [44]:

$$T_{AC,d} = \varepsilon T_W + (1 - \varepsilon)T_{C,d} \tag{6}$$

where ε and T_w represent the air cooler effectiveness and the cooling water inlet temperature, respectively. The air cooler effectiveness is assumed to be a polynomial function of the air cooler air mass flow rate. The specific enthalpy of the air exiting the air cooler is calculated by using the respective temperature as derived by Eq. (6).

The turbine is modelled using its swallowing capacity and efficiency maps, which allow the calculation of turbine flow rate and efficiency through interpolation. The turbine pressure ratio is calculated according to the following equation, by taking the exhaust pipe pressure losses into account, which are considered to be proportional to the square of the exhaust gas flow rate:

$$pr_T = \frac{p_{ER}}{p_{amb} + \Delta p_{ep}} \tag{7}$$

where the subscripts *ER* and *ep* refer to the exhaust receiver and the exhaust pipe, respectively. The exhaust gas outlet temperature is calculated by using the turbine efficiency definition equation [44], whereas the turbine torque is derived by using the following equation:

$$\tau_T = 30\dot{m}_T(h_{T,u} - h_{T,d})/(\pi N_{TC}) \tag{8}$$

The specific enthalpy of the exhaust gas exiting the turbine is calculated by using the respective temperature, whereas the specific enthalpy of the exhaust gas entering the turbine is taken from exhaust receiver element connected upstream.

2.3. Inlet and exhaust receivers

The flow receiver elements (inlet and exhaust receiver) are modelled using the open thermodynamic system concept [44–46]. By applying the mass and energy conservation laws considering that the working medium is an ideal gas, which can be represented by its pressure, temperature and equivalence ratio, and neglecting the dissociation effects and the kinetic energy of the flows entering/exiting the receivers, the following equations are derived for calculating the mass and temperature time derivatives:

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{9}$$

$$\frac{dT}{dt} = \frac{\dot{Q}_{ht} + (\dot{m}h)_{in} - (\dot{m}h)_{out} - u\frac{dm}{dt}}{mc_v} \tag{10}$$

where $\dot{m}$ and $(\dot{m}h)$ represent the mass and energy flow rates and the subscripts *in* and *out* denote the flows entering and exiting the flow receiver, respectively. Heat transfer is not considered for the inlet receiver, whereas for the case of exhaust receiver, the heat transferred from the gas to the ambient is estimated by using the temperature difference, the exhaust receiver surface and the heat transfer coefficient. The latter is calculated using a typical Nusselt–Reynolds number correlation for gas flowing in pipes [47]. The pressure of the working medium contained in the engine receivers is calculated by using the ideal gas law equation.

The properties of the working medium (air for the inlet receiver; exhaust gas for the exhaust receiver) are calculated by using the respective temperatures and the equivalence ratio for the case of exhaust receiver.

2.4. Engine cylinder modelling

The engine cylinders are considered to be a hybrid element that combines functionalities from the mean value and the zero dimensional approaches as explained below.

2.4.1. Open cycle modelling

The open part of the engine cylinders cycle (gas exchange period) is modelled by using the mean value approach. In this respect, the mass and energy flows entering and exiting the cylinders are calculated in per cycle basis. The mass flow rate of air entering the cylinders is calculated by considering the pumping mass flow rate and the scavenging flow rate (during the valve overlap period), as follows:

$$\dot{m}_a = \dot{m}_{pump} + \dot{m}_{scav} \tag{11}$$

The pumping mass flow rate is derived by using the following equation as function of the engine cylinders volumetric efficiency, the density of the inlet receiver, the engine displacement volume and the engine speed:

$$\dot{m}_{pump} = \frac{\eta_{vol}\rho_{IR}V_D N_E}{60k} \tag{12}$$

where k denotes the number of revolutions per cycle.

The engine volumetric efficiency of the process is calculated according to the following equation as suggested in [46] as a function of the engine compression ratio (r_c) and the temperature upstream inlet valve, which is considered equal to the temperature in the inlet receiver (and therefore is taken from the inlet receiver block):

$$\eta_{vol} = \frac{r_c}{r_c - 1}\frac{T_{IR}}{313 + \frac{5}{6}(T_{IR} - 273.15)} \tag{13}$$

The scavenging mass flow rate is calculated according to the following equation, which was derived assuming subsonic flow consideration through the valves [44,45] during the valve overlapping period:

$$\dot{m}_{scav} = c_d A_{eq}\frac{p_{IR}}{\sqrt{R_a T_{IR}}}\sqrt{\frac{2\gamma_\alpha}{\gamma_\alpha - 1}\left(\frac{p_{IR}}{p_{ER}}^{\frac{2}{\gamma_\alpha}} - \frac{p_{IR}}{p_{ER}}^{\frac{\gamma_\alpha+1}{\gamma_\alpha}}\right)} \tag{14}$$

The equivalent cylinders flow area (A_{eq}) can be estimated using the instantaneous area variations for an engine cycle of the intake and exhaust valves, as follows:

$$A_{eq} = \frac{z_{cyl}}{\Delta\phi_{cy}}\int_0^{\Delta\phi_{cy}}\frac{A_{IV}(\phi)A_{EV}(\phi)}{\sqrt{A_{IV}^2(\phi) + A_{EV}^2(\phi)}}d\phi \tag{15}$$

The mass flow rate of the exhaust gas exiting the engine cylinders is calculated as the sum of the air and fuel flows entering the cylinders, i.e.:

$$\dot{m}_e = \dot{m}_a + \dot{m}_f \tag{16}$$

The fuel mass flow is calculated by using the injected fuel mass per cylinder and per cycle, which is regarded as a function of engine fuel rack position. The latter is adjusted by the engine governor and it is modelled using a proportional–integral (PI) controller law with torque and scavenging pressure limiters, as commonly used by engine manufacturers for protecting the engine integrity during fast transients [48].

The exhaust gas equivalence ratio is calculated by using the fuel and air flow rates as well as the fuel–air stoichiometric ratio, which is a property of the utilised fuel. This is fed to the exhaust receiver and used for calculating the exhaust gas properties.

The energy rate entering the engine cylinders is calculated by using the air mass flow rate derived from Eq. (11) and the specific enthalpy of air, which is taken from the inlet receiver block. The energy rate exiting the engine cylinders is calculated by applying the energy balance to the cylinders block, as described in the following equation:

$$\dot{H}_e = \dot{m}_f LHV \eta_{comb} + \dot{H}_a - \dot{W} - \dot{Q}_w \tag{17}$$

For taking into account the effects of incomplete combustion, the combustion efficiency is considered a function of exhaust gas equivalence ratio. The engine cylinders indicated power and heat transfer rate for the entire engine cycle are required in Eq. (17) and therefore, this calculation is performed for each engine cycle. Eq. (17) can be compared to the way that the respective parameter is calculated for the case of MVEM, which is based on the fuel energy chemical proportion in the exhaust gas exiting the engine cylinders. The latter has to be provided as input and needs to be calibrated based on the available experimental data [29,49].

2.4.2. Closed cycle modelling

The closed part of the cylinder cycle is modelled according to a 0-D approach by considering the following phases: compression, injection, combustion and expansion. Each phase is modelled by considering the mass and energy conservation equations along with the ideal gas state equation, the working fluid properties and the appropriate submodels to represent the engine combustion and heat transfer.

By considering the energy conservation neglecting the kinetic energy and assuming for the working medium ideal gas and homogeneous mixture, the state of which can be determined by using its pressure, temperature and composition, the following equation is derived for calculating the cylinder working internal energy time derivative:

$$\frac{du}{dt} = \frac{\frac{dQ_f}{dt} - p\frac{dV}{dt} - \dot{Q}_w - u\frac{dm}{dt}}{m} \tag{18}$$

In the above equation, the heat release rate (dQ_f/dt) is calculated by using the combustion model described below. The properties of the working medium (either air or exhaust gas) are calculated as functions of temperature and gas composition [45]; dissociation effects are not taken into account. The cylinder volume and the volume derivative are calculated based on the engine kinematic mechanism particulars [45].

The ignition delay is calculated by using the following equation as proposed by Sitkei [50]:

$$\Delta\phi_{id} = 6\cdot 10^{-3}N\left[a_{id} + b_{id}e^{\frac{7800}{6.9167RT}}(1.0197p^{-0.7}) + c_{id}e^{\frac{7800}{6.9167RT}}(1.0197p^{-1.8})\right] \tag{19}$$

where N represents the engine speed in r/min, T the gas temperature in K, and p the pressure inside the cylinder in bar; a_{id}, b_{id}, and c_{id} are constants estimated as suggested in [51] for large Diesel engines.

Combustion is modelled according to a Vibe curve, which is often referred to as a good approximation for heat release of engines burning a single fuel [51]:

$$\frac{dQ_f}{d\phi} = Q_{f,tot}a(m+1)\left(\frac{\phi - \phi_{SOC}}{\Delta\phi_{comb}}\right)^m e^{-a\left(\frac{\phi-\phi_{SOC}}{\Delta\phi_{comb}}\right)} \tag{20}$$

where ϕ represents the crank angle in degrees, Q_f the heat release, $\Delta\phi_{comb}$ the total combustion duration and ϕ_{SOC} the start of combustion. The value of the constant a is related to the combustion efficiency, and was assumed equal to 5 as suggested in [52]. Constants m and $\Delta\phi_{comb}$ are calibrated at the engine reference point and are updated at the other operating points according to the following equations, as proposed by Woschni and Anisits [51]:

$$\Delta\phi_{comb} = \Delta\phi_{comb,ref}\left(\frac{\lambda_{ref}}{\lambda}\right)^{a_{comb}}\left(\frac{N}{N_{ref}}\right)^{b_{comb}} \tag{21}$$

$$m = (m_{ref} + \Delta m)\left(\frac{\phi_{id,ref}}{\phi_{id}}\right)^{a_m}\left(\frac{N_{ref}}{N}\right)^{b_m}\left(\frac{m_{IVC}}{m_{IVC,ref}}\right)^{c_m} - \Delta m \tag{22}$$

The constants a_{comb}, b_{comb}, a_m, b_m and c_m are regarded as the model calibration parameters, since they can sensibly differ amongst various engines types and sizes as reported in [51].

The cylinder heat losses (from working medium to cylinder walls) are calculated using the standard equation for convective heat transfer assuming a constant value for the cylinder walls temperature:

$$\dot{Q}_w = hA(T_{cyl} - T_w) \tag{23}$$

The average of cylinder heat losses over one engine cycle is calculated and used in Eq. (17). For calculating the heat transfer coefficient from cylinder gas to wall, the Woschni correlation is used [19,51]:

$$h = 127.93p^{0.8}w^{0.8}d^{-0.2}T^{-0.53} \tag{24}$$

where p represents the cylinder pressure in bar, d the cylinder diameter in m, T the cylinder gas temperature in K and w is a representative velocity that takes into account the mean piston speed and the combustion induced turbulence.

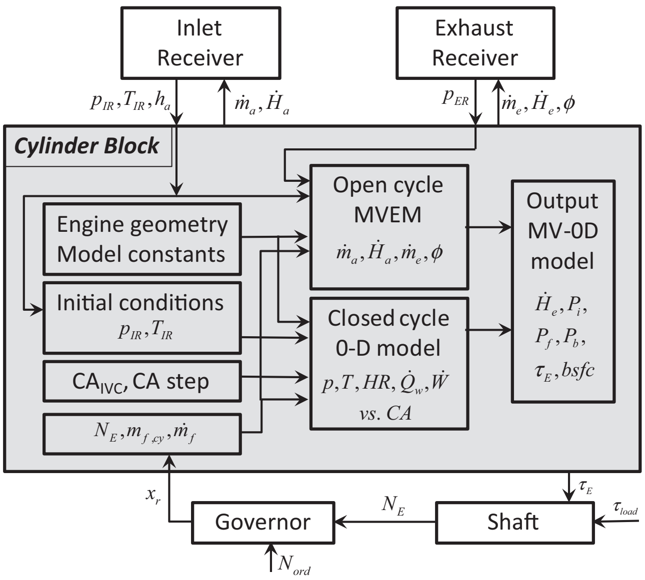

Fig. 2. Graphical representation of the interconnections between the combined MV-0D model and the other engine model elements.

Eqs. (18)–(24) along with the mass conservation and the ideal gas equations form a system of equations that is solved for each crank angle step of the closed cycle from IVC to EVO. A variable time step approach was used with the upper limit equal to 2° crank angle.

2.4.3. Calculation procedure

The graphical representation of the cylinder model including the interconnections between the cylinder block components and the other engine elements are shown in Fig. 2. The input from the adjacent elements include: the pressure, temperature and specific enthalpy from inlet receiver; the pressure from the exhaust receiver; the rack position from governor; and the engine speed from shaft element. The injected fuel amount is calculated by using the rack position and subsequently it is used along with the engine speed for the calculation of the fuel mass flow rate. Using the MV approach, the cylinders air and exhaust gas mass flow rates as well as the energy flow of the air entering the engine cylinders and the equivalence ratio of the exhaust gas exiting the cylinders are calculated.

Based on the inlet receiver pressure and temperature, the initial cylinder pressure and temperature for the start of closed cycle are derived. In specific, the cylinder pressure at the IVC is assumed equal to the inlet receiver pressure; the temperature at IVC is assumed equal to the inlet receiver temperature increased by a reasonable value in order to account for the mixing with the residual exhaust gas [53]; the working medium at IVC is assumed to be air, since the residual exhaust gas fraction is generally small in four-stroke marine Diesel engines [20] and its influence on the prediction of the trapped gas during the compression phase is therefore limited.

The additional input of the 0-D model includes the engine geometry and the model constants as well as the crank angle at IVC and the crank angle step. The pressure, temperature, heat release and heat loss are calculated for the closed cycle and the total closed cycle work and heat loss are derived. The output parameters of the MV and 0-D models are combined for calculating the remaining cylinder performance parameters including the energy flow of the exhaust gas exiting cylinders, the indicated power, the friction power, the brake power, and torque, the brake specific fuel consumption and the engine brake efficiency.

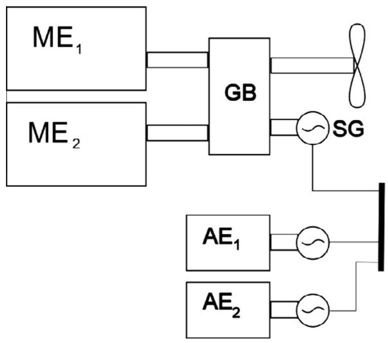

Fig. 3. Schematic representation of ship power plant.

In specific, the indicated mean effective pressure is derived by elaborating the calculated cylinder pressure diagram for the closed cycle and taking into account the pumping work for the open cycle; the latter is the product of the cylinder pressure difference and the engine cylinders displacement volume. The engine brake mean effective pressure is calculated by subtracting the friction mean effective pressure from the indicated mean effective pressure, whereas the engine torque is calculated using the brake mean effective pressure and engine cylinders displacement volume. Several correlations were proposed for the modelling of friction mean effective pressure. In this study, the mean of the correlations proposed by Chen and Flynn [54] and McAuley et al. [55], which are both linear functions of engine speed and maximum pressure, was used.

Then, the calculated parameters are forwarded to the adjacent engine components as shown in Fig. 2. The mass and energy flow rates of the air entering the engine cylinders are provided to the inlet receiver element; the exhaust gas mass and energy flow rates along with the gas equivalent ratio are advanced to the exhaust receiver element; and the engine torque is transferred to the shaft element.

2.5. Model set up procedure and constants calibration

For setting up a new engine morel, the subsequent steps are followed:

- Selection and connection of the blocks representing the engine components.
- Insertion of the required input data in each block.
- Preliminary calibration of the model constants for a reference point.
- Fine tuning of the model constants.

The following input data are needed to set up the model: the engine geometric data, the equivalent area of the cylinder intake and exhaust valves, the steady state compressor and turbine performance maps, the constants of engine combustion model, the propeller loading and the ambient conditions. For integrating the time derivatives of the model governing equations, initial values are also required for the following variables: the engine/propeller and turbocharger rotational speeds, the temperature and pressure of the working medium contained in the engine receivers.

The three main engine parameters describing the combustion (Vibe curve parameters) are not usually known beforehand and need to be determined through a training procedure. The combustion model was initially calibrated versus shop trials performance data for the engine maximum continuous rating (MCR) point, which was considered as the reference point, in order to determine the range for Vibe curve parameters $\Delta\phi_{comb,ref}$ and $m_{comb,ref}$. The parameters defining the shape of the heat release rate function in other engine operating points (Eqs. (21), (22)), are calibrated by

Table 1
Main parameters for the case study vessel and her propulsion engines.

Vessel characteristics	
Deadweight	47,000 t
Design speed	15 kn
Propeller diameter	6.0 m
Number of propeller blades	4
Type/size	Product carrier/handymax
Propulsion engines characteristics	
Type	MaK 8M32C
Bore	320 mm
Stroke	480 mm
Number of cylinders	8
Brake power at MCR	3840 kW
Engine speed at MCR	600 r/min
BMEP at MCR	24.9 bar
Turbocharger units	1 Napier NA 357

Table 2
Calibrated combustion model parameters.

Parameter name	Unit	Lower boundary	Higher boundary	Calibrated value
Heat release, $\Delta\phi_{comb,ref}$	CA degrees	90	110	100.3
Heat release, m_{ref}	(–)	0.2	1	0.2915
Heat release, a_{comb}	(–)	−0.30[b]	0.60[a]	0.9488
Heat release, Δm	(–)	0.00[a]	0.40[b]	0.1919

Both from Ref. [51].
[a] Refers to the suggested values for large diesel engines.
[b] Refers to the suggested values for commercial vehicles direct injection diesel engines.

using the full set of shop trial data. The values proposed by Woschni–Anisits for large two-stroke Diesel engines and heavy duty four-stroke engines are used as boundaries. Finally, all combustion model parameters were fine-tuned at the full engine model. This multiple-level calibration procedure allows for reducing the model set-up time. All calibration steps were performed using a Genetic Algorithm, which had been previously utilised [31] providing promising results.

3. Test case

The four-stroke marine Diesel engine MaK 8M32C was simulated using the described combined MV-0D engine model. The MaK 8M32C is a four stroke, eight cylinders in line, turbocharged engine; one turbocharger unit is used, whereas an air cooler is installed between the compressor and the inlet receiver. The main engine characteristics as well as the required input data were gathered from the engine project guide [56].

Medium speed engines of this size are normally used as propulsion and auxiliary engines in RoRo vessels, small tankers and bulk carriers. For the present study, the propulsion plant arrangement of a handymax size chemical tanker was selected for testing the model predictive ability. In this arrangement, two engines provide power to the ship propeller and one shaft generator unit; a gearbox of two inputs (the shaft of each engine) and two outputs (shaft generator, propeller) is used as shown in Fig. 3. The propeller is of the controllable pitch type, whereas the engine/propeller gearbox ratio is 5.68. Additionally to the shaft generator, the vessel power plant includes two diesel generator sets having a rated power of 690 kWe for providing the required electrical power in cases where the propulsion engines or the shaft generator are not in operation. The main characteristics of the vessel and each propulsion engine are summarised in Table 1.

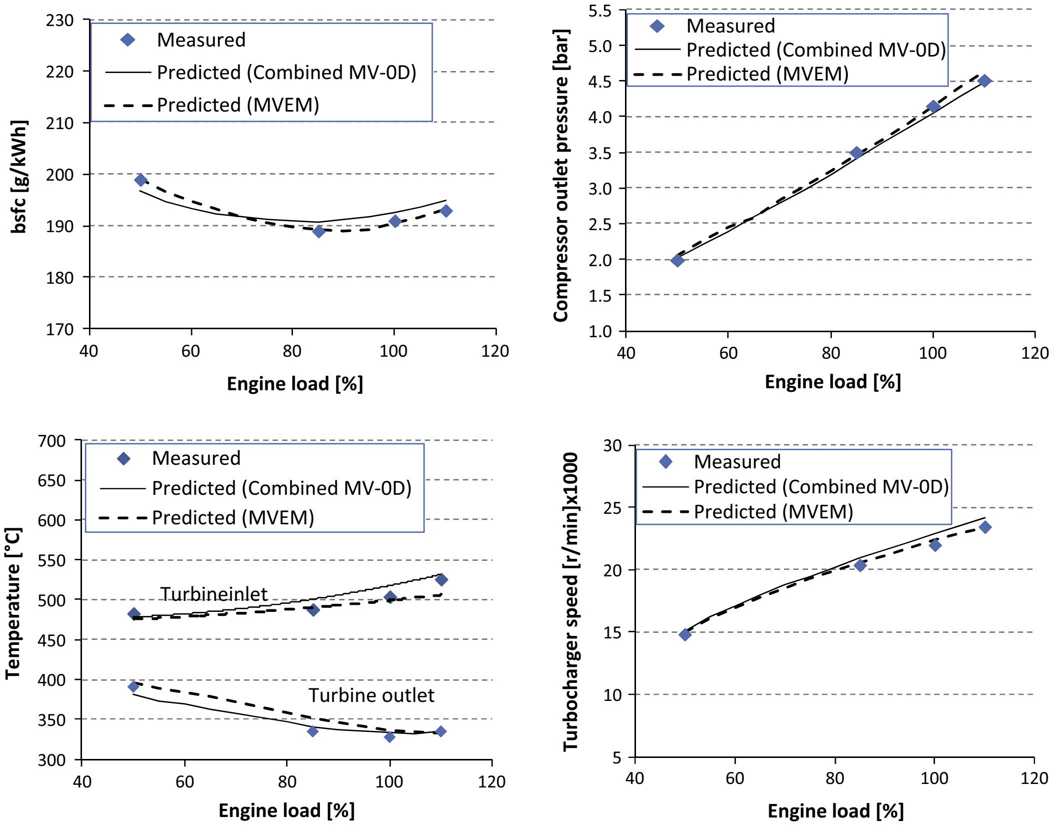

Fig. 4. Steady-state simulation results and comparison with shop trials data.

Table 3
Steady state simulation results, comparison between combined MV-0D model, MVEM and shop trials data.

	50%		85%		100%		110%	
	Combined (%)	MVEM (%)	Combined (%)	MVEM (%)	Combined (%)	MVEM (%)	Combined (%)	MVEM (%)
SFOC	1.16	0.5	−0.94	−0.11	−0.80	0.26	−1.04	−0.10
Inlet pressure	−1.01	−3.00	2.82	1.42	2.51	0.00	−2.31	−2.43
Temperature at compressor outlet	5.66	5.63	4.13	5.13	0.47	2.52	−0.38	2.77
Temperature at turbine inlet	1.26	1.78	−2.17	−0.16	−2.53	1.25	−1.57	3.27
Temperature at turbine outlet	2.48	−1.36	−1.79	−4.78	−1.79	−2.65	0.09	0.42
Turbocharger speed	−1.41	−1.03	−2.40	−0.74	−3.61	−1.35	−2.67	0.45

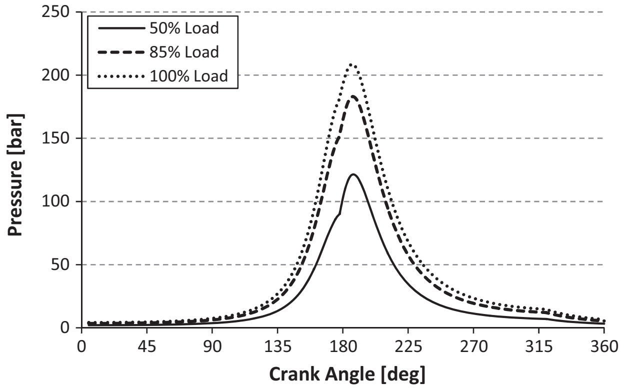

Fig. 5. Combined MV-0D model calculated cylinder pressure diagrams for various engine loads.

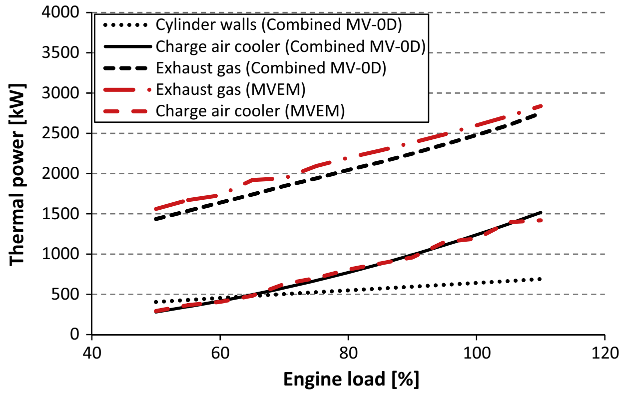

Fig. 6. Calculated cooling power and exhaust gas thermal power versus engine load.

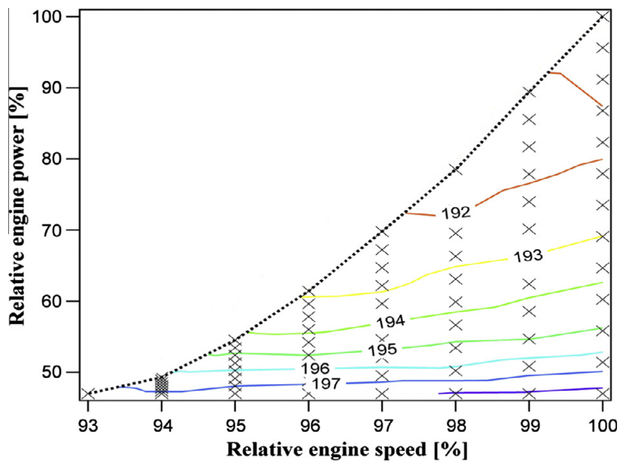

Fig. 7. Engine efficiency map. X marks show the points where the engine efficiency has been calculated using the proposed model. Iso-efficiency lines have been produced through triangulation.

4. Model set up and validation

The engine model was set up providing the required input data, including engine geometric data, turbocharger compressor and turbine steady-state performance maps, model constants and initial conditions. For the investigated engine, the engine shaft speed was used as control variable for the PI regulator and therefore, its initial value was set equal to the desired engine speed value. For the case of the receivers' pressure and temperature, polynomial regressions as a function of engine load were used to estimate the initial conditions derived from the shop trial measurements. The fuel was assumed to be of the marine diesel oil (MDO) type with a lower heating value equal to 42.4 MJ/kg as this fuel type was used in the engine trials. This value was also used for calculating the engine brake specific fuel consumption (bsfc) presented below.

