TECHNOLOGY • RESEARCH • HIGHLIGHTS • VISIONS • SCIENCE
306

Ship energy efficiency technologies – now and the future

Zou Guangrong (Editor)

VTT TECHNOLOGY 306

Ship energy efficiency technologies – now and the future

Zou Guangrong (Editor)

VTT Technical Research Centre of Finland Ltd

ISBN 978-951-38-8567-0 (Soft back ed.)
ISBN 978-951-38-8566-3 (URL: http://www.vttresearch.com/impact/publications)

VTT Technology 306

ISSN-L 2242-1211
ISSN 2242-1211 (Print)
ISSN 2242-122X (Online)
http://urn.fi/URN:ISBN:978-951-38-8566-3

JULKAISIJA – UTGIVARE – PUBLISHER

Teknologian tutkimuskeskus VTT Oy
PL 1000 (Tekniikantie 4 A, Espoo)
02044 VTT
Puh. 020 722 111, faksi 020 722 7001

Teknologiska forskningscentralen VTT Ab
PB 1000 (Teknikvägen 4 A, Esbo)
FI-02044 VTT
Tfn +358 20 722 111, telefax +358 20 722 7001

VTT Technical Research Centre of Finland Ltd
P.O. Box 1000 (Tekniikantie 4 A, Espoo)
FI-02044 VTT, Finland
Tel. +358 20 722 111, fax +358 20 722 7001

Juvenes Print, Tampere 2017

Preface

Given the great opportunity in Tekes[1] Arctic Seas Programme (2014–2017), we were able to form a joint R&D consortium (Ship Energy Efficiency Technologies – SET) to demonstrate a Finnish approach to systematically modelling onboard energy systems of different ship types and to evaluating and optimizing the practical performances of various energy efficiency technologies and solutions, using ship energy flow simulation methods, and to improving the energy efficiency of different ship types under real operating conditions. In this book (also as the final report of the SET projects), we showcased some selected work achieved in the SET projects to hopefully give the readers a flavor of different energy efficiency technologies and methods for their potential applications in the shipping industry and other energy-related industries.

Book chapters

This book is intended, as an introductory and reference book, for general experts, researchers and students interested in the topics related to ship energy efficiency technologies. It comprises one introductory chapter and three technical parts (including 11 chapters): Ship energy efficiency technologies, Ship energy efficiency methods, and Summary of industrial projects. Chapter 1, Introduction to the SET project, is intended to give some background information on the SET consortium and a general introduction to the case ships utilized in the book.

There are three chapters in part I of the book, focusing on ship energy efficiency technologies. Chapter 2 gives a state of the art overview of the practices and recent progresses in passenger shipping energy efficiency. Chapter 3 and 4 introduce two group of energy saving technologies on their potentials for ship waste heat recovery (WHR). Specifically, Chapter 3 first reviews thermal energy storage technology, focusing on latent heat storage, and then carries out a small-scale study on a combined heat storage scenario and compares the WHR potential of different latent heat storage in one case ship. Chapter 4 provides a thorough overview of absorption refrigeration technology, briefly introduces how to model absorption refrigeration processes and further evaluates and optimizes the performances of those technologies for WHR in one case ship.

[1] Tekes – the Finnish Funding Agency for Innovation.

Chapters 5 through 8 constitute the second part of the book, which is dedicated to ship energy efficiency methods. Chapter 5 gives a brief introduction to the ship energy flow simulation method, which has been developed in the former projects and utilized as a testing and optimization platform for different case studies in the SET projects. Chapter 6 gives more details on how to use the energy flow simulation method to model the energy systems of an icebreaker and to further evaluate the potential of different energy saving scenarios. In Chapter 7, pinch analysis is used to optimize the energy systems of a case bulk carrier and further integrate steam Rankine Cycle into the ship energy systems of the case ship. Chapter 8 thoroughly introduces an integrated approach to ship energy system design and operation and further proposes a unified multi-level simulation and optimization framework to better integrate holistic optimization and dynamic simulation of ship energy systems throughout its life cycles.

Part III consists of Chapters 9 to 12, which provide a short summary of four industrial projects run in parallel with the public research project in the SET consortium. In Chapter 9, Deltamarin presents their activities in the SET project and give some example results on how to use simulation-based design method to optimize the energy storage, auxiliary cooling systems and WHR onboard ships. In Chapter 10, ABB shortly takes an overview of their objectives and some example results achieved in their company project specifically on simulation-based marine business innovations. In Chapter 11, Alfa Laval Aalborg briefly presented their activities in the SET project and highlighted their achievement in natural circulation boiler systems in marine applications. In Chapter 12, Aker Arctic introduces their activities in the SET project and further strengthens their Arctic knowhow for designing and optimizing ice going ships.

Acknowledgement

We gratefully acknowledge the generous financial support from Tekes and the SET consortium partners, including Aalto University, ABB Oy Marine and Ports, Aker Arctic Inc., Alfa Laval Aalborg Oy, Deltamarin Ltd and VTT Technical Research Centre of Finland Ltd.

Special thanks are due to the SET project supporting partners who have kindly provided their vessels as case ships in the project, including Arctech Helsinki Shipyard Oy, Arctia Oy, Trafi (Finnish Transport Safety Agency) and Viking Line Abp.

Last but not least, I would like to express my full appreciation to the whole SET steering group and the project group. Without your great support, collaboration and hard work, it would not have been possible to achieve what we have promised in the SET project and make the SET project another success. My big thanks to all of you, including but not limited to

Tekes: Piia Moilanen (Arctic Seas Programme Manager), Rauli Hulkkonen, Maarit Kokko and Ritva Heikkinen;

Aalto University: Kari Tammi, Maunu Kuosa, Juha Vanttola, Waltteri Salmi, Emil Fridolfsson and Tuomo Koivisto;

ABB: Kalevi Tervo;

Aker Arctic: Kari Laukia and Tommi Heikkilä;

Alfa Laval: Pasi Aaltonen;

Deltamarin: Päivi Haikkola (Chair of the SET steering group), Mia Elg, Panu Mäkipeska and Ossi Mettälä;

VTT: Seppo Kivimaa, Johannes Hyrynen, Kari Tammi, Mikko Pihlatie, Mia Elg, Aki Kinnunen, Pekka Rahkola, Tuomas Kyllönen, Heikki Isomoisio, Jukka Junttila, Irmeli Sairiala and Irene Pihlajamäki.

ZOU Guangrong
VTT Technical Research Centre of Finland Ltd

Contents

Abbreviations and acronyms

AC	Alternative current, or air conditioning
AE	Auxiliary engine
AHU	Air handling unit
AIS	Automatic identification system
APCD	Average passenger cruise day
ARC	Absorption refrigeration cycle
BT	Bow thruster
CAC	Charge air cooler
CAPEX	Capital expenditure
CHP	Combined heat and power
COP	Coefficient of performance
CRP	Counter rotating propulsion
DAE	Differential-algebraic equation
DC	Direct current
DE	Diesel engine
DF	Dual-fuel
DG	Diesel generator
ECA	Emission control area
EEDI	Energy efficiency design index
EEE	Energy and environmental efficiency
EEOI	Energy efficiency operational indicator
EG	Exhaust gas
EGB	Exhaust gas boiler

EGE	Exhaust gas economizer
FC	Frequency converter
FW	Fresh water
GCC	Grand composite curve
GEM	Galley energy management
HEN	Heat exchange network
HFO	Heavy fuel oil
HT	Heat temperature
HTF	Heat transfer fluid
HTS	High tensile strength
HVAC	Heating, ventilation and air-conditioning
ICE	Internal combustion engine
IMO	International Maritime Organization
JC	Jacket cooler
LCC	Life cycle cost
LED	Light-emitting diode
LiBr	Lithium Bromide
LNG	Liquefied natural gas
LO	Lubrication oil
LOC	Lubrication oil cooler
LSMDO	Low-sulphur marine diesel oil
LT	Low temperature
MDO	Marine diesel oil
ME	Main engine
MED	Multiple effect distillation
MER	Maximum energy recovery
mmf	Magneto-motive force
MSB	Main switchboard
MSF	Multi-stage flash
NCR	Nominal continuous rating

ODE	Ordinary differential equation
OFB	Oil fired boiler
OIP	Onboard installed power
OIPPP	Onboard installed power per person
OPEX	Operational expenditure
ORC	Organic Rankine cycle
PCM	Phase change material
PT	Power turbine
PTI	Power take-in
PTO	Power take-out
RC	Rankine cycle
RO	Reverse osmosis
ROI	Return on investment
rORC	Regenerative organic Rankine cycle
RPM	Revolution per minute
SEEMP	Ship energy efficiency management plan
SET	Ship energy efficiency technology
SFOC	Specific fuel oil consumption
SG	Shaft generator
SGM	Shaft generator motor
SoC	State of charge
SRC	Steam Rankine cycle
ST	Steam turbine
SW	Seawater
TES	Thermal energy storage
TI	Temperature interval
VFD	Variable frequency drive
VSD	Variable speed drive
WHR	Waste heat recovery
WHRS	Waste heat recovery system

Symbols

c_p	Specific heat capacity (kJ/kg-K or J/mol-K)
ΔT_{min}	Minimum approach temperature (K or $°C$)
Δp	Pressure difference (Pa, kPa, MPa or bar)
ε	The effectiveness of the heat exchanger (%)
E_{tot}	Total energy production or demand (MWh)
η	(turbine, pump or motor) efficiency (%)
h	Specific enthalpy (kJ/kg)
$\dot{m}$	Mass flow (kg/s)
$\dot{n}$	Molar flow (mole/s)
ρ	Density (kg/m^3)
p	Pressure (Pa, kPa, MPa or bar)
P	Pump work (kW)
$\dot{Q}_e$	Cooling power at the evaporator (kW)
s	Specific entropy (kJ/kg-K)
t	Time (s, min or h)
T	Temperature (K or $°C$)
T_a	Temperature at absorber (K or $°C$)
T_c	Temperature at condenser (K or $°C$)
T_e	Temperature at evaporator (K or $°C$)
T_g	Temperature at generator (K or $°C$)
x	Steam quality (%)

1. Introduction to the SET project

Zou Guangrong [1)], Mia Elg[1), 2)], Kari Tammi[1), 3)], Seppo Kivimaa [1)]
[1)] VTT Technical Research Centre of Finland Ltd
[2)] Deltamarin Ltd, [3)] Aalto University

1.1 General information

Climate change is a global issue and shipping is a global industry. The need to reduce marine-borne emissions has been well understood by the shipping industry. As the world trade is projected to grow significantly in the coming decades, emission reduction has been and will be a big challenge to the shipping industry.

Shipping industry, as a key driver to the world's economy, accounts for around 90% of world trade globally and currently contributes to 2–3% of global greenhouse gas (GHG) emissions. As the global effort into GHG emission reduction, the introduction of IMO (International Maritime Organization) energy efficiency measures[1] has already stimulated the development and adoption of novel energy efficiency technologies, practices and innovations through ships' design and operation phases. However, despite the great efforts, a 50–250% increase in maritime CO_2 emissions from international maritime transport is projected by 2050 due to the expected future economic growth and transport demand increase. [1] Further actions are already under consideration to enhance ship energy efficiency and to reduce emissions.

On the other hand, the implemented new technologies and innovative solutions to ship design, building, retrofitting and operating processes have rarely resulted in desired energy efficiency increases. This is partly due to the inefficiency of marine energy systems, especially under dynamic operating conditions and uncertainties, and due to lack of systematic consideration and limited interaction between design and operation of energy systems, which commonly exist in all energy-related industries. It will become even more critical as energy systems get more complex and utilize more flexible energy resources and more advanced energy efficiency technologies in the future.

[1] The IMO measures consist of Energy Efficiency Design Index (EEDI), Energy Efficiency Operational Indicator (EEOI), Ship Energy Efficiency Management Plan (SEEMP) and Emission Control Areas (ECA).

To meet the challenge, we have formed a joint Finnish R&D consortium[1] consisting of four leading marine industrial companies and two high-level research institutions in the marine industry, namely "SET – Ship Energy Efficiency Technologies", with the funding support of the Tekes Arctic Seas programme[2] in 2014–2017. The key partners have had long-term close collaboration in the topics related to ship energy efficiency. Especially, VTT, Deltamarin and ABB have developed a general energy flow simulation platform together to understand energy distribution within different ship systems and to improve ship overall energy efficiency. The adopted multi-domain simulation method enables a simple but feasible representation of ship energy systems at a system level. More importantly, the ship energy flow simulation platform gives some valuable insights into how to design energy-efficient ship power plants and how to operate vessels more efficiently.

1.2 Objectives and scope

By answering three key R&D questions, the main objective of this project is to utilize new energy-saving technologies and to seek new innovative ways so as to improve ship overall energy efficiency. To achieve this target, a forward-looking strategy (current – next step – future) is followed during the project execution, meaning we will not only focus on short-term improvements of ship energy efficiency but also on long-term potential for optimal energy efficient ship solutions. To keep the main interests in this project, we have limited our scope only to ship onboard energy systems, with special focus on the technologies and solutions related to waste heat recovery (WHR) of ships.

The first question we are to answer is how efficient the ship onboard energy systems are running currently. Various energy saving technologies were reviewed and investigated in order to identify their potential in ship applications. The selected case ships are good examples of up-to-date energy efficient solutions, which provide us a great opportunity to review the implemented energy utilization technologies in ships in the last decade.

The second question is how much we are able to improve ship operational efficiency using new technologies, especially for the case ships. This is done by virtually implementing the energy saving scenarios into the energy flow simulators of the case ships without prototyping or installation, which will largely lower the risk of possible investments. Based on the scenario evaluations, the successful implementation would benefit directly the marine industry.

[1] The SET R&D consortium consists of Aalto University, ABB Oy Marine and Ports, Aker Arctic Inc., Alfa Laval Aalborg Oy, Deltamarin Ltd and VTT Technical Research Centre of Finland Ltd.

[2] Arctic Seas programme is a key national R&D programme launched by Tekes – the Finnish Funding Agency for Innovation in 2014, aiming at turning Finland into an internationally attractive centre of Arctic know-how. (https://www.tekes.fi/en/programmes-and-services/tekes-programmes/arctic-seas/)

The third question is how we can design next generation ships. The holistic analytical methods and advanced optimization technologies based on the simulation platform will help us to explore innovative solutions to achieve globally optimal energy efficiency for future ships. This would provide valuable insights for the marine industries into green shipping.

1.3 Waste heat recovery

Shipping has been claimed to be the most energy efficient way of transportation. However, dominated but also limited by internal combustion engines (ICEs) in ship energy systems, even the most energy efficient ships, combined with different energy efficiency technologies, can barely achieve 60% energy efficiency nowadays. In general, another 40–60% of fuel energy has been dissipated, as waste heat energy, into the environment. Besides, even if the waste heat is utilized for certain heating purposes, the current utilization is not necessarily efficient. Hence, waste heat recovery and utilization has been one of the big potential areas to improve ship overall energy efficiency.

Many WHR technologies and solutions have been developed and utilized in the global industries, specifically in process industries. The shipping industry has been also looking into those technologies and taking the potential ones into use. In this project, we reviewed many of the WHR technologies, which are already adopted or available in the shipping industry, mainly including organic Rankine cycle (ORC), steam Rankine cycle (SRC), power turbine (PT), natural circulation boilers, heat pump, absorption refrigeration and thermal energy storage. Based on the initial investigation, we further evaluated their potential and feasibility in improving ship energy efficiency and developed new methods to further optimize the implementation of the technologies already in use and help identify potential technologies for ship applications in the near future.

1.4 Case ships

The SET project adopted an application-centred approach and several state of the art ships were used as case ships for investigating the selected novel energy saving technologies or for the energy system design and operation optimization using process integration and system-level simulation methods. Especially, below are the two case ships that have been intensively utilized in the SET projects.

1.4.1 IB Polaris

IB Polaris, shown in Figure 1.1, is the world's first icebreaker powered by dual-fuel engines capable of using both low-sulphur marine diesel oil (LSMDO) as well as liquefied natural gas (LNG) as fuel. The dual-fuel power plant reduces the vessel's emissions significantly and complies with both IMO Tier III emission limits as well as the special requirements of the Baltic Sea Emission Control Area (ECA), making it also the most environmentally friendly diesel-electric icebreaker ever built. [2] The icebreaker is owned and operated by Arctia Oy.

Technical specifications

Length	110 m
Beam	24 m
Operational draft	8 m
Displacement	3000 t
Installed power	22 MW
Propulsion power	19 MW
Speed	17 knots
Speed at 1.2m ice	6 knots
Pollard pull	214 tn
Crew	16
Endurance	30 days
Nationality	Finland
Classification	Lloyd's Register

Figure 1.1. The icebreaker IB Polaris and its technical specifications. [2]

The icebreaker's power plant consists of two 9-cylinder Wärtsilä 9L34DF and two 12-cylinder Wärtsilä 12V34DF four-stroke medium-speed dual-fuel generator sets fitted with exhaust gas economizers (EGEs). In addition, the vessel has one 8-cylinder Wärtsilä 8L20DF auxiliary engine that can be used to produce electricity when the ship is at port. The combined output of the electrical power plant, which can produce power for all shipboard consumers with any combination of generating sets depending on the power demand, is about 22.5 MW. There is also a separate emergency diesel generator. [3]

IB Polaris features a novel propulsion system consisting of three electrically driven ABB Azipod propulsion units, one of which is located in the bow of the vessel. The stern propulsion units are rated at 6.5 megawatts each while the azimuth thruster in the bow has propulsion power of 6 megawatts. Each ice-strengthened Azipod unit has a four-bladed stainless steel propeller with removable blades and weighs 135 tons. The combined propulsion power of the three propulsion units, 19 MW, makes Polaris the most powerful icebreaker ever to fly the Finnish flag as well as the most powerful icebreaker built specifically for escort operations in the Baltic Sea. [3]

1.4.2 B.Delta 37

B.Delta37, designed by Deltamarin Ltd, has attracted significant attention in the industry due to its best-in-class (handy size segment) performance in terms of low fuel oil consumption and high cargo intake. So far, about 80 B.Delta37 bulk carriers with different cargo hold configurations have been ordered or are already sailing.

The vessels are fitted with one slow speed main engine directly coupled to a fixed-pitch propeller. Electricity is produced by three auxiliary engines, and there is also a separate emergency generator. Steam is produced by multi-inlet composite boiler, including exhaust gas and oil fired sections.

Technical specifications

Length	179.99 m
Beam	30 m
Depth	15 m
Draft, scantling	10.7 m
Draft, design	9.5 m
Deadweight	40600 ton
No. of cargo holds	5
Cargo capacity	50000 m^3
Speed	14 knots
Main engine power	6050 kW
Aux. engine power	3x680 kWe

Figure 1.2. The bulk carrier B.Delta 37 and its technical specifications. [4]

References

[1] International Maritime Organization. Third IMO GHG study 2014. [online] Available: http://www.imo.org.

[2] Arctia Oy. LNG icebreaker IB Polaris. [online] Available: http://arctia.fi/en/ship/polaris/

[3] Wikipedia. Polaris (icebreaker). [online] Available: https://en.wikipedia.org/wiki/Polaris_ (icebreaker)

[4] Deltamarin Ltd. B.Delta37 bulk carrier. [online] Available: https://www.deltamarin.com/references/b-delta37/

Part I
Ship energy efficiency technologies

2. Passenger ship energy efficiency − a review[1]

Tuomas Kyllönen
VTT Technical Research Centre of Finland Ltd

Summary: This chapter briefly reviewed novel methods and technologies currently in use to increase energy efficiency in cruise ships and took a sneak peek at the possible future developing trend in improving energy efficiency on passenger ships in the near future. Some ships and ship classes were studied in terms of energy saving technologies and the development of passenger ship energy efficiency in the early 21st century, followed by future scenarios for ship powering and energy efficiency.

Cruise shipping industry is growing. Meanwhile, passenger ship operators are increasingly taking into account energy efficiency and emission reduction. Efficient ship design and operation deliver fuel savings and promote green images of the shipping lines. Consequently, passenger ships have been getting bigger and more energy efficient technologies and methods have been taken into use in the last decades. More energy efficient passenger ships were designed and equipped with state of the art technologies and products, including novel hull forms and ship structures, latest power generation and energy distribution systems, energy saving and waste heat recovery technologies. Newer ships also utilize other new energy saving technologies, such as air lubrication, efficient lighting and renewable energy technology installed. Energy efficient operations are getting more attention as well. In addition, shipping companies' sustainability reports have indicated some progress on fuel economy.

Besides, an analysis based on installed main engine power per person on-board was performed. From inspected ships, it was found that the onboard installed power per person seems to be generally decreasing on the new builds than in the early 2000s, although it was not definitive.

[1] This chapter is excerpted and revised from the SET research project deliverable with the same name.

2.1 Introduction

As a fast-growing business, the cruise-shipping industry is trying to get eco- and environmentally-friendly images to attract customers. More importantly, cutting costs and environmental regulation drive the industry to more energy efficient design and operation. The industry has made a lot of efforts and good practices to improve energy efficiency of modern large passenger ships in the last 2 decades.

Classification societies and some industrial leaders, such as American Bureau of Shipping (ABS) and DNV GL, have published guidance for energy efficiency in ships. ABS presents five general sections in improving ship energy efficiency, including hull form optimization, energy saving devices, structural optimization, and machinery technology and fuel efficiency of ships in service. [1] Energy saving measures presented by DNV GL include

- hull form optimization
- propulsion efficiency devices
- new materials (lighter weight)
- anti-fouling and coating of the hull to prevent drag increasing over time
- waste heat recovery
- auxiliary engine economizer
- main engine tuning
- electrification of the ship and switching to DC grid
- operational optimization
- weather routing
- speed optimization
- speed control of pumps, fans etc.
- trim optimization
- port/ship logistics
- performance and energy consumption monitoring
- improved power management
- crew training and awareness

Considering the main aspects of the energy saving measures by classification societies, the aim of this study is to have a state of the art review of the energy efficiency in modern large passenger ships, with the focus on ships built after the 2000s. Furthermore, we give a brief overview of the recent progress on ship energy efficiency by the major cruise lines and some further options to consider for improving energy efficiency of passenger ships in the near future.

2.2 Different aspects of passenger ship energy efficiency

2.2.1 Hull form and ship structures

Hull form and its properties affect the drag generation during sailing. Shipyards usually provide readily available standard hull forms and propulsion systems, which, however, are not necessarily optimal for satisfying each client’s needs. Therefore, when the purpose and general loads are known, optimal ship hull form can be decided from readily available options with possible modifications or even completely new design to meet the needs of specific vessels. [1]

One of the sections in hull form that affects the drag drastically is the bulbous bow. Adding a bulb in the front of the vessel reduces the bow wave and consequently reducing drag. Other factors that affect drag are length of the water line, draught and the length-to-fullness ratio. For example, adding a ducktail on the stern of the ship can increase the length of the water line, which can result in 3–7% improvement in total energy consumption in a typical ship. Even larger gains can be achieved if using interceptor trim planes together, which are installed on the width of the aft to bend flow downwards to increase pressure behind propeller. They are cheaper to retrofit and in some cases more effective than a conventional “trim wedge”. According to Wärtsilä, trim plates can lower the propulsion demand by 1–5%, which corresponds to 4% improvement in total energy consumption for a typical vessel. [2] Aft of the ship can be optimized in other ways as well, such as shaft line arrangement, streamlining, skeg shape and trailing edge. For instance, a proper shaft line design can result in 2% energy savings in total energy consumption, and properly streamlined shafts with “wing thrusters” can achieve 8–10% performance improvement. In addition, general optimization of the propeller and hull interaction can improve the performance up to 4%. [2]

Ships displacement in water also affects the drag since the resistance in water is more or less proportional to the displacement of the vessel. Typical ways to lower displacement are to reduce ballast water or overall weight of the ship using lighter materials. Reducing ballast water yields smaller displacement and lower resistance, which, however, has to ensure ship’s stability and sea keeping behaviour and to keep propeller immersed in the water. Using lighter and higher strength materials (high strength steels or composites) can reduce overall weight of the vessel. Due to their high cost, those materials are most likely used in smaller high-speed vessels. [1] It could be, however, possible to use lighter materials in non-critical structures to reduce the weight of the cruise ships. The use of HTS-steel (high tensile strength) could reduce up to 20% of the weight of steel structures. Realistic propulsion power savings can be around 5% due to the use of lighter materials and HTS-steel in many cases. [2]

Coating of the hull affects the friction between water and the hull, which can be reduced by decreasing the size of wetted surface, using low-friction paint on the hull or using hull surface texturing to prevent sea life to grow on the hull. For instance, major cruise liners, such as Disney, Carnival, Celebrity and MSC Cruises, use “foul release” silicone hull coatings. [3] Royal Caribbean also uses low friction hull coatings on its newer ships. Some newer ships feature hull air lubrication, which was initially tested on Celebrity Reflection in 2013, but not permanently installed. Air lubrication could deliver up to 15% energy savings on tankers, but smaller gains on ferries. [2] Quantum of the Seas, along with Anthem and Allure of the seas, will possibly be retrofitted with this technology. [4] According to Seatrade, 7–8% decrease in fuel consumption is expected on the Quantum of the Seas. [5]

Hull cleaning is another important aspect to improve ship energy efficiency during operation by reducing the physical and biological roughness that accumulates in time. Cleaning the hull on a regular basis prevents slime and other things from increasing the hull friction, and hence avoids the increase in the propulsion power demand. [1]

2.2.2 Ship machinery

For ship machinery design and operation, ship level consideration is essential in order to achieve more efficient management and control of power generation, distribution and consumption. This is especially true for modern cruise ships since their machinery is getting much larger and more complex along with the increase of ship sizes and the use of advanced technologies and equipment.

2.2.2.1 Power generation

Internal Combustion Engine (ICE) is still dominating power generation devices of ships. Cruise ships are moving to common rail injection in their engines, which provides better load control and combustion in comparison to older diesel engines and reduces NO_x emissions. First common rail engines on a cruise ship were installed on Carnival Spirit. [6] Allure of the Seas and Oasis of the Seas, run by Royal Caribbean International, also have common-rail engines, along with Celebrity Cruises that is a subsidiary of the Royal Caribbean Cruise Ltd (RCCL). [3]

Flexible combinations of different power and propulsion generating devices can also result in more energy efficient power generation. For instance, the combined diesel-electric machinery is getting popular, which yields 4% potential savings in energy consumption. [2] Hybrid auxiliary power generation can be done with fuel cells, diesel/gas generators, batteries, wind and solar power. The combined heat and power generation (CHP) can further improve energy efficiency especially for cruise ships with huge steam need.

2.2.2.2 Energy distribution

On-board energy distribution systems present losses but a few percent energy savings are achievable by more efficient energy transmission. For example, the use of DC-grid systems in power distribution can decrease total energy consumption of ships by 5–10% in comparison to AC-grid systems. [2] According to a study, DC-grid systems combined with the optimal use of energy storage (battery) can even double the energy savings. [7]

Modern ships also feature advanced energy management and automation systems, which largely helps energy distribution and utilization throughout ship energy systems. The wider use of energy flow regulating devices, such as variable speed drives (VSD) for pumps, fans and valves, also results in more efficient energy distribution onboard ships. Speed control of engine water cooling pumps can save 1% in energy consumption by optimizing flow for current operating condition. [2]

2.2.2.3 Propulsion

According to Wärtsilä [2], potential areas to increase ship propulsion efficiency are:

- CRP (counter rotating propulsion): CRPs are reported as one of the most efficient propulsors and single propeller vessels can achieve 10–15% power reduction with the CRP propulsion. For instance, 13% energy savings have been reported on two Japanese ferries with ABB CRP POD systems. [8]
- Proper propeller design gives 2–4% improvement in propeller efficiency.
- Propeller-rudder combination: by combining propeller and rudder, 2–6% improvement on fuel efficiency can be achieved.
- Adding propeller nozzle/duct can give 5% fuel savings for cruising speed up to 20 knots. Most passenger ships have cruise speed over 20 knots, and hence ducts are not necessarily applicable on them.
- Constant versus variable speed operation on controllable pitch propellers.
- Pulling thrusters can save up to 15% on propulsion power due to undisturbed water flow to propellers and lesser resistance.
- Propeller efficiency measurement can result in 4% fuel savings. Efficient propeller usage for different loading conditions also increases efficiency. [2]

2.2.2.4 Hotel loads

For their speciality, hotels are one of the main energy consumers on board passenger ships. Especially, lighting consumes up to 25% of total auxiliary electricity consumption. To save energy, many vessels have installed more energy efficiency lighting systems, such as LED and fluorescent lighting. For example, Celebrity Equinox features photovoltaic systems with LEDs from bluleu LED Solutions. LED lighting also provides weight reduction by up to 30%, which possibly means a total weight reduction by up to eight tonnes. Taking another example, Navalimpianti's Powergiant 2 illumination system can provide 7,100 kWh energy savings (meaning 1.5 tonnes of fuel savings every day) for cruise ships illuminated with 8,000 halogen lamps. LEDs would reduce power consumption by 300 kW, whereas power consumption for halogen lamps alone in the public areas can be around 370 kW. [10] According to the LEDs Magazine, "a cruise ship with 50,000 light sources, the use of LED solutions can allow a reduction of CO_2 emissions of up to 3,000 tonnes per year and an annual cost saving of up to EUR 200,000." [10]

LEDs also present longer service life of 50,000 hours compared to halogen bulbs. The service life of halogen bulbs is around 6–7 moths in normal applications but usually lower on board ships due to vibrations of the ships. On the contrary, LED technology is not affected by vibrations, and with 10 hours' average use daily, LEDs can remain operational for over eight years. [10]

They have also used high-efficiency appliances to reduce the consumption of energy and water, for example, by adopting the state of the art air conditioning technologies or putting thermostats in neutral position when staterooms are not in use. In the Quantum of the Seas, the new stateroom key requires guests to be in their rooms for activating the lights and air conditioning. [9]

Besides, fresh water consumption on board passenger ships is very high and hence fresh water generation is another big energy consumer. It can be produced using thermal distillation, reverse osmosis, freezing and electrolysis. For direct use of waste heat recovery, thermal distillation is the available method. Reverse osmosis requires less energy to produce fresh water and is getting more popular recently, but water may need to pre-treated before desalination to avoid fouling. [11] There are also systems that capture condensation water from machinery operation, laundry and air conditioning. Water consumption can be also lowered by using water reduction technologies, such as sink aerators, showerheads, reduced-flow dishwashers, low consumption laundry equipment. On the Royal Caribbean's fleet, people consume half less water than average US citizen does. [9]

2.2.2.5 Heating, ventilation, air conditioning and refrigeration

HVAC (Heating, Ventilation and Air conditioning) systems are also a big energy consumer on-board passenger ships. For instance, the air conditioning system on board Viking Grace accounts for around 15% of total electricity consumption. [12] Some of the cruise lines have measures to reduce energy consumption in HVAC systems. Examples are the use of engine waste heat for heating purposes, automatically switching air conditioning on and off in accommodation areas, and insulation of windows. Royal Caribbean ships have tinted windows to keep the ships cooler and to reduce the load on air conditioning systems [9], along with Cunard's new Queen Elizabeth and Celebrity vessels.