The combustion model constants were calibrated as described in Section 2.5 by using the engine performance data from engine shop trials. Measurements for the engine efficiency and the exhaust gas temperature at turbine inlet at 100% load were used for training the model, whereas measurements at other loads (50%, 85%, and 110% of the engine MCR) were used for the model validation. The relative errors for the brake specific fuel consumption and the exhaust gas temperature at turbine inlet were used in both cases for determining the objective function used by the genetic algorithm, as described by the following equation:

$$f_{obj} = \sum_i w_i |\Delta z_{rel,i}| \quad (25)$$

The calibrated combustion model parameters are shown in Table 2. It must be noted that the α_{comb} parameter was optimised at a value outside the original boundaries in order to provide a better match for *bsfc* variation at low loads.

The steady state simulation results at constant engine speed (equal to 600 r/min) are presented in Fig. 4. In addition, Table 3 contains the obtained percentage relative errors between the parameters calculated using the MVEM, the combined MV-0D model and the respective measured data. The results presented in Table 3 show that the combined model presented herein exhibits reasonable accuracy over the entire engine operating range. It can be inferred that the performance of the two models (MVEM and combined MV-0D) are comparable. The combined MV-0D model shows a slight tendency of underestimating engine brake specific fuel consumption at 50% load, with an error which is however lower than the standard tolerance employed by the marine engine manufacturers. The obtained agreement between the predicted and measured values for the MVEM is better, which is attributed to the effective calibration process for this model. However, it must be noted that the MVEM is placed closer to a black-box model than the proposed combined MV-0D alternative.

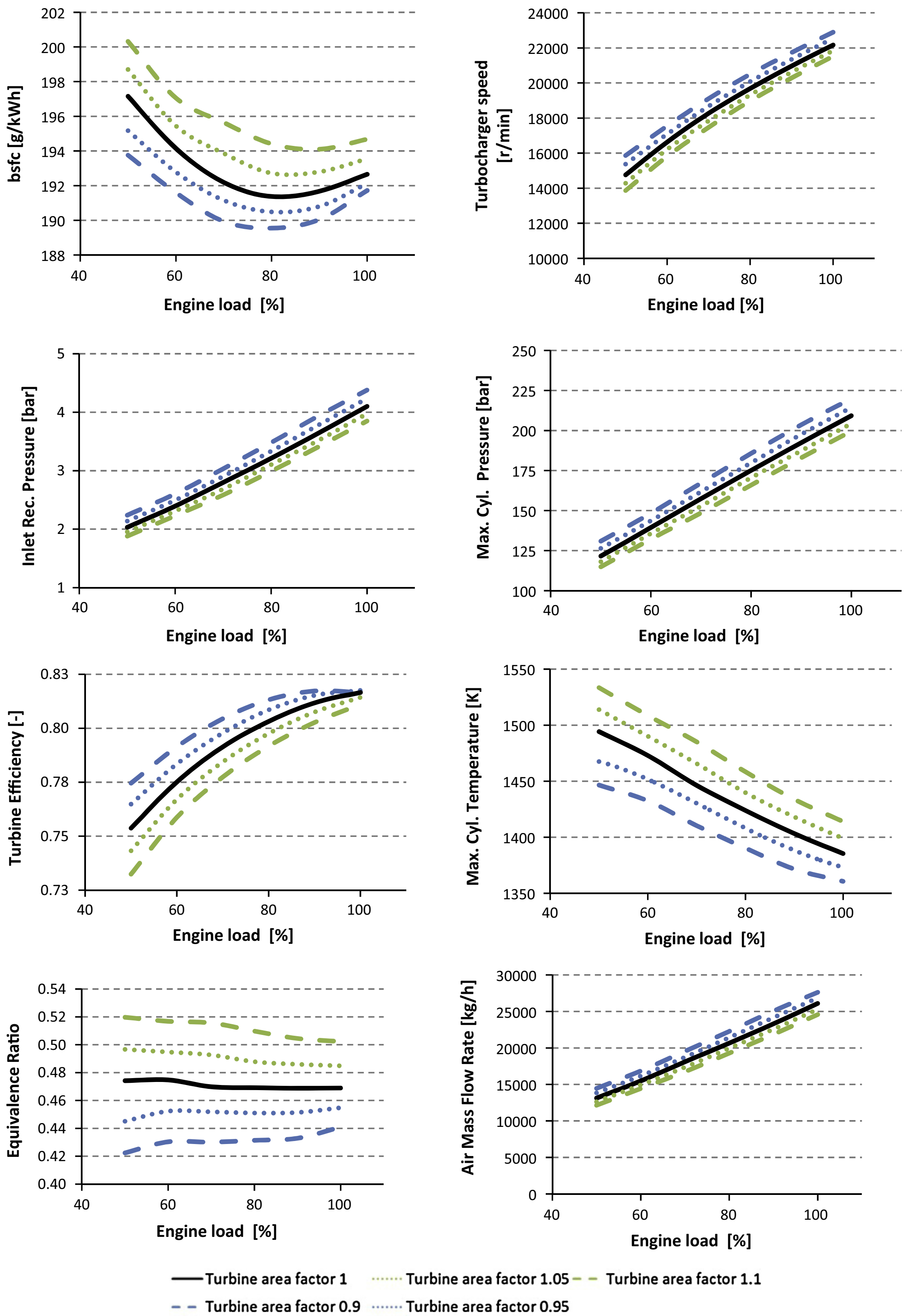

Fig. 8. Combined MV-0D model results for studying the influence of variable turbine area on the engine performance parameters.

5. Model application

Having validated the engine model, it was possible to use it in order to investigate various engine operating cases. The steady-state performance over a range of engine loads from 50% to 110% (using 5% load increase step) at constant engine speed (600 r/min) was first examined. The derived engine parameters including the brake specific fuel consumption (bsfc), the turbocharger speed, the pressure at compressor outlet and the exhaust gas temperature at turbine inlet and outlet are shown in Fig. 4. Values predicted by the combined model are shown, along with the respective predictions obtained using the MVEM model as well as the measured values from shop trials at 50%, 85%, 100% and 110% loads. Once again, good agreement between measured and predicted results by the combined MV-0D and the mean value models can be observed throughout the investigated engine operating envelope. It is inferred from the engine parameters presented in Fig. 4 that the engine was optimised at the high loads region, since the minimum brake specific fuel consumption is obtained at 85% load and the minimum turbine outlet temperature is observed at 100% load. This reflects the market conditions when the ship was designed and built, when slow-steaming and low speed operations were not considered as a possible operating mode.

Fig. 5 shows the cylinder pressure diagrams, which can be calculated by using the proposed combined MV-0D approach and are not available for the case of the MVEM. The cylinder maximum pressure at 100% load was only available and used for initially calibrating the combustion model at this operating point. The relative percentage error that was finally obtained for this parameter at MCR point was 0.8%. As the in-cylinder pressure variation can be predicted by the combined model, this model can be used in the cases were this feature is needed, as an alternative to a 0-D model, which is more complex and computationally demanding. Fig. 6 shows the calculated heat losses to cylinder walls, charge air cooler and exhaust gas for various engine loads. Experimental values for the engine heat losses were not available for validation. Therefore, the results from the proposed combined model are compared with the results from the MVEM; the maximum relative percentage differences for the energy flows of the charge air cooler and the exhaust gas were found to be 9.4% and 7.3%, respectively. The prediction of heat losses to the cylinder walls is a result of the combination of the MV and 0-D models, since the calculation of in-cylinder temperature over the closed cycle allows for the utilisation of the Woschni correlation for the estimation of cylinder heat losses. These results can be of particular interest in the design and optimisation of engine cooling and waste heat recovery systems.

5.1. Engine brake specific fuel consumption map

For the investigated engine, no measurements or data were available at speeds different than the nominal. By using 0-D or the combined model proposed in this work, the estimation of the effect of engine speed on the engine efficiency and on the other parameters can be provided. MVEMs, however, are intrinsically unable to take this effect into account, since the in-cylinder processes are not modelled. The proposed model can provide the required fidelity in the prediction of the influence of engine speed on efficiency.

Fig. 7 shows the results from the application of the model to predict the engine brake specific fuel consumption within the engine power-speed region. It must be noted that the operating envelope of the investigated engine is very narrow and therefore it allows for only limited reduction in engine speed. As expected, the engine speed has the highest impact on engine efficiency in the region close to the upper boundary of the operating envelope, whilst at lower loads the engine efficiency becomes rather insensitive to the engine speed variation. However, a slight bsfc improvement (1–2 g/kW h) could be achieved by operating at an optimal engine speed, which provides the minimum bsfc at each engine load. The utilisation of the proposed model allows for the extension of the efficiency map to the whole engine operating envelope, which is not available from the engine shop tests, the sea trials or the engine project guide. Thus, the more detailed modelling approach followed for the development of the proposed combined model, when compared to mean value models, provides a higher ability of predicting the engine behaviour outside the operational conditions for which the model constants have been calibrated.

5.2. Variable geometry turbine effects on engine performance

The limited size of the allowed speed range for the test-case engine renders a strong limitation for considerable efficiency improvement of the vessel overall propulsion train, since optimal operations at a lower ship speed (e.g. reduction from 15 knots to 12 knots) would require a reduction of the engine speed, which is not permitted for the investigated engine [56]. Reducing turbine area can improve the engine charging process at low loads and consequently can reduce the exhaust gas temperatures. Therefore it is a widely recognised method for broadening the turbocharged engines operating envelope at low loads [15]. Since measurements with a different turbocharger turbine area were not available, the combined model can be used to predict the engine behaviour in this case. On the contrary, MVEMs are not able to fully simulate the effect of increased air charge pressure on engine efficiency and in-cylinder performance parameters. The model derived results using turbine geometric area values in the range from 90% to 110% of its original value for fixed engine speed operation (600 r/min) are presented in Fig. 8.

Reducing turbine inlet area has the main impact of increasing the pressure at turbine inlet, thus providing more exhaust gas energy to the turbine, which increases the turbocharger speed and as a result, the engine inlet receiver pressure. The latter increases the maximum cylinder pressure, which has as a consequence the increase of the engine efficiency and therefore, the decrease of the brake specific fuel consumption. The opposite happens in the case of increasing the turbine geometric area. The above described behaviour is clearly shown in Fig. 8. However, if the turbine geometric area decrease is too large, the engine efficiency may deteriorate and the maximum cylinder pressure might increase above the allowed limit.

The effect of a higher air mass flow rate is also shown in Fig. 8, where the equivalence ratio is shown for different values of turbine inlet area. The increase of the equivalence ratio results in a decrease of the peak temperatures and as a result, in lower engine thermal loading. This behaviour, also shown in Fig. 8, can potentially lead to an extended operational envelope towards lower loads for a given engine speed.

As it was explained above, reducing the inlet turbine area causes an increase in turbocharger speed. At low engine loads, however, the turbocharger speed is far from its high limit and therefore, it should not present a confining factor. In this case, the engine in-cylinder pressure level increases at low loads, which results in a more efficient engine operation.

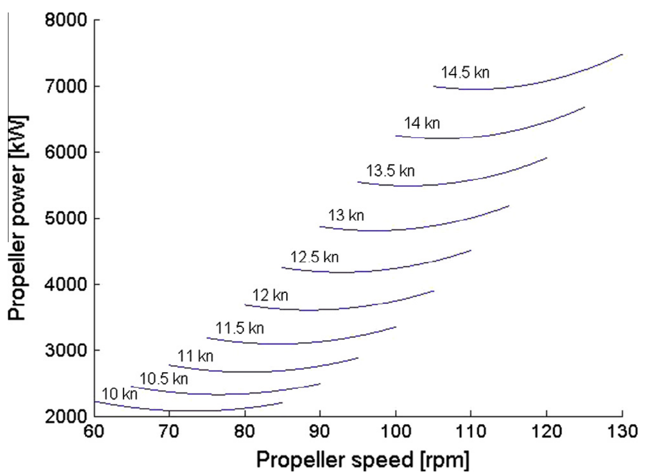

Fig. 9. Propulsion power demand as a function of ship speed and propeller rotational speed for the case study ship considering operation with 15% sea margin.

5.3. Propulsion plant variable speed operation

The model ability to predict the engine performance at different loads and speeds was employed in order to test the potential for improving the operational efficiency of the investigated vessel propulsion plant. The two alternative operational modes that are compared are as follows: (a) the propulsion plant operates with constant engine speed equal to 600 r/min and the shaft generator is used to supply the required electrical power; and (b) the diesel generator sets are used for covering the ship electrical power demand, which allows for operations at variable propeller and engine speeds. The required ship electrical power was set to

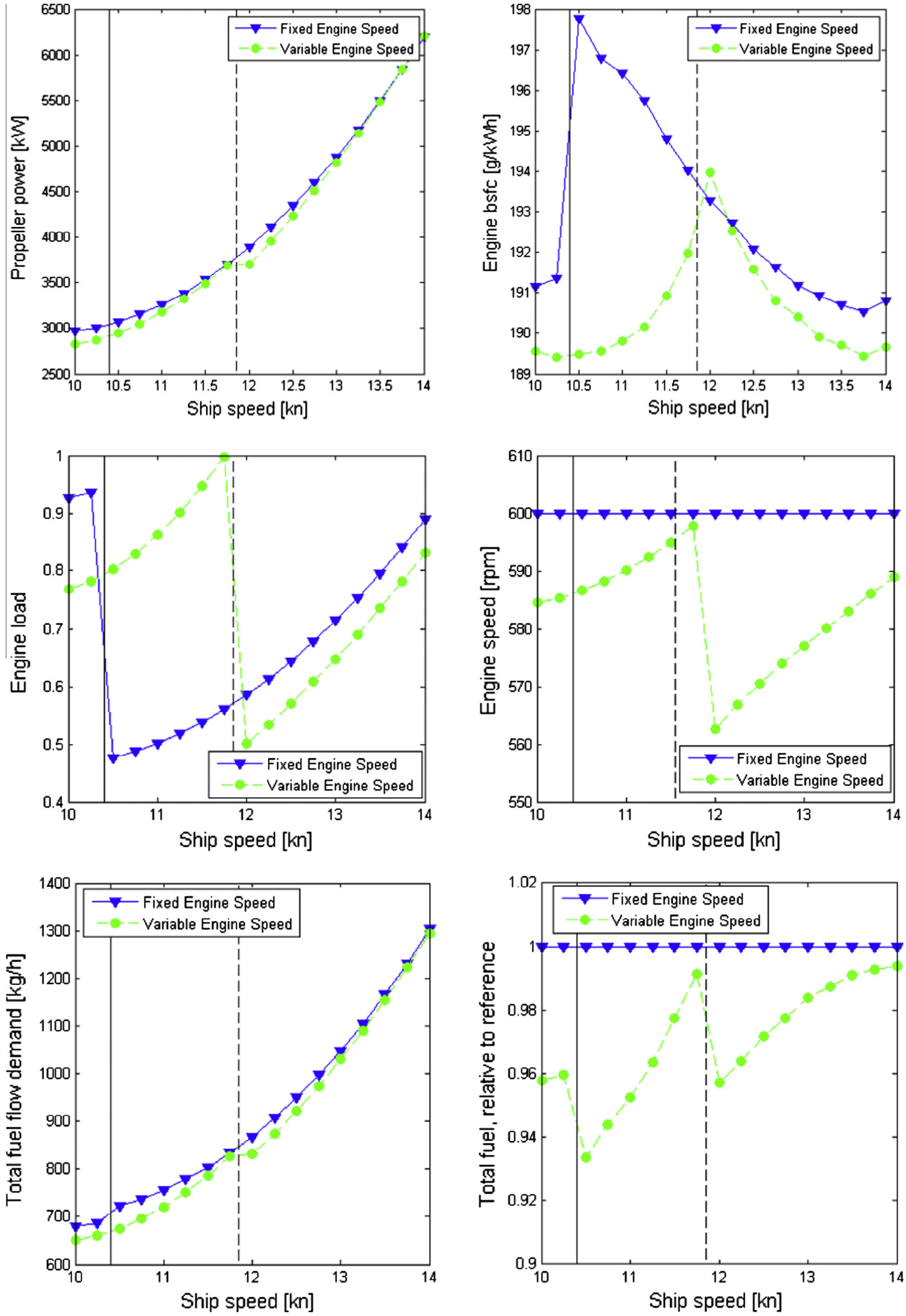

Fig. 10. Derived parameters for the case study ship versus ship speed for constant and variable engine speed operations.

400 kWe based on available measurements from on-board power system.

The rationale for operating at variable propeller speed is attributed to the fact that the propeller efficiency depends on its rotational speed as it is shown in Fig. 9. Hence, operations at constant propeller speed, although allowing for the generation of the ship electrical energy at higher efficiency, lead to lower propeller efficiency at ship speeds lower than the design speed.

The two alternative operational modes were tested for various ship speeds in the range from 10 to 14 knots. The ship resistance was regarded as a polynomial function of the ship speed, whereas the propeller efficiency was derived using the Wageningen series polynomials as described in [57]. The proposed combined MV-0D engine model was used to predict the engine performance under fixed or variable engine speed operating conditions. The following input was additionally used for modelling the two considered cases: (i) gearbox efficiency at full load 98%; (ii) shafting system efficiency at full load 99%; (iii) shaft generator efficiency at rated power 95%; (iv) diesel generators bsfc 210 g/kW h (one diesel generator set operates at 57% load for generating the required electrical power); (v) electric generator efficiency at the considered operating point equal to 95%. The rated efficiencies for the gearbox, shafting system and shaft generator are corrected as proposed in [43] for operation at part loads.

The following additional equations were used for the propulsion system modelling under steady state operating conditions:

$$\eta_{GB}(P_{E1} + P_{E2}) = P_{sh} + P_{SG} \tag{26}$$

$$P_{SG} = P_{el}/\eta_{SG} \text{ and } P_{sh} = P_p/\eta_{sh} \tag{27}$$

$$\dot{m}_{AE} = bsfc_{AE}P_{el}/\eta_{EG} \tag{28}$$

The derived results including the propeller power, the engine bsfc for each operating engine, the engine load and speed, the total fuel flow demand for the operating propulsion engines and diesel generator sets, are shown in Fig. 10. Due to the more efficient propeller operation at the variable speed mode (case b), lower propeller power in this operating mode is required at the entire investigated speed range. It must be noted that for constant engine speed operation (case a), two engines need to operate for ship speed greater than 10.4 knots, whereas for ship speed lower than this, one engine can cover the required total power demand (propulsion and electrical). For the variable engine speed operation (case b), the break point for reducing the number of operating engines from two to one is at 11.8 knots. Therefore, in both cases the maximum engine bsfc point is observed at the break points, since the load of the operating engine increases after switching off one engine unit and consequently the engine bsfc decreases till the engine load reaches the operating point having the minimum bsfc. The engine bsfc is generally lower when operating at variable engine speed, which depends on a combination of the effects of the higher engine load and the lower engine speed.

The calculated total fuel flow demand demonstrates that fuel consumption savings in the range between 1% and 6% can be obtained when operating at variable engine speed even if the ship electrical power is covered by the less efficient diesel generator sets. Fig. 11 shows a breakdown of the different contributions to the total fuel flow demand, which can explain the observed fuel savings. The largest part of the fuel improvement at the variable engine speed mode is attributed to the reduction of propeller power (as a consequence of the propeller efficiency increase). The engine running at different loads, as a consequence of the varying power demand, contributes positively or negatively on the fuel flow demand depending on the ship speed. It must be noted that the contribution at lower ship speeds, which is of the highest interest given the current slow-steaming operations used by shipping companies, is beneficial to reduce the fuel flow demand. As it was expected, the electrical power generation is substantially more efficient when operating at constant speed by using the shaft generator.

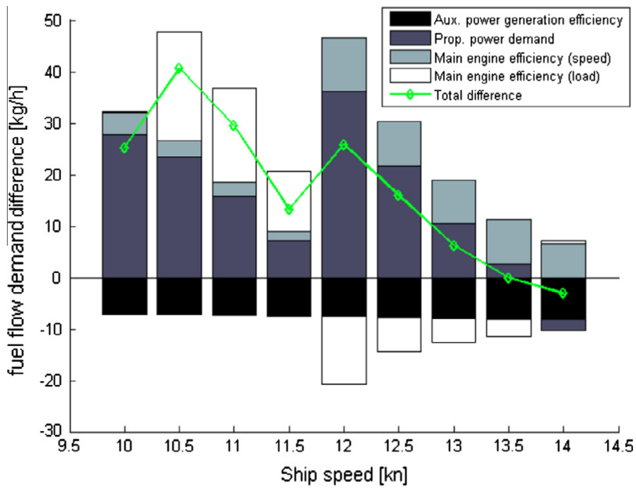

Fig. 11. Breakdown of fuel flow demand difference versus ship speed (positive values refer to higher fuel flow demand in the fixed engine speed case).

It can be concluded from the presented case studies that the developed MV-0D engine model can be used as an alternative to the 0-D models, in studies where varying engine operating conditions in terms of load and speed prevail. The usage of MVEMs is not recommended as calibration of the model constants in a broader engine operating envelope is needed unless extensive experimental data are available.

6. Conclusions

A combined mean-value-zero dimensional model was developed and the simulation of a large marine four stroke Diesel engine was performed. The results were validated against available engine performance parameters measured during engine shop trials and additionally, they were compared with results obtained by using a mean value engine model. Then, the model was used to simulate a number of engine operating points and the results were used for generating the brake specific fuel consumption map in the whole engine operating envelope. Furthermore, cases with varying the turbine geometric area were simulated, so that the model usefulness and superiority against mean value models are presented.

The main conclusions derived from the present work are summarised as follows.

The proposed model can be seen as a compromise between the more empirical mean value models and the more detailed zero dimensional models and can be used in cases where the additional features provided by the 0-D approach are required. The model set up and calibration is faster than the respective one for the 0-D models as only the closed cycle of one engine cylinder is additionally modelled. The calibration of the combustion model parameters is required, which is not too demanding compared with the calibration of the cylinder sub-model for mean value model that requires access to engine performance data and a considerable pre-processing phase.

The model execution time is only slightly greater than the one of the mean value model. In addition, the model prediction capability is quite adequate as it was revealed by the derived parameters comparison against experimental data. The proposed model predicted parameters were comparable with the respective parameters obtained by using the mean value model. The model output parameter set is enhanced, as it additionally includes the in-cylinder parameters, such as pressure, temperature and heat transfer rate. In this respect, the engine efficiency and brake specific fuel consumption can be reasonably predicted in the whole engine operating envelope (with varying load and speed) and in cases where the engine settings change (e.g. variable injection timing and variable geometry turbine).

The model can be used for creating response surfaces for the calculated engine parameters covering the whole operating envelope. The influence of the engine settings on the engine performance are taken into account, since the closed cycle is modelled. Therefore, there is no need for recalibrating the engine cylinder sub-models, as it is required for the mean value models.

The additional case studies of the engine with installed a variable geometry turbine and the vessel propulsion system that can operate in different operating modes revealed the developed model capability of predicting the engine and propulsion system behaviours, respectively. Therefore, the model can effectively assist in the identification of the most efficient engine/propulsion system operations, thus contributing to the quantification of the fuel savings potential.

In conclusion, the combined mean value–zero dimensional model can be used in studies where the mean value model reaches its limitations, especially considering the simulation of electronically controlled versions of marine engines, as well as for simulating engines equipped with variable geometry turbine turbochargers and engine operating in an varying range of operating conditions.

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Paper IV

Energy 80 (2015) 654–665

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

A feasibility analysis of waste heat recovery systems for marine applications

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ARTICLE INFO

Article history:
Received 21 July 2014
Received in revised form
25 September 2014
Accepted 5 December 2014
Available online 6 January 2015

Keywords:
Waste heat recovery
Marine propulsion system
Feasibility analysis
Operational profile
Low carbon shipping

ABSTRACT

The shipping sector is today facing challenges which require a larger focus on energy efficiency and fuel consumption. In this article, a methodology for performing a feasibility analysis of the installation of a WHR (waste heat recovery) system on a vessel is described and applied to a case study vessel. The method proposes to calculate the amount of energy and exergy available for the WHR systems and to compare it with the propulsion and auxiliary power needs based on available data for ship operational profile. The expected exergy efficiency of the WHR system is used as an independent variable, thus allowing estimating the expected fuel savings when a detailed design of the WHR system is not yet available. The use of the proposed method can guide in the choice of the installation depending on the requirements of the owner in terms of payback time and capital investment. The results of the application of this method to the case study ship suggest that fuel savings of 5%–15% can realistically be expected, depending on the sources of waste heat used and on the expected efficiency of the WHR system.

1. Introduction

Shipping business has been expanding almost continuously in the last decades [1], and is today responsible for between 80% and 90% of the overall global trade [2]. When we observe that today global trade, compared to 1950, is more than 100 times larger in terms of volume and value of goods transported [3], the importance and role of shipping in today's economy becomes apparent. However, shipping is now subject to important challenges. Bunker fuel prices are today three times higher than they were in the 80's [4] and fuel costs are estimated to account for between 43% and 67% of total operating costs depending on vessel type [5]. Moreover, upcoming environmental regulations on sulphur oxides, nitrogen oxides and greenhouse gases will exert an additional leverage on fuel costs [6]. This phenomenon will be more pronounced in present and future emission controlled areas, i.e. USA coastal waters, the Baltic Sea, and the North Sea, where regulations will be stricter.

Various fuel saving solutions for shipping are available and currently implemented. Operational measures include improvements in voyage execution, engine monitoring, reduction of auxiliary power consumption, trim/draft optimization, weather routing, hull/propeller polishing, slow-steaming. Design measures can relate to the use of more efficient engines and propellers, improved hull design, air cavity lubrication, wind propulsion, fuel cells for auxiliary power generation, waste heat recovery, pump frequency converters, cold ironing [6]. Several scientific studies have been conducted on these technologies, and a more detailed treatise would be out of the scope of this work, which focuses particularly on the use of WHR (waste heat recovery).

Despite their high brake efficiency Diesel engines waste large amounts of heat to the environment, especially (but not only) in the exhaust gas. Several alternative ways to recover the waste energy produced by Diesel engine on board ships have been proposed and applied in the past [7]. The focus of this paper lies however in the utilisation of WHR for the supply of mechanical/electrical power to the ship. In spite of their still rather limited application in the shipping industry, WHR systems for auxiliary power generation have been widely studied in literature. When four-stroke engines are employed, the relatively high temperature in the exhaust gas (~320 °C [8]) allows the employment of a standard steam cycle. This technology was explored, among others, by Theotokatos et al., who proposed a techno-economic evaluation of the application of a single-pressure steam cycle to bulk carriers [9] and to ferries [10]. Steam-based Rankine cycles have however been proposed also for

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http://dx.doi.org/10.1016/j.energy.2014.12.020
0360-5442/

Nomenclature

Acronyms

AE	auxiliary engine
CMS	continuous monitoring systems
EGE	exhaust gas economiser
GB	gearbox
HT	high temperature (cooling)
JW	jacket water
LO	lubricating oil
LT	low temperature (cooling)
ME	main engine
ORC	Organic Rankine cycle
SG	shaft generator
SW	sea water (cooling)
WHR	waste heat recovery

Subscripts

0	reference conditions
air	air
Comp	compressor
Cool	after cooler
cyl	cylinder
eg	exhaust gas
in	inlet
max	maximum
out	outlet
prod	products
prop	propeller
S	shaft
Turb	turbine

Variables

β	compression ratio
ΔT_{pp}	pinch point temperature difference [°C]
$\dot{m}$	mass flow [kg/s]
$\dot{Q}$	heat flow [kW]
η	efficiency
η_{en}	energy efficiency
η_{ex}	exergy efficiency
η_{pol}	polytropic efficiency
λ	engine load
c	specific heat (for liquids) [kJ/kgK]
c_p	specific heat at constant pressure (for gases) [kJ/kgK]
T_c	cold sink temperature [°C]
EX	exergy [kJ]
h	specific enthalpy [kJ/kg]
m	mass [kg]
N	number
P	power [kW]
s	specific entropy [kJ/kgK]
T	temperature [°C]
V	volume [m^3]

application to two-stroke engines, in spite of the lower temperature of the exhaust gas after the turbocharger (~275 °C, [8]). Ma et al. proposed and evaluated a single-pressure design, both in design conditions and at part-load [11]. A detailed thermo-economic optimization of a WHR system for a two-stroke engine powered containership based on a steam cycle was proposed by Dimopoulos et al. [12], who also investigated the application of exergy analysis as a mean to improve the understanding of the combined cycle (Diesel engine and WHR system) efficiency and the optimization procedure [13]. Grimmelius et al. proposed a modelling framework for evaluating the waste heat recovery potential of Diesel engines and tested it to marine applications [14]. Steam-based WHR systems for both four- and two-stroke engines are available commercially, among others by MAN, Wärtsilä, Mitsubishi and Alfa Laval. Most of the proposed solutions also involve the use of a turbocharger bypass in connection with a power turbine, particularly effective at high engine load [12].

Organic Rankine cycles (ORCs) are considered as an alternative solution in the case of two-stroke engines given the low exhaust temperatures. ORCs are Rankine cycles employing a working fluid different from water in order to adapt evaporation and condensation temperatures to the heat source. Larsen et al. proposed a methodology for the simultaneous optimisation of the process design layout, working fluid and process parameters depending on the temperature of the heat source [15]; Choi and Kim analysed the performance of a dual-loop WHR system for a medium-sized containership under operational conditions [16], while Yang et al. analysed the performance at part-load and transient conditions for a larger vessel [17]. A comparison of conventional steam cycles with ORCs have been proposed by Hountalas et al. [18], while Larsen et al. also included Kalina cycles in the analysis [19,20]. These studies are of particular relevance since two-stroke engines are by far the most employed prime mover in the shipping industry in terms of installed power [21].

As seen in the previous paragraphs the exhaust gas is of major importance for the WHR potential of Diesel engines. Other sources of waste heat are also available, namely the cooling of combustion air after compression CAC (charge air cooling), lubricating oil, and cylinder wall cooling water JW (jacket water). The use of the exhaust gas alone is the most common configuration, and is often suggested both in scientific literature [11,15–17] and as a “baseline” case by manufacturers. The use of charge air cooling water for working fluid pre-heating is also often suggested in literature [9,10,12,14,18]. Finally, Larsen et al. and some manufacturers propose the utilisation of heat from cylinder cooling on top of charge air and exhaust gas [15,22].

With reference to different types of technologies, case studies, and designs, the previously mentioned works witness quite significant possibility for energy saving when WHR systems are employed, ranging from around 1% for single-pressure steam cycles applied to two-stroke engines [9] to more complex systems based on ORCs (up to 10% [18]) or including the cooling systems as a source for waste heat (over 10% [13]).

Despite the acknowledgement of the importance of ship operational profile on WHR systems performance, this aspect is seldom accounted for in the techno-economic or feasibility evaluation. Some of the authors do not include this part in their work [15,17,19,20]; when a techno-economic analysis is included, expected ship performance is often calculated based on a single operating point [11]. The approximation employed by Theotokatos and Livanos [9], Livanos et al. [10], Choi and Kim [16], and Dimopoulos et al. [12,13], which clearly identifies two or three operational speeds, is suitable for the ships operating according to fixed schedules (ferries [10] and container ships [12,13,16]); however, this might not hold true for other kinds of vessels with less predictable operational profiles.

Ships such as tankers, bulk carriers and general cargo ships often operate in volatile markets [1]. Speed has a major influence on ship power demand (as a rule of thumb, propulsive power demand is a function of the third power of ship speed [23]) and is subject to large variations [24], mostly as a consequence of market evolution [1]; draft also remarkably influences ship power demand, and depending on the ship type it can vary sensibly between different voyages [1,24]. Ambient variables, such as wave height and wind speed, are naturally stochastic and can easily double ship power demand for a given speed [25]. Auxiliary power requirement can vary substantially depending on the ship type [23] and on ship operations. Ship variable operations reflect on a wide range of engine loads, which in turn affect the WHR potential both in terms of the availability of waste heat and of recovery cycle efficiency. Steam cycles proposed by Theotokatos et al. show 75% drop in WHR power generation when decreasing engine load from 100% to 50% load [9]. Similar values are reported by other studies including off-design performance evaluations [10–12,16].

As an initial step towards a more thorough inclusion of the complexity of ship operational profile in the design process of WHR systems, this study aims at proposing a simplified method for the evaluation of the WHR potential and of the related fuel savings for ships operating according to complex operational profiles. More specifically, this study aims at identifying whether the installation of a WHR system on board of a specific vessel could be profitable or not depending on the amount of waste heat available and on how the additional power generated by the WHR system can be used.

2. Description of the investigated case study

Operational data from a case study ship are used for demonstrating the application of the method. The selected ship is a Panamax chemical/product tanker. Fig. 1 shows a conceptual representation of the ship energy generation systems, while relevant ship features are presented in Table 1. Measured data are available from a CMS (continuous monitoring system) installed by a private provider. The available data frequency is 1 point every 15 min of operation, derived from the averaging of an original sampling of 1 point every 15 s.

The ship is propelled by two MaK 8M32C four-stroke Diesel engines rated 3840 kW each. The two ME (main engines) are connected to a common GB (gearbox). One of the GB outputs is connected to the controllable pitch propeller with the speed reduced from 600 rpm to 105 rpm. The second output connects the GB to the SG (shaft generator) which provides 60 Hz current to the ship. Auxiliary power can also be generated by two AE (auxiliary engines) rated 682 kW each. Both the SG and the AEs are connected

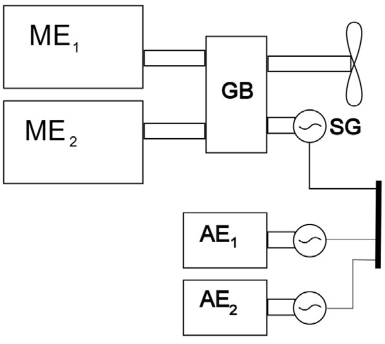


Fig. 1. Conceptual representation of ship propulsion system.

Table 1
Main features of the case study ship.

Ship feature	Value	Unit
Deadweight	47,000	t
Installed power (main engines)	3840 (x2)	kW
Installed power (auxiliary engines)	682 (x2)	kW
Shaft generator design power	3200	kW
Exhaust boilers design steam gen.	1400	kg/h
Auxiliary boilers design steam gen.	28,000	kg/h

to a common switchboard. Auxiliary heat demand is fulfilled by a combination of EGE (exhaust gas economisers) and auxiliary oil-fired boilers.

Referring to the application of WHR, the main sources of waste heat generally are: exhaust gas, charge air cooling, cylinder cooling, and lubricating oil cooling [14]. On the case study ship, the exhaust gas is released in the atmosphere through the funnel after the EGE. All other sources of waste heat are handled by the cooling systems and released to the sea as warm water. The ship is provided with three interconnected cooling systems, namely the HT (high temperature), LT (low temperature), and SW (sea water) cooling. All the heat lost through the charge air cooling, the cylinder walls and the lubricating oil is transferred to these systems, which can therefore be considered as a "secondary" source of waste heat. A graphical representation of the energy flows in the system is provided in Fig. 2, where values of temperature and mass flow are provided for all relevant flows at 85% engine load.

Only operations when the ship is sailing, both in loaded and in unloaded (ballast) conditions, are considered, therefore discarding the time spent in port and manoeuvring. The large variance in the propulsive power requirement showed in Fig. 3a emphasises the need of taking the ship's operational profile in detailed account both in the design and retrofitting process. The auxiliary power demand is more uniform and is below 500 kW for 90% of sailing time (see Fig. 3b). In the remaining 10% of the time, however, demand can increase up to 1500 kW, in connection with operations of e.g. ballast pumps, cargo pumps, and auxiliary boilers.

3. Methodology

The procedure employed in this work is graphically described in Fig. 4 and can be summarized as follows. First, measurements from ship operations are collected from the case study vessel. Second, said measurements are elaborated using ship technical documentation and applying physical principles to calculate the temperatures and mass flow rates of the different sources of waste heat available from the energy system and, consequently, to calculate energy and exergy flows. Third, the WHR system's ability to convert heat to power is represented by an estimated value of its exergy efficiency, and the power generated by the WHR system is calculated. Lastly, the comparison of the available WHR power with the propulsive and auxiliary power demand allows the calculation of the yearly ship fuel consumption with and without the installation of a WHR system. Since this study does not focus on one specific WHR system, steps 3 and 4 are repeated for different values of WHR exergy efficiency, which is used here as an independent variable of the problem.

3.1. Propulsion and auxiliary power demand

Main engine power is calculated according to Equation (1).

$$P_{ME} = \frac{\frac{P_{prop}}{\eta_S} + \frac{P_{SG}}{\eta_{SG}}}{\eta_{GB}} \tag{1}$$

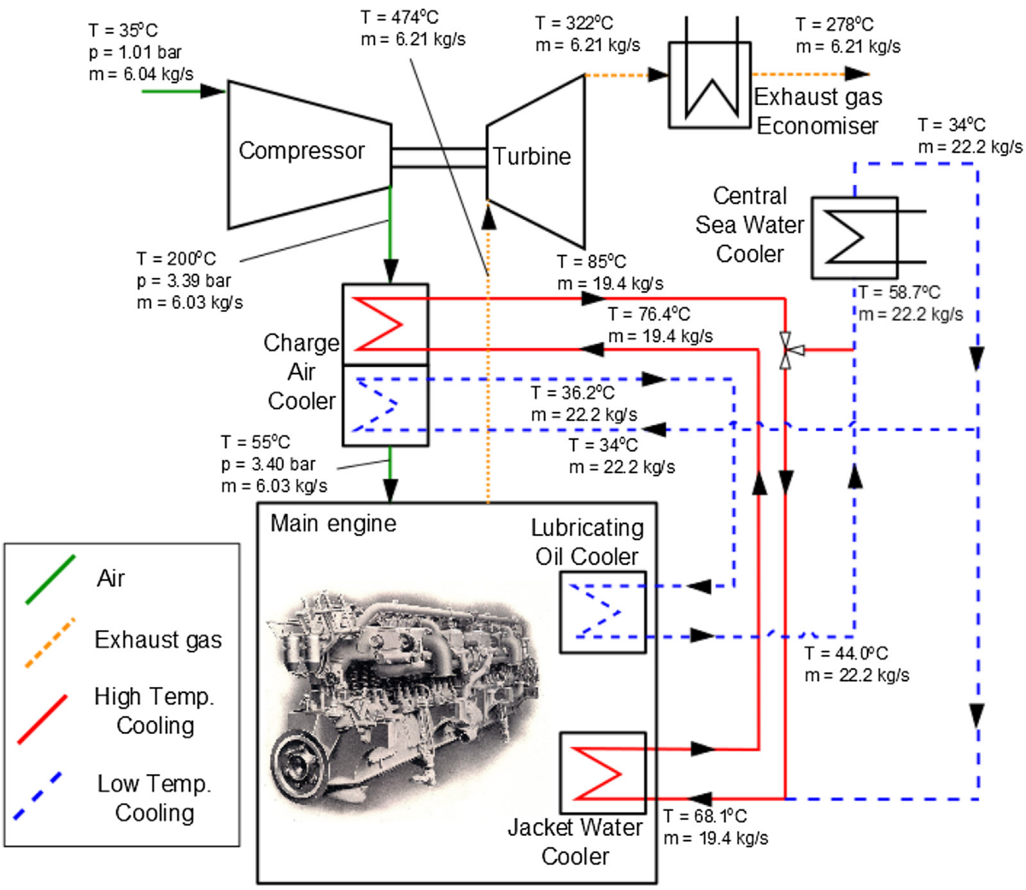


Fig. 2. Graphical representation of main engine cooling systems and heat flows.

where the variables P and η refer to power and efficiency and subscripts prop and S respectively refer to the propeller and the propeller shaft. P_{SG} and P_{prop} are available from the CMS; η_S is assumed equal to 0.99, as suggested by Shi et al. [26]; η_{GB} is assumed equal to 0.983 as reported by the shipyard where the ship was built. Since on the case study ship the SG is dimensioned for high power demand when unloading the cargo, it often operates at low load. A polynomial regression calibrated on the curves proposed in Ref. [27] was therefore used in order to model SG efficiency as a function of load; a value of 95% is assumed for SG design efficiency as reported on technical documentation.

Ships require the generation of both heat and power for auxiliary systems. Auxiliary power demand is measured by the CMS. Auxiliary heat demand was not available from direct measurements and was estimated based on technical data and on available measured values for air and sea water temperatures. Auxiliary heat is generated from waste heat in the exhaust gas by the EGE, and limits the amount of energy and exergy available for recovery.

3.2. WHR power

Waste heat on the case study vessel is available from a number of separate sources. Three alternatives are considered and compared in this study, depending on which of such sources are used for energy recovery: the exhaust gas (A), the exhaust gas and the HT cooling systems (B), and all primary waste heat sources

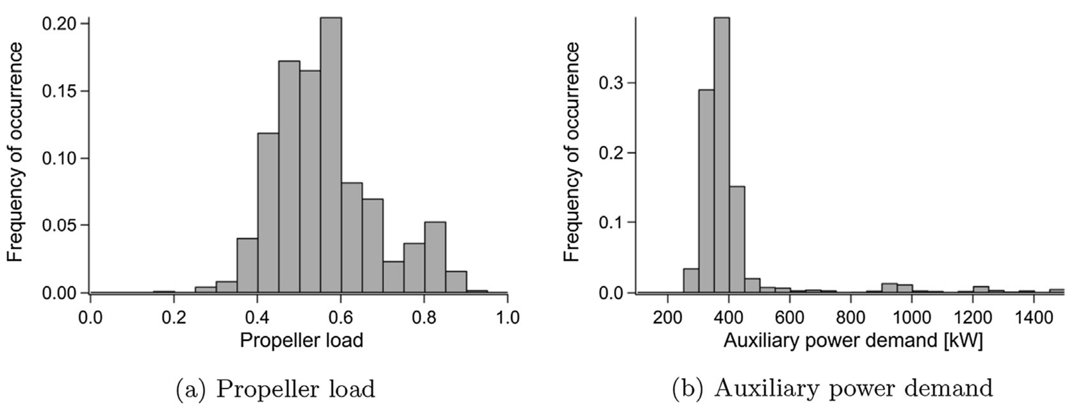


Fig. 3. Load distributions for the propulsion system and the auxiliary power demand.

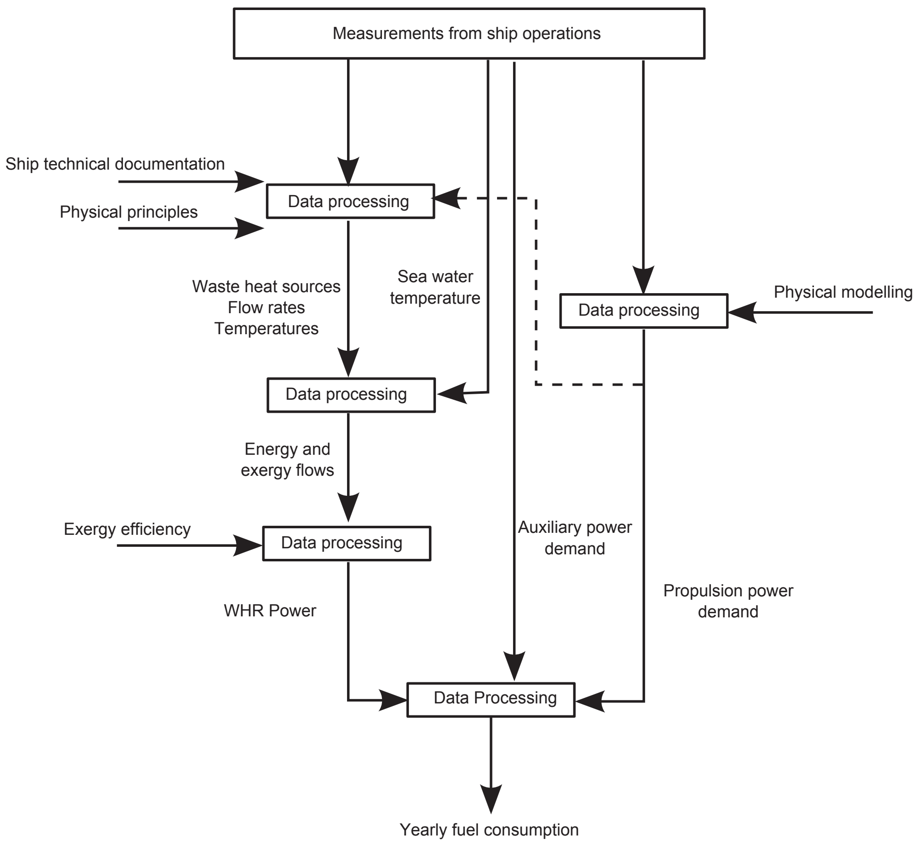


Fig. 4. Methodology.

(Exhaust gas, Charge air cooling, Jacket water cooling, Lubricating oil cooling. Case C). Alternative A represents the most standard and easy-to-retrofit solution; alternative B represents the state of the art [22,28]; finally, alternative C represents the upper boundary, where the highest amount of exergy is available for recovery.

According to the most common arrangement for WHR systems, the recovered power is used for fulfilling auxiliary power demand [9–11]. When additional power is available, the possibility of contributing to main propulsion is often accounted for [12,29]. The additional design/retrofitting effort for allowing this possibility lies in the installation of a shaft generator that can also operate as an electric motor. This solution is not uncommon in the shipping sector, mostly for adding both redundancy and flexibility to the propulsion system [23]. The conversion of the shaft generator to a generator/motor was therefore also explored in the current study.

3.2.1. Exergy analysis

Exergy is defined as the maximum shaft work that can be done by a system in a specified reference environment [30]. For electrical, potential, kinetic, and mechanical energy, exergy and energy flows are normally assumed to coincide; chemical exergy differs from energy only when chemical reactions are involved, which is not relevant in this work. In the case of thermal energy, instead, energy and exergy content are substantially different. For a given amount of matter, its thermal exergy content is defined as showed in Equation (2).

$$\dot{EX} = \dot{m}[(h - h_0) + T_0(s - s_0)] \tag{2}$$

EX, h, and s respectively represent specific exergy, enthalpy, and entropy. The subscript 0 refers to reference conditions, which in this work coincide with measurements of sea water temperature. Starting from the knowledge of the waste heat mass flows and temperatures and from the assumption of all gas flows behaving as perfect gases an alternative form for the calculation of exergy flows can be derived as shown by Equation (3).

$$\dot{EX} = \dot{m}c_p\left[(T - T_0) - T_0\left(\ln\frac{T}{T_0}\right)\right] \tag{3}$$

Only few variables related to waste heat availability are measured by the CMS; the equations, regressions and approximations employed for the calculation of the temperatures and the mass flow rates are provided in Tables 2 and 3; the structure of the main engine and its cooling systems is shown in Fig. 2, where values for flow rates and temperatures calculated according to the method presented in Table 2 are provided for 85% load of the propulsion system. The coefficients related to the amount of energy wasted in the jacket water and lubricating oil are calibrated on engine technical data at full power and are assumed to be load independent.

The set of equations presented in Tables 2 and 3 results in ship waste exergy flows being available as $EX\,(\dot{m}_{fuel}, \lambda_{ME}, T_{SW})$. An example of the resulting flows at a sea water temperature of 20 °C is shown in Fig. 5. The sharp transition that can be observed at 45% load is caused by the shift from one-to two-engines operations.

Fig. 6 shows an extract of 10 days from the dataset used in this work for propulsive power demand, auxiliary power demand, auxiliary heat demand, and for the available exergy from waste flows (exhaust, and whole ship systems). Fig. 6 therefore provides a visualisation of the high variability in both available WHR power and ship power demand.

Table 2
Physical equations, regressions and assumptions employed in the exergy analysis.

Component	Variable	Equation	Eq.#
Air	Temperature, before compressor	$T_{air,Comp,in} = 35\ ^\circ C$ [27]	–
	Temperature, after compressor	$T_{air,Comp,out} = T_{in}\beta^{\frac{k-1}{k\eta_{pol,comp}}}$	(8)
	Temperature, after CAC	$T_{air,CAC,out} = 55\ ^\circ C$ [27]	–
	Mass flow rate	$\dot{m}_{air} = N_{ME}\rho_{in}\frac{\omega_{ME}}{120}V_{cyl,max}N_{cyl}$	(9)
Compressor	Compression ratio	$\beta_{comp} = a_{Beta,0} + a_{Beta,1}\lambda_{ME}$	(10)
	Polytropic efficiency	$\eta_{pol,comp} = a_{eta,0} + a_{eta,1}\lambda_{ME} + a_{eta,2}\lambda_{ME}^2$	(11)
Exhaust gas	Temperature, after turbine	$T_{eg,Turb,out} = a_{eg,0} + a_{eg,1}\lambda_{ME} + a_{eg,2}\lambda_{ME}^2$	(12)
	Temperature, before EGE	$T_{eg,EGE,in} = T_{eg,out} + \frac{\dot{Q}_{aux,heat}}{\dot{m}_{eg}c_{p,eg}}$	(13)
	Temperature, after EGE	$T_{eg,EGE,out} = 150\ ^\circ C$	–
	Mass flow rate	$\dot{m}_{eg} = \dot{m}_{air} + \dot{m}_{fuel}$	(14)
Lub oil	Temperature, after cooler	$T_{LO,Cool,out} = 60\ ^\circ C$	–
	Temperature, before cooler	$T_{LO,Cool,in} = T_{LO,aC} + \frac{\dot{Q}_{LO}}{c_{LO}\dot{m}_{LO}}$	(15)
	Mass flow rate	$\dot{m}_{LO} = 65\frac{m^3}{h}$	–
HT cooling	Temperature, after cooler	$T_{HT,Cool,in} = T_{HT,bC} - \frac{\dot{Q}_{LO}}{c_{HT}\dot{m}_{HT}}$	(16)
	Temperature, before cooler	$T_{HT,Cool,out} = 90\ ^\circ C$	–
	Mass flow rate	$\dot{m}_{HT} = 70\frac{m^3}{h}$	–
LT cooling	Temperature, after cooler	$T_{LT,Cool,out} = 34\ ^\circ C$	–
	Temperature, before cooler	$T_{LT,Cool,in} = T_{LT,Cool,out} + \frac{\dot{Q}_{LT}}{c_{LT}\dot{m}_{LT}}$	(17)
	Mass flow rate	$\dot{m}_{LT} = 80\frac{m^3}{h}$	–

Table 3
Equations and assumptions employed for the calculation of waste heat flows.

Type	Waste heat source	Equation	Eq.#
Primary	Exhaust gas	$\dot{Q}_{eg} = \dot{m}_{eg}c_{p,eg}(T_{eg,Turb,out} - T_0)$	(18)
Primary	CAC	$\dot{Q}_{CAC} = \dot{m}_{air}c_{p,air}(T_{air,Comp,out} - T_{air,CAC,out})$	(19)
Primary	Jacket water cooling	$\dot{Q}_{JW} = 0.414(\dot{Q}_{fuel} - \dot{W} - \dot{Q}_{eg} - \dot{Q}_{CAC})$	(20)
Primary	Lubricating oil cooling	$\dot{Q}_{LO} = 0.444(\dot{Q}_{fuel} - \dot{W} - \dot{Q}_{eg} - \dot{Q}_{CAC})$	(21)
Secondary	HT cooling	$\dot{Q}_{HT} = \dot{Q}_{JW} + 0.776\dot{Q}_{CAC}$	(22)
Secondary	LT cooling	$\dot{Q}_{LT} = \dot{Q}_{CAC} + \dot{Q}_{JW} + \dot{Q}_{LO}$	(23)

3.2.2. *Exergy efficiency*

The exergy related to a specific flow represents the amount of power that could be generated using the flow as the hot source of a Carnot cycle. Exergy efficiency, defined according to Equations (4) and (5), can be used as a representation of the approach of a system to ideal [30]. Compared to energy efficiency, exergy efficiency does not depend on the temperature of the heat source and can be more easily used to compare WHR systems which harvest heat at different temperatures [30].

$$\eta_{ex} = \frac{\dot{EX}_{prod}}{\dot{EX}_{in}} \tag{4}$$

$$\eta_{ex} = \frac{\eta_{en}}{\eta_{Carnot}} \tag{5}$$

In this study the exergy efficiency is used as a parameter which represents the technological level of the recovery system. According to this assumption, WHR systems with low and high exergy efficiency will be respectively referred to as "low-level" and "high level". The exergy efficiency of the systems can be related to different factors: complexity of the thermodynamic cycle, quality of the individual components, size of the heat exchangers. All these factors are supposed to contribute to the total exergy efficiency of the system in relation to their cost (i.e. any improvement that increases exergy efficiency is also expected to increase investment costs proportionally). The existence of a relationship between exergy flows and costs was first proposed by Tsatsaronis and Pisa [31] and is currently often employed under the name of exergo-economic analysis.