Traditionally marine air conditioning units use a direct-expansion evaporator in the air handler to lower supply air's humidity and temperature. A rotary desiccant air conditioning system can utilize latent waste heat from engines and exhaust gasses for cooling exchange air as well as lower the humidity, which increases overall energy efficiency. By 2012, applications on marine air conditioning were still very few since researchers focus more on land applications. However, a study found that two-stage rotary desiccant air conditioning system could save 33.4% of energy compared to a traditional marine system. Analysis was performed based on real data from ship (4900 size pure car truck carrier), including available residual heat and required amount of steam consumption. [13]

HVAC systems can also save energy by utilizing energy transfer between the incoming and the exhausting air. Incoming cold air can be pre-heated with air that is exhausted from indoors environment in cold seasons, or incoming air can be cooled in the same way during warm seasons. [1]

Besides, in buffet areas, chilled rocks can be used, instead of ice, for cooling purposes. Cold seawater can be used in some areas for cooling purposes, which can save 4–5 tons of fuel per day in these areas. [3]

2.2.3 Waste heat recovery

Diesel engines have maximum energy efficiency of around 48–51%, and the rest of fuel energy is lost as waste heat to exhaust gasses, engine cooling water and heat radiation during the combustion process. Due to their high potential, Waste heat recovery systems have been widely used already in the cruising industry. Especially, waste heat recovery from exhaust gas has the potential to cover 15–20% power need. [2]

WHR technologies for ship applications mainly include turbines, Rankine cycles and desalination. In practices, several technologies are used together for better WHR. Turbines are used in turbochargers and power turbines. Turbochargers are used to harness energy from exhaust gas to compress more air into the engine and hence to allow more fuel injection and consequently to increase engine power production. Power turbines uses exhaust gasses to produce electricity or propulsion power, which is also cheaper compared to steam turbine arrangements. When using power turbines together with turbochargers, one has to ensure that exhaust gas flow rate and temperature have to be suitable for efficient operation of the power turbines. [14]

Rankine cycles are another popular application of WHR to produce propulsion or electricity onboard ships, such as in cruise ferries Viking Grace (from Viking Line) and Fjordline's Stavangerfjord. Usually the system consists of a steam turbine, an evaporator, pumps and a condenser. Currently, water is the most commonly used working fluid but organic fluid is getting popular due to its capability of working under lower temperature conditions and vaporization with less heat. Take a study done on cruise ship M/S Birka Stockholm as an example. The proposed ORC system could produce around 390 kW electricity, which equals to 22% percent of on-board power demand. During a round trip, the ORC system could reduce the fuel consumption by 2,030 kg and the CO_2 emissions by 725 kg. Only exhaust gases were considered as the heat source for the proposed ORC-system. Therefore, the integration of additional heat sources, such as the jacket cooling water from the engine and lubricating oil, could reduce the variability of the input temperature and mass flow rates, thus extending the operation of ORC systems during the round trip and increasing the overall net power production. [15] The same researcher group has concluded, in their earlier study in 2015, that a regenerative ORC (rORC) system could cover up to 16% of on-board power demand, if the drop in auxiliary power needs is not taken into account for the same case ship.

When using waste heat for seawater desalination, thermal distillation methods, including multi-stage flash (MSF) and multiple effect distillation (MED), are commonly used to produce fresh water onboard ships. Waste heat can be also recovered using thermal energy storages, such as Viking Grace's water accumulator tanks [16], which could be used on a broader scale in the future.

2.2.4 Onboard energy management

Onboard energy management and automation can remarkably increase ship energy efficiency during operation. For example, efficient power management can produce 5% savings in SFOC by simply operating correct number of electricity generating sets. Ship speed reduction, namely slow steaming, is a very effective way to reduce energy consumption. Half a knot drop in speed could result in 7% drop in energy consumption. [2]

Automation and other ship monitoring systems can make ship operation more energy efficient and can deliver up to 10% fuel savings. [2] Several software providers deliver onboard energy management software for ship owners, such as ABB, Marorka, NAPA and Eniram. Ship energy efficiency can be increased, for instance, by monitoring ship and fleet performance and hence making correct decisions relating to ship operation and maintenance. One of the examples is ABB's EMMA™ software (now called OCTOPUS), which has been installed in many passenger ships including the cruise ferry Viking Grace. The EMMA system can assist in reducing fuel consumption and greenhouse gas emissions, which monitors and compares energy consumption by monitoring internal systems, vessel speed, harbour/dock visits, and outward circumstances. [17] The "Marorka Onboard" software allows monitoring machinery efficiency, fuel consumption, voyage planning and trim. [18] NAPA's energy management software, ClassNK-NAPA Green, has monitoring and optimization capabilities. [19] Finnish company Eniram has also management software for ships and their products have been installed widely on ships.

Maintenance also plays an important role in ship energy efficiency. Regular cleaning and scheduled maintenance ensure good ship operation. [1] For example, propeller surface polishing and hull cleaning can produce up to 10% energy savings. With the help of ship energy efficiency management systems, condition-based maintenance can produce up to 5% energy savings. [2]

There are also other operational measures to lower fuel consumption. For example, optimum trim can result in up to 5% energy savings. Crew operation awareness affects also other operations onboard ships. Ship weather routing develops an optimum track for safe and on-time arrival at destinations, based on the forecasts of weather and sea conditions (waves, currents…). The routes can be updated in real time to respond to possible changes in weather and sea conditions. [1]

2.3 Recent progress on passenger ship energy efficiency

2.3.1 Cruise lines' sustainability reports

As a general trend, passenger ships are getting more energy efficient and environmentally friendly. According to the reviewed sustainability reports from major cruise lines, fuel consumption has been gradually getting lower, at least in terms of fuel consumption per average passenger cruise day (APCD, the number of lower berths on a ship times the number of days that those berths are available to passengers per year). However, as passenger ship fleets are growing, the total fuel consumption of cruise ships is projected to rise and hence it is even more important to look into the practical progress that the world's major cruise liners have recently achieved and to have insights into how to improve passenger ships' energy efficiency and reduce their emissions in the near future. A brief overview of the recent progress of major cruise lines on passenger ship energy efficiency is given in the rest of this section. Please note that the values reported by the reviewed cruise lines are not necessarily comparable due to possible calculation differences, and therefore only cruise line-specific developments and values are presented and not compared with other cruise lines.

Carnival Corporation & PLC, the world's largest cruise company with over 10 subsidiary cruise lines (Carnival, P&O, Costa, etc.), is reported to have lowered total fuel consumption and greenhouse gasses emissions from 2012 to 2015. For example, the fuel consumption rate of the Holland America Lines had lowered by 3.2%, from 2012 to 2013, and total fuel consumption more than 11%, respectively. [20] AIDA cruises reported that their on-board energy consumption per person per day was 0,725 GJ in 2014, which was around 14% lower compared to year 2012. [21] For the whole Carnival Corporation, compared to year 2013, total fuel consumption in 2015 was around 2.6% lower (84,759 metric tonnes). [22]

Royal Caribbean Cruises Ltd., another leading global cruise company, also reported in 2010 to have managed to cut fuel consumption by 4.7% and greenhouse gas emissions by 5.5% per APCD on 2009 levels. According to another stewardship report in 2012, total fuel consumption has lowered 19% per APCD from 2005 level and they were committed to cut greenhouse gas emissions per APCD by one-third until 2015 compared to year 2005. In their 2013 sustainability report, annual energy consumption lowered 1.1% in one year, from 6 481 604 MWh in 2012 to 6 408 756 in 2013. [23]

Other cruise lines have followed the same trend as well. For example, TUI Cruises reported that their year-on-year fleet fuel consumption per passenger per overnight stay was reduced by 11.60% in 2014 [24] and 14.1% in 2015 [25], which were mainly due to introduction of the Mein Schiff 3 and 4 in 2014 and 2015, respectively. Viking Line's annual report showed their LNG consumption by Viking Grace lowered by around 3% in 2015. [26]

2.3.2 Passenger ship energy efficiency evolution (2000-2016)

As an effort to analyse the energy efficiency evolution of recent passenger ships, we introduced a new indicative term, on-board installed power per person (OIPPP, including both passengers (with maximum capacity) and crew). There were 57 passenger ships selected for the analysis, which were around one-third of the large cruise ships built after year 2000 and covered all the main ship classes from major cruise lines, if possible. Most of the ships' general information needed for the analysis was freely available from DNV vessel register. However, it was noticed that on-board installed power and passenger capacity of some ships could be different depending on the information source, which could not be verified from cruise lines or shipyards. Therefore, conclusions from this chapter were only indicative of possible trends in energy efficiency evolution.

The on-board installed power (OIP) referred to the total rated power of the prime movers on-board ships. Most large cruise ships investigated were equipped with diesel-electric power generation components, and Caterpillar, MAN Diesel & Turbo and Wärtsilä were the main engine providers. The rest were powered with GE LM2500+ aero derivative gas turbines, which were popular in the early 2000s, for example as main generators for Radiance- and Millennium-class ships and as auxiliary generators for some ships from Cunard Line, Holland America Line and Princess Cruises. [27]

From cruise lines' perspective, Carnival Corporation & plc had greater OIPPP for ships built before 2008 (years 2000, 2001 and 2006). The Carnival Spirit (built in 2001) from their Carnival Cruise Line had the OIPPP 17.1 kW/person, while their latest ship from their AIDA Cruises, "AIDAprima", had the lowest OIPPP value (10.29 kW/person), which had been marketed as quite innovative in terms of energy efficiency.

For Royal Caribbean Cruises Ltd., their turbine-powered ships built in the early 2000s, including all ships from the Millennium class and the Radiance class, had the OIPPP around 14–15 kW/person. The later Solstice-class cruise ships (built in 2008–2012) from Celebrity Cruises had lower OIPPP (13–14 kW/person) compared to the Millennium- and Radiance-class ships. The OIPPPs of their newest ships (built after 2013) of the Quantum-class from Royal Caribbean International, were below 10.5 kW/person.

The OIPPP of the Norwegian Cruise line's ships built after 2010 was significantly lower in comparison to ships built on 2000–2007, which is mainly due to the rise in passenger numbers. Their Norwegian Breakaway (2013) had the lowest OIPPP (11.1 kw/person). Starting from 2012, TUI Cruises started building three sister ships, Mein Schiff 3, 4 & 5, which were also the only ships they built after year 2000. According to DNV vessel register, these ships were almost technically identical except minor differences in manoeuvring thruster combinations. The Mein Schiff 4 & 5 had lower OIPPP (12.565 kW/person) than Mein Schiff 3 (13.71 kW/person) due to the increase in passenger capacity.

As an overall trend, the average OIPPPs of large cruise ships built after year 2005 tended to decrease with time although it was not definitive, as shown in Figure 2.1. Especially, newer ships built after year 2008 tended had much lower OIPPP in comparison to the older ones built in the early 2000s. This was partly due to the increased ship size and passenger capacity and the adoption of new energy saving technologies (such as air lubrication, waste heat recovery, etc.), although some specific ships did not necessarily followed exactly the same track.

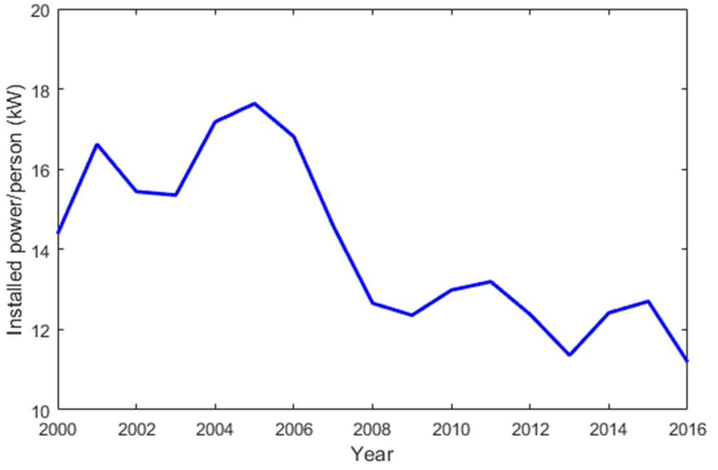

Figure 2.1. The average OIPPPs of large cruise ships built after year 2000.

2.4 Further options for passenger ship energy efficiency

Other than the above-mentioned energy-efficiency technologies and solutions, there are also other potential options already coming or to come to cruise shipping industries, such as renewable energy sources, hybridization and electrification.

Renewable energy sources are possible future solution for ship propulsion. For example, wind power is making a sort of comeback in terms of rotor sails and rigid sails. There are also proposed concepts for hybrids of sails and solar power. Recently, a patent was awarded for integrated ship energy system of rigid sails and solar panels, combined with energy storage modules, which possibly enables emission-free operations while ships are in port. [28] Another example is Norsepower Rotor Sail Solution, which uses Magnus effect to produce lift from side winds with rotating cylinders on deck. Although it is mainly designed for tankers, bulk carriers and ro-ro vessels, some pilot installation on cruise ferry Viking Grace has been released. Other possibilities are to use wind turbines to produce electricity, for example with vertical axis wind turbines. [1] However, using wind for propulsion of cruise ships is still challenging due to limited deck space required.

Battery systems are gaining interest because they can enable flexible operations on ships, for peak shaving or alternative main power sources, and due to IMO's regulations to reduce pollution in shipping. Some ferry operators have reported fuel savings of 20–50%, by switching to electric propulsion, which demonstrates that battery systems can improve environmental performance and achieve financial savings. [29] DC grid solutions, for more efficient energy distribution, have been tested currently on smaller ships but can be viable option for cruise ships in the future.

As LNG is increasing its share in the fuel markets, gas turbines could make a comeback with advanced waste heat recovery, because of lower emissions and also because they take less space with reduced vibrations and noise. Thermal efficiency for gas turbines can achieve 60% with waste heat recovery. [30] Besides, waste heat recovery seems to be currently focusing on the heat providing, but more focus will be on electricity providing in the future with technologies that can utilize lower grade waste heat.

Ports will be a more important part of future ship-n-harbour energy system integration. Ships could utilize “cold ironing” with lesser emissions while in ports. Some ports have been already integrated with power grids, but future ports could have energy production on their own from renewable sources, for example with wind and solar power. Nevertheless, using shore energy eliminates the need to run engines to produce electricity while on port. [3] Solar power system in the Port of Los Angeles was built in 2010, producing power of one MW and giving ships stationed between cruises an alternative power source to their diesel generators. ”According to port authorities, the system will save around $200,000 a year in electricity costs.” [31]

2.5 Discussion

Information on ship technology and properties in this study was collected from public sources, which did not provide detailed or exact information about ships and related energy efficiency technologies on ships. Efficiency of the entire ship systems could not be determined, but some technologies that increase energy efficiency was found. Detailed information was only available on very limited number of ships. Particularly, Scandinavian ships Birka Stockholm and Viking Grace were only ships on which information was relatively easy to find and in more detail.

Anyway, energy efficiency has been taken into account in the cruise shipping industry. New cruise ships have most likely latest machinery in terms of engines and propulsion. Green fuels, such as LNG and methanol, are gaining grounds and WHR technologies are getting more common now. However, more advanced WHR technologies and methods are needed in order to utilize lower grade waste heat on ships. Operational measures also play a huge role on energy consumption, but there is little information available on ship operator practices. Attention is also paid to other areas than just engines and propulsion. For example, many energy efficiency technologies are already in use in the accommodation and entertainment areas for air conditioning, lighting and other measures. Still a lot of potential energy seems to be wasted, and ships inspected cover only narrow area of cruising industry. Efficient designs and technologies may not be so common on all ships.

However, there is no doubt that passenger ships are, and will continue, getting more energy efficiently and more novel technologies and methods for energy efficient design and operation of ship energy systems are to be developed and introduced into the cruise shipping industry in the near future.

References

[1] American Bureau of Shipping, 2013. Ship Energy Efficiency Measures, Status and Guidance. [online] Available: http://ww2.eagle.org/en/news/pressroom-/2013/abs-releases-ship-energy-efficiency-measures-advisory.html
[2] Wärtsilä. Boosting energy efficiency. [online] Available: http://www. shipping-tech.it/PDF/convegni%202010/2tecnologie1/Baan.pdf
[3] Ship-technology.com. Sailing under a green flag. [online] Available: http://www .ship-technology.com/features/feature124994/
[4] Cruise & Ferry. Royal Caribbean perfects the art of sailing on air. [online] Available: http://www.cruiseandferry.net/articles/royal-caribbean-perfects-the-art-of-sailing-on-air
[5] Seatrade. Micro-bubble system helps Quantum to whopping 20% efficiency gain on Oasis. [online] Available : http://www.seatradecruise.com/news/ new s headlines/micro-bubble-system-helps-quantum-to-whopping-20-efficiency-gain-on-oasis.html#headlines
[6] Wärtsilä. Common-rail injection system. [online] Available: http://www.wartsila .com/encyclopedia/term/common-rail-injection-system
[7] Zahedi, B., Norum, L. & Ludvigsen, K. Optimized efficiency of all-electric ships by dc hybrid power systems. *Journal of Power Sources,* Volume 255, pp. 341-354.
[8] Mitshubishi Heavy Industries. Technical review. [online] Available: https:// www.mhi-lobal.com/company/technology/review/pdf/e416/e416338.pdf
[9] Royal Caribbean Cruises. Environment. [online] Available: http://www.rcl-corporate.com/environment/
[10] LEDs Magazine. LED lighting enables economical cruising for giant passenger ships (MAGAZINE). [online] Available: http://www.ledsmagazine.com/articles/ 2011/07/led-lighting-enables-economical-cruising-for-giant-passenger-ships-magazine.html
[11] Khanzada, N., Jamal Khan, S. & Davies, P. Performance evaluation of reverse osmosis (RO) pre-treatment technologies for in-land brackish water treatment. Desalination. Available: DOI: http://dx.doi.org/10.1016/j.desal.2016.06.030
[12] Meyer Turku. Efficient Design on Ferries. Hamburg, International Conference on Ship Efficiency. Available: http://www.ship-efficiency.org/onTEAM/pdf/ 2015/01_Efficient%20Design%20on%20Ferries_final.pdf
[13] Zheng, G., Zheng, C., Yang, G. & Chen, W. Development of a New Marine Rotary Desiccant Airconditioning. Energy Procedia, Volume 16, pp. 1095-1101.
[14] Shu, G. et al. A review of waste heat recovery on two-stroke IC engine aboard ships. *Renewable and Sustainable Energy Reviews,* Volume 19, pp. 385-401.
[15] Mondejar, M., Ahlgren, F., Thern, M. & Genrup, M. Quasi-steady state simulation of an organic Rankine cycle for waste heat recovery in a passenger vessel. Applied Energy. (Article in press).

[16] Lepistö, V., Lappalainen, J., Sillanpää, K. & Ahtila, P. Dynamic process simulation promotes energy efficient ship design. Ocean Engineering, Volume 111, pp. 43-55.
[17] Viking Line. Viking Grace. [online] Available: https://www.vikingline.fi/valitse-matka/laivat/ms-viking-grace/teknologia/
[18] Marorka. Products. [online] Available: http://www.marorka.com/products/marorka-online/
[19] NAPA. ClassNK-NAPA GREEN Optimization. [online] Available: https://www.napa.fi/Ship-Operations/ClassNK-NAPA-GREEN-Optimization
[20] Holland America Line. Energy. [online] Available: http://www.hollandamerica.com/pageByName/Simple.action?requestPage=Sustainability2013&himage=no
[21] AIDA Cruises. AIDA Cares 2015 summary. [online] Available: https://d1ozq1nmb5vv1n.cloudfront.net/fileadmin/user_upload/v4/Unternehmen/Nachhaltigkeit/Tabellen/2015/Tabellen_englisch/Broschuere_AIDA_cares_en_2015.pdf
[22] Carnival Corporation & PLC. Sustainability reports. [Online] Available: http://phx.corporate-ir.net/phoenix.zhtml?c=140690&p=irol-sustainability_env
[23] Royal Caribbean Cruises Ltd. Beyond the Horizon 2012 sustainablity report. [online] Available: http://www.celebritycruises.com/media/en_US/pdf/RCL_2013_Sustainability_Report.pdf
[24] TUI Cruises. ENVIRONMENTAL REPORT 2014. [online] Available: https://www.tuigroup.com/damfiles/default/tuigroup15/en/sustainability/Reporting/tui_cruises_environmental_report_2014-2.pdf-e5cd1303967f6ca3abe2b9-baa7e4e002.pdf
[25] TUI Cruises. UMWELTBERICHT 2015. [online] Available: https://tuicruises.com/ nachhaltigkeit/umweltbericht/
[26] Viking Line. Annual report 2015. [online] Available: http://www.vikingline.com/globalassets/documents/market_specific/corporate/investors/annual-reports/annual-report-2015.pdf
[27] GE Aviation. GE Marine Engines' Gas Turbine-Based System Powers Newly-Launched Brilliance Of The Seas Cruise Ship. [online] Available: http://www.geaviation.com/press/marine/marine_20020731.html
[28] Motorship. US patent for Eco Marine Power. [online] Available: http://www.motorship.com/news101/ships-equipment/us-patent-for-eco-marine-power
[29] World Cruise Industry review. Saft - The balance lies with Li-ion. [online] Available:http://www.worldcruiseindustryreview.com/contractors/world-cruise-industry-review/saft1/
[30] Klein, R. Cruise ship concepts applying LNG fuel. Masters thesis. Aalto University, Espoo, 2015.
[31] Worldcruise-network.com. Solar power: here comes the sun. [online] Available: http://www.worldcruise-network.com/features/featuresolar-power-here-comes-the-sun/

3. Ship waste heat recovery with thermal energy storage[1]

Emil Fridolfsson, Maunu Kuosa
Aalto University

Summary: Waste heat recovery has been playing a very important role in improving ship energy efficiency. One of the potential technologies is thermal energy storage, especially for balancing the heat generation and consumption onboard ships.

In this chapter, some brief introduction is given to the fundamentals of thermal energy storage system design. The focus is more on the latent heat storage systems due to its big advantage over sensible heat storage systems. Especially, the selection and configuration of the phase changing materials is the most critical part of the latent heat storage design, given the big differences in the thermo-physical, kinetic and chemical properties of vast amount of different PCMs.

As an example, we introduced a combined WHR scenario, with ORC and latent heat storage, which can well address the imbalance between the heat generation and consumption during ship operations, as the heat storage ensures more stable and continuous heat output to ORC. One case study was further given to show the potential and benefits of the combined scenario, with substantial fuel savings, in improving ship energy efficiency during operation.

3.1 Introduction

Utilizing the waste heat generated by the engines onboard ships clearly has potential. The difficulty is that the heat generation rarely matches the heat consumption, which is a problem that has to be faced in order to truly improve ship overall energy efficiency. One of the potential ways to handle the mismatch is using thermal energy storage (TES).

Thermal energy storage, also called heat storage, can be divided into sensible heat storage and latent heat storage (or a combination of both). The sensible heat storage is to store heat in a storage material by increasing its temperature; while

[1] This chapter is excerpted and revised from the SET research project deliverable named Ship engine heat recovery and suitable thermal storages.

the latent heat storage is to store heat in a storage material by changing its phase (temperature remains constant during the process). In the latter case, the heat storage material is often referred to as a phase change material (PCM), which allows heat transfer in a relatively constant temperature as heat liquefies the solid PCM. The liquefied PCM can later be used to extract heat for different types of applications, thus increasing overall system energy efficiency. Latent heat storage is more attractive due to its 5-to-10-times larger storage density compared to sensible heat storages [1].

In addition, latent heat storage allows heat transfer at a constant temperature (corresponding to the phase change temperature), which improves heat transfer and simplifies the process. As one example shown in Figure 3.1, the PCM experiences 3 phase changes. The first phase change (B-C) is a solid-solid phase change in which the material's crystalline structure is changed. The second phase change (D-E) is a solid-liquid phase change and the third one (F-G) is a liquid-gas phase change. Although any of the different phase change zones could be utilized theoretically, the solid-liquid phase change is the only practical one. The solid-solid phase change has a small latent heat, reducing its storage potential. The liquid-gas phase change has a larger latent heat, but it's problematic as gases generally occupy a much larger volume than liquids. The solid-liquid phase change offers a high heat storage capacity (reduced system size) and heat transfer a constant temperature. [2]

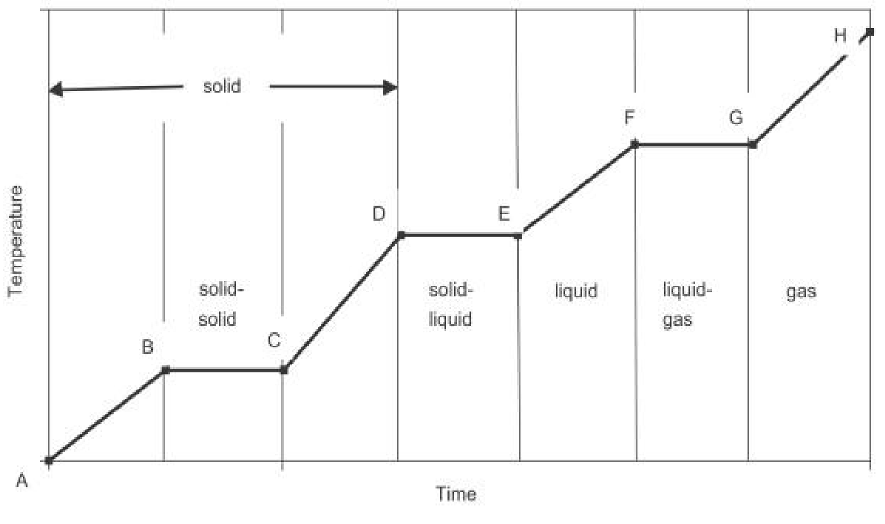

Figure 3.1. The temperature dependency of a material being heated up. [2]

3.2 Configuration

A latent heat storage system typically has four main components, a suitable PCM, a PCM container, a heat transfer surface and a heat transfer medium. [2] It's important to note that the PCM itself cannot serve as a heat transfer fluid (HTF) since its phase changing would cause some flow problems [1]. Thus some other fluid with a high sensible heat, a high thermal conductivity and a low melting point has to be used as the HTF.

In addition, there is not an ideal PCM existing due to an extensive amount of thermo-physical, kinetic and chemical requirements set on the PCM. There are three main categories of PCM heat storage configurations, bulk storage, macro-encapsulation and micro-encapsulation. All have different advantages and disadvantages.