Reference values for the exergy efficiency of WHR systems can be derived from existing publications. Theotokatos and Livanos propose single-pressure steam cycles having design exergy efficiencies of respectively around 35%–38% [9,10]; values for ORCs as proposed by Larsen et al. [19] can reach exergy efficiencies of around 60%; Choi and Kim [16] compare a single-pressure steam cycle and a combination of a steam and an ORC cycle, reporting exergy efficiencies of respectively 37% and 61%. The range of exergy efficiency for the analysis was therefore set between 30% and 70% so to consider from today's standard design practice to latest technological improvement. The efficiency of state of the art ORCs

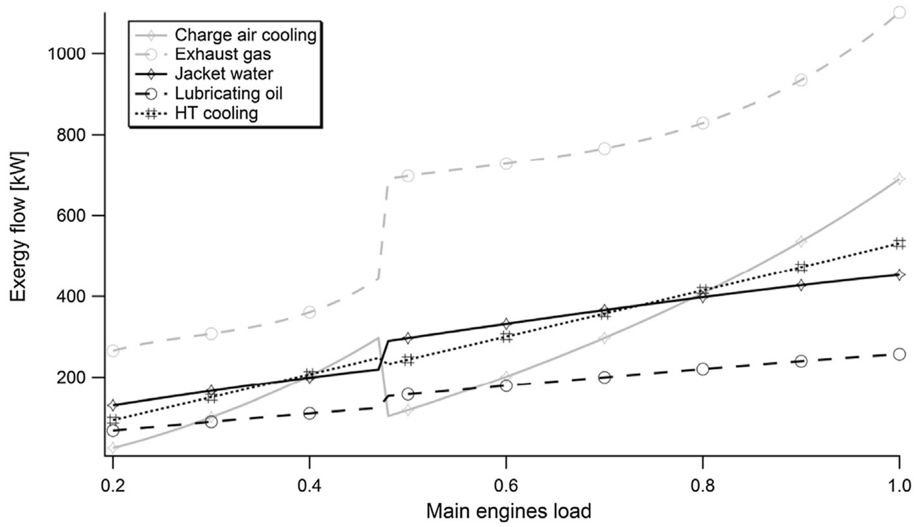


Fig. 5. Waste exergy flows versus main engines load.

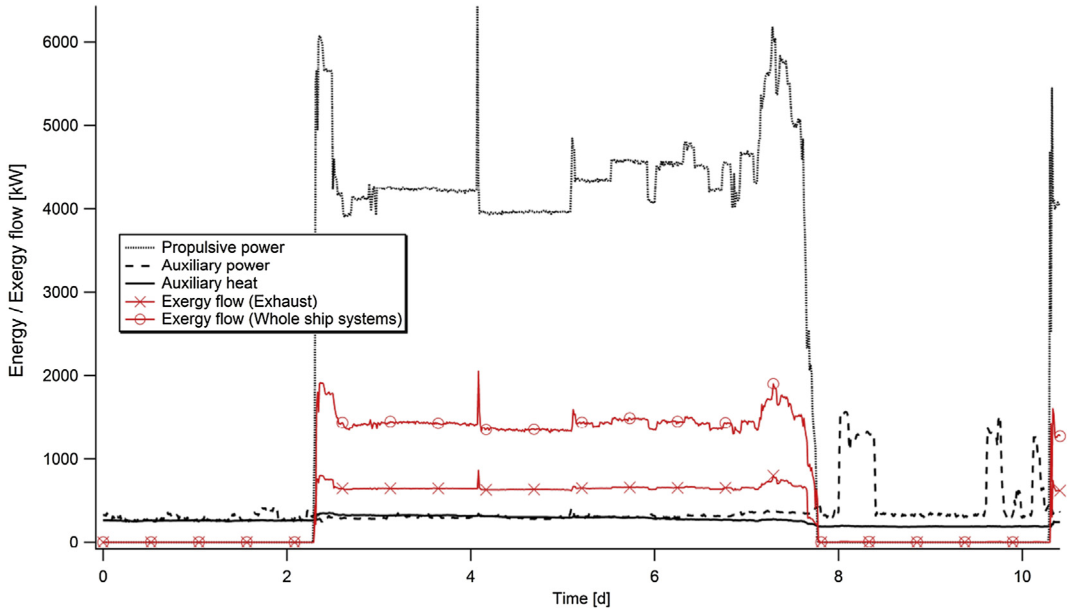


Fig. 6. Extract of 10 days from the available dataset, showing power demand and waste exergy flows.

is estimated using the regressions proposed by Larsen et al. [32] (Equation (6)).

$$\eta_{reg} = a_0 + a_1 T_{in} + a_2 T_{out} + a_3 \eta_{pol} + a_4 \Delta T_{pp} + a_5 T_c \tag{6}$$

where the term η_{pol} refers to the polytropic efficiency of the expander in the recovery cycle, ΔT_{pp} to the minimum pinch point temperature difference and T_c to the temperature of the cold sink. Coefficients for Equation (6) can be found in Ref. [32] and vary depending on the inlet temperature of the heat source.

3.3. Performance parameters

The main performance parameter employed in this study is the reduction of fuel consumption over one year of operations of the selected case study vessel compared to the operations in absence of any WHR system installed.

The percentage of time during which the vessel equipped with a WHR system is able to generate the totality of the auxiliary power demand was also considered; this performance parameter allows to give a better estimation of the increased/reduced need for maintenance connected to the installation of a WHR system. This aspect is particularly of interest because on the selected case study the rotational speed of the engine (and, therefore, of the propeller) is fixed by the requirements of the shaft generator to operate at constant speed. When the WHR system is able to provide all the required auxiliary power and the SG can be shut off, the engine and the propeller can operate at variable speed, thus allowing more efficient operations. This condition is not rare for vessels in the size range as the ship under study, and it has been proved that substantial savings can be achieved by operating CPP propeller ships at variable propeller speed [33,34]. Information such as average load and running time can also be interesting when auxiliary engines are used for auxiliary power generation, as in this case the installation of a WHR system would reduce costs related to maintenance and spare parts.

Finally, an economic evaluation was performed. According to a survey performed by DNV among ship owners, 75% of the respondents to the survey considered 5-years to be the limit to the payback time for a retrofitting technology and 2 years for the remaining 25% [6]. This information can be used in order to calculate an upper boundary for the investment cost of a WHR system given the payback time and its exergy efficiency. The results of the exergy analysis are used to estimate operational savings, under the assumption that other costs and savings than those related to fuel consumption can be considered as negligible. A price of 600 USD/t for marine fuel was employed in the analysis [35]. The maximum purchase price is consequently estimated as shown in Equation (7).

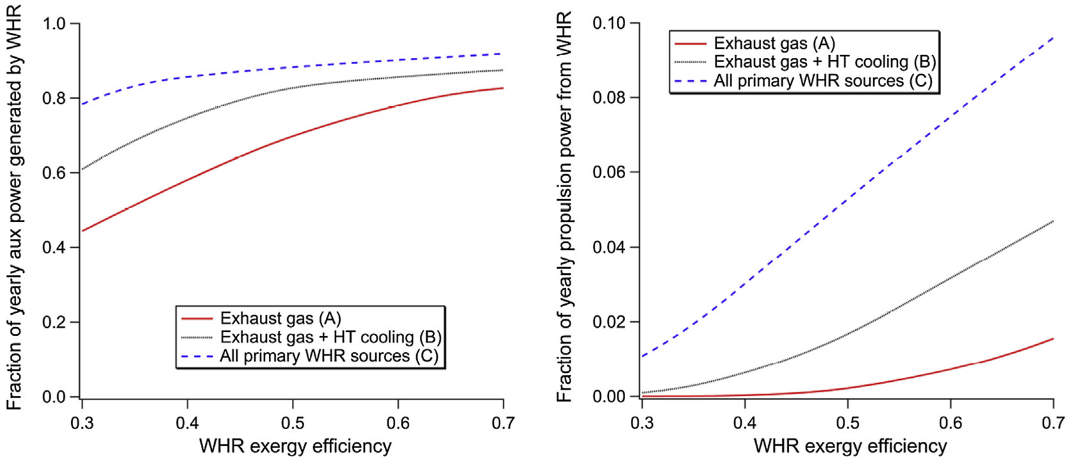
$$C_{\max} = N_{years} \sum_i \dot{m}_{fuel,i} \phi_i \Delta t \tag{7}$$

where N_{years} represents the number of years considered in the payback time calculation, $\dot{m}_{fuel,i}$ the fuel flow and ϕ_i the expected fractional savings at the instant i, and Δt the duration of each time interval.

4. Results

Figs. 7–11 show the results of the application of the method to the case study. All results are presented as a function of the exergy efficiency of the WHR cycle, and for the six alternative arrangements based on the waste heat sources harvested and on the final use of the WHR power as summarised in Table 4.

Fig. 7a shows that between 40% and 90% of the yearly auxiliary energy demand can be expected to be generated by the WHR system. However, Fig. 7a also shows that for higher WHR power the auxiliary power demand tends to get saturated. This phenomenon is clearly shown in Fig. 8, where the curves related to the arrangements where WHR power is solely used for fulfilling auxiliary power demand (A1,B1,C1) tend to reach an asymptote. This phenomenon is more evident for the B and C arrangements, where more WHR power is available. When recovering on the exhaust gas alone, as also shown in Fig. 7b, the power available for propulsion is very limited (less then 0.5% of total propulsion power demand at 0.5 exergy efficiency) and does not justify the installation of means for the utilisation of exceeding WHR power on the only basis of fuel consumption. If the



(a) Auxiliary Power (b) Propulsion

Fig. 7. Fraction of yearly energy demand generated by the WHR system.

results related to the use of all primary waste energy sources (C) represent an ideal maximum for energy recovery, it should be noted that when the HT cooling is also used as waste energy source (B) there is a substantial amount of WHR power that would be lost if no means for using WHR power for propulsion are put in place. For an exergy efficiency of 0.5, almost 2% reduction of propulsive power demand can be achieved in this case.

Fig. 9 represents the expected reduction in yearly fuel consumption compared to the baseline case (no WHR installed) and confirms what presented in the previous figures. Fuel savings from 4% to 8% can be expected in realistic conditions (arrangements A and B, exergy efficiency up to 0.5), while a theoretical maximum of 16% savings for arrangement C at 0.7 exergy efficiency could be reached.

Fig. 10 shows the percentage of sailing time during which the WHR system provides sufficient power for the auxiliaries. Fig. 10 suggests that it is not possible to fulfil both heat and power needs based on the exhaust gas alone unless very complex recovery systems are employed (e.g. dual-cycle ORC cycles with regeneration). A lower exergy efficiency become sufficient when more waste heat sources are recovered; when all available heat sources from the main engines are harvested a rather simple cycle of 0.4 exergy efficiency can suffice to cover the whole auxiliary power demand for more than 80% of the time spent sailing.

The economic evaluation as described in Section 3.3 is presented in Fig. 11. Here, the maximum capital cost which allows a 5-years payback time is shown. The values represented in Fig. 11 should be seen as a guidance for the ship owner interested in the possibility of installing a WHR system for the evaluation of solution proposed by different manufacturers as a function of past ship operations, characteristics of the installed propulsion plant and exergetic performance of the proposed WHR system.

Values for both energy and exergy efficiencies of optimised ORC systems as suggested by Larsen et al. [32] were also shown in

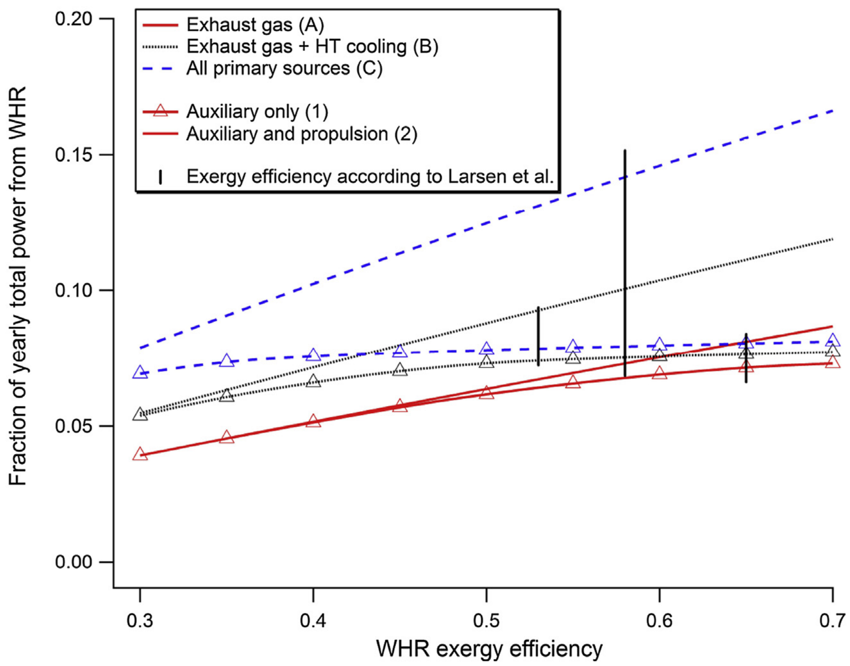


Fig. 8. Fraction of yearly total energy demand generated by the WHR system.

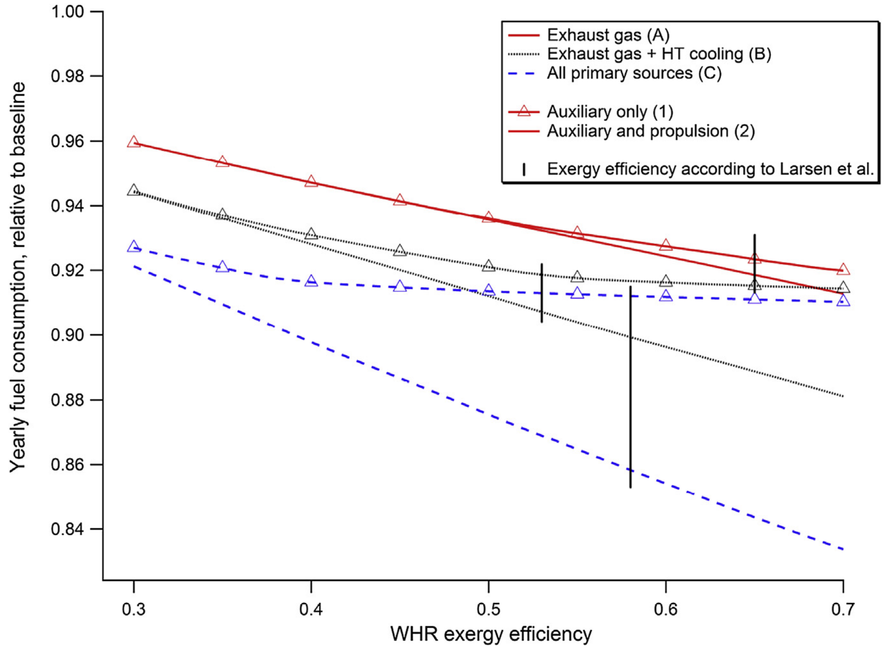


Fig. 9. Expected reduction of yearly fuel consumption.

Figs. 8, 9 and 11 and are summarised in Table 5. The efficiencies that can be reached by ORCs optimised for the specific temperature range from 53% (B case) to 65% (A case). These values represent state of the art ORCs and thus give an indication of today technical limits. If WHR systems with such efficiencies were to be installed on the selected case study, yearly fuel savings of 7–14% could be achieved depending on the selected arrangement (Table 6).

5. Discussion

The results of this study suggest that from an analysis of ship operational profile and of its influence on the potential benefits from the installation of a WHR system it is possible to give an early estimation of how much fuel consumption can be reduced in connection to different alternative systems. This allows to have an initial indication of what type of arrangement should be studied first depending on the type of vessel, operations, and on company investment strategies.

In the case studied during this work the results showed that between 5% and 8% fuel savings can realistically be achieved through the use of WHR systems on the selected vessel. The order of magnitude of these results is in agreement to what presented in other studies found in available literature on the subject [9,10,12,18]. However, the results of the analysis vary sensibly when the operational profile is taken into account. Theotokatos and Livanos [9] propose the installation of a single-pressure system having an average exergy efficiency of 0.35 on a vessel, which is assumed to operate at 85% load for 98% of the sailing time.

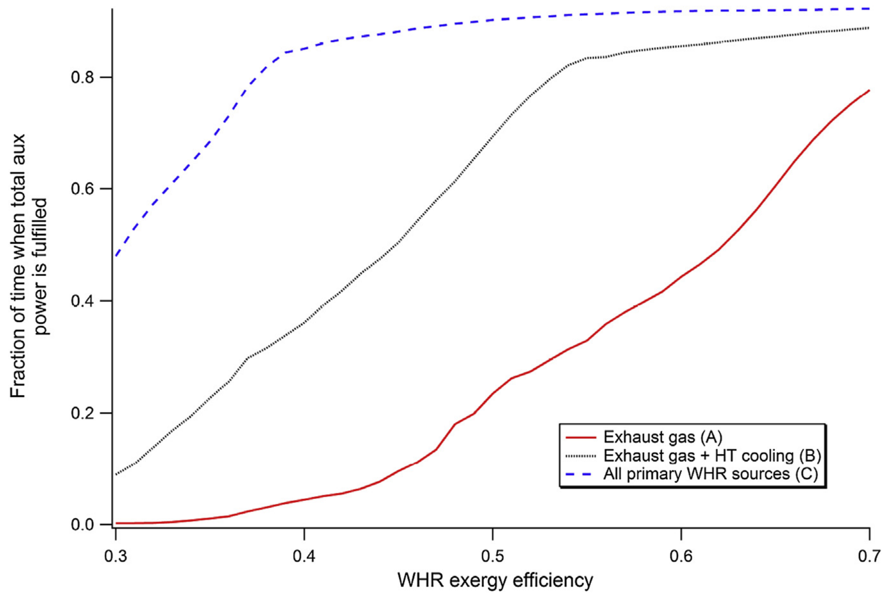


Fig. 10. Fraction of yearly time spent sailing during which the WHR system is able to provide the whole auxiliary power demand.

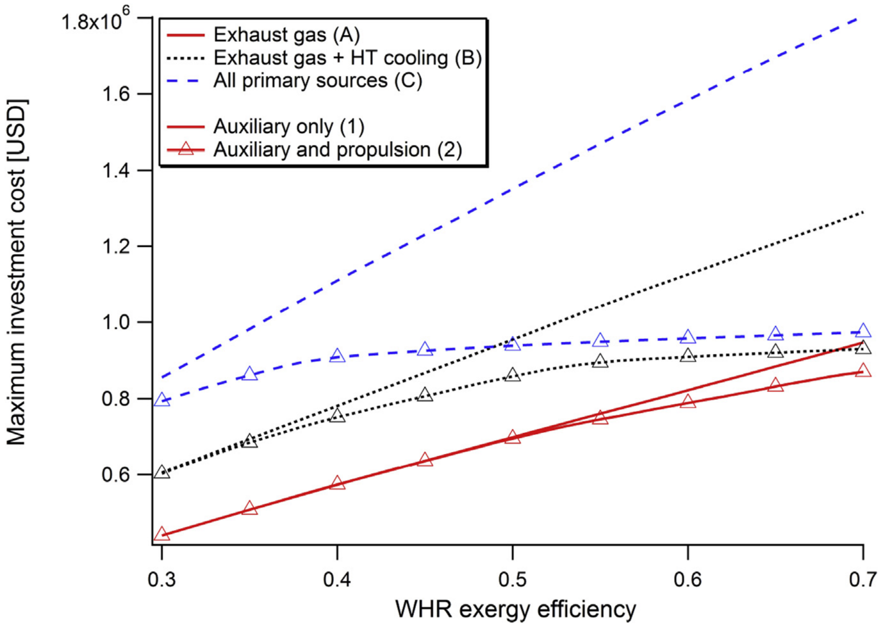


Fig. 11. Maximum capital cost for a 5-years payback time as a function of WHR system exergy efficiency.

Analysing the case study proposed in this work under the same assumptions results in a total yearly reduction in fuel consumption of 263 t (158 kUSD) per year, compared to the 189 t (113 kUSD) per year calculated using a more complex operational profile, giving a substantial 39% difference. Hence, the approximation of constant load proves viable for ferries (as in the case investigated by Theotokatos and Livanos) but not for ships operated on a more variable schedule, such as the case analysed in this study.

The application of the feasibility analysis proposed in this study provided additional insight on the type of installation that should be planned. Two main alternatives are identified that best fit the case study. If resources for new investments are limited, a rather simple WHR system with relatively low efficiency and positioned on the exhaust funnel can be used for the generation of auxiliary power demand on board. Such solution would be relatively cheap and simple and could be performed through a modification of the already existing exhaust gas economiser. If there are possibilities for large investments the use of the HT cooling as a source of waste heat is advised, in particular if connected to the retrofitting of the shaft generator for its possible use as shaft motor. The higher expected investment cost, in this case, would be justified by larger savings. In this second case the possibility of using WHR power for propulsion is suggested, especially if a high-efficiency system is installed.

Table 4
Summary of alternative arrangements.

Case	Waste heat source(s)	WHR power use
A1	Exhaust gas	Auxiliary power
A2	Exhaust gas	Auxiliary power Propulsion
B1	Exhaust gas HT cooling	Auxiliary power
B2	Exhaust Gas HT cooling	Auxiliary power Propulsion
C1	Exhaust gas Charge air cooling Lubricating oil cooling	Auxiliary power
C2	Exhaust gas Charge air cooling Lubricating oil cooling	Auxiliary power Propulsion

This study assumes the exergy efficiency of WHR systems to be constant with load. In this sense, the exergy efficiencies represented in Figs. 9–11 should be seen as average efficiencies of the WHR system. This approximation is justified by noting that exergy efficiency is less load-dependent than energy efficiency; nevertheless, further work should be directed to the accounting of off-design performance.

When compared to previous work in literature, this study employs a high detail in the accounting of the operational profile and it improves the reliability of the results in terms of long-term benefits, as stressed in the previous parts of this work. However, this approach heavily relies on the availability of extensive measurements from the continuous monitoring system. Even though the method is flexible to incomplete datasets, as demonstrated in this study, it relies on inlet data quality for providing insightful analysis and conclusions. In addition, the whole basic assumption that using past ship operations provides a better estimation of future operations can also be challenged and discussed; ship operational pattern can vary substantially over time, as showed for instance by Banks et al. [24].

6. Conclusion

In this paper a method for the estimation of the potential benefits of installing waste heat recovery systems, to be used in the early stages of the retrofitting or the design of a ship, was proposed. The method includes the elaboration of on board measurements in order to calculate the available amount of waste heat, and an exergy analysis for the estimation of the actual amount of useful power that can be recovered. The use of on board measurements ensures

Table 5
Energy and exergy efficiencies for optimal ORC calculated according to Larsen et al.[32]

Waste heat sources	η_{en}	η_{ex}
Exhaust gas	0.29	0.65
Exhaust gas + HT cooling	0.18	0.53
All primary sources	0.18	0.58

Table 6
Numerical results of the feasibility analysis.

		η_{ex}	Alternatives						
			Baseline	A1	A2	B1	B2	C1	C2
Propulsion demand	[TJ]		60.46	60.46	60.46	60.46	60.46	60.46	60.46
Auxiliary demand	[TJ]		6.03	6.03	6.03	6.03	6.03	6.03	6.03
Main engine output	[TJ]	0.3	68.32	65.53	65.53	64.49	64.47	63.23	62.82
		0.4		64.68	64.69	63.51	63.32	62.56	61.21
		0.5		63.88	63.86	62.85	62.20	62.37	59.69
		0.6		63.3	63.1	62.6	61.1	62.2	58.2
WHR energy to auxiliaries	[TJ]	0.3	–	2.67	2.67	3.67	3.67	4.73	4.73
		0.4		3.50	3.50	4.50	4.50	5.17	5.17
		0.5		4.21	4.21	4.99	4.99	5.33	5.33
		0.6		4.71	4.71	5.16	5.16	5.44	5.44
SG energy to auxiliaries	[TJ]	0.3	6.03	3.36	3.36	2.36	2.36	0.00	1.30
		0.4		2.53	2.53	1.53	1.53	0.00	0.86
		0.5		1.82	1.82	1.04	1.04	0.00	0.70
		0.6		1.32	1.32	0.87	0.87	0.00	0.59
WHR energy to propulsion	[TJ]	0.3	–	0	0.00	0	0.06	0	0.66
		0.4		0	0.02	0	0.38	0	1.82
		0.5		0	0.13	0	1.01	0	3.19
		0.6		0	0.44	0	1.91	0	4.53
Total WHR energy	[TJ]	0.3	–	2.67	2.68	3.67	3.73	4.73	5.38
		0.4		3.50	3.52	4.50	4.89	5.17	6.99
		0.5		4.21	4.34	4.99	6.00	5.33	8.52
		0.6		4.71	5.14	5.16	7.08	5.44	9.97
Total fuel consumption	[t]	0.3	4021	3857	3857	3797	3796	3727	3704
		0.4		3808	3808	3742	3731	3684	3609
		0.5		3763	3762	3703	3667	3673	3520
		0.6		3728	3716	3684	3604	3666	3434
Complete fulfilment auxiliary demand	[h]	0.3	–	8	8	366	377	1966	1962
		0.4		182	186	1480	1508	3486	3509
		0.5		962	958	2841	2833	3696	3704
		0.6		1815	1808	3505	3514	3759	3762
Yearly savings	[kUSD]	0.3	–	98	98	134	135	176	190
		0.4		128	127	167	174	202	247
		0.5		155	155	191	212	209	300
		0.6		175	183	202	250	213	352
Maximum installation cost (5-year payback)	[kUSD]	0.3	–	489	489	670	673	881	950
		0.4		638	637	835	868	1009	1233
		0.5		773	777	953	1060	1043	1501
		0.6		876	913	1010	1251	1064	1760

that the effect of the operational profile of the ship on the expected benefits from the installation of a WHR system is accurately captured, an aspect that can become of primary importance in today's volatile shipping market. The application to the case study of a chemical tanker shows that the method can provide important information to the initial phase of the decision making process, when the question lies more in deciding whether the WHR system should be installed or not rather than on which pressure it should operate at.

Using the method proposed in this paper, preliminary results related to the reduction of fuel consumption, avoided use of auxiliary generation equipment and to the capital cost range that would make the WHR installation profitable in a 2- and 5- years horizon could be obtained. In the specific case studied in the paper, fuel savings from 4% to 16% can be achieved, which results in the maximum investment cost ranging between 0.5 and 1.8 MUSD for a 5-years payback time. These results depend on the sources of waste heat employed (exhaust gas, charge air cooling, various types of cooling systems), on the type of complementary auxiliary generation system (shaft generator or auxiliary engines), and on the exergy efficiency of the recovery system. According to the analysis, two main solutions should be considered: either a simple WHR system based on the exhaust gas (low investment cost, low yearly savings) or a more complex system also recovering on the HT cooling systems and with the possibility of using excess WHR power for propulsion (higher investment cost, higher yearly savings).

Acknowledgements

This work is part of a larger project about ship energy systems modeling, performed at the Shipping and Marine Technology Department of Chalmers University of Technology and financed by the Swedish Energy Agency. The authors would also like to thank Bengt-Olof Petersen and Laurin Maritime for their support and discussions. A special acknowledgment is also given to Ulrik Larsen from Chalmers University of Technology and professor Gerasimos Theotokatos from the University of Stratchlyde for the very profitable discussions and reviews of the work.

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Paper V

Ocean Engineering 110 (2015) 85–93

Contents lists available at ScienceDirect

Ocean Engineering

journal homepage: www.elsevier.com/locate/oceaneng

Comparison of different procedures for the optimisation of a combined Diesel engine and organic Rankine cycle system based on ship operational profile

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ARTICLE INFO

Article history:
Received 14 January 2015
Accepted 17 September 2015
Available online 21 October 2015

Keywords:
Waste heat recovery
Marine propulsion system
Organic Rankine cycle
Low carbon shipping

ABSTRACT

At a time of strong challenges for shipping in relation to economic and environmental performance, the potential of waste heat recovery has been identified as among the most important technologies to lower fuel consumption. This paper presents the comparison of four different procedures for the optimisation of a combined Diesel and organic Rankine cycle system with increasing attention to the ship operational profile and to the inclusion of engine control variables in the optimisation procedure. Measured data from two years of operations of a chemical tanker are used to test the application of the different procedures. The results indicate that for the investigated case study the application of an optimisation procedure which takes the operational profile into account can increase the savings of the installation of an organic Rankine cycle from 7.3% to 11.4% of the original yearly fuel consumption. The results of this study further show that (i) simulating the part-load behavior of the ORC is important to ensure its correct operations at low engine load and (ii) allowing the engine control strategy to be part of the optimisation procedure leads to significantly larger fuel savings than the optimisation of the waste recovery system alone.

1. Introduction

In a world where trade is continuously increasing, it is estimated that 80–90% of the goods are transported by sea (UNCTAD, 2012). However, the shipping industry is at present subjected to a challenging transition. Fuel prices have increased three-fold compared to the 80s (Mazraati, 2011) and, although currently at a low point, recent and upcoming restrictions regarding sulphur oxides emissions are expected to further augment fuel prices (DNV, 2012). Furthermore, the recently released regulations on CO_2 emissions from ships (EEDI, SEEMP) will require additional efforts from the industry to reduce its impact on the climate (Devanney, 2011).

Several fuel saving solutions for shipping have been subject of research and development under the aforementioned forces. Operational measures include improvements in voyage execution (Armstrong, 2013), engine monitoring (Sala et al., 2011), reduction of auxiliary power consumption, trim/draft optimisation (Armstrong, 2013), weather routing (Shao et al., 2011), hull/propeller polishing (Khor and Xiao, 2011) and slow-steaming (Armstrong, 2013). Design measures can relate to the use of more efficient engines and propellers, improved hull design, air cavity lubrication (Slyozkin et al., 2014; Mäkiharju et al., 2012), wind propulsion (Schwab, 2005), fuel cells for auxiliary power generation (Sattler, 2000), pump frequency converters, cold ironing (Peterson et al., 2009), and waste heat recovery systems (DNV, 2012). This study focuses particularly on waste heat recovery systems, and in particular on organic Rankine cycles.

Despite their high thermal efficiency, Diesel engines waste large amounts of energy to the environment. Part of this energy is recovered from the exhaust gas to fulfil on board auxiliary heat demand; this demand is however relatively small and leaves potential for further utilisation of the available waste heat for other purposes (Baldi et al., 2014). In particular, WHR systems for the conversion of waste heat to electric power are widely employed in other industrial sectors.

Despite the relative scarcity of industrial applications in shipping, WHR systems for ships have been widely studied in the scientific community. Theotokatos and Livanos proposed a techno-economic evaluation of the application of a single-pressure steam cycle to bulk carriers (Theotokatos and Livanos, 2013) and to ferries (Livanos et al., 2014). Ma et al. (2012) evaluated the part-load performance of a single-pressure design. Dimopoulos et al. (2011, 2012) proposed the thermo-economic optimisation of a

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http://dx.doi.org/10.1016/j.oceaneng.2015.09.037
0029-8018/

steam-based WHR system for a containership powered by a two-stroke engine. Grimmelius et al. (2010) proposed a modelling framework for evaluating the waste heat recovery potential of Diesel engines and tested it to marine applications. Steam distribution systems are widespread in the shipping industry, which makes steam the most ready-to-use solution for ships. Steam based WHR systems for both four- and two-stroke engines are available commercially, among others by MAN, Wärtsilä, Mitsubishi and Alfa Laval. Most of the proposed solutions also involve the use of a power turbine in connection with a turbocharger bypass (Dimopoulos et al., 2011).

Steam-based Rankine cycles are however not suitable for relatively low temperatures (200–250 °C) and power outputs ($<\sim 10$ MW) (Invernizzi, 2013). This makes organic Rankine cycles (ORCs) a competitive alternative in the case of two-stroke engines and smaller propulsion systems. ORCs are Rankine cycles where water is substituted by an organic fluid whose evaporation temperature better fits the available heat source; in addition, some organic fluids have a positively sloped vapour saturation curve, which makes them attractive for avoiding the formation of droplets in the last stages of the expander.

Larsen et al. (2013b) proposed a methodology for the simultaneous optimisation of the ORC process design layout, working fluid and process variables depending on the temperature of the heat source; Choi and Kim (2013) analysed the performance of a dual-loop ORC system for a medium-sized containership under operational conditions, while Yang et al. (2013) analysed the performance at part-load and transient conditions for a larger vessel. Ahlgren et al. (2015) proposed the optimisation of an ORC system for a cruise ship. Soffiato et al. (2015) investigated the use of ORC systems based on low-temperature heat sources (charge air cooler, lubricating oil cooler, and jacket cooling cooler) showing that two-stage ORC configurations can reach much higher net power outputs than simple-cycle ones, at the cost of increased system complexity. Song et al. (2015) compared the recovery of the heat from engine exhaust gas and jacket water cooler with two alternative systems: one made of two separated single stage ORC cycles, and the other made of one two-stage ORC cycle, showing that the latter has only slightly lower performance, but is more economically attractive.

A comparison of conventional steam cycles with ORCs has been proposed by Hountalas et al. (2012), while Larsen et al. (2013a, 2014) also included Kalina cycles in the analysis. These studies are of particular relevance since two-stroke engines are by far the most employed prime mover in the shipping industry in terms of installed power (Haight, 2012).

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

As pointed out by, among others, Banks et al. (2013), ships can have very variable operational profiles, which directly affects the possibilities for energy recovery. In spite of this, to the best of our knowledge, only few studies take the complexity of the ship operational profile into account in the design and optimisation of WHR systems.

In previous work of the authors, we proposed the accounting of the operational profile in a feasibility analysis of WHR systems for ships. This study, however, focused on the required system performance rather than on how to achieve it in terms of optimal cycle variables (Baldi and Gabrielii, 2015). Dimopoulos et al. (2011) identified four operational conditions, and took them into account in the steam-based WHR optimisation problem. Choi and Kim (2013) analysed the operational profile of a case study and identified two main operational conditions of particular relevance for the recovery cycle, and optimised a dual-loop ORC system on those conditions. Kalikatzarakis and Frangopoulos (2014) took the full operational profile complexity into account when optimising an ORC, and showed that different operational profiles have a large impact on the expected economic performance of the system. All other articles previously cited propose an optimisation of the WHR cycle at one well-defined design condition, while part-load operations are simulated, but not taken into account during the optimisation of the WHR system design.

The aim of this paper is to systematically investigate the influence of accounting for the operational profile of the ship in the process of optimising the combined cycle's design and operational variables. We test different optimisation methods, which differ with respect to the amount of part-load points that are evaluated for each objective function evaluation, and with respect to the amount of optimisation variables included in the process.

2. Methodology

In this study, we propose the installation of a WHR system recovering the heat from the main engines exhaust gas for providing additional electric power to the ship. The power generated by the WHR system is primarily used for fulfilling auxiliary power demand; excess power can be converted into propulsive power, using the shaft generator as a shaft motor. This results in the fact that the power generated by the combined Diesel engine–WHR system is equal to the power generated by the Diesel engine with no WHR system installed.

2.1. Description of the case study

The proposed comparison of different optimisation procedures is applied to a Panamax product/chemical tanker. The ship is equipped with a data-logging system which provides measurements of propulsion power and auxiliary power demand with a 15 min frequency. The ship is powered by two MaK 8M32C four-stroke Diesel main engines (ME) rated 3840 kW each. The MEs are connected to a common gearbox (GB), which in turn is connected to a controllable pitch propeller and to the shaft generator (SG, rated 3200 kW). Auxiliary power during port stays is generated by two auxiliary engines (AE) rated 682 kW each. In this study we only consider operations during sea passages, and therefore AEs are not included in the analysis. Conceptual representations of the ship propulsion system in its original version and after the installation of the WHR system are represented in Figs. 1 and 2 respectively.

Fig. 3a and b show the distribution over one year of operation for the propulsion power and auxiliary power demand respectively, while Fig. 4 shows the distribution of the main engine load factor, calculated according to the following equation:

$$P_{tot} = \frac{\dfrac{P_{prop}}{\eta_S} + \dfrac{P_{SG}}{\eta_{SG}}}{\eta_{GB}} \tag{1}$$

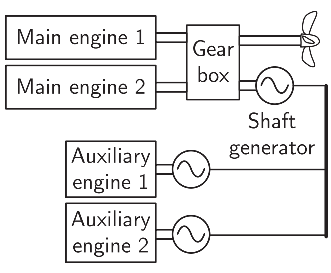

Fig. 1. Conceptual representation of the original ship propulsion system.

where P and η refer to power and efficiency and subscripts *prop* and *S* refer to the propeller and the propeller shaft respectively. P_{SG} and P_{prop} are available from the continuous monitoring system installed on board; η_S is assumed equal to 0.99 (Shi et al., 2009); η_{GB} is assumed equal to 0.983 as reported by the shipyard where the ship was built. In this specific case, the SG maximum power was selected for the simultaneous use of all cargo pumps, which are operated in port during the discharge phase. This however results in the SG operating at very low load factors when the ship is at sea, when power demand is reduced to approximately 400 kW. The expression proposed by Haglind for modelling large ships generator part-load efficiency based on the design point efficiency and the copper loss fraction of the total losses is used (Haglind, 2010):

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

where λ, $\eta_{d,e}$ and F_{cu} represent the load factor, the design efficiency and the copper loss factor of the electric generator respectively. A value of 95% is assumed for SG design efficiency in accordance with technical specifications, while a value of 0.43 for F_{cu} was assumed (Haglind, 2010).

The engine is modelled using a validated zero-dimensional, single zone model (Baldi et al., 2015). The choice of using a zero-dimensional model instead of polynomial interpolations was based on the absence of measured data for engines exhaust gas flow. The model uses a double Wiebe curve for the modelling of heat release, the Woschni correlation for heat losses, and the Chen correlation for friction modelling, as suggested in relevant literature on the subject (Asad et al., 2014; Kumar and Kumar Chauhan, 2013; Scappin et al., 2012). The use of the model allows simulating the engine output in terms of BSFC, exhaust mass flow and temperature as a function of the engine load factor, as shown in Fig. 5. These variables are of particular relevance as they influence the amount of heat available for recovery and its temperature level.

It should be noted that since the propulsion system is composed of two equally sized engines, the definition of the load factor at which the switch between one-engine and two-engine operation happen (λ_{switch}) is of primary importance. This can be observed in Fig. 6 where the engine BSFC and the heat flow in the exhaust gas are plotted versus the load factor of the propulsion system for different values of λ_{switch}.

Under current conditions, where no WHR system is installed, this shift is performed at λ_{switch}=47.5% in order to maximise fuel efficiency and to keep a safe engine load margin. When a WHR system is installed, however, the trade-off between more efficient engine operations and a higher energy flow to the WHR system should be analysed more in detail, as discussed by Larsen (2014). Fig. 6 shows how decreasing the value of λ_{switch} generates a simultaneous decrease of the engine efficiency and increase of the energy flow in the exhaust gas.

2.2. ORC systems modelling

In this paper, a steady-state model of a single ORC WHR system recovering exhaust gas heat from the two main engines is optimised. The model was built in Matlab and uses Coolprop to simulate the working fluid properties. It is here assumed that all the waste heat in the main engines exhaust gas is available for heat-to-power WHR purposes.

A general ORC system is composed of 4 types of components: pump, expander, heat exchangers, and electric generator. The typical structure of an ORC system is presented in Fig. 7.

A detailed description of the modelling approach and equations is provided in Larsen et al. (2015). Hereafter the main equations and assumptions are summarised.

The pump was modelled as shown in Eq. (3) (Quoilin et al., 2011). The coefficients of the regression were adjusted to match the higher efficiency due to larger size (see pump characteristics from commercially available centrifugal pumps described by Manolakos et al., 2001). The isentropic efficiency of the pump (η_p)

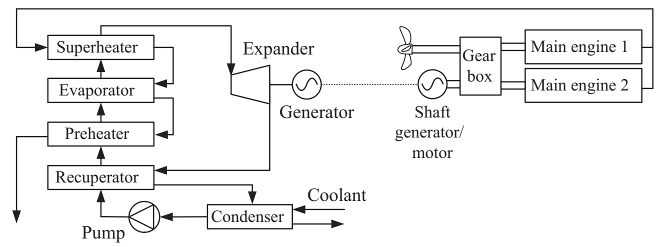

Fig. 2. Conceptual representation of the combined cycle-based propulsion system.

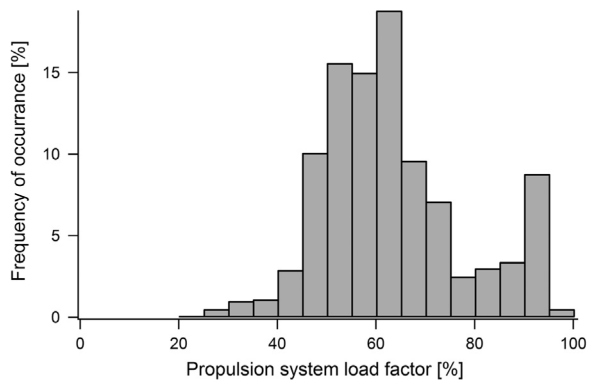

Fig. 4. Yearly distribution of total ship power demand.

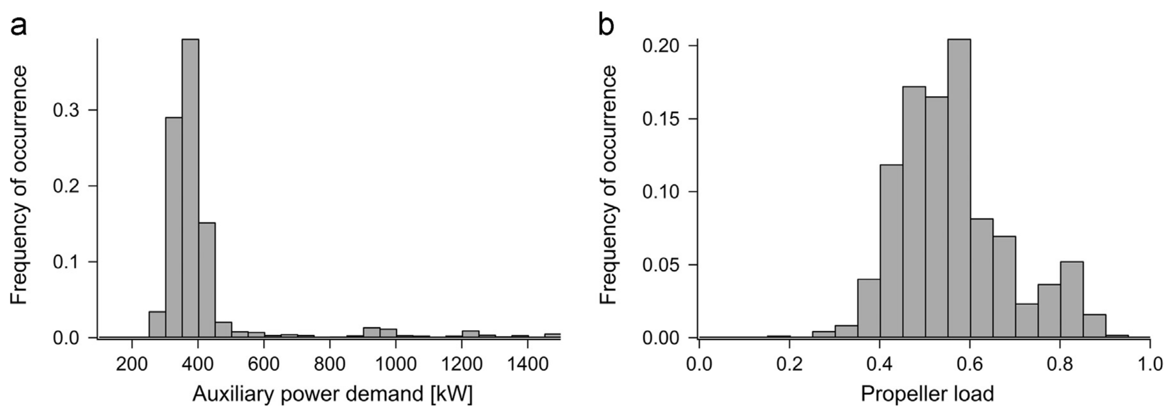

Fig. 3. Yearly distribution of propeller and auxiliary power demand. (a) Auxiliary power and (b) propeller.

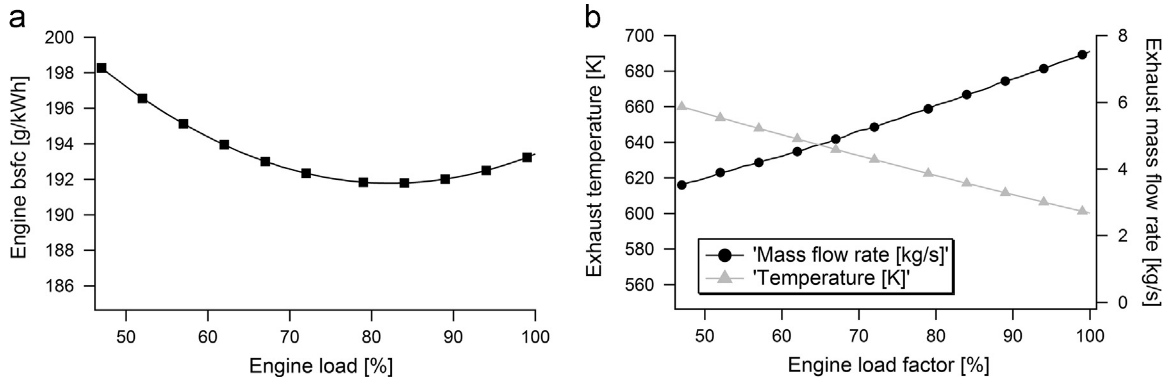

Fig. 5. Diesel engine model output for engine BSFC and exhaust gas. (a) BSFC, and (b) exhaust, mass flow and temperature.

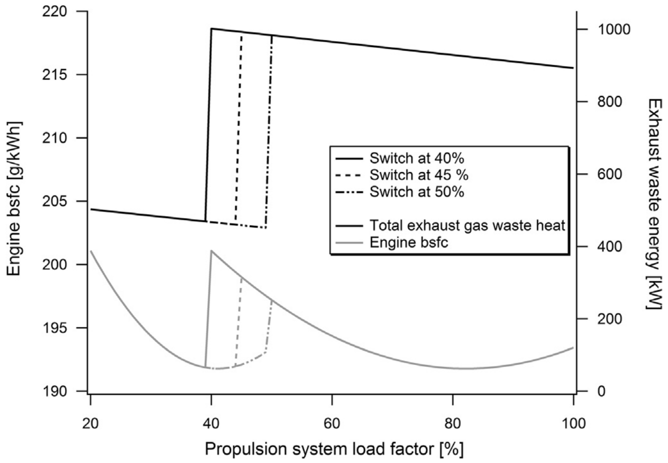

Fig. 6. Engine BSFC and total exhaust waste energy as a function of the load factor for one-to-two engines operation switch.

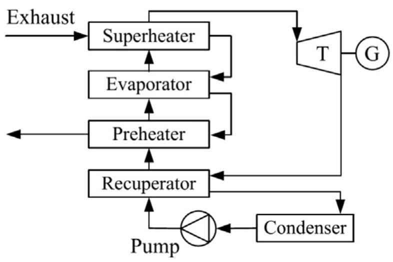

Fig. 7. Sketch of the ORC process.

relative to its design value ($\eta_{p,d}$) can thus be calculated as a function of the pump volumetric flow ($\dot{V}$):

$$\frac{\eta_{pump}}{\eta_{pump,d}} = a_3\left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}}\right)^3 + a_2\left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}}\right)^2 + a_1\left(\frac{\dot{V}_{pump}}{\dot{V}_{pump,d}}\right)^1 + a_0 \tag{3}$$

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

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

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

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

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

where C_{exp} is the turbine characteristic constant

The overall heat transfer coefficient and the heat transfer area are calculated at the design point (UA_d) using the following equation:

$$\dot{Q} = UA\Delta T_{ml} \tag{6}$$

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

$$UA = UA_d\left(\frac{\dot{m}}{\dot{m}_d}\right)^m \tag{7}$$

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

The part-load performance of the ORC depends on the applied control strategy; in the present case a sliding-pressure mode was adopted since it has been suggested that it provides higher efficiency in part-load operations (Weber and Worek, 1993; Adibhatla and Kaushik, 2014). According to a sliding-pressure control mode (as opposed to constant pressure mode) the turbine inlet valves are not throttled in order to keep constant turbine inlet pressure. Instead, the valves are left wide open, and the turbine inlet pressure changes with load adapting to the rest of the system. From a modelling perspective, the part-load evaporation pressures are thus governed by the Stodola equation, by the heat transfer processes, and by the characteristic curve of a pump equipped with a variable frequency motor (respectively Eqs. (5), (7) and (3)).

For the simulation of part-load performance, we imposed the constraint of constant exhaust gas outlet temperature after the recovery boiler (160 °C). This assumption was made because lower exhaust temperature would lead to sulphuric acid condensation and subsequent corrosion of the heat exchangers and because operating at higher temperature after the boiler would lower the amount of heat recovered in the WHR system, and therefore reduce the power output.

Finally, it was assumed that the ORC system cannot operate at a higher load for which it was designed for, due to mechanical and thermal limitations on its components.

It should be noted that the aim of our study was to calculate the full potential for heat-to-power recovery from the exhaust gas. We therefore assumed that other sources of waste heat from the main engines, such as the charge air cooler or the jacket water cooler, can provide the required heat for auxiliary purposes. Although this is not a common choice in normal merchant vessels, where the exhaust gas provides more than enough heat for auxiliary purposes, this is common practice in other applications, such as in the case of cruise ships, where the demand for auxiliary heat is larger.

2.3. Optimisation procedures

We compare four optimisation procedures for the selection of the optimal design variables for an ORC system applied to the case study ship: two based on the performance of the system at design point (DP, DP_+) and two based on the system's performance over its operational profile (OP, OP_+). The generic optimisation procedure is based on the application of a genetic algorithm and can be summarised in the following steps:

1. The ORC state points are resolved using the optimisation variables (see Table 2).
2. Simultaneously, the main engine load factor is adjusted so that the combined cycle outputs the same power as in the reference case with the engine operating alone. This is also done at part-loads.
3. The components are then defined by the heat exchangers global heat exchange coefficient and the turbine constant.
4. The part-load performance is calculated.
5. The objective function is calculated according to Eqs. (8)–(10) depending on the optimisation procedure. The required part load evaluations for each optimisation procedure are summarised in Table 1.

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

The DP (design point) optimisation represents the "baseline" type of optimisation procedure where the objective function is only calculated at the propulsion system's design point:

$$f_{obj,DP} = BSFC_{tot,d} \tag{8}$$

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

$$f_{obj,DP+} = \begin{cases} BSFC_{tot,d} & \text{if } P_{ORC,\lambda=50\%} \neq 0. \\ Inf & \text{otherwise.} \end{cases} \tag{9}$$

In the OP (operational profile) optimisation the objective function is based on the ORC performance at load factors from 50% to 100% with 5% intervals, for a total of 11 cycle calculations per objective function evaluation. The contribution of the fuel consumption for each load factor to the objective function is weighted based on the measured frequency of ship operations at that specific load factor:

$$f_{obj,OP} = f_{obj,OP+} = \sum_i BSFC_{tot,i} w_i P_i \tag{10}$$

where w_i represents the weight of the i-th load and is based on the load distribution presented in Fig. 4.

The range of engine load factor at which the ORC system is operated ($\lambda_{ORC,min} - \lambda_{ORC,max}$) has a large influence on the combined system efficiency, together with λ_{switch}. For this reason we developed an additional optimisation procedure (OP_+), in which these three variables were also optimised. This optimisation procedure requires a maximum of 11 function evaluations but involves three more optimisation variables and therefore requires a larger number of function evaluations in order to converge to an optimal solution.

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

Table 2
Boundary values for optimisation variables.

Variable name	Unit	Range
p_{ev}	bar	1–0.9p_{crit}
$\Delta T_{PP,rec}$	K	10–250
$\Delta T_{PP,eva}$	K	10–250
λ_{switch}	%	27.5–57.5
$\lambda_{ORC,max}$	%	70–100
$\lambda_{ORC,min}$	%	55–30

Table 1
Summary of the analysed optimisation procedures.

Procedure name	Loads in f_{obj}	variables
Design Point (DP)	100%	Design evaporation pressure (p_{ev})
Design Point (DP_+)	100%, 50% (check)	Recuperator $\Delta T_{PP,rec}$
Operational Profile (OP)	100% to 50%, 5% intervals	Evaporator $\Delta T_{PP,eva}$
		Fluid
Operational Profile plus (OP_+)	Variable range	Design evaporation pressure
		Recuperator Pinch point ($\Delta T_{PP,rec}$)
		Evaporator pinch point ($\Delta T_{PP,eva}$)
		Fluid
		Engine operational switch load factor (λ_{switch})
		Max prop. system load factor for WHR on (cut-in load factor, $\lambda_{ORC,max}$)
		Min prop. system load factor for WHR on (cut-out load factor, $\lambda_{ORC,min}$)

temperature range and were therefore shown in the results: R236ea, R245fa, MM, MDM, benzene, toluene and cyclopentane.

3. Results

Fig. 8 shows a comparison between the four different optimisation procedures and the baseline case for the case study ship in terms of the total fuel consumption over one year of operations. Values are presented in percent, with reference to the "ME stand-alone" baseline case.

The optimal variables, together with a summary of systems performance, are displayed in Table 3 for the evaluated optimisation. Table 4 shows the part-load performance of the optimal OP_+ ORC system.

The performance of the DP optimisation procedure leads to estimated fuel savings of 7.3% compared to the baseline case with no ORC system installed (see Fig. 8). This result is influenced by the inability of the system to be operated when only one engine is running.

The performance of the DP_+ optimised system brings yearly savings up to 9.9%. The improvement compared to the *DP* results is mainly connected to the ability of the system to run at lower loadfactors, as ensured by the additional step in the optimisation procedure (see Fig. 9). This advantage outweights the lower ORC power output available at higher propulsion system loadfactors (see also Fig. 10) because the ship under study operates at low load factors for long periods of time, as shown in Fig. 4.

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

In optimisation procedures DP, DP+ and OP, the values of λ_{switch}, $\lambda_{ORC,max}$, and $\lambda_{ORC,min}$ are treated as constants, and fixed to 47.5%, 50% and 100% respectively (see Table 3). However, the results of our investigation shown in Fig. 11 indicate that these variables have a large influence on the performance of the combined engine–WHR system. These three variables were therefore included in the set of optimisation variables in the OP_+ optimisation procedure.