3.2.1 Bulk storage

The bulk storage is similar to normal tank heat exchangers. The HTF flows in pipes through a large container filled with PCM. This is clearly a really simple solution, which comes at a cost. The problem with this type of solutions is poor heat transfer, due to a small heat transfer area. Considering that the latent heat of PCM’s are typically high, different heat transfer improvements are commonly implemented, such as attaching fins on the piping surface, using different metal structures, mixing high conductivity particles to the HTF, inserting fibers on the PCM side and the rolling cylinder method (rotating heat storage to induce stirring and thus enhancing heat transfer). [2]

3.2.2 Macroencapsulation

Macroencapsulation stores a large quantity of PCM capsules inside a larger tank. This allows the HTF to flow through the voids surrounding the capsules, thus significantly increasing the heat transfer rate compared to the bulk storage. The shape of the capsules can be rectangular, spherical or more or less any shape. The capsule size ranges from grams to kilograms of PCM contained. The important part of this type of storage is to find a proper capsule shape and size, which fits well for the specific application. Another benefit of macroencapsulation is its smaller capsule size compared to the bulk storage. This decreases the phase separation as the PCM phase varies according to the capsules and drastically decreases the heat transfer rate (e.g. when extracting heat, the PCM solidifies on the heat transfer surface, thus reducing total heat transfer). In addition, a storage design made of several smaller containers improves the mechanical strength of the heat exchanger. As shown in Figure 3.2, a possible macroencapsulation configuration consists of small spherical capsules filled with PCM. The HTF flows through the voids surrounding the capsules. [2]

Macroencapsulation is the most common storage configuration type. It can be used for a large variety of different applications if the capsule geometry, capsule material and PCM are properly chosen. This is also the most viable option for storing waste heat produced by ship engines. [2]

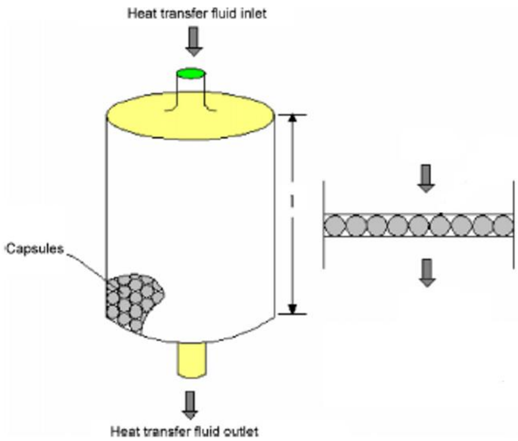

Figure 3.2. A macroencapsulation configuration with spherical capsules filled with PCM. [2]

3.2.3 Microencapsulation

Microencapsulation is similar to macroencapsulation, except that the capsule size is significantly smaller, typically on a micrometer scale, as shown in Figure 3.3. The capsule size and form are slightly random, but the most possible shape is naturally spherical. These extremely small capsules bring certain benefits and drawbacks. The smaller capsule size diminishes the heat transfer effect arisen in the macroencapsulation storage. On the other hand, manufacturing microscale capsules is more challenging in comparison to macro scale capsules. The microencapsulated PCM is typically mixed into different matrices (e.g. building blocks). This is often handy and can be exploited in several different applications. The downside is that the heat transfer is significantly reduced, as there is no convective heat transfer on the surface of the PCM capsules. [2][3]

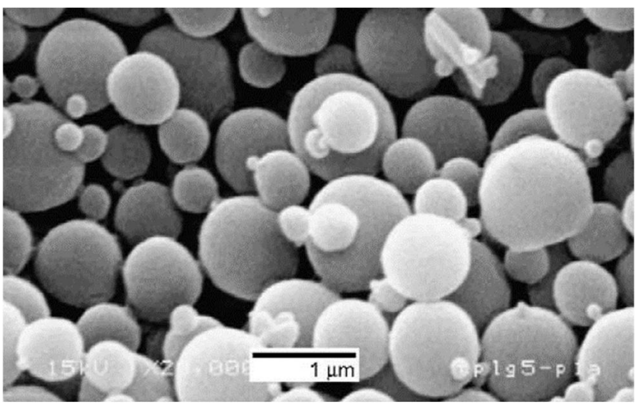

Figure 3.3. A close-look image of microencapsulated PCM. [4]

3.3 Phase change materials

The PCM is the most vital part of the latent heat storage system as it's where heat is stored. Hence, the PCM selection is key in designing latent heat storage. There's a vast amount of possible PCMs to consider if only the melting point is considered. However, in practice, several other equally important characteristics have to be taken into account when choosing the PCM. These characteristics can be divided into three main categories: thermo-physical, kinetic and chemical properties. In addition to these properties, it's important that the PCM is cheap and with large supply capacity. The different properties, which should be taken into consideration when choosing the PCM, are listed below:

- **Thermo-physical properties:**
  - Desired operating temperature (i.e. melting point)
  - High latent heat (i.e. high energy storage density)
  - High sensible heat to provide additional heat storage capacity
  - High thermal conductivity in both solid and liquid phases (i.e. improved heat transfer)
  - Small volume change in solid-liquid phase change, as a large volume change requires a more complex configuration to avoid extreme pressure condition which may harm the system
  - Small vapor pressure at operating pressures to reduce the containment problem (i.e. a small vapor pressure only cause small amounts of PCM to vaporize, which is beneficial as the vaporized PCM easily escapes the PCM containers)
  - Congruent melting to ensure constant storage capacity of each freezing/melting cycle
- **Kinetic properties:**
  - High nucleation rate to avoid super cooling in liquid phase
  - High rate if crystal growth, to ensure that the system can meet the heat recovery demand from the storage system
- **Chemical properties:**
  - Chemical stability
  - Reversible freeze / melt cycles
  - No degradation after a large number of freeze / melt cycles
  - Non-corrosive to the storage container

An ideal PCM is impossible to find because of the large amount of requirements set on the PCMs, which severely cut down the amount of possible PCMs. The potential PCMs can be divided into three main categories: organic, inorganic and eutectics, as illustrated in Figure 3.4. Due to the vast amount of PCMs, only the most common and promising PCMs are further discussed in detail.

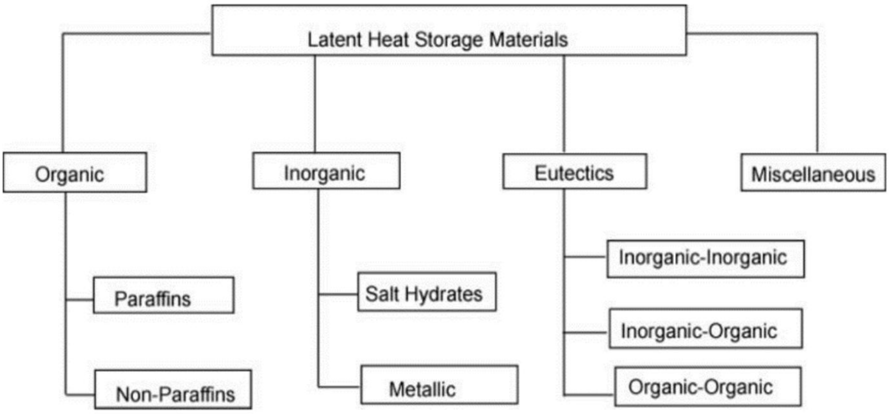

Figure 3.4. Categorization of phase change materials. [1]

3.3.1 Paraffins

Paraffins are a family of saturated hydrocarbons, generally of the form C_nH_{2n+2}, which commercially are the most commonly used heat storage PCMs. Paraffins with more than fifteen carbon atoms are generally waxy solids (i.e. smaller ones are liquids). The melting point of these waxy solids ranges from 23 to 67˚C. They are obtained from petroleum distillation and are thus available from many manufacturers. [1]

The advantages of paraffins are that they do not have the tendency to segregate and are chemically stable. In addition, their thermal properties remain good even after a large number of melting and freezing cycles. Paraffins also have high latent heat. However, due to oxidation, paraffins have to be stored in closed containers. They are safe and non-reactive, and can be stored in most metal containers but not in plastic containers as paraffins may soften the plastic walls over time. The biggest disadvantage of paraffins is their low thermal conductivity in the solid phase, which consequently requires high heat transfer rates during the freezing cycle. This, however, can be obtained by using different fins or aluminium honeycombs. A volumetric expansion rate when melting is common for all PCMs and is especially large for paraffins, which causes many challenges to container design. Paraffins are flammable and more expensive than salt hydrates. [1]

3.3.2 Organic non-paraffins

The organic non-paraffins category is the biggest PCM category. The materials suitable for energy storage in this category are, for instance, esters, fatty acids, alcohols and glycols. Such a large group of materials with different chemical structures bring a large variation in properties. The organic non-paraffins are all flammable and should not be exposed to either really high temperatures or oxidizing agents. The fatty acids subcategory, with properties similar to paraffins, is one of the most promising among all the mentioned subcategories. In comparison to paraffins, the benefit with fatty acids is their much sharper phase transformation. In other words, the phase change goes straight from solid to liquid, without going through a so called "soft solid" (i.e. wax-like) phase. The downside of fatty acids is the price, typically two or three times higher compared to paraffins. Fatty acids are also slightly corrosive, which have to be taken into account when designing the PCM containers. [1]

3.3.3 Salt hydrates

Salt hydrates include some type of salt which is either completely or partially dissolved in water. The salt water mixture creates a crystalline matrix when it solidifies. The melting point of these salt hydrates range from 15 to 117 ˚C. Salt hydrates are the oldest heat storage PCM and have thus been extensively studied for a long period of time. Salt hydrates can be used either alone or in eutectic mixtures (mix of several PCMs, all with same melting point). Salt hydrates can be divided into three groups depending on their melting behavior: congruent, incongruent and semi-congruent melting. Congruent melting occurs when the salt has totally dissolved in the water and therefore there's one specific temperature at which the solution melts. Incongruent melting occurs when the water is saturated at its melting point before all the salt has been dissolved. Thus two different melting points exist, one for the salt water mixture and the other for the undissolved salt. Semi-congruent melting occurs when a material melts and forms a new liquid and a new solid (i.e. two new materials), which combine into one single form, if further cooled [1] [5].

The main advantages of salt hydrates are the abundant availability and low price of the materials. They also have sharp melting points and high thermal conductivity (compared to other PCMs), allowing increased heat transfer when charging and discharging. The high latent heat of salt hydrates and the low volume change reduce the system size and hence simplify the container design.

The biggest problem with salt hydrates is segregation. In other words, salt hydrates tend to form other hydrates or dehydrated salts with lower latent heats. The heat storage capacity is thus heavily compromised. It has for instance been shown that the overall latent heat of $Na_2SO_4.10H_2O$ is reduced by even up to 73% after 1000 melting/freezing cycles. This problem can be avoided to some extent by using gelled or thickened mixtures, although this has a negative influence on the heat transfer characteristics and the material will degrade over time anyway. Thus salt hydrates might have to be replaced every once for a while (considering them being cheap and abundantly available). A hermetically sealed (i.e. watertight) system has been shown to be crucial, as otherwise degradation might occur just after a few melt/freeze cycles. Salt hydrates also tend to super-cool, which however, can be avoided by using a proper nucleating material to start the crystal growth. The compatibility of the storage container and the salt hydrate has to be verified, as salt hydrates have a tendency to cause corrosion (to the most common metals used in container walls). [1]

3.3.4 Eutectics

A eutectic is a mixture of two or more materials which freeze or melt congruently. When frozen, they form a mixture of component crystals. Segregation is uncommon, as eutectics typically freeze to an inmate mixture of crystals (i.e. leaving only a small possibility for components to separate). All the components liquefy simultaneously, thus rarely having the possibility to segregate. [1]

An advantage with eutectics is that the melting point can easily be adjusted by changing the ratios of the different materials of eutectics. In addition, it has been shown that their thermal characteristics do not degrade over time (i.e. over multiple melting/freeze cycles). [6]

3.4 Case studies

There are several possible ways to utilize heat storage combined with other technologies onboard ships to improve overall system energy efficiency, such as organic Rankine cycles (ORCs) and various HT/LT water applications. Especially, ORCs clearly have potential because both the surplus steam and high temperature water could be used as heat sources. A promising concept would be to store the engine waste heat into a latent heat storage and then use the heat storage to power the ORC. This would address the imbalance of heat production and consumption and give flexibility and adjustability to the power production of the ORC, due to the fact that the ORC power production would no longer be dependent on the current engine waste heat recovery (i.e. a more or less constant ORC heat input would be achieved when the heat storage works as the heat source).

In the rest of the section, we will carry out a small-scale study on the combined WHR scenario (ORC and heat storage) and compare the potential of different latent heat storage. The study is made on one case ship listed in Chapter 1, MS Oasis of the Seas. The main idea is here to give the reader a concrete example of the physical size and potential of heat storage combined with an ORC onboard these two different ship types.

3.4.1 Case ship operating profiles

The real performance of ship energy systems largely depends on the operating profiles, which vary quite a lot depending on the ship type. Below, we give typical operating profiles of one case ships, MS Oasis of the Seas, and further assess the surplus heat that could be potentially utilized for the combined WHR scenario.

This data used here is based on the summer (2016) timetables of the case ship, which, however, are supposed to work as an example only. The cruiser is assumed to sail for 38 hours followed by 11 hours in port. The velocity profile (i.e. share of velocities) used is illustrated in Figure 3.5. [7]

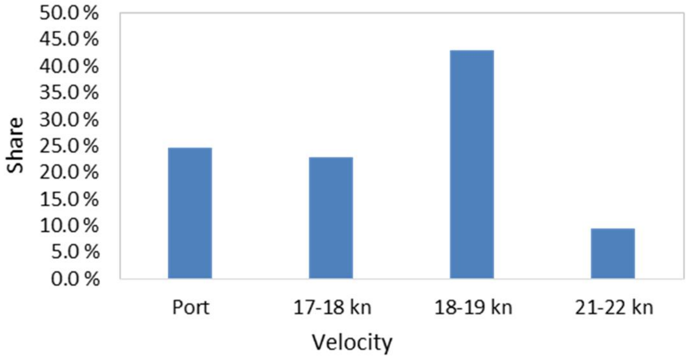

Figure 3.5. The estimated velocity profile for the cruiser. [7]

The heat generation and consumption is based on data found in [7]. The only difference from the original data is that the potable water is generated using reverse osmosis (i.e. no heat need), which seems to be the future trend. Figure 3.6–8 show the heat generation, consumption and surplus heat as a function of velocity.

It can be seen in Figure 3.6 that the heat production increases more or less linearly as the velocity increases. Heat generation exists during port time as well, as some auxiliary generator is running to meet the electricity demand in port. As shown in Figure 3.7, the unusually high demand in HT water during port time is mainly because the increased heat need for air conditioning and potable water generation. The use of steam is significantly higher when sailing mainly due to preheating of the heavy fuel oil. Galleys and laundries are other applications that use steam as their heat source. The use LT water remains constant at all velocities, which is mostly used to heat up swimming pools. The surplus heat shown in Figure 3.8 is simply calculated as the difference between the heat generation and consumption. In this case, the surplus heat remains positive at all times and hence always available for some additional applications.

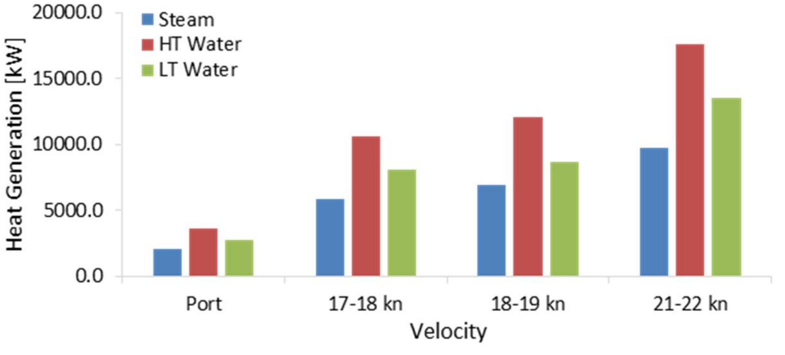

Figure 3.6. The total heat generation onboard the cruiser.

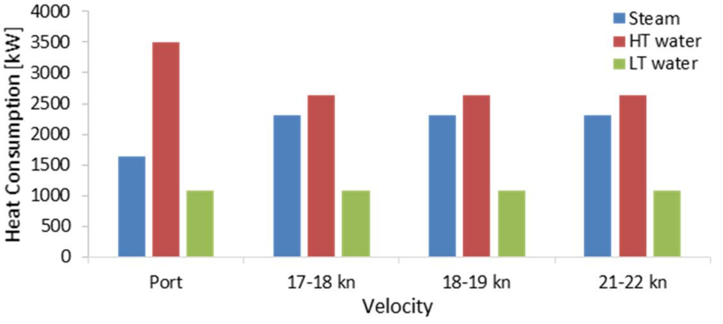

Figure 3.7. The total heat consumption onboard the cruiser.

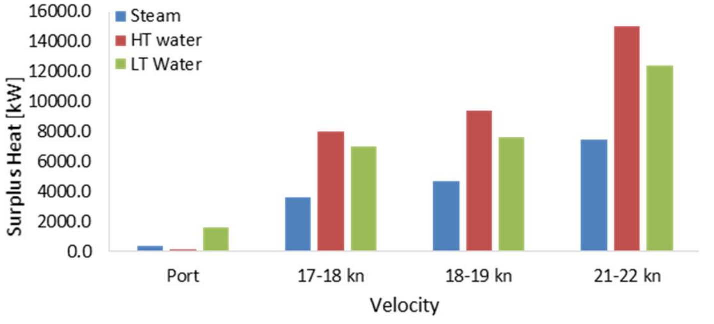

Figure 3.8. The total surplus heat onboard the cruiser.

3.4.2 The combined WHR scenario

As mentioned earlier, the focus of the case studies is on the WHR scenario combined ORC and heat storage, which could potentially better utilize the waste heat onboard ships. Both the ORCs and heat storage are potential technologies for low-grade waste heat recovery. Specifically, different from traditional Rankine cycle (RC), the ORCs use some organic fluid, which allows evaporations at much lower temperatures and hence lowers the temperature requirement on heat source. The combined scenario would better address the imbalance of the heat production and consumption onboard ships and improve the overall system energy efficiency, as the heat storage ensures more stable and continuous heat output to ORC.

Below are some initial assumptions adopted in this section.

- The ORCs use the heat from latent heat storage for heating its working fluid.
- The boiling point of the ORC working fluid is assumed approximately 100 ˚C, but no further considerations were made regarding the working fluid.
- Based on the selected capacity of the ORC and heat storage, the excess heat is stored and used by the ORCs without heat losses.
- The ORC efficiency adopted is 10%, which is at middle range for low-grade heat recovery applications.
- When calculating fuel savings, the SFOC adopted is 0.21 kg / kWh.

3.4.2.1 MS Oasis of the Seas

Figure 3.9 shows the total surplus heat remaining after one typical trip with the earlier-mentioned profiles (i.e. sailing for 38 hours and resting 11 hours in port) as a function of the ORC power output. The maximum ORC capacity that allows continuous production is found when all the excess heat is used after the trip (i.e. when the surplus heat reaches zero in the figure). In this case, the nominal ORC capacity is 1095 kW. In practice, the real nominal power will be slightly lower due to the heat losses. Hence, the fuel savings are expected to be approximately 230 kg/h.

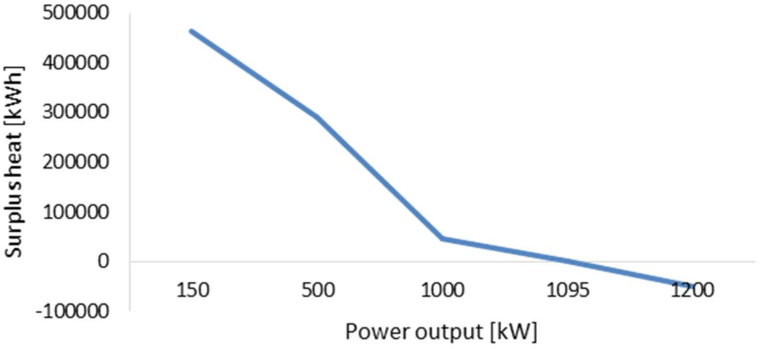

Figure 3.9. Total surplus heat after the ORC use as a function of power output.

3.4.3 Latent heat storage comparison

In the last section, we assume all the surplus heat needed for ORCs is stored but the selection of heat storage and the calculation of its capacity remains untouched. In this chapter, we will further select, calculate and compare the capacity of different latent heat storage based on the heat demand for the dimensioned ORC of these two case studies. Below are some initial assumptions adopted in the calculations.

- The heat storage is scaled to be able to store all the surplus heat, even in the case where all the heat generation during a trip is done consecutively (i.e. no heat storage discharge occurring). This assumption is in most cases justified, as the larger velocities (i.e. heat generating velocities) occur when cruising on open waters and the smaller velocities occur close to port.
- The HT water and surplus steam work as HTFs when charging the heat storage. This is considered valid as it is already partly in use in existing installations.
- When transferring heat from the latent heat storage to the ORC, the needed HTF is not taken into account in this study, despite the fact that the HTF can work as sensible heat storage and thus reduce the amount of PCM needed to satisfy the planned storage capacity.
- The heat stream out of the ORC condenser is led straight into the ocean, even though it could be potential to lead it back into heat storage or to use it

in some other applications (e.g. LT water). This is a subject to be further studied and verified.

- The PCM only works as latent heat storage, where a saturated solid goes through the phase change into a saturated liquid (i.e. zero sensible heat is stored).

The heat storage is dimensioned based on the theoretical maximum heat storage capacity needed. The theoretical maximum is mathematically the integration of all the (positive) surplus heat collectable at a certain velocity over the operating hours during the whole trip.

The pre-selected PCMs to be evaluated in the studies are listed in Table 3.1, including their thermal properties such as their melting point, density and latent heat. RT90 and RT100 are commercially available organic PCMs sold by Rubitherm Technologies GmbH. E117 is a commercially available eutectic mixture provided by Environmental Process Systems Ltd.

Table 3.1. The different phase change materials compared and their thermal properties. [1]

PCM	Category	Melting point [˚C]	Density [kg/m^3]	Latent heat [kJ/kg]
Acetamide	Organic non-paraffin	81	1159	241
Alpha Napthol	Organic non-paraffin	96	1095	163
Glautaric acid	Organic non-paraffin	97,5	1429	156
Dimethyl Fumarate	Organic non-paraffin	102	1045	242
Catechol	Organic non-paraffin	104,3	1370	207
Quinone	Organic non-paraffin	115	1318	171
$MgCl_2.6H_2O$	Salt hydrate	116	1450	167
RT90	Commercially available	90	930	194
RT100	Commercially available	99	940	168
E117	Commercially available	117	1450	169

For each case of the studies, water is used as a sensible heat storage, and its capacity needed for the cases is further calculated and utilized as a baseline when comparing different latent heat storage. Hence, for the comparison purpose, the water properties are listed in Table 3.2, including its density, sensible heat and the operational temperature range for the sensible water heat storage. The temperature difference is the total temperature difference within the temperature range, which is used to calculate the heat stored in the sensible heat water storage.

Table 3.2. The properties of water at 85˚C. [8]

Heat storage material	Temperature range	Temperature difference [˚C]	Density [kg/m3]	Sensible heat [kJ/kgK]
Water	75-95 ˚C	20	968,6	4,1997

3.4.3.1 MS Oasis of the Seas

According to the initial calculation, for the cruise ship, the heat storage is being discharged only in port and being charged for the rest of the velocity points listed in Figure 3.5. Hence, the total heat storage capacity was calculated to be approximately 454 MJ. Table 3.3 presents the mass and volume of needed for heat storage of different PCMs. The mass and volume comparison reveals that a latent heat storage is significantly smaller than a sensible heat water storage. The mass of PCMs varies between 1870–2910 tons and the volume between 1600–2880 m^3. A significant difference can thus be made by choosing a proper PCM. However, it is also important to validate that other properties of the PCM comply with the requirements set on the PCM.

According to the calculation, Acetamide and Dimethyl Fumarate seem to be good PCMs for heat storage, based on the smaller mass and volume required in comparison to other PCMs. Acetamide can be recognized as C2H5NO and Dimethyl Fumarate as C6H8O4 [9][10]. It is important to keep in mind that the melting point of Acetamide is 81 ˚C, which is relatively low. Thus Dimethyl Fumarate, with a melting point of 102 ˚C, is recommended in this scenario.

Table 3.3. The PCM mass and volume needed to store 454 MJ of heat, as well as a comparison made to the sensible heat water storage.

Phase change material	Mass [tons]	Volume [m^3]	Mass reduced compared to Water	Volume reduced compared to Water
Acetamide	1884.5	1626	34.9%	29.1%
Alpha napthol	2786.4	2544.6	51.5%	45.6%
Glautaric acid	2911.4	2037.4	53.8%	36.5%
Dimethyl Fumarate	1876.8	1795.9	34.7%	32.2%
Catechol	2194.1	1601.5	40.6%	28.7%
Quinone	2656	2015.2	49.1%	36.1%
$MgCl_2.6H_2O$	2719.6	1875.6	50.3%	33.6%
RT90	2341.1	2517.3	43.3%	45.1%
RT100	2703.4	2876	50%	51.5%
E117	2687.4	1853.4	49.7%	33.2%
Water	5407.2	5582.3	1	1

3.5 Conclusions

Thermal energy storage is a potential technology for onboard ship WHR and hence for improving ship overall energy efficiency, especially when combined properly with other promising technologies onboard ships, such as organic Rankine cycles (ORCs). Especially, the latent heat storage is more promising due to its 5–10 times larger storage capacity in comparison to sensible heat storage.

Phase change materials (PCMs), as the heat storage media, are the vital part of the latent heat storage system. The selection and configuration of the PCMs is the most critical part of the latent heat storage design, given that there are big differences in the thermo-physical, kinetic and chemical properties of vast amount of different PCMs. Especially, a proper melting point of PCM is the key requirement and has to be primarily considered when selecting PCMs. Macroencapsulated PCMs are the most viable storage configuration for onboard ship WHR.

Both the ORCs and heat storage are potential technologies for low-grade waste heat recovery. The combined WHR scenario, with ORC and latent heat storage, can well address the imbalance between the heat generation and consumption during ship operations, as the heat storage ensures more stable and continuous heat output to ORC. In the two case studies on a cruise ship and a cruiseferry, the ORC capacities are 1095 kW for the cruiser and 770 kW for the cruiseferry, respectively. Dimethyl Fumarate, with a melting point of 102 ˚C, was recommended as a heat storage PCM for the cases, and the corresponding masses of the heat storage are around 1870 tons for the cruiser and 250 tons for the cruiseferry respectively. The fuel savings from the combined WHR scenario are approximately 230 kg/h for the cruiser and 162 kg/h for the cruiseferry during summertime (reduced to 102 kg/h during wintertime) respectively.

Heat storage is a potential application on board ships, but there are also some challenges to be further deal with next before make it more mature for marine applications. For example, accurately modelling heat transfer of heat storage systems using PCMs is especially difficult due to the complex geometry of the storage system and the PCM phase change when heat is either extracted or absorbed. Related to marine applications specifically, the main challenge is to integrate the heat storage systems into the whole energy system and hence to design and operate the whole energy systems more efficiently under dynamic operating conditions.

References

[1] Sharma S D, Sagara K. Latent Heat Storage Materials and Systems: A Review. International Journal of Green Energy, 2:1, 1-56, 2005. DOI: 10.1081/GE-200051299

[2] Regin A F, Solanki S C, Saini J S. Heat Transfer Characteristics of Thermal Energy Storage System Using PCM Capsules: A Review. Roorke, India, Indian Institute of Technology Roorke, 2007.

[3] Tyagi V V, Buddhi D. PCM Thermal Storage in Buildings: A State of Art. Indore, India, Devi Ahilya University, 2015.

[4] Hawlader M N A, Uddin M S, Khin M M. Microencapsulated PCM Thermal-energy Storage System. Singapore, Singapore, National University of Singapore, 2003.

[5] Mehling H, Cabeza L F. Heat and Cold Storage with PCM: An up to date introduction into basics and applications. 2008. ISBN: 978-3.540.68556.2.

[6] Sari A, Kaygusuz K. Thermal Performance of a Eutectic Mixture of Lauric and Steatic acids as PCM Encapsulation in the Annulus of Two Concentric Pipes. Tokat, Turkey, Gaziosmanpaşa University, 2002. DOI: 10.1016/S0038-092X(02)00026-9.

[7] Elg M. Analyzing the Energy Balance and Optimizing the Waste Heat Utilization on a Passenger Cruise Ship. Helsinki, Finland, Helsinki University of Technology, 2007.

[8] Kotiaho V W, Lampinen M J, Seppälä A. Termodynamiikan ja lämmönsiirto-opin taulukoita, Raportti 136. Espoo, Finland, Teknillinen Korkeakoulu, 2004. ISBN: 951-22-7259-8.

[9] ChemSpider: Search and Share Chemistry. Available: www.chemspider.com /Chemical-Structure.173.html (Referred 8.5.2016).

[10] PubChem: Open Chemistry Database. Available: pubchem.ncbi.nlm.nih.gov/ compound/Dimethyl_fumarate#section=Top (Referred 8.5.2016).

.

4. Ship waste heat recovery with absorption refrigeration[1]

Waltteri Salmi, Maunu Kuosa
Aalto University

Summary: This chapter briefly investigated the potential of waste heat recovery using absorption refrigeration technologies. It presented a steady-state model of absorption refrigeration cycles with water-LiBr and ammonia-water working fluid pairs for ship application purposes. Coefficient of performance was studied with different generator and evaporator temperatures in ISO and Tropical conditions.

Different energy sources, including exhaust gas, jacket water and scavenge air, were examined on their potential for two mentioned absorption refrigeration cycles. Optimal generator temperatures for different refrigerant temperatures were found for absorption refrigeration systems using different waste heat sources. All of these values were used to estimate the cooling power and energy production possibilities in a case ship. Theoretical possibilities to save 70% of compression electricity usage in ISO conditions and 61% in tropical conditions were found. The estimated annual fuel savings were 46.7 and 95.0 tons, respectively. Moreover, WHR from the engine jacket water with water-LiBr absorption refrigeration system had potential to provide 2.2-4 times more cooling power than needed during the sea operations in ISO conditions, depending on the main engine load.

4.1 Introduction

Chilling processes are necessary for different applications onboard ships, such as air-conditioning, ice-making and medical or food preservation. [1] They are usually electricity-powered, which hence increases the electric load of ship energy systems. Meanwhile, large amount of heat energy is wasted during the power generation process. In this chapter, we are investigating the potential of using waste heat for refrigeration with absorption refrigeration technology so as to improve ship overall energy efficiency.

[1] This chapter is excerpted and revised from the SET research project deliverable named "Using waste heat of ships as energy source for an absorption refrigeration system".

Absorption refrigeration technology was discovered by Nairn in 1777 and the first commercial application was built in 1858 by Ferdinand Carré. Refrigeration system using lithium bromide water was commercialized in the 1940s and used in 1950s as chillers for air-conditioning in large buildings. [2] Since then, due to its ability of using heat energy to drive the cooling process, it has been gradually adopted in many industrial and household applications. Below are two recent examples specifically related to its applications in ships.