The results of the OP_+ optimisation procedure show that by manipulating the aforementioned engine control variables it is possible to achieve savings of up to 10.8% from a value of 9.9% when they are not taken into account. This result is obtained for values of $\lambda_{ORC,max}$, λ_{switch}, and $\lambda_{ORC,min}$ respectively set at 95%, 40% and 32.5% (See Table 3. Note that the values for λ_{switch} and $\lambda_{ORC,min}$ differ because a minimum difference between the two must be kept in order to prevent oscillations in the control system when the WHR is started). It should be noted that this improvement in the ORC performance is achieved without any additional investment cost when compared to the results of the *DP* and DP_+ procedures.

The results presented in Figs. 8–11 refer to the use of the fluid that provides the lowest combined fuel consumption in each case. Fig. 12 shows the combined efficiency relative to baseline in the

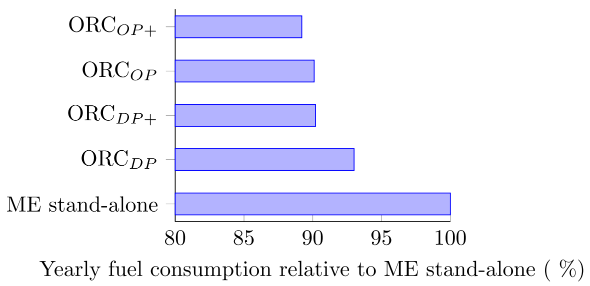

Fig. 8. Yearly fuel consumption of the engine + ORC system relative to engine-alone baseline system.

Table 3
Optimal design variables for the ORC WHR system using the OP and DP optimisation procedures.

Variable	Unit	Optimisation procedure		
		DP	DP_+ and *OP*	OP_+
λ_{switch}	%	47.5	47.5	32.5
$\lambda_{ORC,min}$	%	–	50	40
$\lambda_{ORC,max}$	%	100	100	95
ORC fluid		cyclopentane		
ORC des. evaporation pressure	bar	34.4	24.5	33.7
ORC des. evaporation temperature	°C	218.1	193.6	216.7
ORC condensation pressure	bar	0.42	0.42	0.42
ORC des. working fluid flow rate	kg/s	4.25	4.34	4.08
ORC des. power	kW	799	767	763
ORC des. efficiency	%	30.7	26.6	30.0

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

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

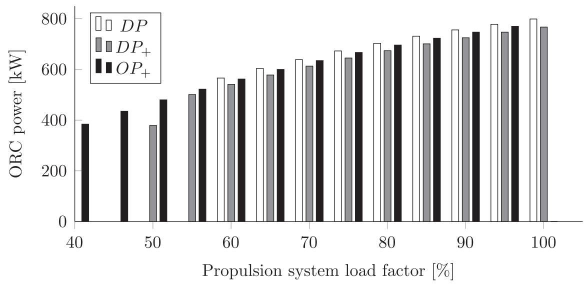

Fig. 9. ORC outlet power versus whole propulsion system load factor for the different optimised systems.

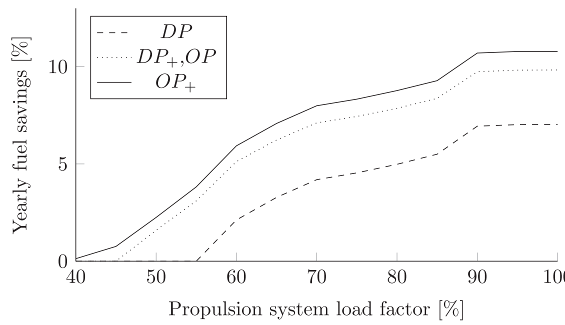

Fig. 10. Cumulative fuel yearly savings for different optimisation procedures compared to the baseline.

OP_+ optimal case for the 7 fluids showing the highest performance in terms of reduction of yearly fuel consumption. These results indicate that apart from cyclopentane, also benzene is a promising fluid to be used in this specific case.

3.1. Economic evaluation

The net present value (NPV) and the payback time (PBT) of the OP_+ optimised ORC system are shown respectively in Figs. 13 and 14 as a function of the expected fuel prices (ranging between 400 and

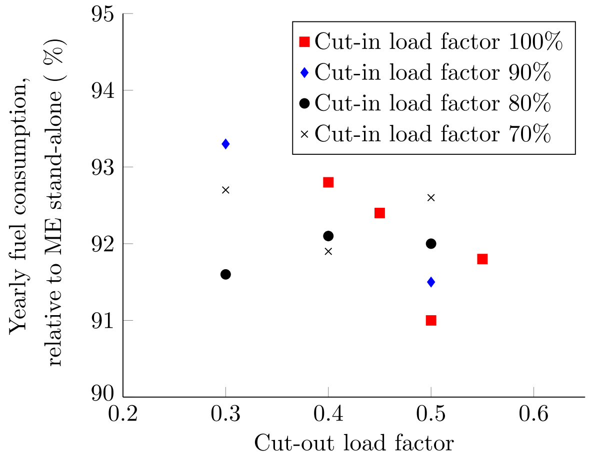

Fig. 11. Influence of the cut-in and cut-out load factor of the ORC on the yearly fuel consumption of the engine+ORC system.

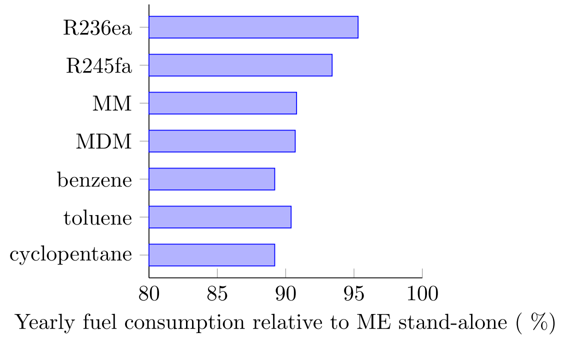

Fig. 12. Influence of the working fluid on the yearly fuel consumption of the engine+ORC system.

800 EUR/ton) and for three different specific investment cost (1500, 2000 and 2500 EUR/kW), which was based on the investigations by Quoilin et al. (2013). For the NPV a time horizon of 10 years and an interest rate of 10% have been selected. It can be noticed that the NPV is positive for all scenarios, regardless of the fuel price and the required investment cost. The payback time is in line with what normally estimated as an acceptable payback time for retrofitting projects (DNV, 2012), ranging from 1.5 to 5.2 years in the best and worst case scenarios.

4. Discussion

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

The main contribution of this paper can be identified in showing the potential of including the operational profile in the optimisation of ORC-based WHR systems.

The comparison of the optimisation procedures indicate that improvements in ORC system efficiency can be achieved when moving from an approach only based on design point performance (*DP*) to one that also takes into account part-load performance (DP_+, *OP*, OP_+). In the case studied in this paper, this leads to an increase from 7.0% to 10.8% yearly savings. In particular, savings could be improved from 7.0% to 9.0% using the DP_+ approach, which requires only one

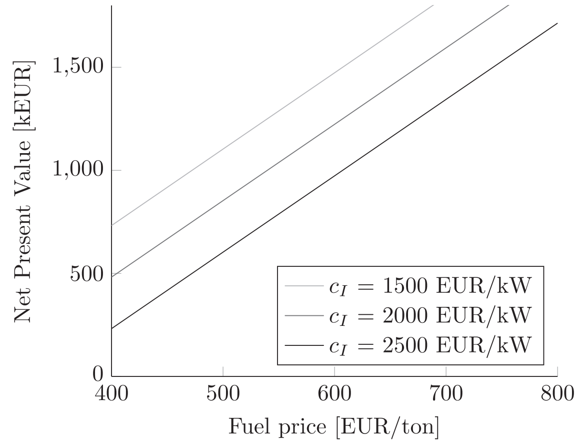

Fig. 13. Expected net present value of the optimised ORC system versus fuel price for different specific investment costs.

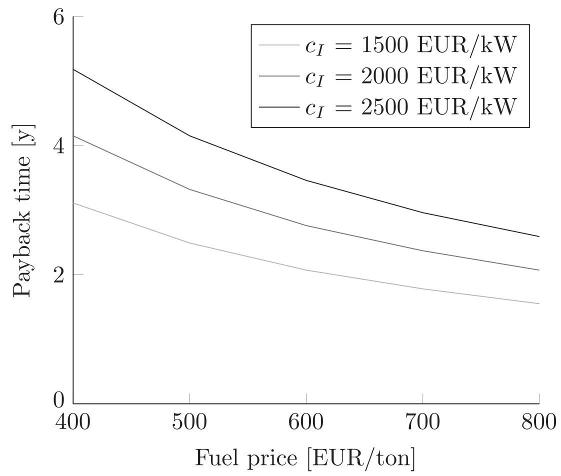

Fig. 14. Expected payback time of the optimised ORC system versus fuel price for different specific investment costs.

additional system simulation per evaluation of the objective function in the optimisation in order to ensure efficient operations at low load. This result has a twofold explanation. On the one hand, the switch between one- to two-engines operations brings a discontinuity in the boundary conditions for the ORC (exhaust mass flow rate and temperature) which the ORC system cannot handle if it has not been included in the design process. On the other hand, the relatively large amount of time the ship spends at low load increases the importance of the ORC power output in these conditions.

The complexity of the operational profile led us to test an optimisation procedure where the performance of the ORC at part-load was included in the evaluation of the objective function (*OP*). This required a larger number of system simulation per objective function evaluation (11 compared to 2 in the DP_+ case). Surprisingly, however, this led to the same optimal cycle as evaluated with the DP_+ procedure. This result can be explained by the monotonic dependence of the ORC system power output on propulsion system load factor. This result indicates that the use of a DP_+ procedure can be a good compromise between efficiency and computational time when designing ORC systems for WHR in shipping. This result partly confirms the validity of the approach presented by Choi and Kim (2013), who identified two main operational modes of the original propulsion system based on the ship's operational profile and used them as the base for the optimisation of the WHR system.

In the presence of two engines in the original propulsion systems, the values of λ_{switch}, $\lambda_{ORC,min}$ and $\lambda_{ORC,max}$ have an effect on both the efficiency of the Diesel engines and of the WHR system, and they were therefore taken into account in the optimisation procedure. This leads to an additional increase of the yearly fuel savings from 9.9% to 10.8% for the case study vessel. On the one hand, this improvement is achieved at the cost of a higher computational effort. On the other hand, being designed for a lower maximum load factor, the OP_+ optimised system is expected to require a lower capital investment.

In addition to their importance in relation to the choice of the optimisation procedure, the results presented in this paper also show that the best efficiency for the combined Diesel engine–ORC WHR system is obtained when the load factor of the switch from one- to two-engines operations is set at only 32.5%. This result confirms what already observed by the authors in a previous study (Larsen et al., 2015), i.e. that penalising engine performance may lead to improved combined cycle efficiency.

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

In addition, there is room for improving the simulation of the WHR cycle at part-load conditions, in particular with reference to possible technical means to allow the system to operate at very low loads. However, when the system is optimised for high-load operations, it experiences a very significant drop in performance at low-load, as can be observed in the results of the DP+ optimisation (see Fig. 9). This result indicates that, although an updated model for part-load operations could improve the reliability of the results, this would not impact the main findings of this paper.

In the case of ORCs the working fluid plays an important role. The results of this study confirmed what was observed in previous studies (Larsen et al., 2013b) identifying cyclopentane and benzene as the most suitable fluids for the application of ORCs to medium temperature heat sources (Larsen et al., 2013b; Lai et al., 2011).

The choice of the working fluid influences the system's efficiency. On the other hand, organic fluids can be toxic or flammable and their use on board requires the proper safety precautions and equipment. This could possibly be solved by either using fluids with higher standards in terms of toxicity and flammability (Larsen et al., 2013b), or by adding an intermediate fluid in the system setup. Both solutions would however generate a reduced system efficiency, and the trade-off between expenses related to safety procedures and high efficiency are still today a matter of discussion.

It should be also noted that the results presented in this study are based on the assumption that all the heat from the exhaust gas of the main engines is available for the heat-to-power WHR system. In most vessels today heat from the exhaust gas is used to fulfil on board auxiliary heat demand, which would potentially decrease the amount of heat available for the heat-to-power system. In this study, we assumed that on board auxiliary heat demand can be fulfilled by using low-temperature waste heat sources such as the charge air cooler or the jacket water cooler. This would however require an additional investment (although limited to the required heat exchangers) which is not included in the economic evaluation that we presented in this work.

5. Conclusion

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

The main conclusions of this paper are the following:

- An optimisation approach that accounts for part-load performance provides significant benefits in terms of yearly fuel consumption when compared to a case where only the performance at the design point is considered.
- Ensuring the ability of the system to operate at low load is a good compromise for increasing the reliability of the optimisation while not significantly increasing the computational effort. An efficient ORC optimisation procedure should therefore also test the ORC performance at the lowest load factor at which the propulsion system is expected to operate for a significant amount of time.
- The operational variables related to the combined engine and ORC system are important for the optimisation of the combined system and should therefore be taken into account in the optimisation procedure.

Estimated savings improved from 7.0% to 9.0% when the objective function in the optimisation procedure included also the expected fuel savings at part-load; when variables related to the interaction between the engine and the ORC system (engine load factor switching between one- and two-engines operation, minimum and maximum propulsion system load factor at which the ORC system is required to operate) were included in the optimisation procedure, the expected savings increased to 10.8%. In addition to the higher fuel savings, the proposed optimised system is designed for a lower maximum power output, therefore being smaller in size and less expensive to build.

Acknowledgements

The authors are grateful to the Swedish Energy Agency (Project number: (31886-3) for the financial support to the project “Systems modeling for sustainable, energy efficient shipping” and to Laurin Maritime for providing the operational data used in this work.

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Paper VI

Optimal load allocation of complex ship power plants

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Abstract

In a world with increased pressure on reducing fuel consumption and carbon dioxide emissions, the cruise industry is growing in size and impact. In this context, further effort is required for improving the energy efficiency of cruise ship energy systems.

In this paper, we propose a generic method for modelling the power plant of an isolated system with mechanical, electric and thermal power demands and for the optimal load allocation of the different components that are able to fulfil the demand.

The optimisation problem is presented in the form of a mixed integer linear programming (MINLP) problem, where the number of engines and/or boilers running is represented by the integer variables, while their respective load is represented by the non-integer variables. The individual components are modelled using a combination of first-principle models and polynomial regressions, thus making the system nonlinear.

The proposed method is applied to the load-allocation problem of a cruise ship sailing in the Baltic Sea, and used to compare the existing power plant with a hybrid propulsion plant. The results show the benefits brought by using the proposing method, which allow estimating the performance of the hybrid system (for which the load allocation is a non-trivial problem) while also including the contribution of the heat demand. This allows showing that, based on a reference round voyage, up to 3% savings could be achieved by installing the proposed system, compared to the existing one, and that a NPV of 11 kUSD could be achieved already 5 years after the installation of the system.

1. Introduction

The shipping industry, despite its low contribution to global anthropogenic CO_2 emissions today (2.7% of the total as of 2012 [1]), will have to face increasingly stronger challenges in the future in relation to its contribution to global warming [1]. Most predictions suggest that shipping volumes (and, therefore, emissions) are expected to increase in the foreseeable future [1]. On the other hand, it has been shown that for achieving the 2°C climate goal shipping should reduce its CO_2 emissions by more than 80% by 2050 compared to 2010 levels [2].

International regulations, such as the revised version of the MARPOL [3], have started to put limits on ship emissions. Even further efforts are expected to be required if local regulations will be implemented. The European Union, for instance, is planning actions for achieving a 40-50% reduction in CO_2 emissions from ships visiting European harbours by 2050 [4], and in Sweden the fairway dues soon might be calculated against the clean shipping index which includes CO_2 emissions.

1.1. Energy efficiency in shipping

Many new practices and technologies are being introduced for improving energy efficiency in the shipping sector. These measures are normally subdivided between operational and design.

Operational measures include efforts that do not require the installation of new equipment on board. Optimal voyage planning allows maximising the cargo transported while reducing the length of ballast legs [5], while adapting routes for avoiding conditions of bad weather can reduce the negative impact of high waves and strong winds on ship fuel consumption [6,7]; improving trim and draft setting, together with optimising the schedules and practices for hull and propeller polishing, lead to reduced ship resistance for a given speed [8–10]; slow steaming can also dramatically reduce the fuel bill: as the amount of cargo transported decreases linearly with the speed, while the power demand from the engines roughly depends on the cube of the speed, the advantage is obvious [11,12].

Retrofit and design measures, on the contrary, refer to physical technical solutions. This connects to the development of the performance of individual parts of the systems, such as the engine [13–16], the propeller [17,18], and the hull [19]. Additional energy sources can be used both for propulsion (e.g. sails and rotors [20,21]) and for auxiliary power generation (e.g. fuel cells [22]). Waste energy on board can be recovered in different ways, among other for heating, power [23–25], and cooling [26,27].

1.2. Challenges of ship on board energy management

Differently from a number of land-based systems, ships can operate in many different conditions and, hence, with large variations in power demand. This is even more challenging in the case of some specific ship types, such as cruise ships, where demand of energy in different forms (mechanical, electric, thermal) and of comparable size are observed. When in port, mechanical power demand for propulsion is virtually zero, while it can be predominant in sailing conditions, depending on the speed of the vessel. Demand for thermal energy can depend on the outer temperature of air and water, as well as on the number of passengers on board. Electric power demand can similarly vary as a function of environmental and operational conditions. These conditions require the ship power plant to be able to handle many combinations of energy demands with high efficiency.

Different types of hybrid propulsion systems (i.e. systems where the systems for the generation of propulsive and electric power are interconnected) are gaining ground in the sector, as they allow for increased flexibility in fulfilling both propulsive and electric power demand. Such systems proved to allow fuel savings of 1-2% [28]. These systems require however additional effort both in the design phase [29] and in the definition of the control strategy [30,31], as the increase in the number of connections between different parts of the system allows for the load to be fulfilled using a potentially high number of combinations of engines running at different loads.

In most ships, the waste heat available from the engines is largely sufficient for fulfilling on board demand for thermal energy [32], and further uses for waste heat are today a common research topic [33–35]. On cruise ships, however, thermal energy demand is higher than on other ship types [36]. However, to date, there is little evidence of scientific work aiming at the optimisation of the on board energy demand which also includes thermal energy as part of the definition of the constraints and of the objective function.

1.3. Aim

In this paper, we propose a method for optimising the load configuration of the energy converters of a ship power plant. More in general, the proposed load-optimisation method can be applied to energy systems with a time-dependent demand of mechanical, electric and thermal power with no connection to external energy networks.

The method proposed in this paper is applied to the energy system of a cruise ship. In particular, the method is to a proposed retrofit to the existing system where all engines are allowed to contribute to fulfil both the mechanical and the electric power demand. The application of the proposed method allows handling the increased complexity of the load-allocation problem and, therefore, evaluating the expected fuel savings derived by the system retrofit.

2. Method

The method presented in this study aims at being applicable to most of ship energy systems available today. The system configuration presented in Figure 1 is seen as sufficiently general with respect to the standard practice in today's shipbuilding industry. The system configuration therefore includes:

- Two propulsion lines: the most general system configuration on today's ships involves only one propulsion line, which can be seen as a particular case of the more general configuration of Figure 1
- Two main engine blocks, each composed by n_{ME} engines
- Two auxiliary engine blocks, each composed by n_{AE} engines, to accommodate for purely Diesel-electric systems equipped with engines of up to two different sizes.

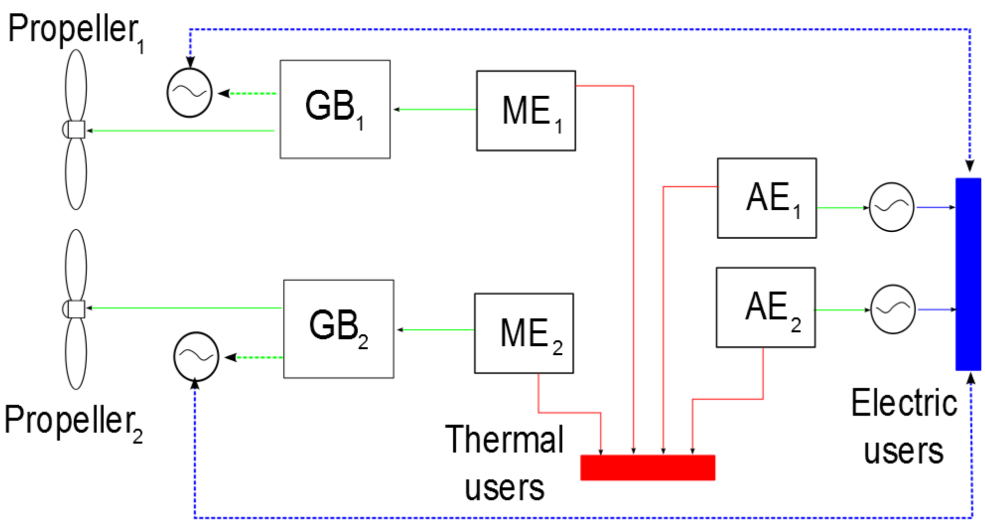


Figure 1: Schematic arrangement of a hybrid propulsion system

2.1. MINLP problem setting

In all alternative systems, more than one possible configuration of engines running could be employed in order to fulfil the electric power demand. However, for each propulsion/electric power pair, there is one combination of main and auxiliary engines which fulfils the demand with the lowest fuel consumption. The selection of such combination requires an optimisation process, particularly in all cases where a hybrid propulsion system is available, since any engine can fulfil any demand.

The optimisation problem can be seen as a mixed integer nonlinear programming (MINLP) problem, where the integer variables are the number of engines running in each engine group and the continuous variables are the load of each engine group. The simplification of optimising the load of engine groups instead of individual engines is based on the assumption that, given that all the engines of the group have the same size and performance, it is most efficient to run all engines at the same load [37].

The optimisation problem can therefore be summarised as follows:

Min		$f(\bar{x})$	(1)
s.t.		$g_{eq}(\bar{x}) = b_{eq}$	(2)
		$g_{neq}(\bar{x}) < b_{neq}$	(3)
	For *i* in (1,5)	x_i integer	(4)
	For *i* in (6,18)	$0 \leq x_i \leq 1$	(5)

Where the subscripts *eq* and *neq* refer to equality and non-equality constraints respectively. The objective function is defined as:

$$f_{obj} = \dot{m}_{ME1} + \dot{m}_{ME2} + \dot{m}_{AE1} + \dot{m}_{AE2} + \dot{m}_{AB} \tag{6}$$

Where each of the mass flows is calculated according to:

$$\dot{m}_i = \frac{\lambda_i \, MCR_i \, n_{i,on}}{\eta_i(\lambda_i) \, LHV_{fuel}} \tag{7}$$

Where the LHV of the fuel is that of marine heavy fuels, assumed equal to 40.7 MJ/kg [38], while λ_i and η_i represent the load and the efficiency of the i-th group of engine/boilers, respectively. Each of the mass flows in Equation (6) can therefore be defined as a function of the load of the component block (elements 6-18 in $\bar{x}$), and on the number of elements in the component block that are running (elements 1-5 in $\bar{x}$).

The first five elements of $\bar{x}$ represent the number of engines/boilers that are running in each of the groups (e.g. $n_{ME1,on}$ for the first group of main engines) while the elements from 6 to 18 represent the load of each of the engine/boiler groups connected to a specific demand. For instance:

$$x_6 = \lambda_{ME1\rightarrow prop1} = \frac{P_{ME1\rightarrow prop1}}{MCR_{ME1}} \tag{8}$$

Represents the share of the mechanical power generated by the 1st group of main engines which is used by the 1st propeller. Therefore, for each group of engines the following three elements in the $\bar{x}$ vector are represented:

$$\lambda_{i\rightarrow prop1} \quad \lambda_{i\rightarrow prop2} \quad \lambda_{i\rightarrow el} \tag{9}$$

Finally, element 18 represent the auxiliary boiler(s) load:

$$x_{18} = \lambda_{AB\rightarrow th} = \frac{\dot{Q}_{AB}}{\dot{Q}_{ab,max}} \tag{10}$$

The nonlinear equality conditions represent the requirement that the system is able to fulfil the totality of the mechanical power demand from the two propeller and the electric demand:

$$\begin{cases} P_{prop1} = \sum_i P_{i\rightarrow prop1}\eta_{i\rightarrow prop1} & (11) \\ P_{prop2} = \sum_i P_{i\rightarrow prop2}\eta_{i\rightarrow prop2} & (12) \\ P_{el} = \sum_i P_{i\rightarrow el}\eta_{i\rightarrow el} & (13) \end{cases}$$

The nonlinear inequality conditions represent the requirement that each of the engines/boilers in the systems is not loaded above its maximum load and below its minimum load.

(14)

$$\begin{cases} P_{ME1,min}n_{ME1,on} & \leq & P_{ME1\rightarrow prop1} + P_{ME1\rightarrow prop2} + P_{ME1\rightarrow el} & \leq & P_{ME1,max}n_{ME1,on} & (15) \\ P_{ME2,min}n_{ME2,on} & \leq & P_{ME2\rightarrow prop1} + P_{ME2\rightarrow prop2} + P_{ME2\rightarrow el} & \leq & P_{ME2,max}n_{ME2,on} & (16) \\ P_{AE1,min}n_{AE1,on} & \leq & P_{AE1\rightarrow prop1} + P_{AE1\rightarrow prop2} + P_{AE1\rightarrow el} & \leq & P_{AE1,max}n_{AE1,on} & \\ P_{AE2,min}n_{AE2,on} & \leq & P_{AE2\rightarrow prop1} + P_{AE2\rightarrow prop2} + P_{AE2\rightarrow el} & \leq & P_{AE2,max}n_{AE2,on} & (17) \end{cases}$$

The last of the inequality conditions requires the sum of the available waste heat from the engines and the heat generated by the boilers to be larger than the total heat demand

$$\dot{Q}_{WH,ME1} + \dot{Q}_{WH,ME2} + \dot{Q}_{WH,AE1} + \dot{Q}_{WH,AE2} + \dot{Q}_{AB} \geq \dot{Q}_{th} \tag{18}$$

Where the waste heat from each engine is calculated as explained in Section 2.2.

MINLP problems can be solved in different ways depending on the structure of the different parts of the optimisation problem [39]. In this case, the optimisation problem was solved using a sequential quadratic programming (SQP) algorithm (built-in Matlab® NLP solver *fmincon* in SQP mode) for the solution of the NLP programming. A branch-and-bound method was implemented by the authors for handling integer variables [40].