4.1.1 Notable examples

4.1.1.1 Cargo ship

Cao and his colleagues from University of Maryland have recently conducted a simulated investigation of waste heat powered absorption cooling system onboard a cargo ship. As shown in Figure 4.1, the refrigeration system uses the waste heat energy recovered from engine exhaust gas via exhaust gas economizers (EGEs) for cabin air-cooling. A simulation model was developed and evaluated under different weather conditions. Under Miami conditions, the COP was 0.64 COP for the absorption refrigeration cycle and 0.6 for the whole system. The fuel consumption was 62% less compared to vapour compression. When the simulation was done under Abu Dhabi conditions (warmer), the fuel consumption increased up to 68%; while under Baltimore conditions (colder), it decreased to 38%. [3]

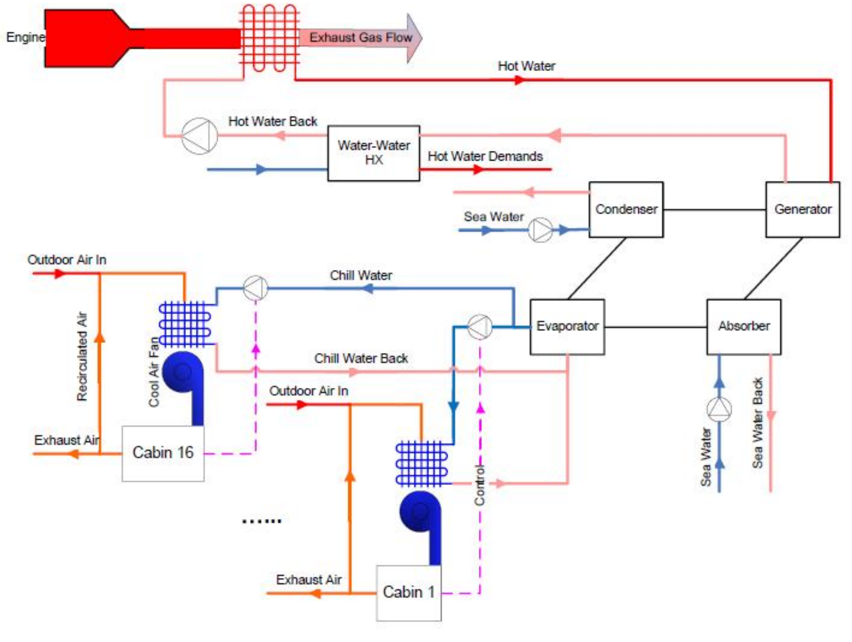

Figure 4.1. A waste heat powered absorption-cooling system for a cargo ship. [3]

4.1.1.2 Naval ship

The second example, for a naval ship application, presented a single effect water/Lithium Bromide absorption and a subcritical CO_2 vapour compression coupled cycle, as shown in Figure 4.2. The cascaded absorption-vapour compression used the exhaust heat (175–275 °C) of the onboard gas turbine to power the generator. The absorption refrigeration cycle is used to meet the medium-temperature cooling (5 °C) need for electronics and air-conditioning; while the subcritical CO_2 cycle is used to meet the high-load (megawatt scale) cooling need for advanced naval electronics. The COP of the individual absorption cycle was estimated to be 0.7803 and for the combined total-energy-input based COP it was estimated to be 0.594. Electrical COP was estimated to be 5.698. The cascaded absorption-vapour compression cycle was estimated to save up to 31% of the electricity consumption in the case ship studied, compared to an equivalent two-stage vapour-compression system. [4]

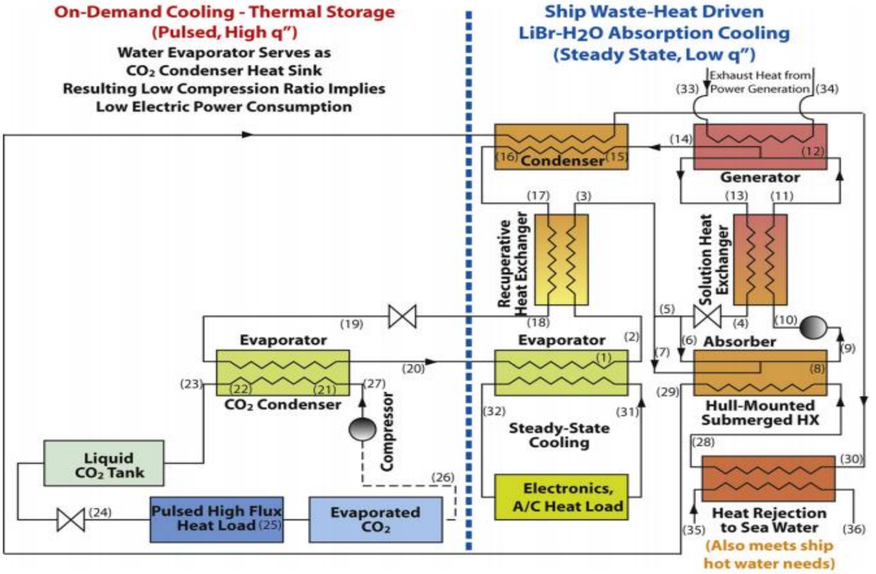

Figure 4.2. A single effect absorption refrigeration and CO2 vapour compression coupled cycle. [4]

4.2 Operation principles of absorption refrigeration

For both absorption and compressor refrigeration, a refrigerant with a low boiling point is used to take some heat away (when it evaporates), hence providing the cooling effect. Instead of forced circulation in the compressor refrigeration systems, the operation of absorption refrigeration systems is driven by heat, which causes the pressure difference due to the boiling point difference between a refrigerant and the solution. It can be described in three phases, evaporation (where the liquid evaporates in a vacuum tank when heat is received), absorption (where the vapour is passed and absorbed in another solution tank) and regeneration (where the vapour condenses, by extracting heat, and recirculates to supply the evaporation phase). Absorption refrigeration systems use regenerative cycles to maintain the pressure difference and the cooling effect of the evaporator. [5]

4.2.1 Single-effect water/Lithium Bromide refrigeration cycle

The simple and commonly used absorption refrigeration systems are of a single-effect water-LiBr (Lithium Bromide) cycle, where water is the refrigerant and LiBr is the absorbent. As described in Figure 4.3, when heat is taken away from the absorber where the latent heat in water vapour is absorbed into the solution, the thermodynamic equilibrium is broken and consequently pure water vapour flows in from the evaporator. To maintain the process, the weak water-LiBr solution in absorber is pumped through a recuperative heat exchanger (to improve the system performance) to the generator where high temperature heat is added to separate pure water from the solution. The evaporated water vapour is condensed back to liquid state in the condenser, while the strong solution of Water-LiBr in the generator is led back to the absorber (through expansion valve) to maintain the LiBr concentration constant. After the condenser, liquid water flows through expansion valve back to the evaporator, where low-grade heat is added to evaporate water (refrigeration effect). [6]

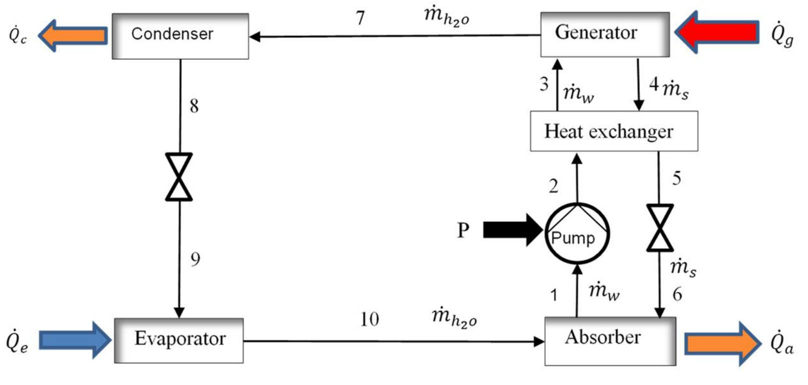

Figure 4.3. A single effect water-LiBr absorption refrigeration cycle.

4.2.2 Refined ammonia-water refrigeration cycle

The principle of an ammonia-water absorption refrigeration cycle is shown in Figure 4.4. In this case, water is the absorbent and ammonia the refrigerant, which enables the cooling process to reach below zero temperatures in the evaporator. The process cycle is quite similar to the water-LiBr cycle described previously. The difference is that the ammonia-water refrigeration cycle needs a rectification part after the generator to condensate water existing in the ammonia vapour after the generator. The mixture in the absorber is pumped through the rectifier to condensate water in the vapour, which is led back to the generator. [6]

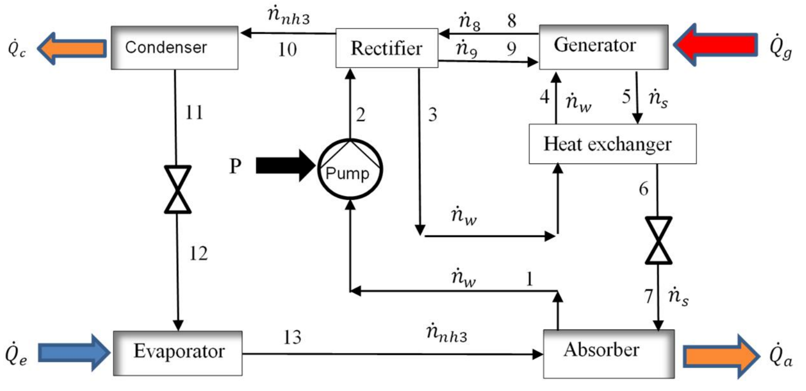

Figure 4.4. A schematic view of an ammonia-water absorption refrigeration cycle.

4.3 Process modelling of the absorption refrigeration

4.3.1 Process modelling

For the ease of the process modelling, some initial assumptions have been adopted in this section:

- the refrigeration cycle is of a steady state,
- the temperatures at different components are uniform,
- all pumps and expansion valves are adiabatic,
- there is no heat loss to surroundings,
- there is no pressure loss through pipes,
- the refrigerant is saturated vapour after the evaporator and saturated liquid after the condenser, and
- for ammonia-water cycle, all water is assumed to condense at the rectifier.

The pressure levels of the system are calculated using the function,

$$p = 10^5 exp\left(\frac{11.78(T - 372.79)}{(T - 43.15)}\right) \tag{4.1}$$

Where T (K) is the temperature of the condenser or evaporator.

Enthalpies (kJ/kg) of the solutions are calculated as,

$$h(T,X) = \sum_{n=0}^{4}(a_n X^n) + T\sum_{n=0}^{3}(b_n X^n) + T^2\sum_{n=0}^{2}(c_n X^n) + T^3 d_0 \tag{4.2}$$

$$s.t., 273 \leq T \leq 463, 40 \leq X \leq 65$$

The constants of the equation are presented in the Table 4.1. Table 4.2 shows the simplified modelling process of the absorption refrigeration cycles.

Table 4.1. The constants for enthalpy calculations of water-LiBr solution. [7]

n	a_n	b_n	c_n	d_n
0	-954.8	-3.293E-1	7.4285E-3	-2.269E-6
1	47.7739	4.076E-2	-1.5144E-4	
2	-1.59235	-1.36E-5	1.3555E-6	
3	2.09422E-2	-7.1366E-6		
4	-7.689E-5			

Table 4.2. The modelling process of the absorption refrigeration cycles in brief.

	Water/Lithium bromide cycle	**Ammonia – water cycle**
Initial values	$\dot{Q}_e\ (W)$ - cooling power at the evaporator T_g (K) - temperature at generator T_e (K) - temperature at evaporator ε - the effectiveness of the heat exchanger	 T_c (K) - temperature at condenser T_a (K) - temperature at absorber
Concentration (*strong* or *weak*)	$X_s = \dfrac{49.04 + 1.125T_g - T_c}{134.65 + 0.47T_g}$	$X_w = \dfrac{49.04 + 1.125T_a - T_e}{134.65 + 0.47T_a}$
Mass flows $\dot{m}$ - mass flow $\dot{n}$ - molar flow	Based on mass balance of the system	
	$\dot{m}_w X_w = \dot{m}_s X_s$ $\dot{m}_w = \dot{m}_{h2o} + \dot{m}_s$ $\dot{m}_{h20} = \frac{\dot{Q}_e}{h_{10}-h_9}$ (water cycle)	$\dot{n}_w X_w = \dot{n}_s X_s$ $\dot{n}_w = n_{h2o} + \dot{n}_s$ $\dot{n}_{nh3} = \frac{\dot{Q}_e}{h_{13}-h_{12}}$ (ammonia cycle) $\dot{n}_8 = \dfrac{\dot{n}_{nh3}}{Y_a}$ $\dot{n}_9 = \dot{n}_{nh3} - \dot{n}_8$
Enthalpy	Based on energy balance of the system	
	h_1, h_3, h_4, h_5 - calculated using equation 4.2. h_7, h_8, h_{10} - calculated using REFPROP [11]. $h_2 = \dfrac{\dot{m}_s}{\dot{m}_w}(h_5 - h_4) + h_3$ $h_6 = h_5$ $h_9 = h_8$	h_1, h_5, h_8, h_9 - calculated using REFPROP [11]. $h_7 = h_6$ $h_4 = \dfrac{\dot{n}_w h_3 + \dot{n}_s h_5 - \dot{n}_s h_6}{\dot{n}_w}$ $h_6 = \dfrac{\dot{n}_w h_3 + \dot{n}_s h_5 - \dot{n}_w h_4}{\dot{n}_s}$ $h_3 = \dfrac{\dot{n}_8 h_8 + \dot{n}_w h_2 - \dot{n}_9 h_9 - \dot{n}_{nh3} h_{10}}{\dot{n}_w}$ $h_2 = h_1$
Heat transfer	$\dot{Q}_a = \dot{m}_{h20} h_{10} + \dot{m}_s h_6 - \dot{m}_w h_1$ $\dot{Q}_g = \dot{m}_{h20} h_7 + \dot{m}_s h_4 - \dot{m}_w h_3$ $\dot{Q}_c = \dot{m}_{h20}(h_7 - h_8)$	$\dot{Q}_a = \dot{n}_{nh3} h_{13} + \dot{n}_s h_7 - \dot{n}_w h_1$ $\dot{Q}_g = \dot{n}_8 h_8 + \dot{n}_s h_5 - \dot{m}_w h_4 - \dot{n}_9 h_9$ $\dot{Q}_c = \dot{n}_{nh3}(h_{10} - h_{11})$
COP	$COP = \dfrac{\dot{Q}_e}{\dot{Q}_g}$	

4.3.2 Model validation

To evaluate its correctness and accuracy, the thermodynamic model is validated against a model presented by Ebrahimi et al. [6] The input data series 1 and 2 used for comparison is presented in Table 4.3, and the result comparison is shown in Table 4.4. As it can be seen, the model is very accurate. The maximum deviation against data series 1 is 0.7 % whereas it is 0.1 % against data series 2. The model of Ebrahimi et al. [6] has 5 % deviation against data series 1.

Table 4.3. Input data used for water-LiBr absorption refrigeration calculations.

Input data	Symbol (unit)	Data series 1	Data series 2
Evaporator cooling load	$\dot{Q}_e(kW)$	201.29	-
Generator temperature	$T_g(°C)$	84.8	90
Condenser temperature	$T_c(°C)$	39.8	35
Evaporator temperature	$T_e(°C)$	8.6	5
Absorber temperature	$T_a(°C)$	35.5	35
Effectiveness of heat exchanger	ε (%)	70.7	60
Generator heating load	$\dot{Q}_g(kW)$	-	100

Table 4.4. Results comparison of water-LiBr absorption refrigeration calculations.

Output results	Symbol (unit)	Data series 1	Present study	Deviation (%)	Data series 2	Present study	Deviation (%)
Generator heating load	$\dot{Q}_g(kW)$	259.55	257.91	0.6	input data	iterated	-
Evaporation cooling load	$\dot{Q}_e(kW)$	input data	input data	-	75.865	75.777	0.1
Condenser cooling load	$\dot{Q}_c(kW)$	213.37	213.41	0.0	80.981	80.874	0.1
Absorber cooling load	$\dot{Q}_a(kW)$	247.47	245.80	0.7	94.885	94.904	0.0
Pump work	$P(kW)$	-	-	-	0.001	0.001	0.0
COP		0.7755	0.7805	0.6	0.7587	0.7578	0.1

The ammonia-water model is validated against the COP graphs, as a function of the generator temperature, of Ouadha and El-Gotni [8]. As shown in Figure 4.5, the graphs are performed for three different condensing temperatures (20°C, 30°C and 40°C). The results of the model presented in this study are presented on the left side. It can be noted that the curves are in great agreement when the temperatures are within approx. 20 degrees region over that critical point, but in the higher temperature region, the curves of our model are descending while the curves of

Ouadha and El-Gotni [8] are ascending. This is mainly because the ammonia vapour after the generator is assumed to be purely saturated in their model, while, we assume that the vapour is not saturated after the generator in our model and a rectifier is used to remove the evaporated water. Consequently, in our model, more water is vaporized and more energy is needed for water condensation at higher temperatures, which is closer to the real condition and hence ensure our model having higher accuracy.

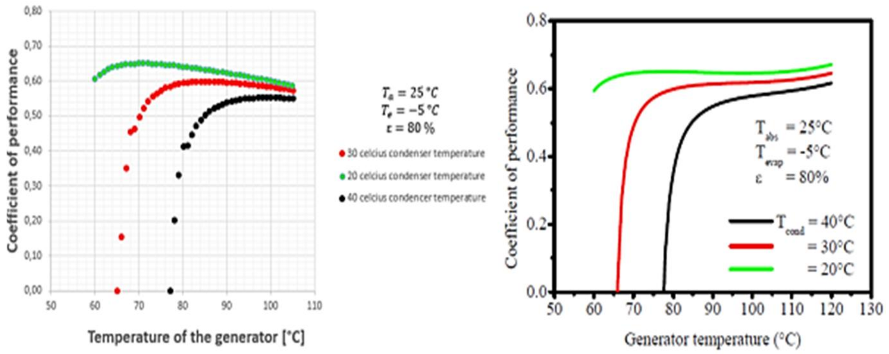

Figure 4.5. Performance comparison of ammonia-water absorption refrigeration models. [8]

4.3.3 The system performance and critical temperatures

By changing some initial values of the process model, we are able to evaluate the performance of the system. As an example, we will check the COPs of the system with only the change of the generator temperature (keeping others constant). The results, for three different evaporator temperatures, are shown in Figure 4.6 for water-LiBr refrigeration and in Figure 4.7 for ammonia-water refrigeration, respectively. In both cases, the temperatures of the condenser and absorber were set to be 30 °C.

As shown in Figure 4.6, the COP of the water-LiBr system is stable but drops fast when approaching the critical generator temperature where the refrigeration power is zero. While the COP of the ammonia-water working pair, shown in Figure 4.7, does not drop so dramatically. It starts to decrease at approximately 15 degrees before the critical generator temperature, which is less than 10 degrees before the critical temperature of the water-LiBr system. The critical generator temperatures are higher for colder evaporator temperatures. For 10°C evaporator temperature the critical value is 52°C in the water-LiBr system and for -20 °C evaporator temperature in ammonia-water system the critical temperature is 90°C.

It was also noted that the COPs of the systems decrease and the critical temperatures increase if the condensation and absorption temperatures increase. The critical temperatures of the generator mentioned previously increased to 74°C and 114°C respectively when the absorption and condensation temperatures were increased by 10°C. Cooler seawater temperature enables better COP compared to tropical conditions. Hence, one important factor for system performances is the seawater temperature, which is used for cooling the absorber and the condenser. For better coefficient of performance, these temperatures should be kept close to the seawater temperature.

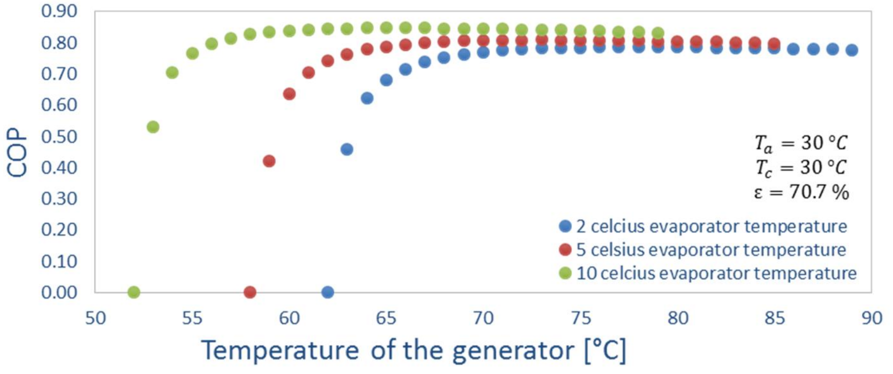

Figure 4.6. The performance of the water-LiBr absorption refrigeration cycle COPs against the generator temperature.

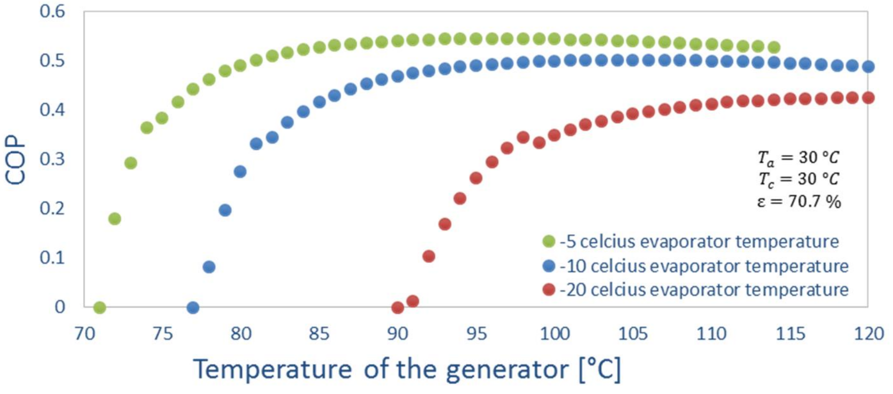

Figure 4.7. The performance of the ammonia-water absorption refrigeration cycle COPs against the generator temperature.

One disadvantage of the water-LiBr working pair is that it cannot produce below zero degree coolant. It also has a risk of crystallization at high temperatures when LiBr concentration is high. The crystallization limit is shown in the Figure 10. [3]

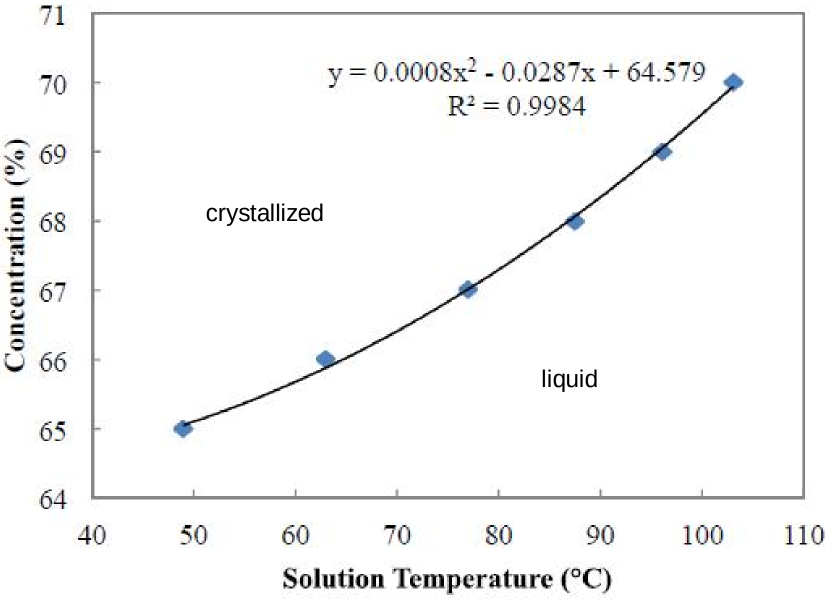

Figure 4.8. Crystallization limit in a water-LiBr solution between 45 – 103 °C. [9]

4.3.4 The optimal temperatures under ISO condition

To get the optimal temperatures of the absorber, condenser and generator with different evaporator temperature levels is a multi-variable optimization problem. In this study, we use evolutionary optimization methods in one Excel-Solver to find the optimal temperatures for maximal COPs of the water-LiBr absorption refrigeration cycle. During the optimization process, all these temperatures were set to be variable but the cooling power and the effectiveness of the heat exchanger were fixed.

Figure 4.9 and Figure 4.10 show the optimal temperatures of the generator and evaporator and the corresponding optimal COPs of the water-LiBr absorption refrigeration cycle. It is noted that the temperatures of the absorber and the condenser should be kept as low as possible to maximize the COPs. In this case, the lowest temperatures are approximately 30 °C, the seawater temperature under ISO conditions. It is also noted that the system performs better (higher COPs) at higher cooling temperatures than lower temperatures and the optimal generator temperature decreases linearly as the temperature of the evaporator rises. Hence, warmer generator temperatures are needed in order to obtain colder refrigerant more efficiently.

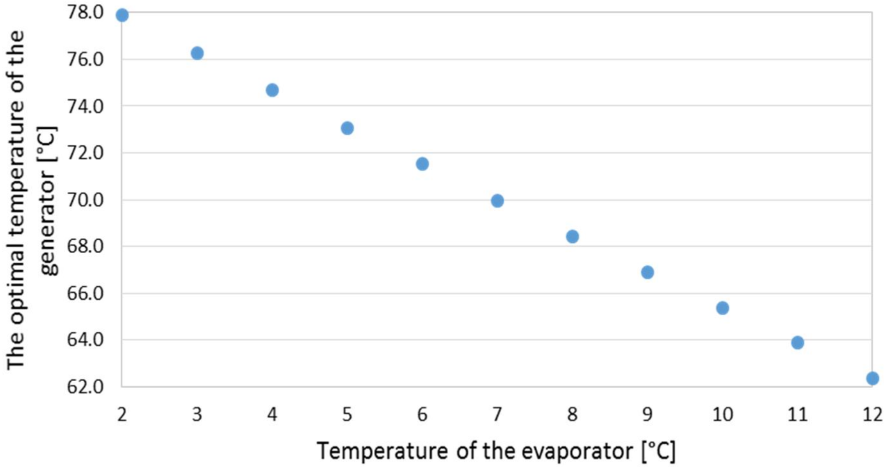

Figure 4.9. The optimal generator temperatures and corresponding COPs of the water-LiBr absorption refrigeration cycle.

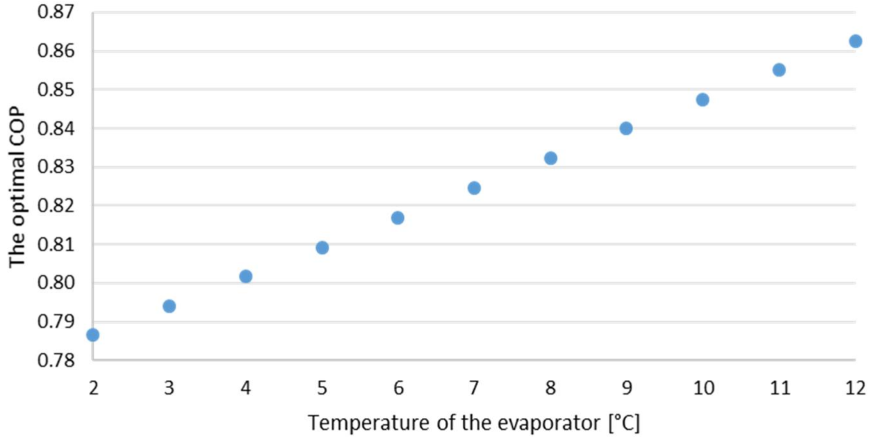

Figure 4.10. The optimal evaporator temperatures and corresponding COPs of the water-LiBr absorption refrigeration cycle.

4.4 Case studies

In this section, a case study is briefly carried out to explore the cooling potential with the two above-mentioned absorption refrigeration systems using waste heat available from the water cooling system and exhaust gas system onboard a case ship.

4.4.1 The case ship specifications

The bulk carrier B.Delta 37, listed in Chapter Case ships1.4, is used as the case ship in this study. Figure 4.11 shows the simplified diagram of the case ship water cooling system. The system integrates the high temperature (HT) and low temperature (LT) cycles together. The HT water cycle is mainly for main engine jacket cooling and fresh water generation, while the LT water cycle is mainly used to cool the auxiliary engines, scavenge air of the main engine, air-conditioning (AC) chillers, lubrication oil (LO), HT water cycle and to condensate the surplus steam from the boilers. The central cooler is used to take away the excessing heat in the system. In the AC chiller, the compressed refrigeration is currently used for air-conditioning, which is the only cold refrigeration need where absorption refrigeration system could be useful in this case. [10]

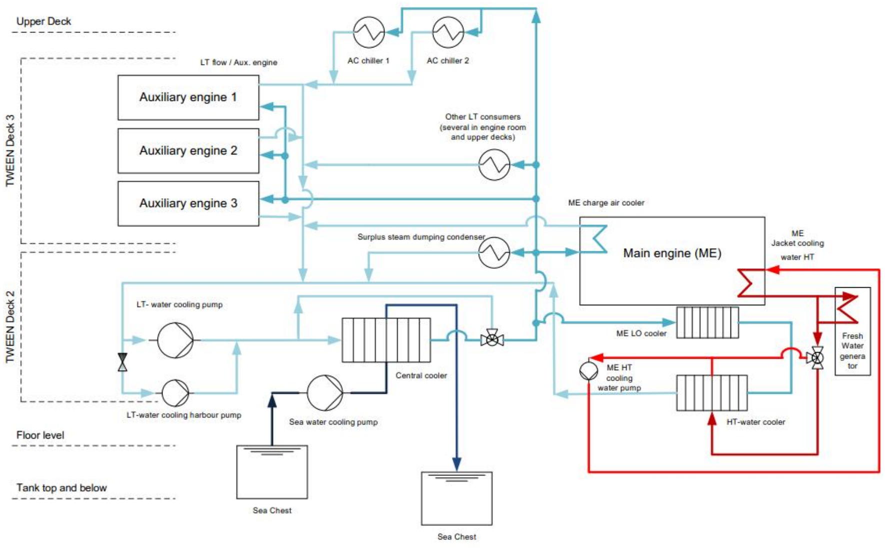

Figure 4.11. The water cooling system of the case ship. [11]

Table 4.5 shows the load of ship engine cooling water systems under specified operating conditions. Specifically related to this study, the engine loads and operating hours are used for calculating total cooling energy potential of absorption refrigeration systems. AC chiller cooling power presented in the table shows the cooling need of the condenser in vapour compression cycle that leads to 350 kW and 150 kW evaporator cooling power with the 4.5 COP of the existing cooling system. [10]

Table 4.5. The load of ship engine cooling water systems under specified operating conditions. [10]

	Yearly operation hours	Jacket water cooler	Lubrication oil cooler	Scavenge air cooler	AC chiller	Surplus steam dumping condenser	Auxiliary diesel engine	Other LT consumers	Central cooler
Dimensioning case, tropical conditions	-	950	600	1980	432	450	583	166	5662
ISO, 100% ME load	-	930	590	1940	183	10	465	66	4046
ISO, Scantling draught 74,4% ME load	1226,4	760	540	1490	183	10	465	66	3366
ISO, Scantling draught 39,0% ME load	1226,4	520	420	440	183	5	465	66	1951
ISO, Design draught NCR load 67,8% ME load	919,8	720	520	1220	183	5	465	66	3031
ISO, Design draught slow steaming 35,1% ME load	919,8	485	400	330	183	0	465	66	1781
ISO, Ballast draught 63,7% ME load	919,8	690	500	1100	183	0	465	66	2856
ISO, Ballast draught slow steaming 25,0% ME load	919,8	415	360	100	183	0	465	66	1441
ISO, Maneuvering	96	0	0	0	183	0	498	66	776
ISO, Harbour loading	936	0	0	0	183	0	1026	66	616
ISO, Harbour unloading	1176	0	0	0	183	0	1026	66	776
ISO, Harbour stand still	456	0	0	0	183	0	423	66	516

According to the cooling power need and temperature levels, the potential energy sources for absorption refrigeration are jacket-cooling water and scavenge air. High temperature exhaust gases of main engine and auxiliary engines are potential energy sources too. The amount of surplus steam has potential in tropical conditions but is minor in ISO conditions. Table 4.6 shows the potential (at different temperature levels) of different heat sources in ISO conditions for absorption refrigeration. It can be seen that both the exhaust gas and scavenge air have high operating temperatures. The exhaust gas seems to be the best energy source but the disadvantage is that it cannot be cooled below 167°C due to the risk of corrosion [10]. In the rest of the section, all these sources are explored with respect to the gained refrigerating power and total cooling energy suitable for cooling purposes in the case ship.

Table 4.6. The heat sources in ISO conditions for absorption refrigeration. [10]

Temperature	**208–177 °C**	**203–68 °C**	**85°C**
Engine load	**Exhaust gas (kg/s)**	**Scavenge air (kg/s)**	**Jacket water (kg/s)**
100%	11.8	11.6	14.8
75 %	10.3	10.1	12.1
65 %	9.1	9.0	11.0
40 %	6.1	6.0	8.3
35 %	5.4	5.3	7.6
25 %	4.0	3.9	6.5

4.4.2 Assumptions and Initializations

For the ease of the calculation, other than the assumptions listed in Section 4.3.1, some more specifications have been made as follows.

- The heat exchanger efficiency is assumed to be 70.7%, which is also used in the Data series 1 in Table 4.3.
- The condenser and absorber temperatures are set to be constant, 30 °C in ISO conditions.
- Three evaporator temperature settings are used in the calculation. They are, 2°C, 5°C and 10°C for the water-LiBr absorption refrigeration system, and -5°C, -10°C and -20°C for the ammonia-water system, respectively. However, only one case for each absorption system is presented here.
- The generator temperature and the cooling power are the model variables.

4.4.3 The LiBr-water absorption refrigeration cycle

4.4.3.1 Exhaust gas WHR

The temperature of the exhaust gases and mass flows varies depending on the engine load. Based on the case ship data in ISO conditions, the available heat is calculated assuming that the exhaust gas is cooled down to 167 °C. The maximum cooling power in different loads can be calculated using the optimal COP generator temperatures and the exhaust gas data of the case ship. The calculated results are shown in Table 4.7. Specifically, the heating potential of the exhaust gas (for the generator) is calculated, based on its molar flow and enthalpy difference over the generator, as

$$\dot{Q}_g = \sum \dot{n}_{gas}(h_{in} - h_{out}) \tag{4.3}$$

The cooling power (for the evaporator) is calculated based on the equation in Table 4.2 and the optimal COPs in Figure 4.9, as

$$\dot{Q}_e = \dot{Q}_g * COP \tag{4.4}$$

It can be noted that the cooling demand for air-conditioning (approx. 150 kW) of the case ship can be satisfied by the exhaust gas flow under all load conditions. During the scantling draught, with 74.4% main engine load, it is possible to produce about 530 kW cooling power for AC needs, which is more than 3.5 times of the need for cabins chilling in the case ship. Even under the lowest main engine load (25%), the produced cooling power (about 250 kW) is more than 1.5 times of the AC chiller's need. The cooling demand from the condenser and absorber (about 1190 kW) can be met by LT-water or seawater.

Table 4.7. The WHR potential of exhaust gas for the water-LiBr absorption refrigeration in ISO conditions.

(COP = 0.81)	GENERATOR	EVAPORATOR	CONDENSER	ABSORBER
Temperature	***73 °C***	***5 °C***	***30 °C***	***30 °C***
Load	**Heating potential $\dot{Q}_g$ [kW]**	**Cooling power $\dot{Q}_e$ [kW]**	**Cooling demand $\dot{Q}_c$ [kW]**	**Cooling demand $\dot{Q}_a$ [kW]**
100 %	750	607	639	717
95 %	725	587	618	694
90 %	703	569	599	672
85 %	699	565	596	668
80 %	685	554	584	655
75 %	659	533	562	631
70 %	626	507	534	599
65 %	591	478	504	566
60 %	560	453	477	536
55 %	525	425	448	503
50 %	490	397	418	469
45 %	454	367	387	434
40 %	414	335	353	396
35 %	379	306	323	362
30 %	345	279	294	330
25 %	312	252	266	298

4.4.3.2 Engine jacket water WHR

Due to its relatively high temperature (85°C), the main engine jacket cooling water can be used as an energy source for the absorption refrigeration. The generator could be placed before the HT-water cooler as shown in Figure 4.12. To simplify the case, the energy consumption of the fresh water generator is not taken into account.

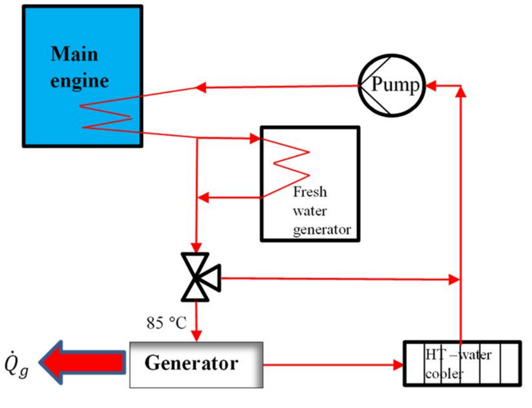

Figure 4.12. The generator arrangement in the engine jacket cooling water system.

The operating temperatures of the HT water circuit are 85°C and 70°C (return and supply water of the main engine) [10]. In this case, the generator temperature was set to be 65°C to ensure 5°C temperature difference in the heat exchanger. It can be noted from Figure 4.6 that the generator temperature (65°C) is close to the critical value of 2°C evaporation temperature. The COP value, 0.68, is used in calculations, but even a small temperature change can drop the system performance dramatically. For 5°C and 10°C evaporator temperatures, the COP values are 0.79 and 0.85 respectively. The cooling power (for the evaporator) is calculated based on the equation in Table 4.2. The calculated results are shown in Table 4.8 below. According to the results, the ME jacket cooling water is even more potential than the ME exhaust gas. Specifically, the cooling capacity of the ME jacket cooling water at 74.4% ME load is approximately 600 kW, which is 50 kW more than the potential in the exhaust gas WHR. During scantling draught at 39 % ME load, the cooling potential of the ME jack-cooling water is 75kW more that the potential of the ME exhaust gas. Overall, the ME jacket cooling water has 16 % more cooling potential in average than the ME exhaust gas.

Table 4.8. The WHR potential of engine jacket cooling water for the water-LiBr absorption refrigeration in ISO conditions.

(COP = 0.79)	**GENERATOR**	**EVAPORATOR**	**CONDENSER**	**ABSORBER**
Temperature	***65 °C***	***5 °C***	***30 °C***	***30 °C***
Load	**Heating potential** $\dot{Q}_g$ **[kW]**	**Cooling power** $\dot{Q}_e$ **[kW]**	**Cooling demand** $\dot{Q}_c$ **[kW]**	**Cooling demand** $\dot{Q}_a$ **[kW]**
100 %	930	733	768	896
95 %	900	710	743	867
90 %	860	678	710	828
85 %	830	655	685	799
80 %	800	631	660	770
75 %	760	599	627	732
70 %	730	576	603	703
65 %	690	544	570	664
60 %	660	520	545	636
55 %	620	489	512	597
50 %	590	465	487	568
45 %	550	434	454	530
40 %	520	410	429	501
35 %	480	379	396	462
30 %	450	355	372	433
25 %	410	323	339	395

The problem of the system is that the generator temperature is quite close to the boundary limit (in low evaporator temperature). In tropical conditions, the water-LiBr absorption refrigeration system cannot even work at any evaporator temperatures between 2°C and 10°C due to the high absorber and condenser temperatures (approx. 40°C) caused by the high seawater temperature.

4.4.3.3 Scavenge air WHR

Depending on the engine load, the scavenge air has substantial amount of waste heat to be utilized for absorption refrigeration system. The heat input is calculated as

$$\dot{Q}_g = \dot{m}_{in}(h_1 - h_2) \tag{4.5}$$

where $\dot{m}_{in}$ is the massflow rate of the scavenge air and $h_1$and $h_2$are the enthalpies of the inlet and outlet flows. The pressure loss over generator is assumed zero and the outlet temperature is 10°C higher than the generator temperature to ensure good heat exchange ratio. The generator temperature is optimized so that the overall refrigerating energy is maximized for yearly operation. The calculated results are shown in Table 4.9.

Table 4.9. The WHR potential of engine scavenge air for the water-LiBr absorption refrigeration in ISO conditions.

(COP = 0.78)	GENERATOR	EVAPORATOR	CONDENSER	ABSORBER
Temperature	***64 °C***	***5 °C***	***30 °C***	***30 °C***
Load	**Heating potential $\dot{Q}_g$ [kW]**	**Cooling power $\dot{Q}_e$ [kW]**	**Cooling demand $\dot{Q}_c$ [kW]**	**Cooling demand $\dot{Q}_a$ [kW]**
100 %	1563	1218	1274	1508
95 %	1423	1109	1160	1373
90 %	1282	998	1044	1235
85 %	1264	984	1029	1218
80 %	1236	963	1007	1192
75 %	1095	853	892	1056
70 %	914	712	745	881
65 %	745	580	607	718
60 %	612	477	499	590
55 %	480	374	391	463
50 %	361	281	294	348
45 %	247	193	202	239
40 %	149	116	121	144
35 %	71	55	58	68
30 %	14	11	12	14
25 %	0	0	0	0

As presented in Figure 4.6, the optimal generator temperature for 5°C evaporator temperature is 73°C in order to achieve the best COP of the water-LiBr absorption refrigeration cycle. However, depending on the case ship operating profiles, the optimal generator temperature of the absorption system for the scavenge air WHR is 64°C (listed in Table 4.9) for 5°C evaporator temperature. It is also seen that the cooling potential is reasonable only when the engine load is higher than 30%. As shown in the table, when the ME load is over 65%, the scavenge air has more cooling potential than the ME exhaust gas and jacket cooling water. Specifically under scantling draught conditions (with 74.4% ME load), the cooling power generated from the scavenge air is 60 % and 42% higher than the exhaust gas and the jacket cooling water, respectively. It would be even higher if the cooling power were optimised for high engine loads and not taking overall energy production into account. However, the cooling capacity drops fast and with 39.0 % scantling draught only around 116 kW of chilling power can be produced, which cannot meet the whole cooling demand of 150kW.

4.4.3.4 Full WHR potential of absorption refrigeration and fuel savings

Based on the calculations above, the full WHR potential of the water-LiBr absorption refrigeration system (5°C evaporator temperature) from different waste heat sources with respect to different engine loads is shown in Table 4.10.

Table 4.10. The full WHR potential of the water-LiBr absorption refrigeration in ISO conditions.

ENGINE LOAD [%]	**YEARLY OPERATION TIME [H]**	**AC COOLING DEMAND [KW]**	**COOLING POTENTIAL [KW]**		
			exhaust gases	**jacket water**	**scavenge air**
74.4	1226.4	150	533	599	853
39	1226.4	150	335	410	116
67.8	919.8	150	478	544	580
35.1	919.8	150	306	379	55
63.7	919.8	150	478	544	580
25	919.8	150	252	323	0
0	2664.0	150	0	0	0

The yearly cooling potential of the water-LiBr absorption refrigeration from WHR can be hence calculated as

$$E_{tot} = \sum \frac{Q_e * t}{1000} \tag{4.6}$$

where E_{tot} (MWh) is the total energy production or potential and t (h) is yearly operation hour. The calculated yearly WHR potential is shown in Figure 4.13.

It can be noted that the cooling potential is much higher than the cooling demand of the bulk carrier. The cooling potential, using the exhaust gas or jacket cooling water, covers solely all cooling needs during the main engine operation hours. This means that approximately 70% of the cooling need can be covered with these solutions yearly when taking into account the ME-off time. The scavenge air WHR system covers around 50% of the total cooling need. It has a lot of cooling potential during high engine loads, but in this case, it cannot be fully utilized because the cooling demand remains constant.

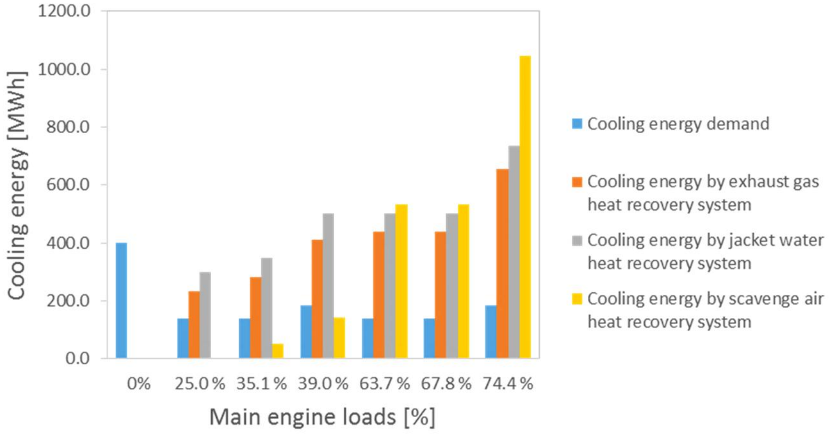

Figure 4.13. The yearly cooling demand and WHR potential of the water-LiBr absorption refrigeration system using ship waste heat sources in ISO conditions.

Currently, the cooling power of the case ship is produced by vapour compression, which consumes 33.3 kW electricity. [10] Hence, the yearly electricity consumption is approximately 293 MWh. Using the water-LiBr absorption refrigeration system with WHR from exhaust gas or jacket water, it is possible to save around 204 MWh electricity, which is 70% of the electricity consumption of vapour compression (if the pumping power increase for the absorber and condenser is not considered). Hence, the saving potentials are significant. If the exhaust gas of auxiliary engines is utilized for cooling purposes while the main engine is off, the cooling potential can be possibly increased over 70%.

If we assume that the efficiency of electricity production using auxiliary diesel generators is 37%, the yearly fuel savings with absorption refrigeration for the case ship could be approximately 46.8 tons (42.7 MJ/kg).

4.4.4 The ammonia-water absorption refrigeration cycle

According to section 4.2.2, water is used as absorbent in the ammonia-water cycle. Hence, the generator temperature should be higher than the water evaporation temperature. Consequently, the exhaust gas would be the suitable waste heat source for the absorption refrigeration cycle. The calculation process of the ammonia-water absorption refrigeration system with heat recovery from the ME exhaust gases is similar to the one presented for water-LiBr system previously, except that the generator temperature is different. The calculated results (with -10°C evaporator temperature) are shown in Table 4.11. The optimal temperature values (calculated previously) are used to obtain maximal cooling power generation. The system COP is 0.5, which means that half of the exhaust gas energy can be turned into refrigeration power. As shown in the table, the cooling power production is high enough for air-conditioning demands during all engine loads and the surplus cooling power capacity during high engine loads could be used for ice production. Around 330 kW of cooling power is generated during the scantling draught when the ME load is 75%. This means that approximately 2.7 tons of ice can be produced within an hour for various purposes, such as fish preservation. Even when the ME load is relatively low, the ice production capacity stays still high. Approximately 1.7 tons of ice can be produced within an hour when the ME load is 40%.

Table 4.11. The WHR potential of engine exhaust gas for the ammonia-water absorption refrigeration in ISO conditions.

(COP = 0.50)	**GENERATOR**	**EVAPORATOR**	**CONDENSER**	**ABSORBER**
Temperature	***107 °C***	***-10 °C***	***30 °C***	***30 °C***
Load	**Heating potential $\dot{Q}_g$ [kW]**	**Cooling power $\dot{Q}_e$ [kW]**	**Cooling demand $\dot{Q}_c$ [kW]**	**Cooling demand $\dot{Q}_a$ [kW]**
100 %	750	376	460	666
95 %	725	364	444	645
90 %	703	353	431	625
85 %	699	351	428	621
80 %	685	344	420	609
75 %	659	331	404	586
70 %	626	314	384	556
65 %	591	297	362	526
60 %	560	281	343	498
55 %	525	264	322	467
50 %	490	246	300	436
45 %	454	228	278	403
40 %	414	208	254	368
35 %	379	190	232	337
30 %	345	173	212	307
25 %	312	157	191	277

4.5 Conclusions

Both the water-LiBr and ammonia-water absorption refrigeration systems have large potential in ship WHR to improve overall ship energy efficiency. The water-LiBr absorption refrigeration system is potential using the waste heat from the engine exhaust gases, jacket water or scavenge air as energy source for onboard chilling functions, such as air-conditioning. All the cooling demand can be easily satisfied by the system using the exhaust gas or jack water. In ISO conditions, engine jacket water is the best to provide stable cooling power during all engine loads for the water-LiBr absorption refrigeration. The light temperature variation can be easily compensated by changing the mass flow of jacket water through the generator. In tropical conditions, the absorption refrigeration systems using exhaust gases are the best solution. The ammonia-water absorption refrigeration system is more suitable for the below-zero-Celsius applications using the waste heat from the engine exhaust gas for air-conditioning or ice-production. The cooling power productions can easily the cooling demand of cabins in the case ship, and the power could be used for ice-production purposes, for example, for medical and food preservation needs. It is estimated that approximately 1.4 tons of ice could be produced within an hour while chilling the cabins during high load (at 74.4%) of the main engine.

As shown in the study, the cooling potential of absorption refrigeration systems can be estimated with steady state calculations. However, dynamic simulations are needed for more accurate performance estimation and evaluation. Another challenge of using absorption refrigeration systems onboard ships is the motion of the ship. For example, the generator should be horizontal and the movement caused by the waves may deteriorate the performance of the system. [12] Any condition that is off-design increases the irreversibility of the absorption cycle, which affects harmfully on the performance of the system. [3] However, this study shows that, even with 40% lower performance of the system, it is still possible to produce enough cooling power to meet the needs of a bulk carrier in ISO conditions.

References

[1] Shu G, Liang Y, Wei H, Tian H, Zhao J, Liu L. A review of waste heat recovery on two-stroke IC engine aboard ships. Renewable and Sustainable Energy Reviews, Vol. 19, 2013, pp. 385-401

[2] Manzela A A, Hanriot S M, Cabezas-Domez L, Sodre J R. Using engine exhaust gas as energy source for an absorption refrigeration system. Applied Energy, Vol 87, 2010, pp. 1141-1148.

[3] Cao T, Lee H, Hwang Y, Radermacher R, Chun H. Performance investigation of engine waste heat powered absorption cycle cooling system for shipboard applications. Applied Thermal Engineering, Vol. 90, 2015, pp 820-830.

[4] Garimella S, Brown A M, Nagavarapu A K. Waste heat driven absorption/ vapor-compression cascade refrigeration system for megawatt scale, high-flux, low-temperature cooling. International Journal of Refrigeration, Vol 34, 2011, pp. 1776-1785.

[5] Watanabe H., “Field experience with large absorption heat pump (Type I) and heat transformers (Type II)”, Sanyo Electric Tokki Co., Ltd., Tokyo, Japan.

[6] Ebrahimi K, Jones G, Fleischer A S. Thermo-economic analysis of steady state waste heat recovery in data centers using absorption refrigeration. Applied Energy, Vol. 139, 2015, pp. 384-397.

[7] Kaita Y. Thermodynamic properties of lithium bromide-water solutions at high temperatures. International Journal of Refrigeration, Vol. 24, 2001, pp. 374-390.
[8] Ouadha A, El-Gotni Y. Integration of an ammonia-water absorption refrigeration system with a marine Diesel engine: A thermodynamic study. Procedia Computer Science, Vol. 19, 2013, pp. 754-761.
[9] Eyringer V, Kohler H W, Lauer A, Lembper B. Emissions from international shipping: 2. Impact of future technologies on scenarios until 2050. Journal of Geophysical Research 2005;110:D17306.
[10] Elg M, Kuosa M, Tammi K, Mäkipeska P, Kinnunen A, Rahkola P. Supporting the energy efficient ship design with energy flow simulations: case efficient cooling water system. Energy Efficient Ships 2015, 4th November 2015, Rotterdam, The Netherlands
[11] Elg M, Kuosa M, Lampinen M, Lahdelma R, Mäkipeska P, Raita J, Zou G, TAMMI K. Advanced auxiliary cooling system for energy efficient ships. Eemods'15 Conference, Energy efficiency in motor driven systems, 15-17 September, Helsinki.
[12] Kim K H, Yang K W, Jeong S, Nam K, Chang D. Absorption refrigeration system utilising engine exhaust gas for bulk gas carriers. Ships and Offshore Structures, Vol. 9, 2014, pp. 380-386.

Part II
Ship energy efficiency Methods

5. Ship energy flow simulation[1]

Zou Guangrong [1)], Aki Kinnunen [1)], Mia Elg [2)], Kalevi Tervo [3)], Kari Tammi [1)]
[1)] VTT Technical Research Centre of Finland Ltd
[2)] Deltamarin Ltd, [3)] ABB Oy Marine and Ports

Summary: Modern ship energy systems are large and complex, involving various energy processes and systems, which makes it more challenging to design and operate ship energy system energy-efficiently and environmentally friendly. In this chapter, a ship energy flow simulation method was introduced to thoroughly understand ship energy systems and to find potential ways to improve ship overall energy efficiency.

Due to the involvement of different physical domains in the energy processes, it adopted a multi-domain simulation approach to represent the ship energy systems systematically. All the main energy processes were modelled as subsystems only at a general and system level, and built as simple but feasible as possible to facilitate the dynamic interaction among different subsystems. After a brief introduction to the method, some earlier studies were presented as examples to show its feasibility and reliability for simulating the energy flow throughout a modern cruise ship with complex energy systems.

More importantly, it could serve as a very useful platform to get valuable insights into how to design an energy-efficient ship power plant and how to operate the vessel efficiently. Furthermore, it could be utilized to evaluate new technologies, and hence to support ship owners and managers in their business-critical decisions for both the existing and new-building ships.

5.1 Introduction

New technologies and innovative solutions have been widely applied to ship energy system design, retrofitting and operation processes in the last decades, which, however, rarely resulted in the expected energy efficiency improvements in practice. Lack of systematic consideration is one of the main reasons, especially when

---

[1] This chapter is excerpted and revised from two conference papers published in CIMAC 2013 [2] and Compit 2014 [3].

modern ships are getting larger and more complex, including various components and subsystems. For instance, RMS Queen Mary 2, the largest passenger ship, in terms of on-board installed power, has a nameplate capacity of 117.32 megawatts [1], supplied by four diesel engines and two gas turbines and equipped with numerous energy transferring components and a vast number of accommodating and entertaining facilities. Without a systematic understanding of how energy flows through ship energy systems, it is impossible to design and operate such complex energy systems, like Queen Mary 2, energy-efficiently at a ship level. Unfortunately, due to their complexity, there still is rarely a clear and thorough understanding of how energy is exactly distributed and consumed throughout the whole ship energy system, which is especially true for the steam-powered subsystems.

On the other hand, ship energy systems are dynamic engineering systems including mechanical, electrical, fluid, thermal, thermodynamic, chemical-process and mixed systems, which can be naturally represented with components and subsystems of specific physical domains and their interconnections at a system level. Besides, to consider ship energy systems at a system level would also give advantages to catch the big picture of energy system performances under different conditions and uncertainties, especially at the early design phase when detailed component information is not yet available and during real operating conditions when very detailed component information is not of high interest.

One challenge to model ship energy systems is how to efficiently represent the interactions between the components or subsystems of different physical domains and to construct unified mathematical models of the multi-physical systems at a system level. In this chapter, we explain a generalized approach to model and simulate these energy systems. In particular, we use the concept of *energy flow* and *energy flow simulation method*, which are applicable to all such systems, as the basis for defining a new set of domain variables and for describing the system components and their interactions at a system level in a unified manner.

5.2 Energy flow in general

Energy flow here refers to the flow or transfer of energy in different forms through a series of components and interconnected processes of an energy system, like the energy system of a building, a modern cruise ship, an industrial plant, or even a country. As the widely accepted approach, we consider *state* would be sufficient to describe the dynamics of an energy system at a system level, which are related to the energy exchange between connected components at any given instant. The state variables, or domain variables, of specific physical domains usually work as a variable pair, whose product is the energy flow or power in watts. For example, the state variables are voltage and current for electric domain, force and velocity for translational mechanical domain, and pressure and volume flowrate for hydraulic domain. Exceptions are thermal energy related domains where the commonly used state variable, heat flowrate, by itself is energy flow in watts. Similarly, the product of the magnetic domain variable pair is energy, not power. Besides, one more variable pair, pressure and mass flowrate, is needed in order to catch the dynamic phenomena of the pressure-effective thermal physical domains, including gas domain, thermal fluid domain and two-phase fluid domain.

Furthermore, for complex energy systems, there are many across-domain components, which transform energy from one domain to another one or more domain. For example, electric generators transform the energy from mechanical domain to electric domain, with some loss as heat energy in thermal domain due to their inefficiency. Using the well-accepted concepts, we introduce generalized variables and components so that we can have “equivalent” variables for each domain variable pair. As a generalization from the electric domain to all the domains, we use generalized *flow* and *effort* to name two variables of each pair respectively. For instance, current is flow variable and voltage is effort variable in the electric domain, and heat flowrate is flow variable and temperature is effort variable in the thermal domain. Mathematically, energy flows of different physical domains are exchangeable and of the same type. The more detailed description of each physical domain is shown in Table 5.1. There are several slightly different definitions of the generalized domain variables, such as those specified in [4]. Here, we define the domain variables as specified to meet the specific need in our work.

Table 5.1. List of physical domains with corresponding domain variables.

Physical domain	**Domain variable pairs**	
	Flow variable	**Effort variable**
Electric	Current	Voltage
Electric AC	AC Current, complex phasor	AC Voltage, complex phasor
Magnetic	Flux	Magneto-motive force (mmf)
Mechanical (translation)	Force	Translational velocity
Mechanical (rotation)	Torque	Angular velocity
Hydraulic (incompressible)	Pressure	Volume flowrate
Gas	Temperature, Pressure	Heat flowrate, Mass flowrate
Thermal	Temperature	Heat flowrate
Thermal fluid	Temperature, Pressure	Heat flowrate, Mass flowrate
Two-phase fluid	Specific enthalpy, Pressure	Heat flowrate, Mass flowrate
Chemical	Molar flowrate	Chemical potential

Using the generalized analogy, we can easily define generalized domain libraries of each physical component, e.g., generalized resistance, capacitance, inertance and flow or effort source, which lay the foundation for the multi-domain energy flow simulation method.

5.3 Ship energy flow simulation method

Ship energy flow simulation method is a model-based physical network approach to systematically represent the dynamic interactions among different components and subsystems of ship energy systems. It is a system-level approach simplifying the description of the complex energy systems into a topology consisting of a set of discrete components to approximate the behaviour of the spatially distributed physical systems. Specifically, as a general methodology, instead of representing every detail of ship on-board energy systems, it takes into consideration only key components and subsystems at a system level, aiming to build the energy flow simulators as simple but feasible as possible to facilitate the interactions among different main subsystems. Lumped-parameter models are constructed to simplify the physical energy systems to an acceptable level so as to address specific problems properly. It is a multi-domain approach combining the system components of different physical domains into one model, which allows obtaining a clear big picture of the whole energy system. Using the generalized analogy, the energy flows of different domains can be seamlessly integrated together, which helps to discover the cause and effect of specific components to other components and to the whole energy system.

Mathematically, the lumped-parameter model is a simplified topological network with only a finite number of parameters, which can adequately describe the physical system, using ordinary differential equations (ODE) (mostly first-order), algebraic equations, or differential-algebraic equations (DAEs). Given the initial conditions of the system, the generalized Kirchhoff's circuit laws can be employed to analyse the network and hence to discover how energy of different forms transfers and flows through every component in the network under dynamic conditions.

5.3.1 Simscape

There are different approaches utilized for practical implementations of physical system modelling in different environment. In this chapter, we are going to introduce one renowned software, Simscape, which Mathworks developed for modelling physical systems in their Simulink® environment. It employs the Physical Network approach, which differs from the standard Simulink modelling approach and is particularly suited to simulating systems that consist of real physical components. [4]

Simscape consists of a set of block libraries of different physical domains and special simulation features. The domains available in Simscape currently are electric, gas, mechanical, hydraulic, magnetic, thermal, thermal liquid and two-phase liquid. For each domain libraries, there are some foundational blocks available for use. Moreover, there are also more advanced libraries with a number of component blocks available for modelling common engineering systems, including Simscape Driveline, Simscape Electronics, Simscape Fluids, Simscape Multibody and Simscape Power Systems. In case the domains or the components you work with are not available in the domain libraries yet, you can use the Simscape language to easily set up customized domains and develop new component blocks. For example, we have built our own domains and a large number of key components to simulate the ship energy systems. The readers can find more detailed information on the Simscape software in its user manuals. [4]

5.4 Case examples

In practical applications, ship energy flow simulators, developed using ship energy flow simulation method, can give good insights into understanding of the energy distribution of ship energy systems and hence finding potential and globally optimal ways to design and operate ship energy systems more energy efficiently and environmentally friendly. In the rest of the chapter, we give two short examples (without the details) of how we utilize the simulators for some specific purposes.