2.2. Diesel engines

The efficiency of the Diesel engines, both main and auxiliary engines, are calculated using 2nd degree polynomial regressions based on data from the engines shop trials. A fixed penalty term of 1.05 is considered to account for the engines not operating in ISO conditions; an additional penalty term of 1.05 is arbitrarily assigned in order to represent engine performance degradation during 10 years of operations. Engine efficiency is therefore calculated according to Equation (19):

$$\eta_{eng} = 1.05^2\ P_2(\lambda) \tag{19}$$

Where λ represents the engine's load. Regression curves for MEs and AEs are represented in Figure 2. Waste heat flows from both the MEs and the AEs needs to be modelled for on board WHR.

Most large marine Diesel engines are equipped with a compressor bypass, which allows part of the air flow after the compressor to bypass the engine cylinders and be directly mixed with the exhaust gas before the turbine. In order to account for a variable bypass flow, the following system of equations is proposed.

Starting from the engine load, the air flow through the cylinders can be calculated according to Eq. (20):

$$\dot{m}_{air,cyl} = \eta_{vol}\frac{p_{ca}}{R_{air}T_{ca}}V_{cyl,max}n_{eng}N_{cyl} \tag{20}$$

Where η_{vol} represents the volumetric efficiency of the engine, and is calculated as suggested by Hiereth and Prenninger [41]; p_{ca} represents the charge air pressure, and is calculated as a function of the engine load based on polynomial regressions of measured data obtained from on board alarm systems; T_{ca} represents the charge air temperature, which is generally regulated by the cooling systems control to a value of approximately 50-60°C; $V_{cyl,max}$ represents the cylinder maximum volume, n_{eng} the engine speed and N_{cyl} the number of cylinders.

The equations related to the energy balance of the turbine and of the mixer of flows 3,5,6, as well as the mass balance in splitter and mixer must be solved simultaneously:

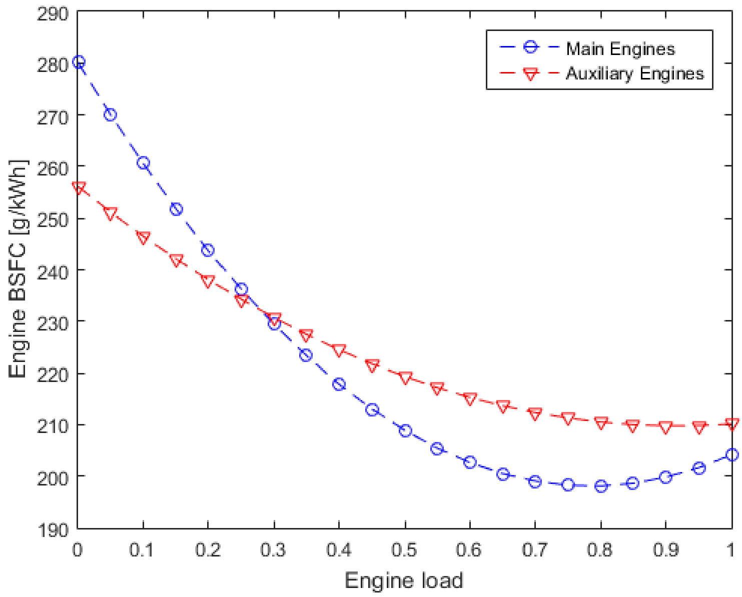


Figure 2: Main and auxiliary engines BSFC versus engine load, from regressions

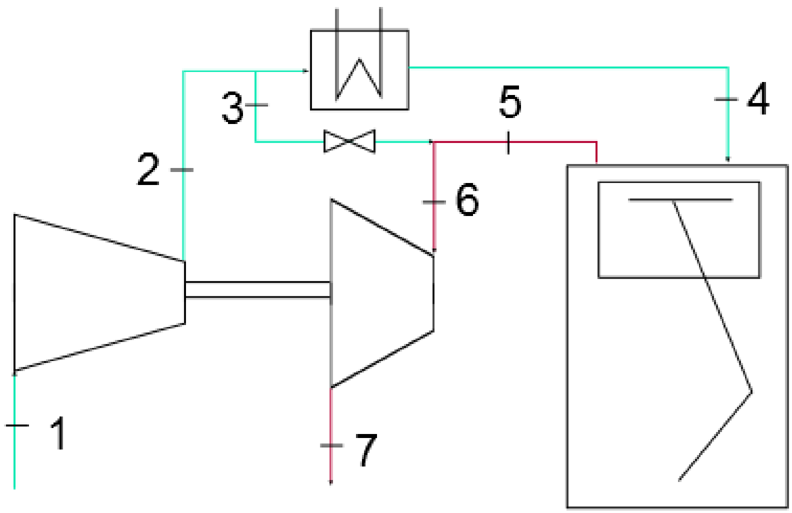


Figure 3: Schematic representation of the main engines turbocharging system, with bypass

$$\begin{cases} \dot{m}_1 c_{p,air}(T_2 - T_1) = \eta_{mech,TC}\dot{m}_6\, c_{p,eg}(T_6 - T_7) & (21) \\ \dot{m}_6 c_{p,eg}(T_6 - T_0) = \dot{m}_5 c_{p,eg}(T_5 - T_0) + \dot{m}_3 c_{p,air}(T_3 - T_0) & (22) \\ \dot{m}_1 = \dot{m}_3 + \dot{m}_4 & (23) \\ \dot{m}_6 = \dot{m}_5 + \dot{m}_3 & (24) \end{cases}$$

Where $\eta_{mech,TC}$ represents the mechanical efficiency of the turbocharger, assumed equal to 0.98, and $c_{p,air}$ and $c_{p,eg}$ the specific heat of air and exhaust gas, assumed constant with temperature and equal to 1.02 and 1.08 kJ/kg respectively. In the formulation of Equations (21) and (22) it is assumed that the mass flow of the exhaust gas leaving the cylinders is sufficiently larger than the bypass flow, thus allowing to assume that the specific heat of the mixed flow is equal to that of the pure exhaust gas flow from the cylinders.

The system of equations (21)-(24), together with equation (20), requires four of the variables to be determined in advance in order to be solved. The definition of these four variables depends on the availability of measured data in each individual case. The assumptions for the specific test case presented in this paper will be presented further in Section 3.

The aforementioned procedure allows calculating all physical flows in the engine, and therefore the energy leaving in the form of exhaust gas and available for recovery ($\dot{Q}_{eg,ME}$):

$$\dot{Q}_{eg,ME} = \dot{m}_7 c_{p,eg}(T_7 - 433) \tag{25}$$

where the lower limit of 433 K is assumed based on the need to avoid sulphuric acid condensation in the exhaust gas [42].

Marine engines are generally cooled using both a high temperature (HT) and a low temperature (LT) cooling system. The temperature of the HT cooling systems ranges between 70 and 90°C, while the temperature in the LT cooling systems generally ranges between 30 and 50 °C. For this reason it is assumed that only the heat transferred to the HT cooling systems is available for recovery.

The calculation of the thermal power available in the HT cooling systems is based on the energy balance over the engine:

$$\dot{Q}_{HT} = f(\lambda)\left[\dot{m}_{fuel} LHV_{fuel} + \dot{m}_1 c_{p,air}(T_1 - T_0) - P_{mech} - \dot{m}_7 c_{p,eg}(T_7 - T_0)\right] \tag{26}$$

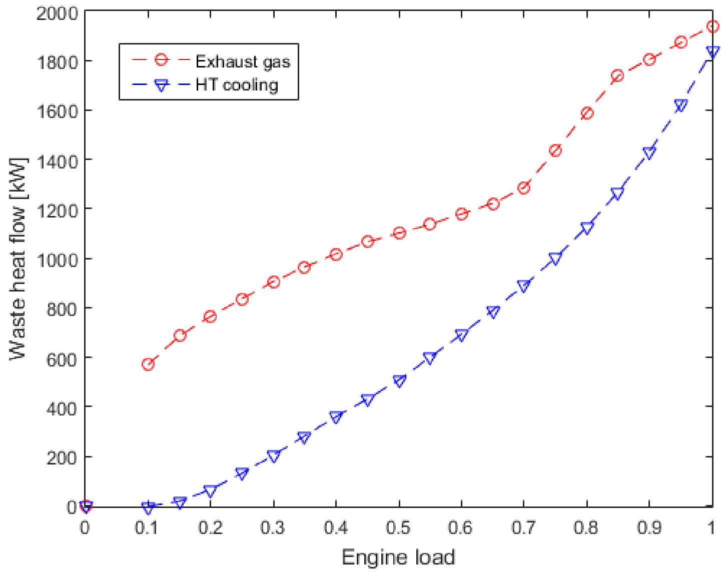


Figure 4: Main engines waste heat versus engine load

where f(λ) represents the fraction of the residual heat that is transferred to the HT cooling system, and is calculated as a polynomial regression of data available from the engine technical documentation.

An example of the resulting flows of thermal energy available for recovery from the exhaust gas and the HT cooling of the main engines of the test case are represented in Figure 4.

It should be noted that if the engines are not equipped with a bypass system, the system of equations (20)-(24) is largely simplified.

2.3. Oil-fired Boilers

Marine oil-fired boilers are generally dimensioned for providing high performance even at very low loads. For these types of systems it is therefore assumed that the part-load efficiency can be modelled using a linear step-wise interpolation (see Figure 5) as based on the efficiency curves for marine boilers presented by Cohen [43]:

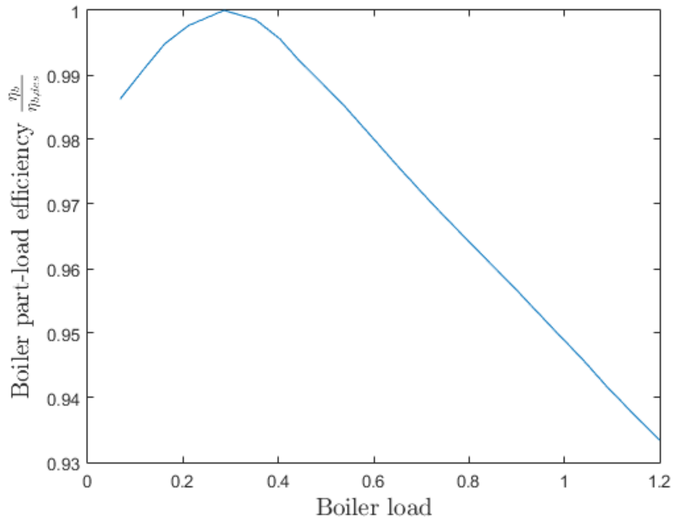


Figure 5: Boiler efficiency, relative to design, versus load

The design efficiency can be assumed to 90% in absence of more specific information.

2.4. Other components

All other components on board are modelled according to the following approximation:

$$\eta = \eta_{des} f_{corr}(\lambda) \tag{27}$$

Where η_{des}, λ and f_{corr} represent the efficiency at the design point, the load of the component, and the correction factor that represents the off-design behaviour.

f_{corr} is calculated using a 2nd degree polynomial approximation both for mechanical components (gearboxes) and for electrical components (generators, motors, and frequency converters) based on [44]. The evolution of the efficiencies for electrical machines and the gearbox with load is provided in Figure 6. The efficiency of the frequency converter, the switchboard and the shaft are assumed to be constant with load. The design efficiencies of all the components in design conditions are presented in Table 1

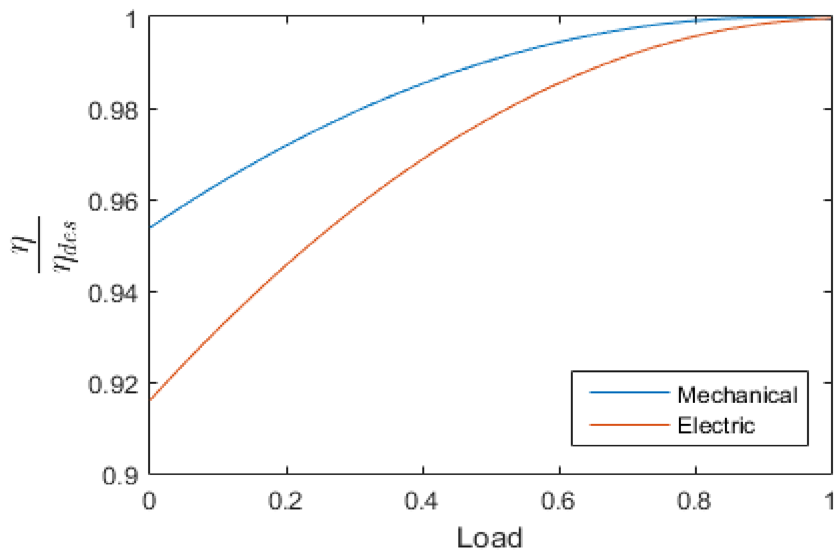


Figure 6: Off-design efficiency correlations for mechanical and electric machines

Table 1: Design efficiencies for mechanical end electric machines

Component	η_{des}	
Gearbox	0.98	[45]
Generator	0.97	[46]
Motor	0.96	[46]
Frequency converter	0.98	[47]
Switchboard	0.99	[47]
Shaft	0.99	[45]

3. Test case

3.1. Description of the case study

The case study ship is a cruise ship which operates daily tours in the Baltic Sea between Stockholm on Swedish mainland and Mariehamn on the Åland islands. The ship was built in 2004 and is 176.9 m long and 28.6 m wide. It can accommodate up to 1800 passengers and is equipped with restaurants, night clubs and bars, as well as saunas and pools. Typical ship operations, although they can vary slightly between different days, are represented in Figure 1. The ship leaves at around 18 from Stockholm, until it reaches the open sea, where it stops for the night before reaching Mariehamn early in the morning. The ship then leaves Mariehamn at around 9 AM and arrives back to Stockholm at around 4 PM (see Figure 7).

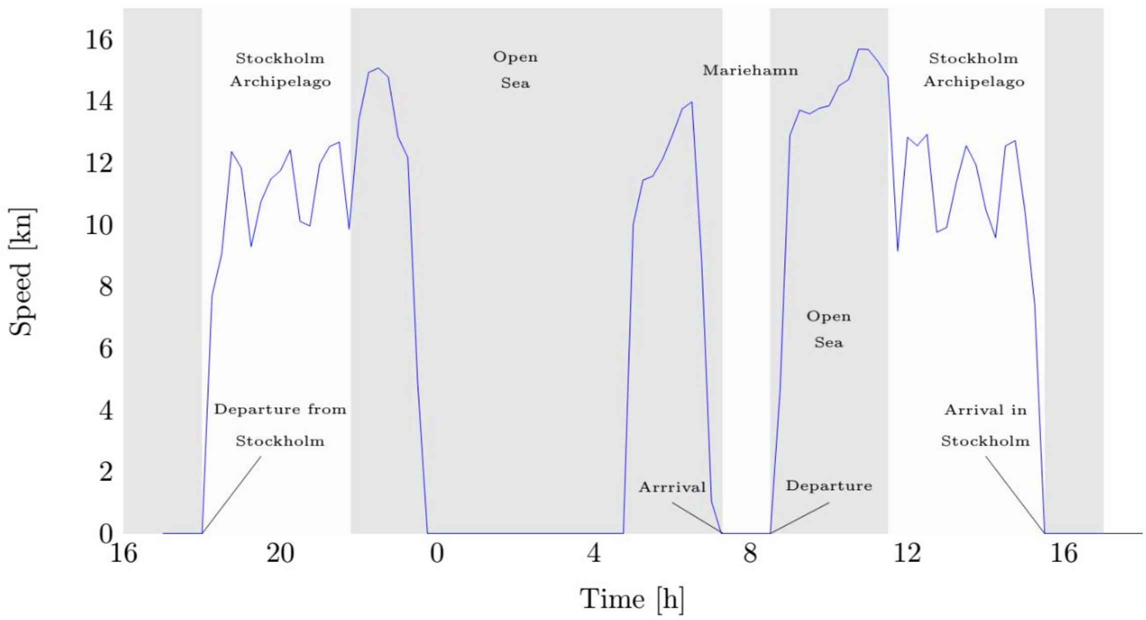


Figure 7: Typical operational profile (ship speed) for the selected ship.

The ship energy system is summarized in Figure 8. The propulsion system is composed of two propulsion lines composed of two main engines, a gearbox, and a propeller each. The main engines (ME) are four Wärtsilä 4-stroke Diesel engines rated 5850 kW each. The ship is also equipped with four Wärtsilä auxiliary engines (AE) for electricity production, rated 2760 kW each. All AEs and one ME for each propulsion line are also equipped with heat recovery steam generators (HRSG); in addition, two auxiliary boilers (AB) are installed on board; finally, the high-temperature (HT) cooling systems of all engines are connected to a heat recovery system which allows to use such heat for accommodation heating. All engines are equipped with selective catalytic reactors (SCR) for NO_X emissions abatement.

Propulsion power is provided by the MEs connected to the two propulsion lines, and is needed whenever the ship is sailing. Auxiliary power is needed on board for a number of alternative functions, from pumps in the engine room to lights, restaurants, ventilation and entertainment for the passengers. Auxiliary heat demand is mostly fulfilled by the EGBs and by the heat recovery from the HT cooling; ABs are mainly used when in port, or during winter. The heat is needed for passengers and crew accommodation, as well as for the heating of the highly viscous heavy fuel oil used for engines and boilers.

Table 2: Summary of energy sources on board of the case study vessel

Component block	Number	Design power [kW]
Main engines – 1	2	5850
Main engines – 2	2	5850
Auxiliary engines	4	2760
Auxiliary boilers	2	4500

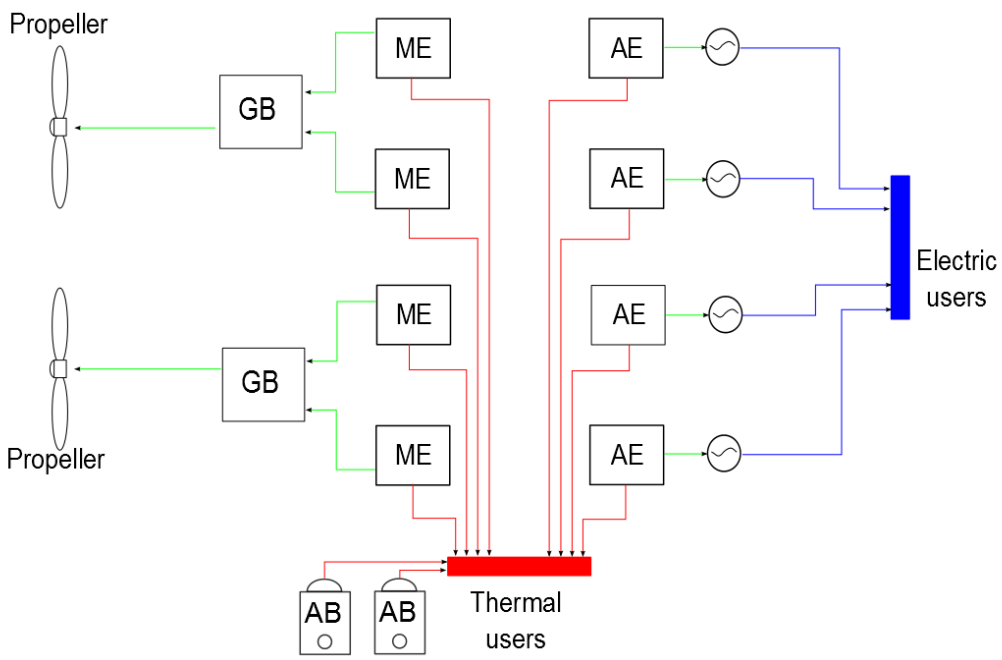


Figure 8: Schematic representation of the case study's energy system

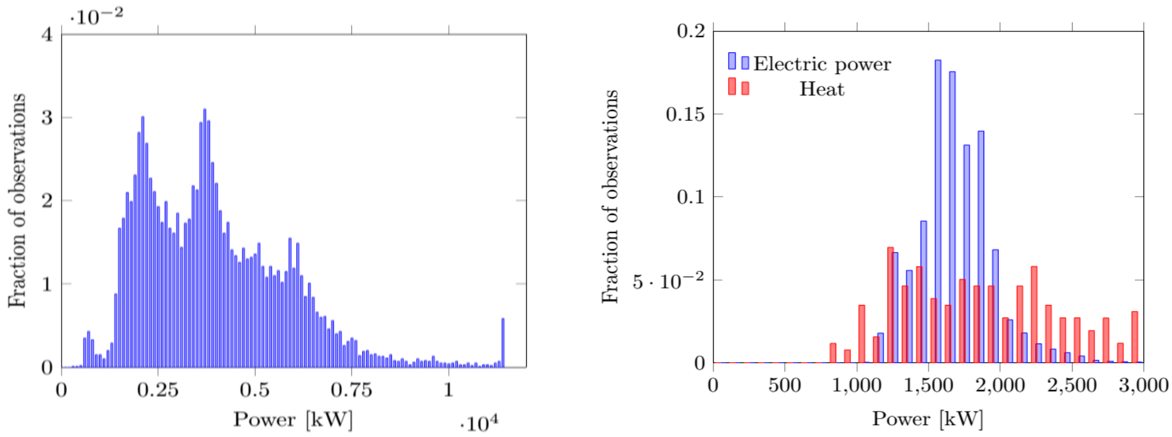
3.2. On board power demand

For testing the application of the aforementioned method to the case study ship, the evolution of on board demand for propulsion, auxiliary power and auxiliary heat needs to be provided. This was determined based on data collected from the on board monitoring system.

- Propulsive power demand was determined based on measurements of MEs fuel rack position and speed. The combination of these two variables allows to estimate fuel mass flow rate, which in turn, using technical documentation related to engines performance, was used to determine the instantaneous propeller power demand.
- The electric power generated by the auxiliary generators was directly measured on board.
- Thermal power demand was determined based on the estimated thermal power recuperated from the EGBs and on measured fuel consumption from the ABs. The method employed in this phase could only lead to an estimation of the daily energy demand. For this reason, we

employed the assumption that the on board thermal power demand follows a similar trend as what represented in Figure 10, which refers to the thermal power demand of a hotel.

A detailed description of the methodology employed for the determination of the demand of propulsive, electric and thermal power is proposed in [36]. A representation of propulsive and electric power distribution is provided in Figure 9. It can be seen that, from the analysis of the data collected from the ship on board systems, it resulted that the ship rarely sails at full speed, and most of the time it only needs only a small fraction of its design power.



(a): Propulsive power (b) Auxiliary power

Figure 9: Power demand distribution

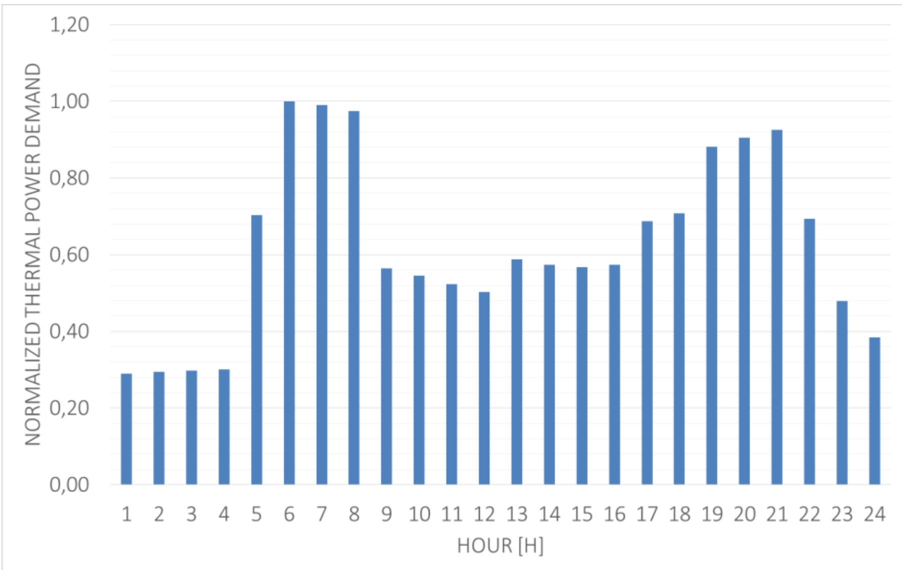


Figure 10: Hourly distribution for on board thermal energy demand

3.3. Shaft generator/motor installation

Given that both main and auxiliary engines on board of the selected vessel are often operated at low load, and therefore at sub-optimal conditions, the possibility of improving the efficiency of the system through the installation of shaft generators/motors connected to the main engines was explored. This would allow a larger freedom in the utilisation of the power generation system, as any engine could provide power to any of the demands. This comes however at the cost of increased conversion losses in the electrical components. The hybrid propulsion system proposed for retrofit on the selected vessel is shown in Figure 11:

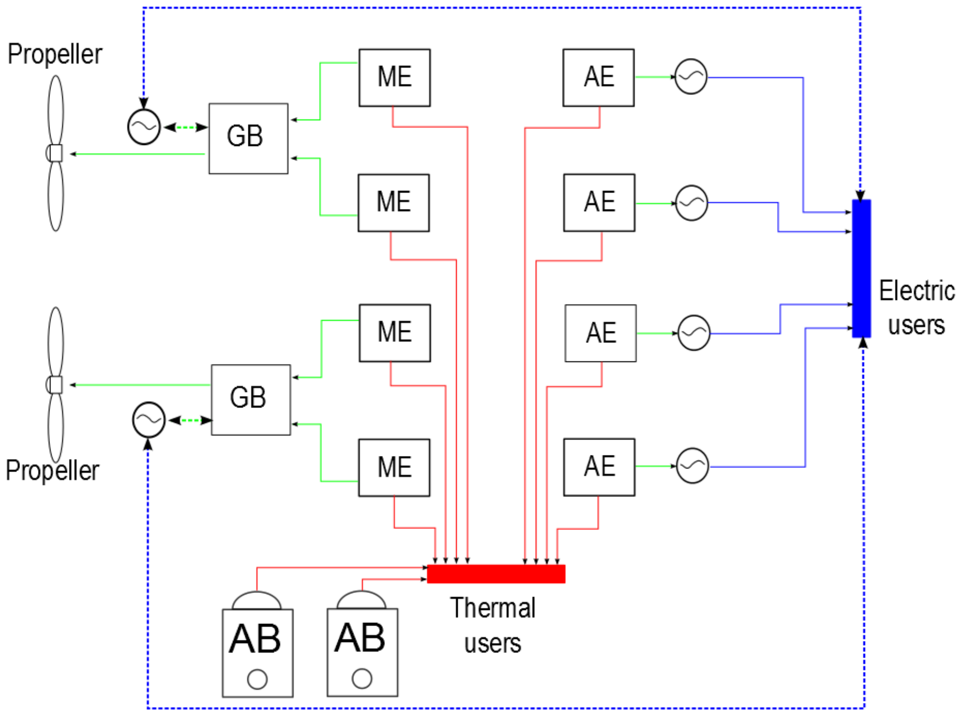


Figure 11: Schematic representation of the case study ship energy system after retrofitting of shaft motors/generators

4. Model application

The model proposed in this paper was tested for the optimisation of the power management of the selected case study over one reference voyage. The model input, namely the power demand for propulsion, auxiliary electric and auxiliary heat, is shown in Figure 12 and in Table 3. The trip represents a reference round-voyage of a total duration of 24 h including one stop in Stockholm (start- and end-destination, 15-18), one in Mariehamn (intermediate destination, 7-8)) and one, stop in the sea of Åland (0-5) where the ship is not in port but drifting at sea, waiting in order to avoid arriving in Mariehamn too early in the morning.