5.4.1 Container ship energy flow simulator

Despite the fact that modern giant container ships are equipped with most efficient diesel engines, engine efficiency could barely go over 50% under specific operating conditions. In practice, there is still around 50% of the fuel energy wasted and taken away by engine exhaust gas, cooling water (both HT and LT) during the combustion process. Hence, waste heat recovery technologies have become popular and viable options for improving ship energy efficiency and reducing emissions, due to their long success in land-based combined heat and power (CHP) applications. In the first example, we developed a ship energy flow simulator for one of the largest and most energy efficient container ships to investigate its WHR potential.

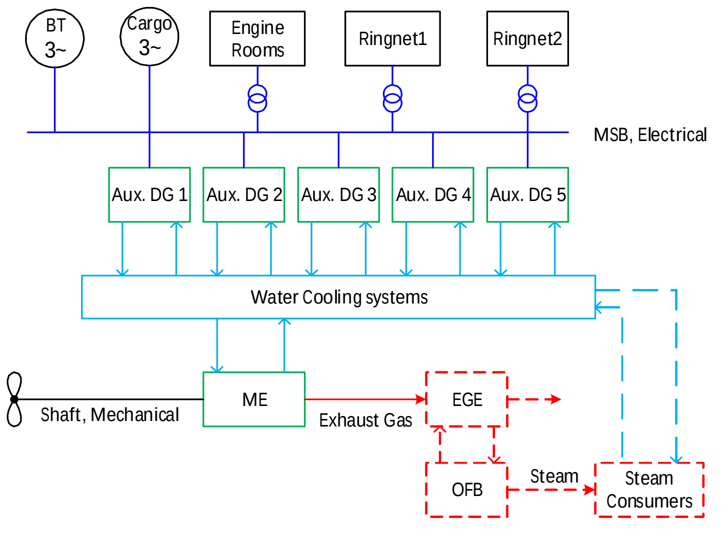

Figure 5.1. The schematic diagram of a container ship's main energy systems. [3]

Figure 5.1 shows a top-level schematic diagram of the container ship's main energy subsystems. The main propulsion system is diesel-mechanical, including a giant two-stroke diesel engine (ME) and a fixed-pitch propeller, directly connected through a main shaft. Five auxiliary diesel generators (Aux. DG) supply electricity to different electric consumers through main switchboards (MSBs), such as bow thruster (BT), lighting, cargo handling and engine rooms. Besides, an exhaust gas economizer (EGE) and an oil-fired boiler (OFB) are installed on-board to meet the need of steam consumers.

Using the developed container ship energy flow simulator as a baseline model, different WHR scenarios have been designed and evaluated under dynamic operating conditions, from the techno-economic perspective, to achieve the optimal overall energy efficiency at the ship level. Specifically, a combined energy saving scenario, including a dual-pressure EGE, a steam turbine (ST), a power turbine (PT) and a shaft generator (SG), has proven to be very promising. The implemented energy saving scenario has been highlighted (yellow and orange) in Figure 5.2.

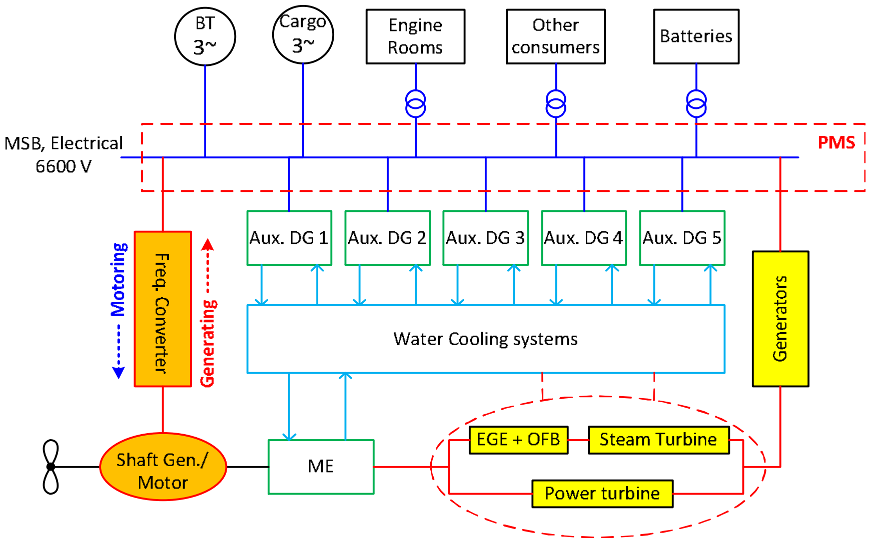

Figure 5.2. The schematic diagram of a modern container ship's energy system with implemented energy saving scenarios. [3]

The combined WHR system is able to recover the heat energy from ME exhaust gas and its fresh water cooling systems. Specifically, a PT system is employed to recover heat energy from ME by-pass exhaust gas and an advanced 4-stage dual-pressure EGE is in conjunction with a 2-stage steam turbine to enable extra heat recovery from the ME exhaust gas. The SG system is integrated with main shaft between the propeller and ME output. The SG system starts working when ME's RPM is equal to or larger than the RPM reference, 40% of its nominal RPM. The transmitted power is limited by both the absolute power difference (between the propeller shaft power and the ME admissible power reference as of its RPM) and SG's power as of ME's RPM. The SG system is working at the generator mode when the power difference is positive, otherwise working at the motoring mode. The combined scenario is able to achieve 8–20% energy efficiency increase depending on the specific operating conditions of the energy system.

5.4.2 Cruise ship energy flow simulator

Cruise ships are of high interests for evaluating different energy saving technologies due to its complexity and the induced potential for energy efficiency improvement. In this example, another ship energy flow simulator has been developed for a recently-built large cruise ship to test and evaluate different energy saving solutions. As shown in Figure 5.3, the ship energy system includes power generation and transmission system (e.g., engines, generators, shafts), electrical system (e.g., propulsion, lighting), water related system (e.g., water supply, fresh water production, engine cooling), steam powered system, etc. There are numerous different components included in each subsystem and interacted with each other as a whole energy system.

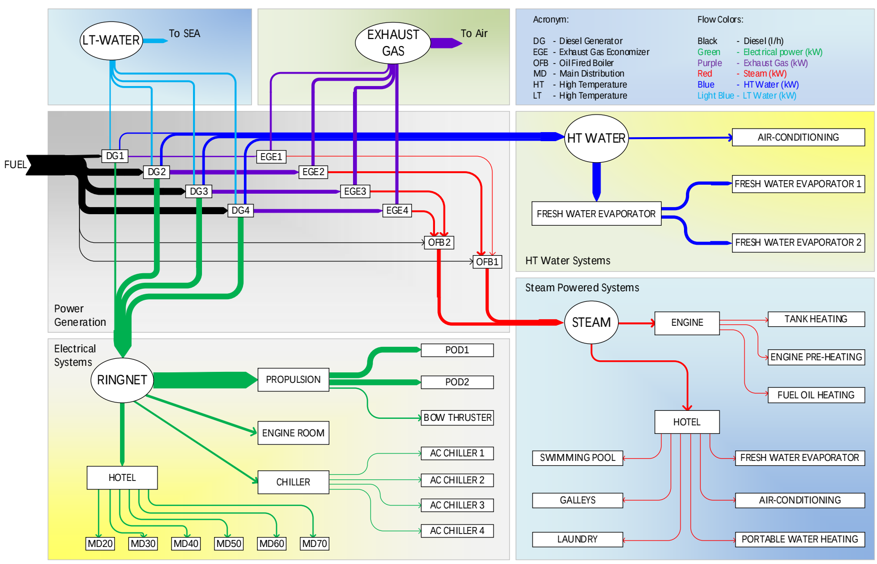

Figure 5.3. The schematic diagram of a cruise ship's main energy systems. [2]

With the help of the developed energy flow simulator, we can gain valuable insights into how to design an energy-efficient ship and how to operate a vessel efficiently. Here is one example related to typical fresh water cooling systems of diesel engines. Figure 5.4 shows the partial diagram of fresh cooling water systems extracted from the Wärtsilä 46 engine project guide. [5] The fresh water cooling systems, including one HT cooling circuit and a LT cooling circuit, are designed to take away extra heat energy from diesel engines to guarantee their working safety and best working efficiency. In the red-line HT water circuit, the temperature is controlled using a 3-way mixing valve 4V02 and a 3-way splitting valve 4V01, which bypasses a fresh water generator 4N02 (the WHR unit), to maintain the temperature at the engine outlet constantly at 91 ˚C and to protect the engine from excessive thermal load. Currently the set point of valve 4V02 is constantly at 74 ˚C for the safety of operation. Consequently, a large amount of heat energy has been wasted to LT cooling water circuit and further dissipated to the environment.

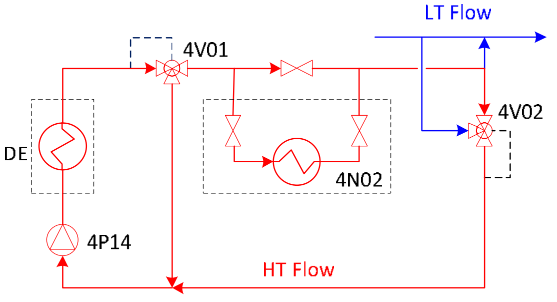

Figure 5.4. A partial schematic diagram of the HT cooling water system of one Wärtsilä 46 marine engine. [2]

Using the developed energy flow simulator, we have evaluated the potential of WHR via advanced valve control. Specifically, by adjusting the opening of the temperature control valve 4V02 according to the engine load, about 15–45% extra heat energy could be recovered from the HT cooling water circuit under different engine load conditions, as shown in Figure 5.5.

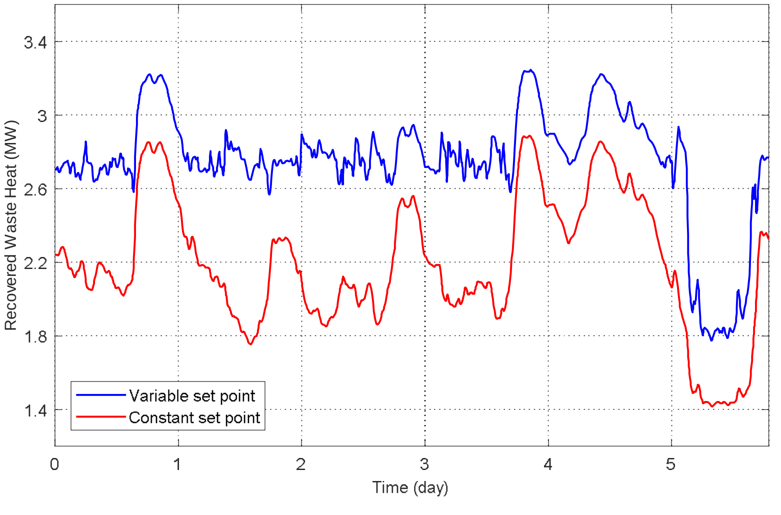

Figure 5.5. The comparison of the WHR from the HT cooling circuit under two different operating conditions of the temperature control valve 4V02. [2]

5.5 Conclusions

Ship energy flow simulation has been gaining the popularity due to the simpler and more efficient representation of ship energy systems and due to its capability to catch the real-life performances of ship energy systems under dynamic operating conditions and uncertainties. In comparison to the traditional simulation methods, the developed ship energy flow simulators can give very valuable insights into system-level ship energy system design and operation optimization, serving as more useful platforms to evaluate novel energy saving technologies and solutions without changing the real systems and helping to find promising ways to improve the energy efficiency of both the existing and new-building ships. More importantly, the simulators could support ship owners and managers in their business-critical decisions and hence lower the risk of their investments and improve the competitiveness of customers and their partners in the booming market.

This approach is very general and can be easily applied to different energy system and even to a wider variety of engineering systems.

References

[1] Wikipedia. RMS Queen Mary 2. [online] Available: https://en.wikipedia.org/wiki/RMS_Queen_Mary_2

[2] Zou G, Elg M, Kinnunen A, Kovanen P, Tammi K, Tervo K. Modeling ship energy flow with multi-domain simulation, Proc. CIMAC 2013

[3] Zou G, Kinnunen A, Tervo K, Orivuori J, Vänskä K, Tammi K. Evaluate Ship Energy Saving Scenarios Using Multi-Domain Energy Flow Simulation, Proc. Compit 2014

[4] Matheworks. Simscape User’s guide (2016b). [online] Available: http://www.mathworks.com/help/pdf_doc/physmod/simscape/simscape_ug.pdf

[5] Wärtsilä. Wärtsilä 46 engine project guide. [online] Available: https://www.wartsila.com/products/marine-oil-gas/engines-generating-sets/diesel-engines/wartsila-46

.

6. Ice going vessel energy system modelling and simulation[1]

Pekka Rahkola, Aki Kinnunen
VTT Technical Research Centre of Finland Ltd

Summary: In this chapter, a system-level multi-domain simulation model of an icebreaker energy system is developed to study its energy efficiency and further to investigate the energy improving potential of different energy saving scenarios under different load conditions and operation modes.

The dynamic simulation model is developed systematically based on mechanical and thermodynamic first principles, which well represented the onboard energy system of the case icebreaker, with all the key subsystems included. Based on the developed baseline model, different energy saving scenarios were investigated in order to evaluate their potential in improving energy efficiency of the icebreaker. Based on the specification, different configuration of variable speed pumps of the seawater system did not bring in substantial benefits. However, 2.6% energy savings can be possibly achieved only with the optimized generator load control. Up to 4.8% of savings may be reached if combined with battery systems.

6.1 Introduction

To be able to design more energy efficient ships and to operate ships in an energy efficient way, it is essential to study the whole energy system with all key subsystems included. This could help capture the cross behaviour of subsystems which has an effect of the overall functionality and efficiency of the system.

In this study, we use ship energy flow simulation method to develop a dynamic simulation model of the energy system of the case icebreaker IB Polaris, as listed in Section 1.4, so as to study its functionality and efficiency under different load conditions and operation modes. The system-level simulation model represents all the main ship energy subsystems, including electricity production, propulsion machinery and heat processes. Using the developed simulation model as a

[1] This chapter is excerpted and revised from the SET research project deliverable with the same name.

baseline, we are able to study the efficiency of the case ship and further to investigate the applicability and potential of different energy saving scenarios for the case ship, including variable speed pumps, electric energy storages and engine operation optimization.

6.2 Energy system modelling

Based on the case ship configuration, using a multi-domain modelling approach, a simplified simulation model is developed to represent the key processes of the ship energy system in the Matlab/Simulink/Simscape simulation environment. Physical domains included in the model are mechanical domain for the engine, electrical domain for the generator and electric network, and fluid domain for the cooling system. The main input to the simulation model is the time dependent power consumption in different domains including electricity and heat. The outputs of the simulation model are the energy flows from engine, exhaust gas heat recovery, cooling circuits, and electric network as a function of time. The model includes engines, generators and electric network, exhaust gas waste heat recovery, engine cooling system including engine and auxiliary system circuits, heat exchangers and seawater cooling circuit, as shown in Figure 6.1.

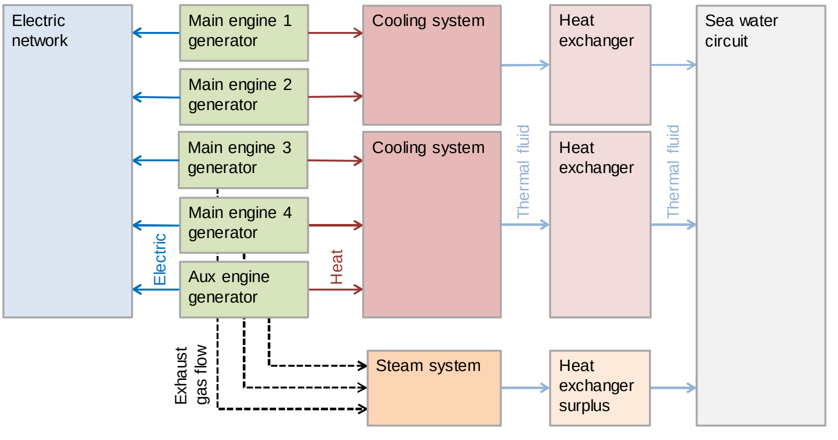

Figure 6.1. Schematic view of the energy systems of the case ship.

6.2.1 Main engines, generators and electric systems

The case ship features a diesel-electric propulsion system, equipped with dual-fuel engines that run on liquefied natural gas (LNG) or marine diesel oil (MDO). There are four main engines (ME, 2 x 6 MW and 2 x 4.5 MW) and an auxiliary engine (AE, 1.3 MW) onboard the ship. The propulsion machinery consists of three azimuth thrusters with total power of 19.5 MW. [1]

The developed power subsystem model, including engine, generator and electric system, is shown in Figure 6.2. It consists of the mechanical system, which includes the rotating mass of the engine and output torque acting on it, and the electrical system, which includes the converter from the rotational domain into the electric domain. The engine model is equipped with a speed controller, which defines torque value to maintain the engine speed under different loads coming from the electric network. The outputs of the engine are the heat variables that are connected to the cooling water system, exhaust gas mass flow and temperature that are connected to the steam system, and variables describing the state of the engine, for example the engine load.

The mechanical output of the engine is connected to the AC generator block. The flag signal of the engine activation is used to couple and uncouple the generator set from the electrical network and to define the target speed for the engine speed controller. The generator output is connected to the electric network, which consists of main electric consumers including three propulsion units and other consumers. The electric domain uses a simplified phasor representation with a fixed voltage level and thus it has no dynamics involved. The power need of the consumers defines the current in the electric network, which gives the load for the generators and mechanical load to the engines.

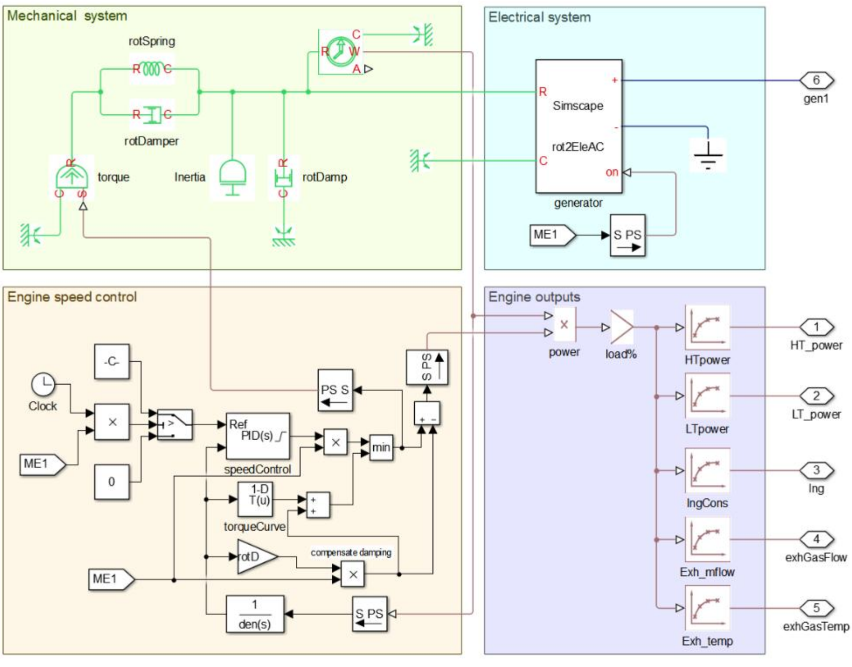

Figure 6.2. The engine and generator model of the case ship.

6.2.2 Cooling system

The cooling system is modelled using Simscape's Thermal Liquid domain, which models the liquid properties as a function of liquid temperature, for example liquid density and viscosity. [2] The modelling approach also captures the phenomena of liquid thermal expansion and warming in a pipe due to viscous losses. State variables of the domain are pressure, temperature, mass flow, and heat flow of the fluid. We assume that that the fluid stays liquid and no phase change occurs.

6.2.2.1 Engine cooling circuit

An example of an engine cooling circuit model is shown in Figure 6.3. The cooling system is divided into internal engine cooling circuit and external cooling circuit, which covers the rest of the system. The internal engine cooling system represents the cooling circuit, which are physically part of the engine installation, for heat extraction from cylinder jackets, charge air coolers and lubrication oil. The circuit is divided into high temperature (HT) and low temperature circuits (LT) with own engine driven circulation pumps. HT circuit has a thermostatic valve after the engine jacket to keep the temperature after the engine jacket at 96 °C. Input temperatures of the HT and LT flows to the engine are controlled. In the external cooling circuit, there is a heat recovery valve, which transfers heat from HT to LT circuit by mixing HT and LT water to ensure the engine HT inlet water at the temperature of 72 °C. In the LT circuit, there is a heat exchanger control valve to ensure the engine LT inlet water at the temperature of 36 °C. This heat exchanger valve controls the heat transfer from LT circuit to seawater circuit by mixing LT water to the cooled water coming after the heat exchanger. [3]

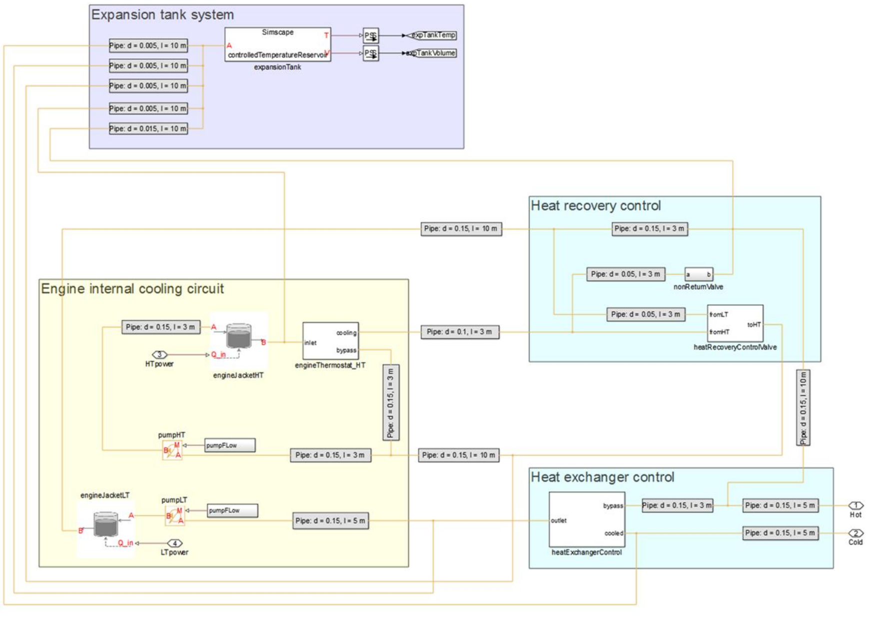

Figure 6.3. The engine cooling circuit model.

When physical modelling of thermal fluid is used, the system needs to have some components that are not relevant for the system simulation point of view but are essential for the physical system to work properly. Because the thermal expansion of the fluid is included, the connection to expansion tank system is needed to ensure realistic pressure levels for the system. The engine cooling circuit shown in Figure 6.3 has five connections to the expansion tank, each connected with a small diameter pipe. The expansion tank itself has a constant hydrostatic pressure due to the location of the tank, which is above other components of the system. Similarly, the non-return valve between HT and LT circuits in the heat recovery control unit ensures the proper functioning of HT and LT water mixing.

6.2.2.2 Heat exchangers

The engine cooling circuit is connected from the heat exchanger control unit to a heat exchanger, which transfers the heat from the engine cooling circuit to the seawater cooling circuit. The modelled heat exchanger is a counter current flow type, where fluids are flowing in opposite directions. An example of the heat exchanger model is shown in Figure 6.4, where the model is constructed with four pipe pairs connected with a heat transfer block. Heat is transferred between fluid volumes of pipe pairs using a convective heat transfer model.

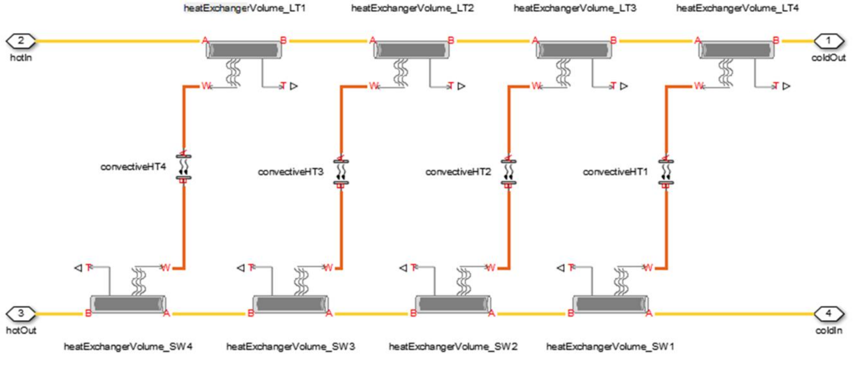

Figure 6.4. The heat exchanger model.

In the energy system model, the engine cooling circuit is connected to a seawater circuit, which has an inflow with constant temperature. An example of heat exchanger simulation results with varying LT inlet water temperature is shown in Figure 6.5. The simulation is done using a model with six pipe segments. The lowest plot in the simulation results show the capability of a counter current flow heat exchanger, where the output temperature of the LT water is lower than the outlet temperature of the seawater. This plot showing the temperature distribution of LT and seawater sides as a function of pipe segment number is made from one time instant of the simulation.

Heat exchanger models are validated against the full power capacity of the units. This means that heat power at known mass flow is at right magnitude, and heat exchanger dimensions are realistic, which give the calculated pressure loss over the heat exchanger. Dynamics of the heat exchanger are, however, unknown, which means that the response time against input temperature or mass flow changes may differ.

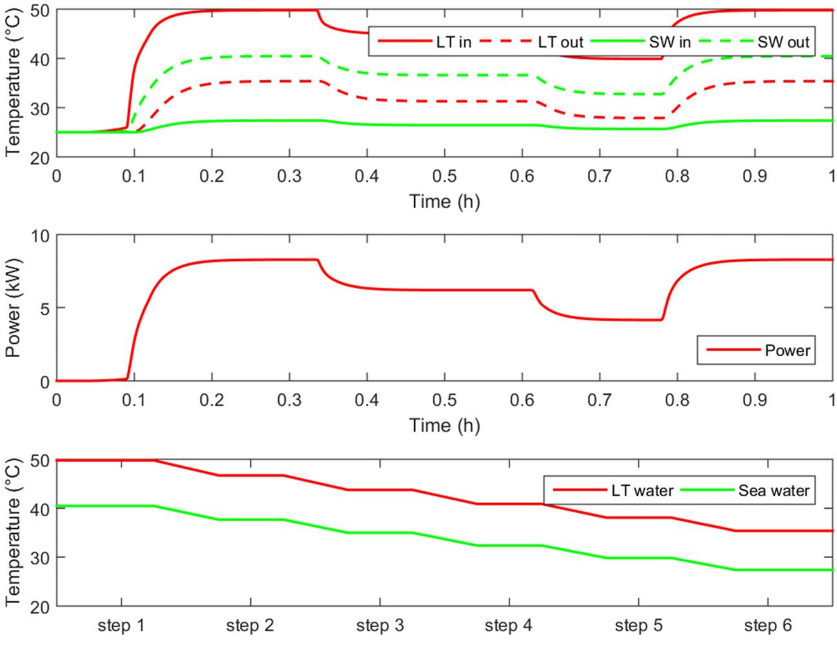

Figure 6.5. The simulation results of the heat exchanger model.

6.2.2.3 LNG Vaporizer

LNG is vaporized using heat from the HT cooling circuit. LNG vaporisers are tube heat exchangers connected to the main engine HT circuits. Glycol-water mixture acts as a medium for the heat transfer from HT circuit to LNG evaporation. The heat from the ME1 and ME2 HT circuits is used for starboard side (SB) and the heat from the ME3, ME4 and AE circuit for port side (PS) evaporation.

Pressure and temperature in the LNG tank are assumed to be 10 bar and −162 °C. Pressure and temperature requirements at the engine inlet are 452 kPa and 0–60 °C, respectively. The density, enthalpy and entropy of methane at specified state points of the LNG tank and the engine inlet are shown in Table 6.1. Energy needed for the LNG vaporization can be defined as the change between enthalpy values of the state properties. An average enthalpy change from the tank to the engine inlet is 920 kJ/kg.

Table 6.1. Methane properties at different temperatures and pressures in the LNG tank and in the engine inlet.

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)
111	0.50	423.7	-1.7	-0.02
111	1.00	424.2	-1.0	-0.03
111	1.50	424.6	-0.3	-0.03
273	0.50	3.6	849.9	5.64
303	0.50	3.2	917.0	5.87
333	0.50	2.9	986.0	6.09

The LNG system implementation in the simulation model is shown in Figure 6.6. An example of simulation results with LNG vaporization heat extraction for one engine-generator set is shown in Figure 6.7.

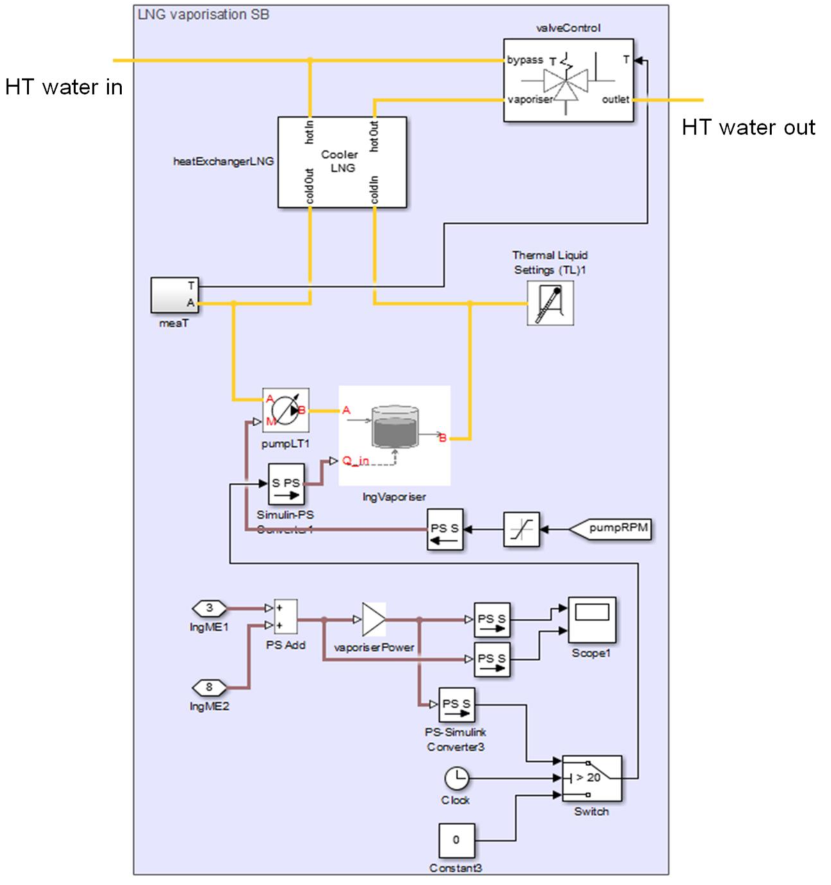

Figure 6.6. LNG evaporation system in the simulation model.

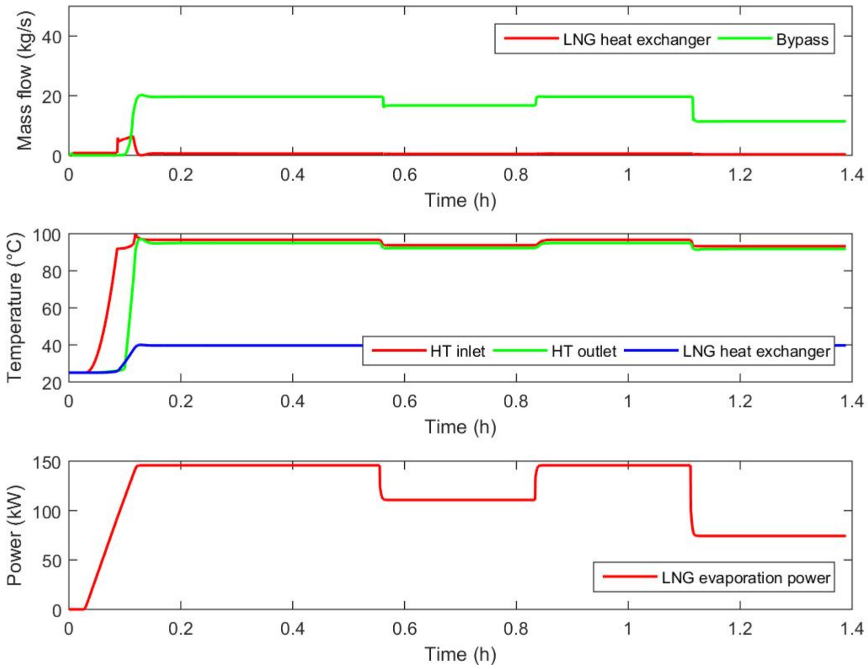

Figure 6.7. The example results of the LNG vaporization using heat extracted for one DG set.

6.2.3 Steam system

The steam production onboard the case ship is done primarily using natural-circulation exhaust gas boilers (EGBs), using an oil fired boiler (OFB) only when more steam than what EGBs can generate is needed. The main steam consumers onboard are deck device de-icing and hotel consumers. Main components in the actual steam system are hotwell, steam drum, three exhaust gas economizers (EGEs), OFB, and surplus condenser. As seen Figure 6.8, a pump drives water from the hotwell to the steam drum. Water in the steam drum is fed into the EGEs using natural circulation. Steam from the steam drum at the pressure of 7 bar is fed into consumers. The water level in the drum is kept at a certain level by controlling the feed water flow rate under varying steam consumption conditions. This feed water control also regulates the pressure in the drum. The rise of the pressure in the steam drum is controlled by feeding the excess steam to the surplus condenser, which extracts the heat into seawater cooling circuit.

The dynamic model of the steam system is based on the article of drum boiler dynamics [4]. The system is described using six state variables, six inputs and written using six differential-algebraic equations. State variables of the system model are the drum pressure, total water volume in the system, volume of steam in the drum and mass fraction of the steam in the three risers. The simulation model of the boiler system is implemented as a scripted system function (S-function) in Simulink. The calculated heat flow from the exhaust gas, the steam flow from the boiler, and the feed water flow into the boiler are input values to the boiler S-function. As an output, the S-function gives the pressure and the water level in the drum as well as the mass flows of water and steam coming out of the riser. The simulation model of the steam system is restricted to the boiler system only while the heating of the hot well tank for the feed water is left out.

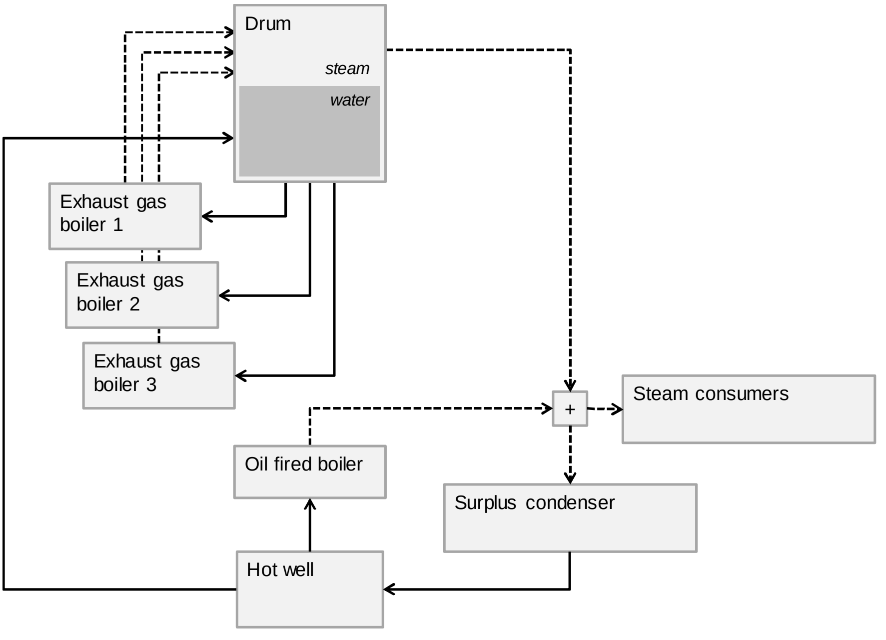

Figure 6.8. The schematic system diagram of the steam system.

The heat flow from the exhaust gas to the boiler is determined by the boiler manufacturer Alfa Laval Aalborg. Starting from the boiler dimensions and heat exchanger area, the heat transfer coefficients from exhaust gas to the steam boiler and further to the water are defined. The heat transfer is dependent on the exhaust gas and water velocities and densities. The temperature decrease of the exhaust gas in the evaporator is defined using the heat transfer coefficient, inlet temperatures of the exhaust gas and water, and heat flows of the exhaust gas and water. Using the temperature difference, the mass flow and the specific heat of the exhaust gas, the evaporator thermal power can be defined.

The boiler model is first tuned for the biggest boiler. Exhaust gas inlet and outlet temperatures and mass flow were set according to the boiler data sheet. After that, boiler dimensions and parameters are determined according to Alfa Laval Aalborg theory. The same parameter tuning procedure is applied for other boilers as well. Boiler data needed for the simulation model is shown in Table 6.2.

Table 6.2. Boiler data sheet information used in the modelling.

Parameter	Boiler 1 / ME3	Boiler 2 / ME4	Boiler 3 / AE
Mass flow (kg/s)	9.1	6.8	2.1
Temp in (°C)	380	380	365
Temp out (°C)	244	244	212

An example of the boiler simulation model is shown in Figure 6.9. At the instant 300 s, engines starting with full load causes the rise in the drum pressure. In the beginning, the steam consumption is zero and all produced steam is fed to the surplus condenser. At the instant 500 s, steam consumption starts to increase and the amount of condensed steam decreases. After the instant 700 s, the needed steam amount exceeds the capacity of the steam boiler and the extra amount is supplied using the OFB.

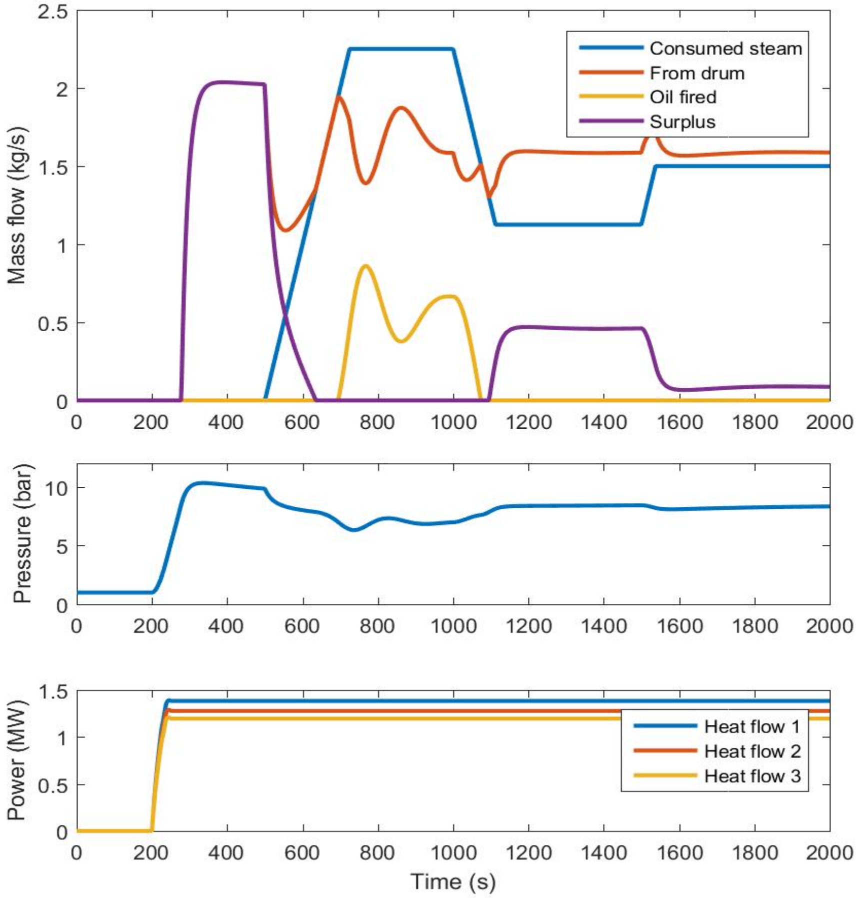

Figure 6.9. An example of simulation results of the boiler system.

6.3 Baseline simulations

Before energy saving scenario implementation, the baseline simulation model is tested under six operation modes: (1) navigation at open sea with full speed, (2) navigation at open sea with 12 knots speed, (3) manoeuvring, (4) icebreaking, (5) harbour and (6) standby at sea. The testing scenario is presented in Figure 6.10, including different operation modes, engine numbers and loads, and power output.

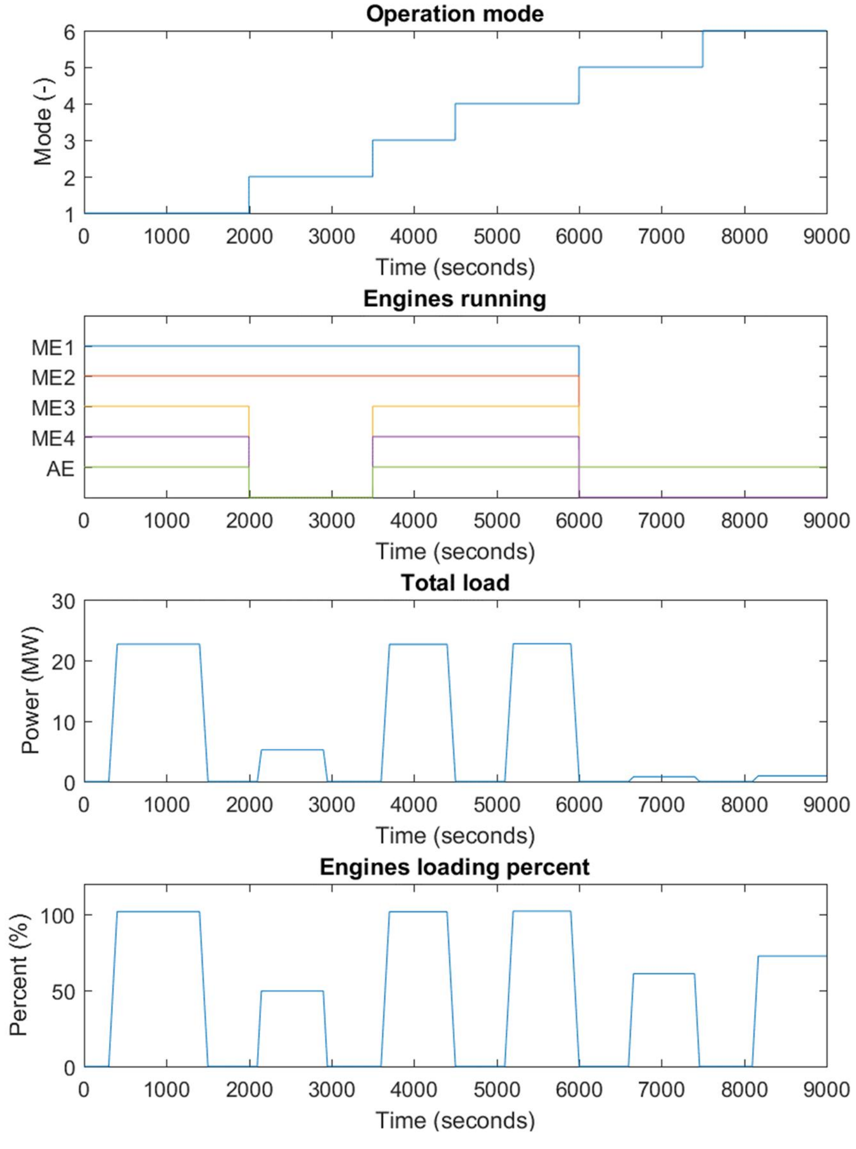

Figure 6.10. The testing scenario of the baseline simulation model.

As the examples, based on the presented testing scenario, some simulation results of the fresh water cooling circuit of the main engines ME1 and ME2 and the seawater cooling circuit are shown in Figure 6.11 and Figure 6.12, respectively.

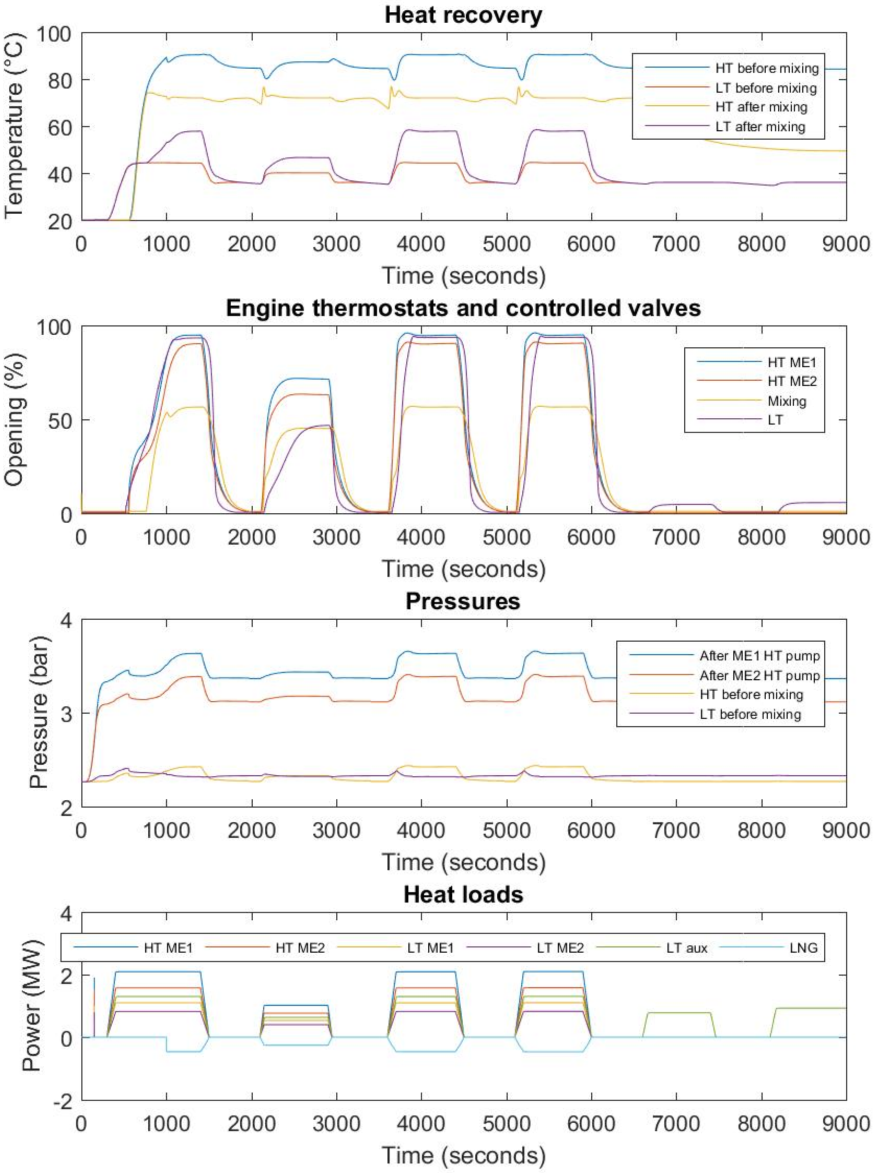

Figure 6.11. The example results of the fresh water cooling circuit of the main engines ME1 and ME2, including temperatures at heat recovery system, openings of engine thermostats and controlled valves, pressures in the system, and heat loads.

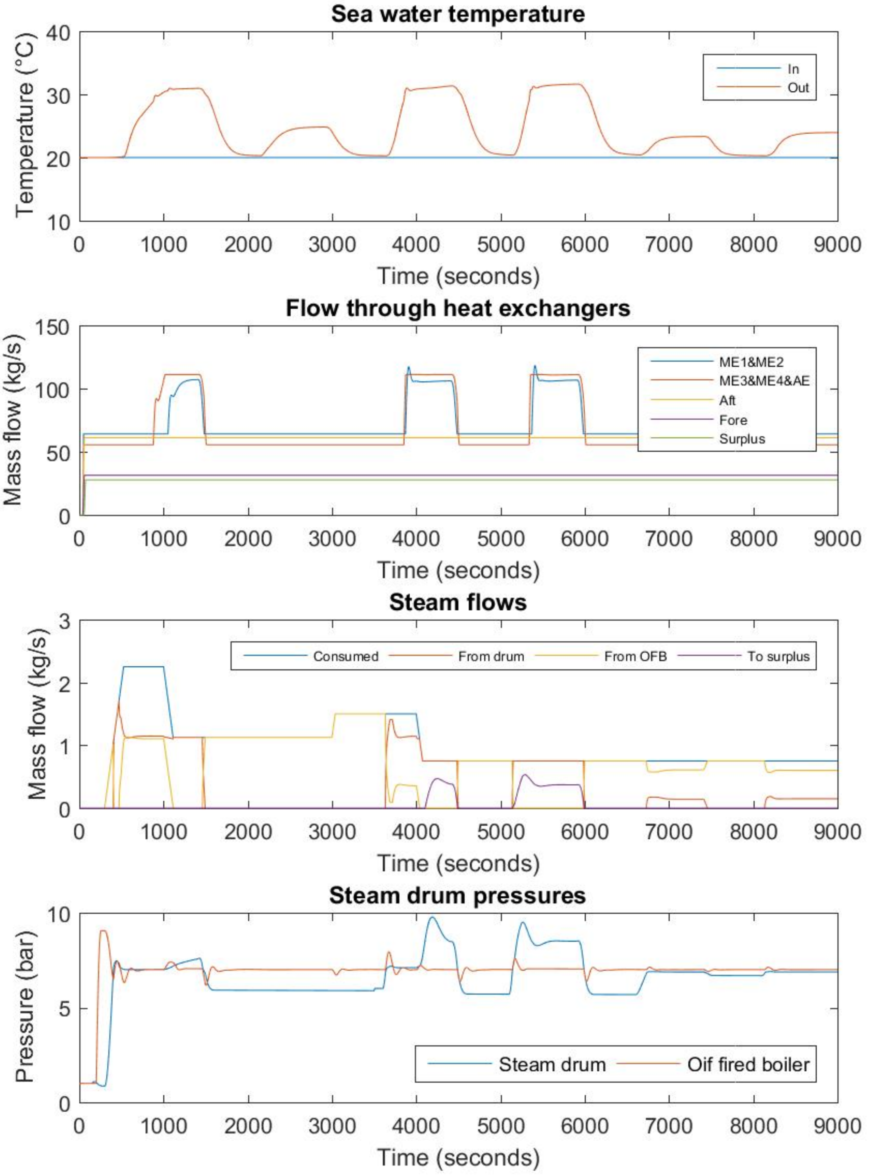

Figure 6.12. The example results of the seawater cooling circuit, including temperatures, mass flows, and pressures.

6.4 Energy saving scenarios

In this section, several energy saving scenarios are implemented into the baseline simulation model to evaluate their potential for improving ship energy efficiency.

6.4.1 Operating profiles

The operating profile is constructed from measurement data of a similar icebreaker, which shows the high variation of its electric propulsion. As it can be seen in Figure 6.13, the power consumption of the electric systems for one typical working day of an icebreaker is constructed and used as the input to the model.

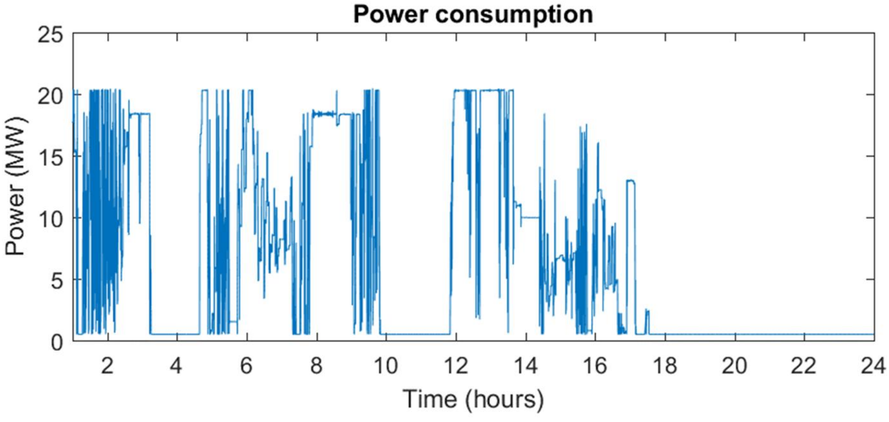

Figure 6.13. Total electric power demand during one "icebreaker working day".

As an example, simulation results of the fresh water cooling circuit of the main engine ME1 with the dynamic loading are shown in Figure 6.14. It can be seen that the fast transients in the engine power are filtered somewhat in the valve opening signals and even more in the outlet temperature of the seawater circuit.

6.4.2 Variable speed pumps

In this section, the power consumption of seawater circulating pump is studied to find the saving potential and margins regarding the seawater mass flow. The studied seawater circuit is by default equipped with variable speed pumps. The control region is 50 - 100% of the nominal flow rate. When the lower limit is the seawater pump speed is met, the heat exchanger control valve in the LT circuit is adjusted to decrease the mass flow rate from the LT circuit to the heat exchanger.

The required pumping power is estimated by using equation

$$P = \frac{\dot{m}}{\rho}\frac{\Delta p}{\eta}, \tag{6.1}$$

where $\dot{m}$ is mass flow through the pump, ρ is density of the fluid, Δp is pressure loss over the pump, and η is the pump efficiency, which is assumed to be constant 70 %. The pressure loss over pump comes from the flow resistance through the heat exchanger, which is taken from the heat exchanger data sheet.

The simulated results of the power consumption of one seawater pump in the seawater circuit is shown in Figure 6.15. During the simulation process, the heat load of the LT circuit comes from two main engines, which are kept running at constant load of 100% (solid line) and 50% (dashed line) from the instant 1000 s onwards. Besides, the seawater temperature varies as a function of time from 25 °C to 1 °C. It can be seen that, when seawater gets colder, the seawater mass flow through the heat exchanger decreases. When the engine load is 100%, the seawater pump reaches the lowest possible output at seawater temperature of 5 °C. At the same time, the estimated seawater pumping power is 5 kW. When the engine load is 50%, the seawater pump reaches its lowest possible output with seawater temperature of 20 °C.

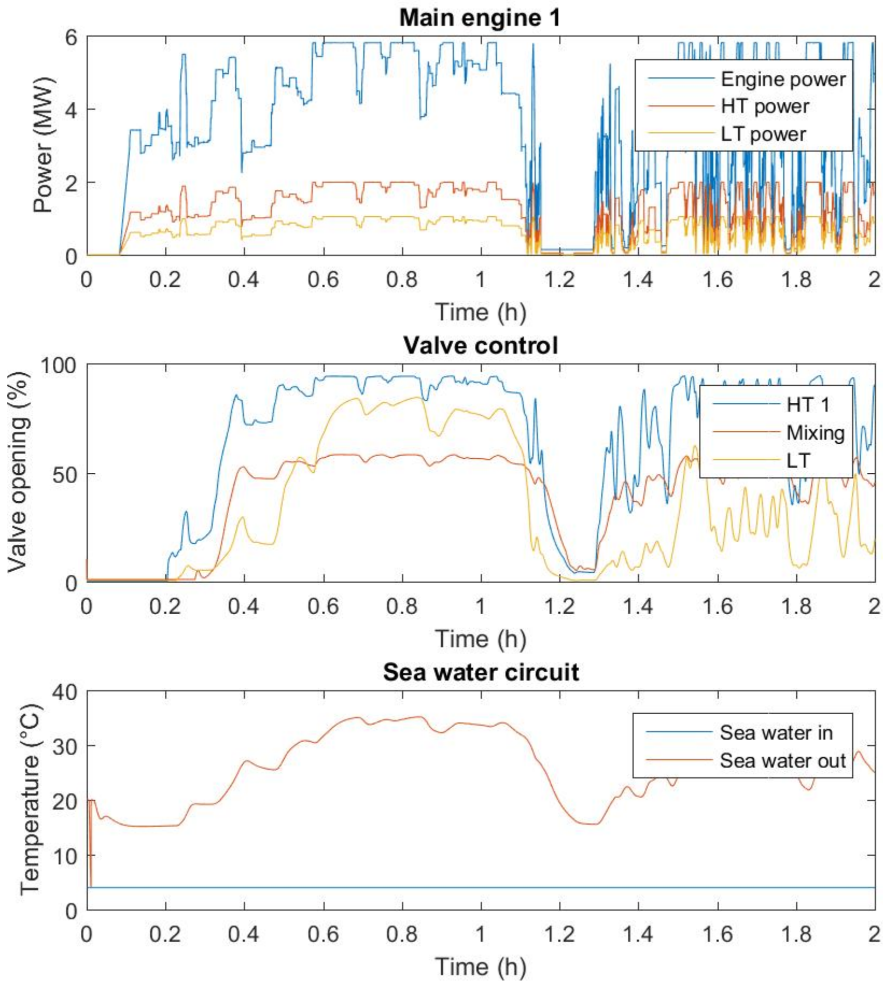

Figure 6.14. Example results for the main engine cooling circuit and valve controls with the transient load data.

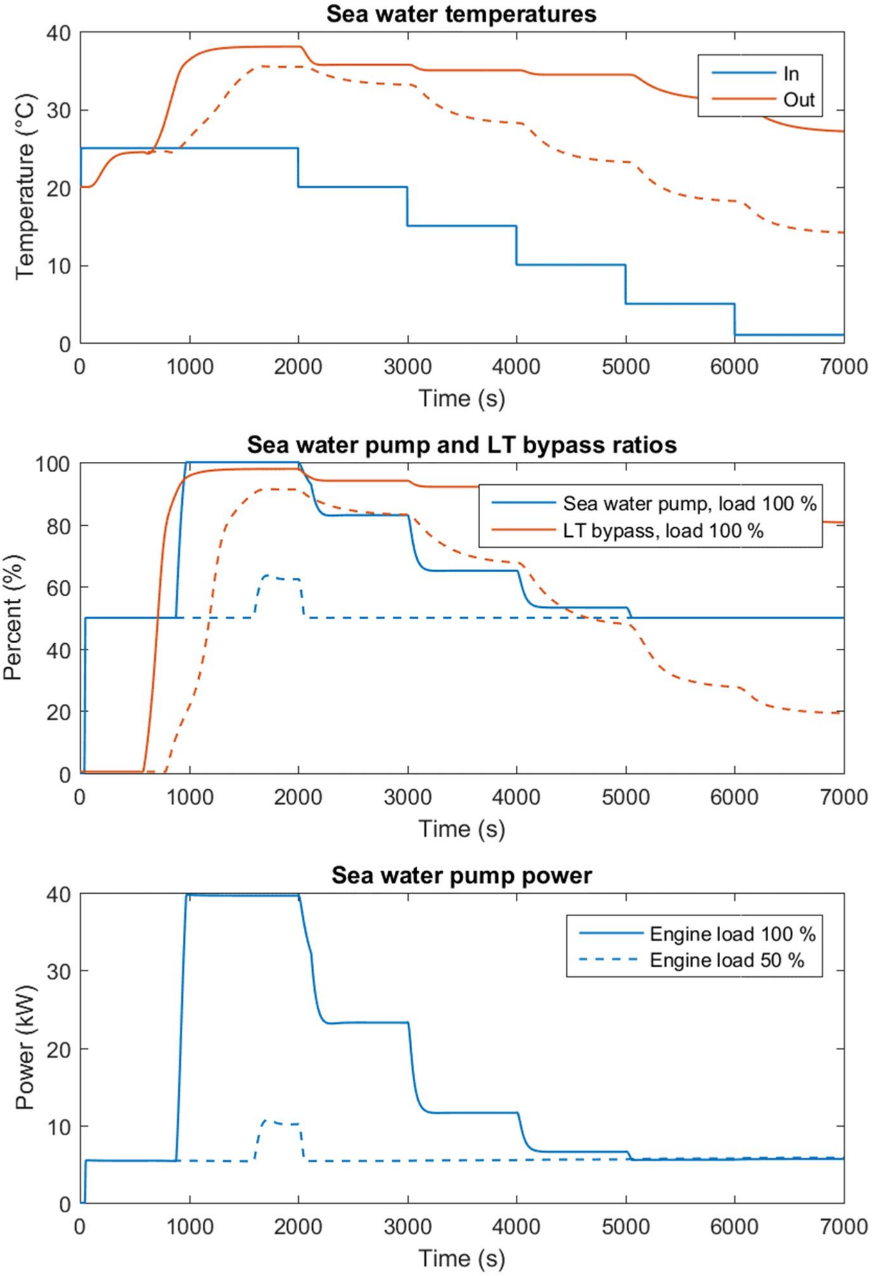

Figure 6.15. The simulation results of variable speed pump functionality with different seawater temperatures and engine loads.