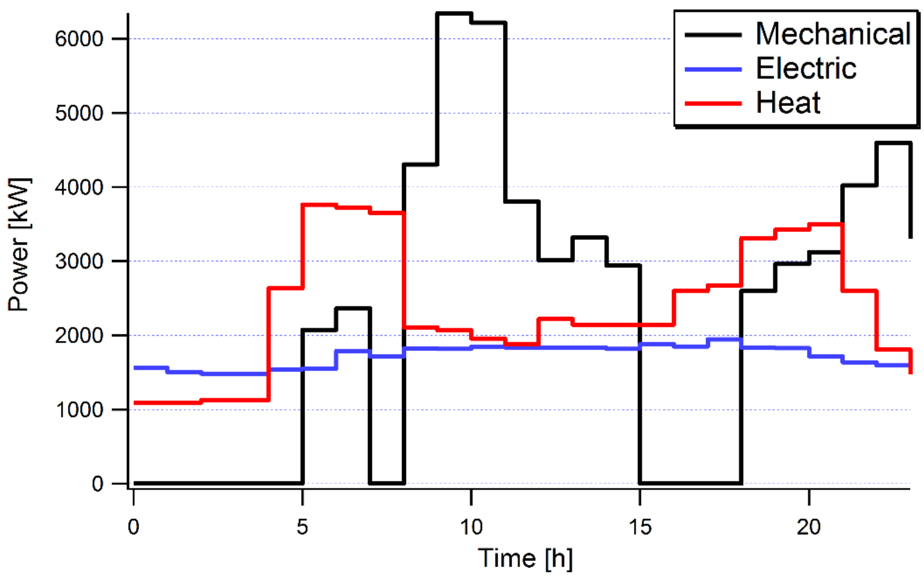


Figure 12: Reference round voyage

Table 3: Summary of the power demands for the reference voyage

Time [h]	P_{mech}	$P_{electric}$	P_{heat}
0	0	1563	1091
1	0	1502	1091
2	0	1485	1129
3	0	1476	1129
4	0	1540	2634
5	2075	1551	3763
6	2363	1790	3726
7	0	1713	3650
8	4305	1824	2107
9	6343	1817	2070
10	6218	1844	1957
11	3800	1835	1882
12	3012	1834	2220
13	3320	1834	2145
14	2940	1815	2145
15	0	1880	2145
16	0	1849	2597
17	0	1945	2672
18	2601	1830	3312
19	2961	1828	3425
20	3125	1717	3500
21	4023	1638	2597
22	4598	1593	1806
23	3303	1549	1473
24	0	1563	1091

4.1. MINLP problem setup

The MINLP problem was setup as shown in Section 2.1. Given that the power plant of the proposed test case has four AEs of the same model, only one group of AEs is considered. The elements of the optimisation independent variable *x* are summarised in Table 4

Table 4: Summary of the elements of the optimisation vector for the test case vessel

Main engines (1)			Main engines (2)			Auxiliary engines			Auxiliary boilers		
N	T	M	N	T	M	N	T	M	N	T	M
x_1	I	# running	x_2	I	# running	x_3	I	# running	x_4	I	# running
x_5	C	$\lambda_{ME1\to prop1}$	x_8	C	$\lambda_{ME2\to prop1}$	x_{11}	C	$\lambda_{AE\to prop1}$	x_{14}	C	$\lambda_{AB\to heat}$
x_6	C	$\lambda_{ME1\to prop2}$	x_9	C	$\lambda_{ME2\to prop2}$	x_{12}	C	$\lambda_{AE\to prop2}$			
x_7	C	$\lambda_{ME1\to el}$	x_{10}	C	$\lambda_{ME2\to el}$	x_{13}	C	$\lambda_{AE\to el}$			

N = Variable name

T = Variable type

M = Variable meaning

I = Integer

C = Continuous

4.2. Results analysis

The results of the application of the proposed method to the test case (presented in Section 3, according to the reference voyage as presented in Figure 12) are shown in Table 6 Table 5 and Table 6Table 8 for the baseline and the hybridised power plants, respectively. In particular Table 5Table 7 and Table 8Table 6 Table 6 show, for every time step (fixed to 1 hour), the operational mode (OM) and the number of components running for each of the groups (variables x_1 to x_4 for ME1, ME2, AE and AB respectively), the fraction of the MCR of each group going to the corresponding demand (x_5, x_6 and x_7 for the fraction of the MCR of Group ME1 that goes to the first propeller, the second propeller, and the electric demand respectively, and similarly to the other main engines and for the auxiliary engines).

The hybridisation of the system via the installation of reversible shaft generators/motors provides savings, in relation to the reference voyage, of up to 2.9% compared to the baseline system.

Each point was classified in 4 operating modes:

- ST (standard) mode: Main engines provide power for propulsion, auxiliary engines provide auxiliary electric power
- AE mode: All propulsive and auxiliary power is provided by the auxiliary engines
- ME1 mode: All propulsive and auxiliary power is provided by one group of main engines
- ME2 mode: All propulsive and auxiliary power is provided by two group of main engines
- MIX mode: Power is provided by a mix of main and auxiliary engines

The baseline power plant can only operate in ST mode, while the hybridised power plant allows a larger number of alternative operational modes. Table 6Table 6 shows that the ST mode is the optimal one only when there is no propulsion power demand. This also comes as a consequence of the fact that the auxiliary power demand, both in port and at sea, never exceeds the MCR of one auxiliary engine, and it makes it therefore more convenient not to use the main engines in this mode.

It can be noted that the optimal loads, when both groups of main engines are operated, are not always equally distributed on the two engine groups. This comes as a consequence of the nonlinearity of the efficiency curve of the engines, which can make it more convenient to operate one engine closer to its most efficient point, while leaving the other at low load, and low efficiency.

The resulting lower fuel consumption can be related to a contribution of several factors, as shown in Table 7Table 7. The comparison of the losses in the baseline and hybrid case, broken down to the different groups of components, shows the improvement in the performance of the main and the auxiliary engines, which are operated closer to their most efficient load point. On the other hand, it is also shown that the advantages resulting from the hybridisation of the system are reduced by increased transmission losses, both in the motors/generators and in the gearbox, as well as from additional losses from the boilers. This last point comes from the fact that the engines, being operated at higher efficiency, generate lower losses and therefore lower availability of thermal energy for further use on board.

It should be noted that, in this specific application case, it was imposed that only one of the two ME blocks is able to provide exhaust gas heat. In theory, two EGBs are installed on the MEs on board: one per propulsion line, making it possible to recover heat from the exhaust gas of two ME simultaneously. This is, in practice, never done on board in order to ensure that all the four main engines are operated for a similar amount of time.

Table 5: Detail of the load and efficiency of all component groups for the case study vessel and for the reference voyage. Baseline power plant.

Time	OM	Main engines (1)					Main engines (2)					Auxiliary engines					Aux. Boilers		
		x_1	x_5	x_6	x_7	η	x_2	x_8	x_9	x_{10}	η	x_3	x_{11}	x_{12}	x_{13}	η	x_4	x_{14}	η
1	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.62	0.43	0	-	-
2	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.59	0.43	0	-	-
3	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.59	0.43	0	-	-
4	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.58	0.43	0	-	-
5	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.61	0.43	1	0.36	0.85
6	ST	1	0.19	0	0	0.37	1	0	0.19	0	0.37	1	0	0	0.61	0.43	1	0.14	0.85
7	ST	1	0.22	0	0	0.38	1	0	0.22	0	0.38	2	0	0	0.36	0.43	1	0.29	0.85
8	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.67	0.43	2	0.30	0.85
9	ST	1	0.39	0	0	0.42	1	0	0.39	0	0.42	1	0	0	0.72	0.43	0	-	-
10	ST	1	0.58	0	0	0.45	1	0	0.58	0	0.45	1	0	0	0.71	0.43	0	-	-
11	ST	1	0.56	0	0	0.45	1	0	0.56	0	0.45	1	0	0	0.72	0.43	0	-	-
12	ST	1	0.35	0	0	0.41	1	0	0.35	0	0.41	1	0	0	0.72	0.43	0	-	-
13	ST	1	0.27	0	0	0.40	1	0	0.27	0	0.40	1	0	0	0.72	0.43	0	-	-
14	ST	1	0.30	0	0	0.40	1	0	0.30	0	0.40	1	0	0	0.72	0.43	0	-	-
15	ST	1	0.27	0	0	0.39	1	0	0.27	0	0.39	1	0	0	0.72	0.43	0	-	-
16	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.74	0.43	1	0.20	0.85
17	ST	0	-	-	-	-	0	-	-	-	-	2	0	0	0.74	0.43	1	0.22	0.85
18	ST	0	-	-	-	-	0	-	-	-	-	1	0	0	0.76	0.43	1	0.32	0.85
19	ST	1	0.24	0	0	0.39	1	0	0.24	0	0.39	1	0	0	0.72	0.43	0	0.16	0.85
20	ST	1	0.28	0	0	0.39	1	0	0.28	0	0.39	1	0	0	0.72	0.43	0	0.16	0.85
21	ST	1	0.29	0	0	0.40	1	0	0.29	0	0.40	1	0	0	0.67	0.43	0	0.19	0.85
22	ST	1	0.37	0	0	0.42	1	0	0.37	0	0.42	1	0	0	0.64	0.43	0	-	-
23	ST	1	0.42	0	0	0.43	1	0	0.42	0	0.43	1	0	0	0.63	0.43	0	-	-
24	ST	1	0.30	0	0	0.40	1	0	0.30	0	0.40	1	0	0	0.61	0.43	0	-	-

Table 6: Detail of the load and efficiency of all component groups for the case study vessel and for the reference voyage. Hybridised power plant

Time	OM	Main engines (1)					Main engines (2)					Auxiliary engines					Aux. Boilers		
		x_1	x_5	x_6	x_7	η_{ME1}	x_2	x_8	x_9	x_{10}	η_{ME2}	x_3	x_{11}	x_{12}	x_{13}	η_{AE}	x_4	x_{14}	η_{AB}
1	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.62	0.43	0	-	-
2	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.59	0.43	0	-	-
3	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.59	0.43	0	-	-
4	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.58	0.43	0	-	-
5	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.61	0.43	1	0.36	0.85
6	MIX	1	0.19	0.00	0.00	0.37	0	-	-	-	-	2	0.00	0.25	0.31	0.42	1	0.14	0.85
7	MIX	1	0.22	0.00	0.00	0.38	0	-	-	-	-	3	0.00	0.19	0.24	0.41	0	-	-
8	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.67	0.43	2	0.30	0.85
9	ME2	1	0.39	0.00	0.34	0.46	1	0.00	0.40	0.03	0.43	0	-	-	-	-	0	-	-
10	ME2	1	0.58	0.00	0.18	0.46	1	0.00	0.58	0.18	0.46	0	-	-	-	-	0	-	-
11	ME2	1	0.56	0.00	0.19	0.46	1	0.00	0.56	0.19	0.46	0	-	-	-	-	0	-	-
12	ME2	1	0.35	0.00	0.34	0.46	1	0.00	0.35	0.03	0.42	0	-	-	-	-	0	-	-
13	MIX	1	0.27	0.00	0.32	0.45	1	0.00	0.28	0.00	0.40	1	0.00	0.00	0.10	0.37	0	-	-
14	ME2	1	0.30	0.00	0.34	0.46	1	0.00	0.31	0.03	0.41	0	-	-	-	-	0	-	-
15	ME2	1	0.27	0.05	0.29	0.45	1	0.00	0.23	0.08	0.40	0	-	-	-	-	0	-	-
16	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.74	0.43	1	0.20	0.85
17	ST	0	-	-	-	-	0	-	-	-	-	2	0.00	0.00	0.74	0.41	1	0.22	0.85
18	ST	0	-	-	-	-	0	-	-	-	-	1	0.00	0.00	0.76	0.43	1	0.32	0.85
19	MIX	1	0.24	0.00	0.00	0.39	0	-	-	-	-	2	0.00	0.31	0.36	0.43	0	-	-
20	MIX	1	0.27	0.00	0.00	0.39	0	-	-	-	-	2	0.00	0.35	0.36	0.43	0	-	-
21	MIX	1	0.29	0.00	0.00	0.40	1	0.00	0.10	0.00	0.35	2	0.00	0.24	0.34	0.43	0	-	-
22	MIX	1	0.37	0.00	0.27	0.46	1	0.00	0.37	0.00	0.42	1	0.00	0.00	0.13	0.38	0	-	-
23	ME2	1	0.42	0.00	0.32	0.46	1	0.00	0.42	0.00	0.43	0	-	-	-	-	0	-	-
24	ME2	1	0.30	0.00	0.31	0.45	1	0.00	0.30	0.00	0.40	0	-	-	-	-	0	-	-

Table 7: Breakdown of the conversion and transmission losses during the reference voyage, comparison between baseline and hybrid system. All values are presented in GJ

Losses in:	Baseline	Hybrid	Difference
Main engines	333,00	432,00	-28,600
Auxiliary engines	231,00	103,00	
Auxiliary boilers	4,25	4,25	0,000
Gearbox	10,50	12,99	2,500
SG/SMs	6,38	6,71	0,330
Frequency converters	0,00	1,49	1,490

4.3. Shaft motor/generator dimensioning

The results presented in the previous section relate to the installation of a SG/SM rated 2000 kW on each propulsion line. The analysis of the optimal dimensioning of the size of the SM/SG is hereby presented.

The maximum savings are achieved for an installed SG/SM power of 2000 kW, as for higher installed SG/SM power the effect of decreasing efficiency when operated at part-load becomes relevant and is not balanced by increased benefits (see Figure *13*).

Figure 13 also shows the results of the installation of an SG/SM on only one of the two propulsion lines. This allows only one of the two groups of main engines to be used for auxiliary electric power generation, while not allowing engine group 1 to transfer mechanical power to propeller 2 and vice versa. It can be noticed that although savings are lower, it would still be possible to achieve an estimated 2.3% fuel savings.

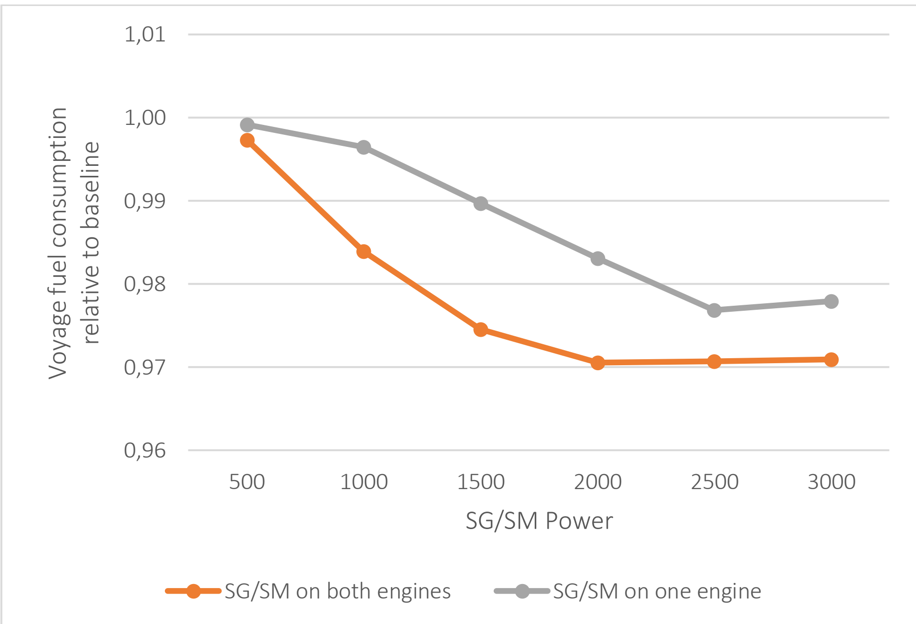


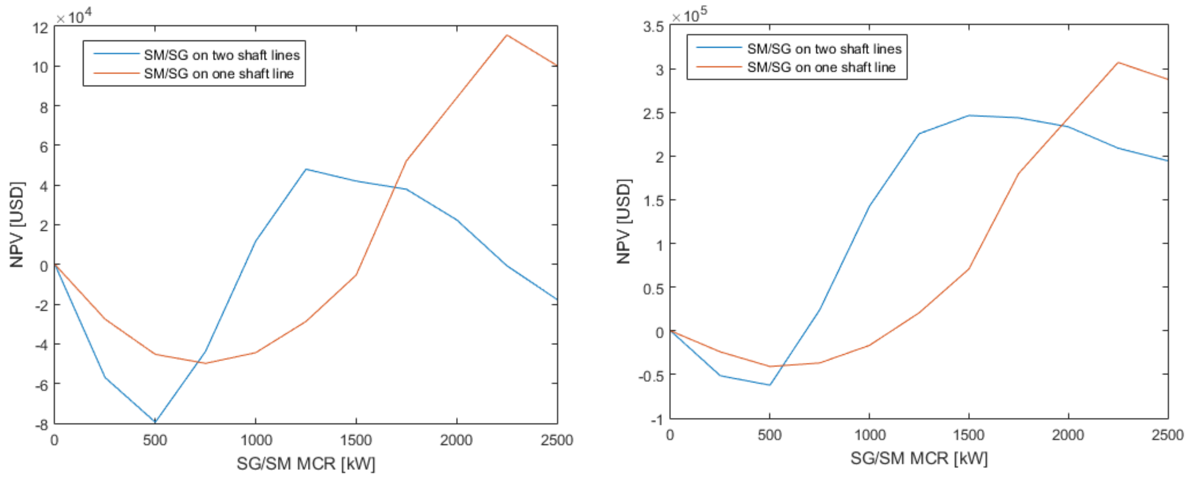
Figure 13: Calculated voyage fuel consumption versus installed SG/SM power

The energetic performance of the hybrid system suggests that the largest benefits can be achieved when SG/SM are installed on both shaft lines, for an installed power of approximately 2000 MW per machine. The solution with the SG/SM on only one shaft line, although promising, does not lead to a decreased fuel consumption in the same magnitude.

The NPV of the system retrofit was calculated based on a 5% interest rate [48], a fuel price of 300 USD/ton and a time horizon of 5 and 10 years. The cost function proposed by Astolfi et al. [49] for electric generators was used to estimate the dependence of capital costs on installed power, not considering the cost of installation of considering a retrofit.

From results of the economic analysis, it appears that (see Figure 14):

- For having a positive NPV it is required to install at least a certain size for the SG/SM (above 1000 kW each for installation on two shaft lines, above 1600 kW for installation on only one shaft line in the 5 year case).
- The installation of a SG/SM on only one of the two shaft lines is more economically convenient for the investigated time horizons. The difference is more pronounced for the 5-years horizon.
- For the case where the SG/SM is installed on only one propulsion line, the optimal MCR does not change with the time horizon and is located around 2250 kW of installed power.
- For the case where the SG/SM is installed on both propulsion lines, the optimum point moves towards higher installed power (from approx. 1250 kW to approx. 1500 kW) with increased time horizon



(a) 5-years horizon (b) 10-years horizon

Figure 14: NPV of the hybridised system as a function of the installed power of the SG/SM

4.4. The relevance of thermal power demand

Compared to the work already published in scientific literature, such as what presented by Kanellos et al. [31], the present work proposed as element of innovation the inclusion of the thermal energy demand in the constraints to be fulfilled by the load-allocation algorithm, and the fuel consumption from the oil-fired boilers to the objective function to be minimised.

Figure 15 presents the comparison of the fuel consumption resulting from the use of the optimisation algorithm with and without the presence of the oil-fired boilers contribution to the objective function. The results presented in Figure 15 refer to the hybridised system: the conditions imposed in the baseline system the load allocation problem do not allow sufficient flexibility for benefiting from an improved load allocation procedure.

When the ship is sailing, more than sufficient waste heat is available from the main engines as to make boilers operations unnecessary in both cases. When the ship is in port, however (see. 5-6 AM and 6-8 PM in Figure 15: Comparison of the fuel consumption for the reference voyage. The), only the auxiliary engines are running and, therefore, the oil-fired boilers are used. In these conditions, the optimisation of the load allocation is also influenced by the need of satisfying the demand for thermal power on board.

The improvement in the performance of the system is estimated to reach up to roughly 3%, compared to an optimisation where the thermal element of on board energy demand is not considered. This can also be seen in the different load allocation in the two cases, as shown in Table 8. It can be seen in particular in the cases of 6-8 PM that the load allocation algorithm that takes into account the fuel consumption of the boilers adjusts the load on the engines to avoid running the boilers.

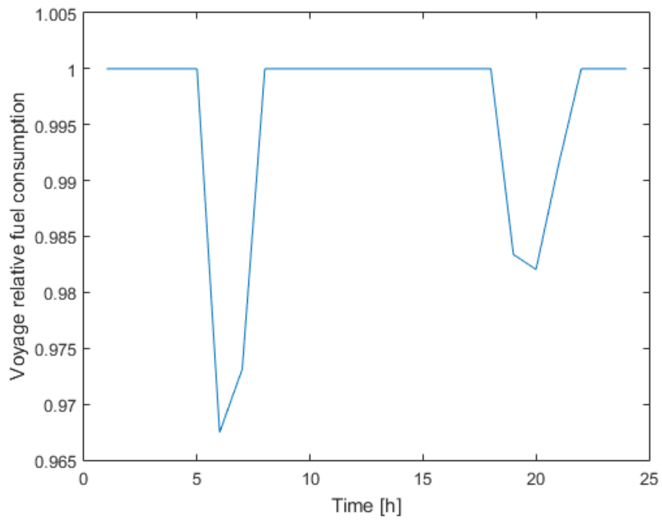


Figure 15: Comparison of the fuel consumption for the reference voyage. The value on the Y axis represents the ratio between the ship fuel consumption when the load sharing is optimised including boilers fuel consumption, and the ship fuel consumption when the boilers' fuel consumption is not included in the optimisation.

Table 8: Comparison of the load distribution when the fuel consumption of the auxiliary boiler is included or not.

	Load							
Time	$\dot{m}_{AB}$ not included in f_{obj}				$\dot{m}_{AB}$ included in f_{obj}			
	ME (1)	ME (2)	AE	AB	ME (1)	ME (2)	AE	AB
6	50% (1)	19% (1)	0% (0)	27% (1)	19% (1)	19% (1)	61% (1)	14% (1)
7	56% (1)	24% (1)	0% (0)	20% (1)	22% (1)	22% (1)	68% (1)	0% (0)
19	58% (1)	27% (1)	0% (0)	10% (1)	52% (1)	24% (1)	17% (1)	0% (0)
20	61% (1)	30% (1)	0% (0)	10% (1)	57% (1)	27% (1)	15% (1)	0% (0)

4.5. Further considerations

The results presented in Sections 4.2 to 4.4 related to the sole aspects related to energy and fuel consumption. Hereafter some issues are discussed that, although not strictly related to the efficiency of engines and boilers, have an impact of the behaviour of the whole power plant.

Maximising the safety of ship operations often brings to conflicting interests with energy efficiency. On the specific ship under study, for instance, it has been observed that when the ship is operating in some conditions (for instance when manoeuvring in port) it is preferred to run two auxiliary engines at low load instead of one at high load in order to provide redundancy in the failure of one of the two engines.

This type of operational constraints, which can vary from ship to ship, are not included in the presented load-optimisation method, and the possibility to add such a constraint to the problem formulation should be seen as a way to further improve the validity of the results.

Engine operations at low load were restricted by imposing a low limit on the engine loading, which was namely not allowed to be lower than 10% of the MCR. In practice, low-load operations are in general demanding on the engines and should be avoided in order to reduce engine wear. The method presented in this work does not account for this aspect, and the addition of soft constraints for disincentivising low-load operations can be also consider as a further addition to the model.

The importance of the heat demand, especially for those conditions in which it is required to run the auxiliary boilers at very low load, was highlighted by the results of this work. This would suggest that additional solutions for improving energy efficiency on board from the heat demand side could be implemented. In particular, energy storage (as proposed for a different ship type in previous work by the authors [50]) and demand-side management are seen as possible solutions that could bring to improvements in the efficiency of the power plant with limited costs and efforts.

5. Conclusion

In this paper, a method for optimising the load allocation of the different prime movers for an isolated system characterised by independent demands of mechanical, electric and thermal power is presented. The method consists of the use of simplified nonlinear correlations for the efficiency of the individual components, and of the definition of a MINLP problem to be solved with a combination of SQP and branch-and-bound methods.

The proposed method was applied to the power plant of a cruise ship sailing in the Baltic Sea, characterised by a time-dependent demand of mechanical power (for propulsion), electric power (for on board auxiliaries and accommodation) and thermal power (for accommodation and other facilities).

The application of the method showed its ability to identify the optimal load allocation for the different prime movers. The results showed the importance of including thermal power demand into account, which can lead to fuel savings of up to 3% compared to the case where this demand is not accounted for.

Furthermore, the method was applied to the possibility of improving the efficiency of the power plant by installing shaft generators/motors. The proposed method allows handling the increased level of complexity of the system, where the additional interconnections within the system would allow for the on board power demands to be fulfilled in many different combinations of prime movers. This allowed evaluating the performance of the retrofitted system, which showed the possibility for up to 3% yearly savings in fuel consumption, and identifying the optimal installed power for maximising the NPV of the system.

Acknowledgments

The authors of this paper would like to thank the Swedish Energy Agency (Energimyndigheten) and the Swedish Maritime Administration (Sjöfartsverket) for their financial support. The authors of this paper would also like to thank Eckerö Rederiet, and in particular the crew of MS Birka Stockholm, for sharing the data, information and experience that was crucial for this work.

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