6.4.3 Generator load control and battery system

In this section, we investigate how the electric load division between generators and a battery system could affect ship energy efficiency during operation. The energy saving potential comes from the fact that, with LNG fuel, the higher the engine load, the lower the specific fuel consumption. If the load of each generator could be set individually, in optimal situation only one engine at a time would be running at a partial load while others run at a full load. The situation could be even better if the electric network is further supplied by a battery system, which could balance the power consumption and generation so that the partial load either could be increased to store the excess energy or reduced using the stored energy. As one scenario shows in Figure 6.16, when the power demand increases, the main engines ME1, ME2, and ME3 are run at full load and the battery system supplies the partial load. When the battery state of charge (SoC) falls below 10%, the main engine ME4 is run at full load to charge the battery until the battery's SoC reaches 90%. Using this approach, all the main engines are driven all the time at full load and its best efficiency.

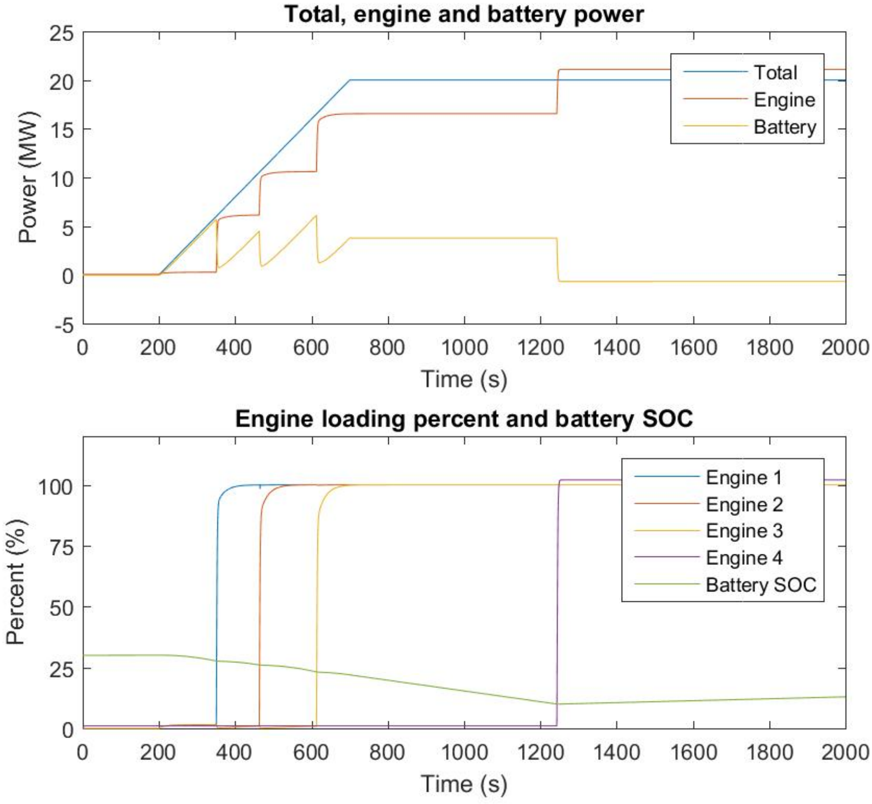

Figure 6.16. The example results of the combined energy saving scenario, generator load control and battery system.

The presented energy saving scenario is further tested using the operating profiles representing one “icebreaker working day”. As one example, the battery system is configured with a capacity of 5 MWh and an efficiency factor of 0.97. Battery dynamics or needed power conversions between AC and DC grids are not taken into account.

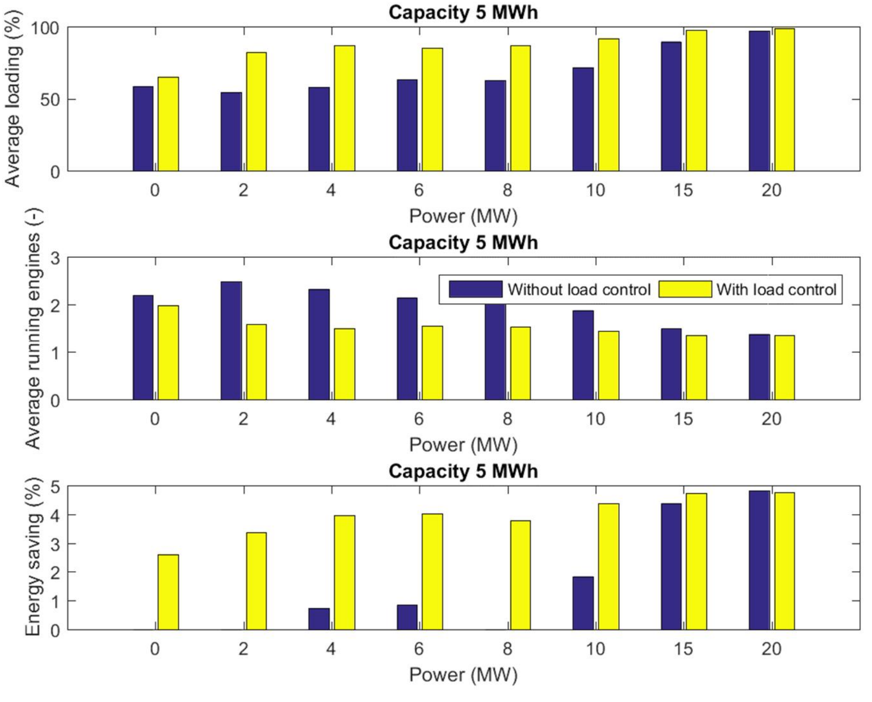

Figure 6.17. Energy saving potential, when battery and load control is applied.

Figure 6.17 shows the comparison of the results between different scenarios, specifically with or without the battery system combined with or without the load control. It can be seen that when the load control is applied the average load percentage is about 20% higher and the number of running engines about half units lower in comparison to the situation where load control is not applied. This gives energy saving from 2.6% to 4.8% in comparison to the baseline configuration.

6.5 Discussion

A dynamic simulation model of ship energy systems can be used easily to simulate and study its overall energy efficiency during design and operation, for design, optimisation and evaluation of various energy saving scenarios. It gives an efficient way to study the effects of design parameters on the behaviour of large physical systems, for example, for dimensioning and optimisation of system components or energy management systems, and verification of concept designs for next vessels. Using real operation data, it can be validated, updated and further used to study the system operation efficiency in different operating conditions and operating modes.

Biggest uncertainty at the cooling system model is the restriction of the pipes, which has an effect on the pressure levels at the circuit, and time constants of the system regarding the heat transfer rate between different media and control of valves to get the dynamics of the cooling system at right magnitude. Besides, the simulation of large physical systems may have numerical troubles. When the fluid compressibility is taken into account, it results in a stiff system model, which mathematically can be very different and slow to solve.

6.5.1 Variable speed pumps

Seawater circulating pumps are the most significant pump consumers of the ship, while HT and LT pumps are engine driven. Using the developed baseline simulation model, the idea was to evaluate the energy saving potential if the pumping power is further adjusted according to the seawater temperature. However, according to the simulation results, the system dimensioning and the control logic of the variable speed drives have been done energy efficiently. The seawater pump runs at low flowrate when the engine load and/or seawater temperature are low. At the lowest pump control value, the pumping power is 6 kW for one seawater pump. Hence, the saving potential is negligible even if one seawater pump unit is replaced with two smaller units.

6.5.2 Generator load control and battery system

Since the LNG-fuelled engine is most energy efficient when running at the highest load, we use the developed baseline model to test how potential in energy savings the combined load control and battery system could be. According to the simulation results, the easiest and biggest potential way is load control. By applying the control without battery, energy saving of 2.6% can be achieved for defined icebreaker working day. When the battery system is applied, the energy saving could potentially increase up to 4% with moderate battery capacity of 4 MW. The maximum saving of 4.8% can be achieved if the battery system works as the only prime mover.

References

[1] Wikipedia. Polaris (icebreaker). Available: https://en.wikipedia.org/wiki/Polaris_(icebreaker)

[2] Mathworks. Simscape manual. 2015.

[3] Wärtsilä 34DF Product guide. 180 p. Vaasa, 2015.

[4] Drum-boiler dynamics. Åström, K. J., Bell, R. D. Automatica, 2000. Pp. 363-378.

7. Pinch analysis of a case bulk carrier ship and integration of a steam Rankine cycle[1]

Juha Vanttola, Maunu Kuosa
Aalto University

Summary: The main purpose of this chapter was to study the waste heat recovery (WHR) potential of a case ship, bulk carrier B.Delta37, when operated in ISO-ambient conditions, in order to find potential ways to improve its energy efficiency. For that purpose, the pinch analysis method had been adopted to evaluate the theoretical "Maximum Energy Recovery" (MER) of the case vessel, which indicates the maximum amount of energy that can be possibly recovered with a heat exchanger network from the thermodynamic processes of the case ship, without violating the second-law of thermodynamics. As a result, by efficiently utilizing the internal waste heat recovery of the case ship, we achieved the theoretical minimum demand of external heating and cooling utilities for operating the ship under ISO conditions, which are currently supported by burning fuel oil in an oil fired boiler (for heating) and circulating big amount of seawater (for cooling). Besides, we also evaluated the possibly integration of a steam Rankine cycle to the renewed MER-heat exchanger network of the case vessel for additional electricity production. The obtained electricity was used in reducing the use of auxiliary engines, used in electricity generation, hence to further improve the overall operational energy efficiency of the case ship.

As a result, by optimal arrangement, the developed theoretical MER-heat exchanger network could potentially recover the waste heat to support all of the heating requirements and hence reduce the need for an oil-fired boiler on the case ship. The annual potential could reach approximately 11 metric tons of fuel savings for the case ship. Furthermore, the integrated steam Rankine cycle would further reduce the energy consumption of the auxiliary engine demand at least by 181 metric tons annually, which represents 18.8% fuel savings of the auxiliary engines, and 4.7% of the fuel consumption of the whole ship.

[1] This chapter is excerpted and revised from the SET research project deliverable with the same name.

7.1 Introduction

The sea bound cargo transportation is considered one of the most energy efficient transportation methods. The extensive development work of the marine diesels has yielded very energy efficient engines for the sea transportation; the most efficient slow speed engines can operate at higher than 50% efficiency, which means that more than half of the fuel's chemical energy content is put to work. However, the other side is that half of the fuel's work potential is lost as waste heat.

This study aims to provide some insight into how much of the engine waste heat is theoretically convertible to shaft power. To achieve that, the pinch analysis method is utilized to analyse the waste heat sources and various heat consumers on ships. As a result, the "maximum energy recovery" (MER) of the case ship is derived, which yields the theoretical minimum demand for external heating and cooling utilities for the case vessel by organizing the heat exchanger network efficiently. The second target is to study the potential of a steam Rankine cycle (SRC), integrated with the MER-heat exchanger network. The SRC is used to generate electricity from the waste heat sources and lessen the load of auxiliary engines (AEs) that currently consume slightly over three-fourths of the vessel's fuel. [1]

7.2 Pinch analysis and grand composite curve

Pinch analysis, as an early-stage design optimization tool for process integration, has received a firm placeholder amongst optimization methods in process industry. It has been suggested that pinch analysis is one of the most important contributions to the development of the process industries. Besides, pinch analysis is one of the rare optimization tools for process industry that takes the second-law of thermodynamics into account in process design. The result of a correctly implemented pinch analysis is a thermodynamically or economically optimized heat exchange network (HEN). In the thermodynamically optimized system, the internal heat recovery of a process is maximized and the need of external heating utility, such as additional steam production or cooling seawater, is minimized. In pinch analysis, the heat rejecters and heat consumers of the process are separated into "hot" and "cold" streams. The "hot" and "cold" process streams are the streams with heat surplus and heat deficiency respectively. In other words, the cold streams require heating and the hot streams require cooling. In pinch analysis, these streams are matched in such a way that the hot streams provide heating to the cold streams, and the cold streams provide cooling to the hot streams without the need of external heating or cooling utility, such as high-pressure steam, or cooling water. The result of the analysis is a network of heat exchangers that minimizes the demand for the external heating and cooling resources. The process streams, such as, "hot" charge air, or a "cold" un-preheated fuel, are divided into their own separate temperature intervals. In the analysis, the heat is "cascaded" from higher temperature to colder, as shown in Figure 7.1.

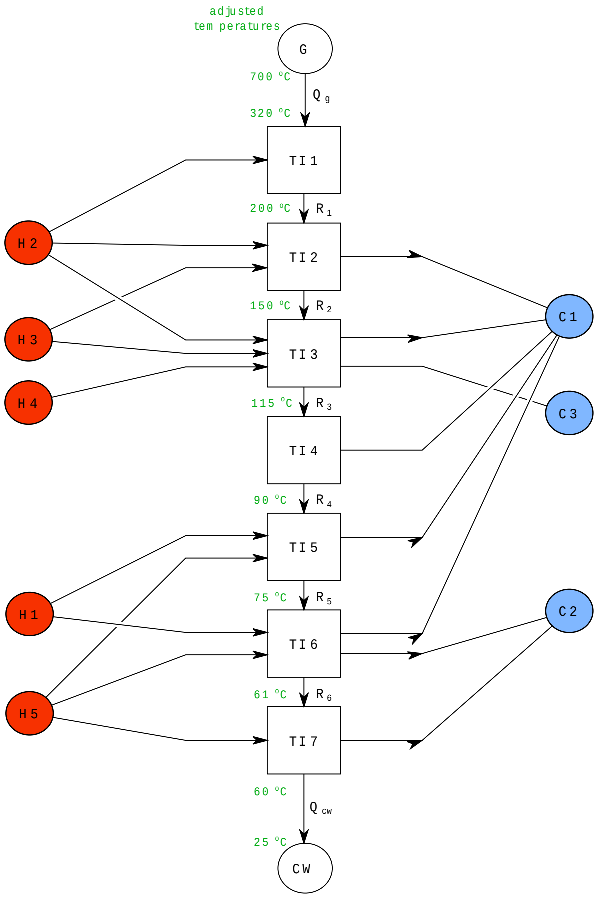

Figure 7.1. A heat cascade example of an integrated process with five heat sources (red) and three heat sinks (blue), having seven temperature intervals TI1–TI7.

The temperature intervals, as seen in the figure, are defined by the "supply" and "target" temperatures of the streams. The supply temperature is the initial temperature of a stream, and the target temperature is the end temperature of the stream after heating or cooling. If the target temperature is higher than the supply temperature, the stream is cold, i.e. it requires heating, and if the target temperature is lower than the supply temperature, then the stream is hot, i.e. it requires cooling.

In pinch analysis a "minimum approach temperature", ΔT_{min}, is assigned for the process streams. This represents the minimum temperature difference between hot and cold stream in a heat exchanger. The minimum approach temperature represents the smallest temperature difference between hot and cold streams. Either a global ΔT_{min} can be chosen for all of the process streams (Figure 7.2a), or alternatively, it can be composed individually for different fluids that have different heat transferring coefficients, such as gases or liquids (Figure 7.2b).

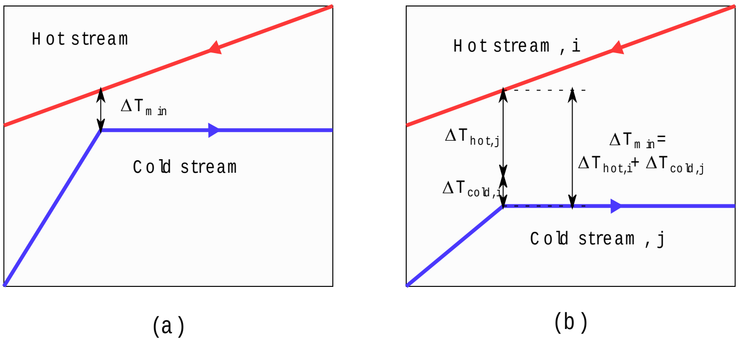

Figure 7.2. (a) Global ΔT_{min} for all the streams in the network; (b) The individual contribution of hot stream, *i*, and cold stream, *j*, to ΔT_{min}.

Furthermore, pinch analysis yields a very compressed and insightful picture of the heat exchanges as a Grand Composite Curve (GCC). The GCC shows the heat surplus in each temperature interval on the horizontal axis, and the "adjusted temperature" on the vertical axis. A pinch point is normally identified, where the heat from hotter streams cannot be cascaded to the colder streams. This is the point where the GCC touches the vertical axis, and the heat sources from the higher temperature process streams are depleted. The heat exchangers associated with the pinch point are referred as the "pinch heat exchangers".

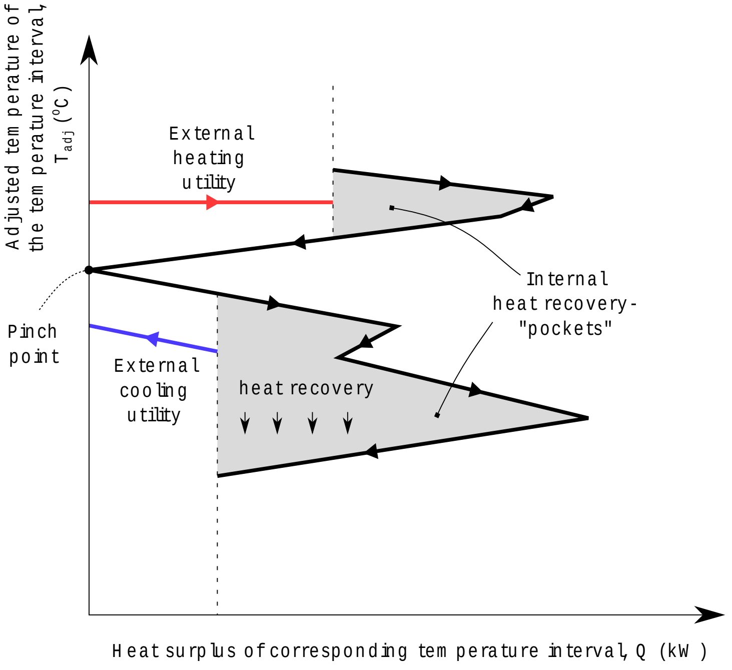

Figure 7.3. Illustration of a grand composite curve.

7.3 Steam Rankine cycle model

Steam Rankine cycle (SRC) is a Rankine cycle using water/steam as working fluid, which converts high-grade heat into mechanical work while rejecting low-grade heat into a heat sink. Figure 7.4 shows a simple model of a steam Rankine cycle used onboard a ship. The cycle begins when superheated steam is fed into the steam turbine (state point 1 in Figure 7.4). A portion of the steam's potential energy is converted into shaft power for the generator, shown as G in Figure 7.4. The expanded low-pressure steam is discharged from the turbine and condensed in a seawater condenser (state points 2–3). The low-pressure steam is then fed back into the hotwell and further pumped into the boiler using a feedwater pump (state points 3–6). In the boiler, the feedwater is heated, evaporated and further superheated before feeding into the steam turbine again (state points 5-6-1).

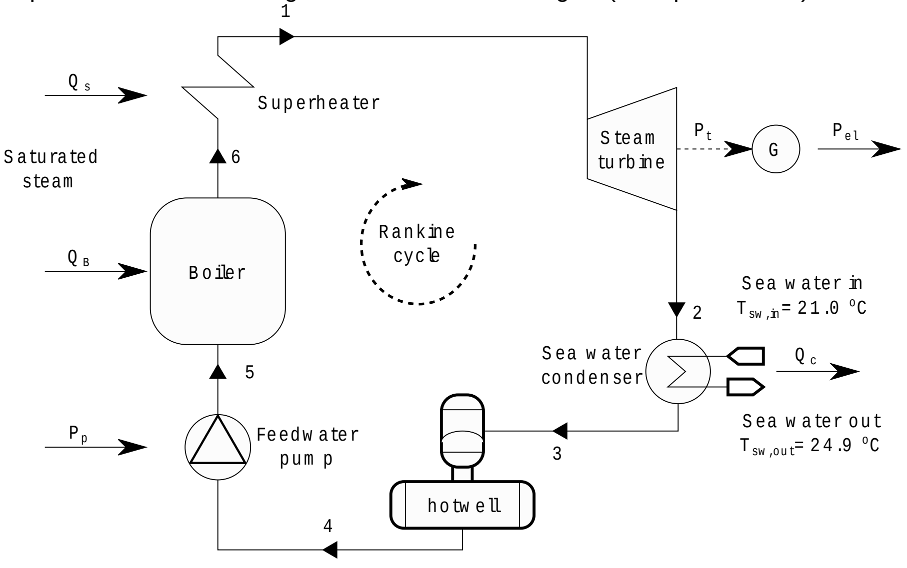

Figure 7.4. An example of steam Rankine cycle used onboard a ship.

The turbine and pump isentropic efficiencies, η_T and η_p, are assumed as follows,

$$\eta_T = \eta_p = 0.7 \tag{7.1}$$

which represent typical efficiency values for steam turbines and feedwater pumps. Based on energy balance of the system, for enthalpy change in the pump, we have

$$h_5 = h_4 + \frac{h_{5s} - h_4}{\eta_p} \tag{7.2}$$

where h_4, h_5 and h_{5s} represent the enthalpies of circulation water at the inlet and outlet of the feedwater pump (Figure 7.4). $h_{5s} - h_4$ represents the enthalpy change associated with of ideal pumping process. h_{5s} can be derived, using REFPROP fluid properties database, [2] as

$$h_{5s} = h_{water}(s_4, p_B)\ , \tag{7.3}$$

where s_4 is the entropy of circulation water at state point 4 and p_B is the boiler pressure. Entropy of water at sate point 4 is calculated with REFPROP as

$$h_4 = s_{water}{}'(T_C)\ , \tag{7.4}$$

where $s_{water}{}'(T_C)$ represents saturated liquid water at condenser temperature.The liquid pump's shaft power and electrical power consumption, P_p and $P_{p,el}$ can now be calculated as

$$\begin{cases} P_p = \dot{m}_{st}(h_5 - h_4) \\ P_{p,el} = \dfrac{P_p}{\eta_{el}} \end{cases}\ , \tag{7.5}$$

where η_{el} is the efficiency of the pump's motor (as well as the steam turbine's generator), for which we assign a following commonly used value η_{el} = 0.95.

Furthermore, the heat exchange in the boiler Q_B composes of desubcooling and evaporation of the circulation water, which can be calculated as

$$Q_B = \dot{m}_{st}(h_6 - h_5), \tag{7.6}$$

where $\dot{m}_{st}$ is the mass flow rate of the steam, and h_6 is the enthalpy of saturated water vapour, derived with REFPROP as follows

$$h_6 = h_{water}{}''(p_B), \tag{7.7}$$

where $h_{water}{}''(p_B)$ is the enthalpy of saturated water vapor at boiler pressure, p_B.

For the superheating of the steam (state points 6 and 1 in Figure 7.4), represented by the heat flow Q_s, applies

$$Q_s = \dot{m}_{st}(h_1 - h_6), \tag{7.8}$$

where h_1, the enthalpy of the vapour, is governed by the boiler pressure, p_B, and the superheating temperature, T_1. From REFPROP we have

$$h_1 = h_{vapour}(p_b, T_1) \tag{7.9}$$

The enthalpy change in the turbine can be calculated in a similar manner as in the liquid pump. For the isentropic efficiency of the steam turbine, applies

$$h_2 = h_1 - 0.7(h_{2s} - h_1)\ , \tag{7.10}$$

where h_{2s} and h_2 are the enthalpies of the circulation water in isentropic and the real process, respectively. h_{2s} is derived from REFPROP using p_C and the entropy of s_1 as input values

$$h_{2s} = h_{water}(s_1, p_C)\ , \tag{7.11}$$

The entropy s_1 is derived from REFPROP using the superheating temperature and the enthalpy h_1 as input values

$$s_1 = s_{water}(T_S, h_1)\ . \tag{7.12}$$

Furthermore, the boiler pressure p_c is calculated from the condensing temperature T_C of the circulation at condenser with REFPROP

$$p_c = T_{water}''(T_C). \tag{7.13}$$

The condenser is cooled with 21°C seawater that is heated up in the condenser by 3.9°C. As the pinch temperature of the boiler is selected as 2°C, hence for T_C we have

$$T_C = (21 + 3.9 + 2)℃ = 26.9\ ℃. \tag{7.14}$$

The steam turbine shaft power production P_T can now be calculated as

$$P_T = \dot{m}_{st}(h_5 - h_4) \tag{7.15}$$

The steam turbine's electrical power output $P_{T,el}$ is calculated as

$$P_{T,el} = P_T \eta_{el}\ , \tag{7.16}$$

The net power output of the SRC is thus calculated as

$$P_{SRC,el} = P_{T,el} - P_{P,el}\ . \tag{7.17}$$

Moreover, some simplifications were done for the SRC model. The heat and pressure losses in the circulation water pipelines of SRC are not taken into account.

$$\begin{cases} Q_B + Q_S = P_T + Q_C \\ p_5 = p_6 = p_1 = p_B \\ p_2 = p_3 = p_4 = p_C \end{cases} \tag{7.18}$$

The adjusted temperature absorbing profile of the SRC is merely "fitted" under the grand composite curve in such a way that the net power output of the device is maximized. The fitting of the SRC below the GCC, as seen in Figure 7.5, depends mainly on three aspects. Firstly, the mass flow rate of the steam $\dot{m}_{st}$ in the SRC, defines the "width" of the SRC cycle in the direction of the horizontal axis of below the grand composite curve. Secondly the boiling temperature, T_6, defined by the boiler pressure p_6, defines the height of the "knee" of the SRC. Finally, the superheating temperature T_1 defines the peak temperature of the SRC. As a common practice, a suggested minimum of the steam quality at the low-pressure side of the turbine is x_2 = 0.85 in order to avoid possible corroding effect on the turbine blades. [3]

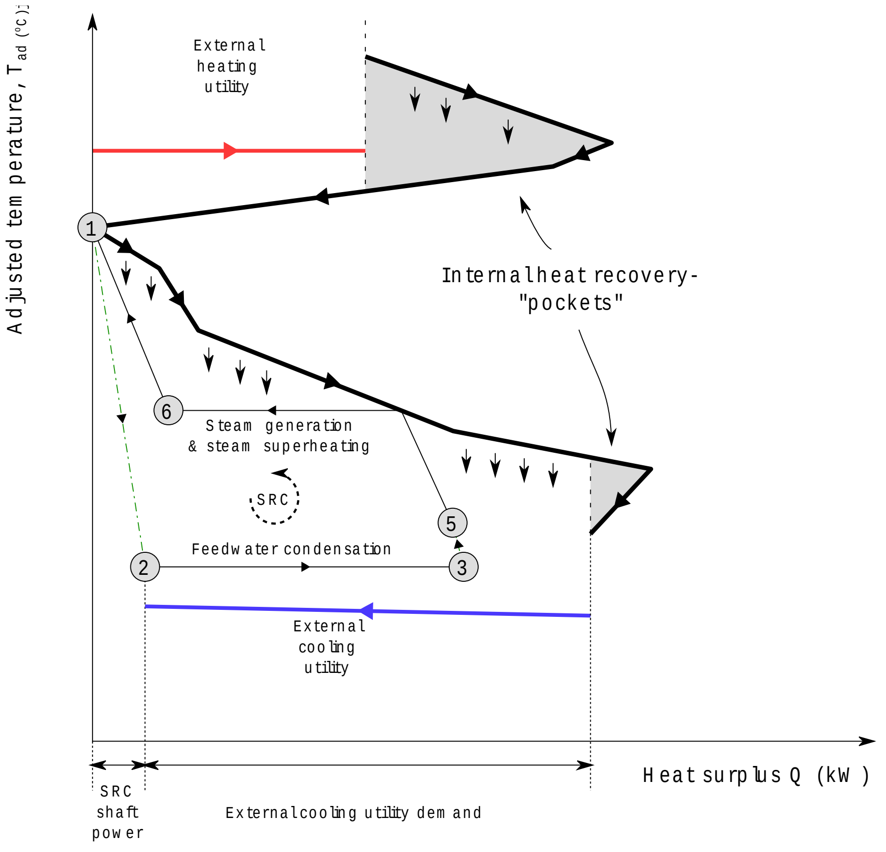

Figure 7.5. Illustration of a "Maximum Energy Recovery" (MER) network's Grand composite curve, obtained by pinch analysis, along with an integrated steam Rankine cycle (SRC)

The process integration can be considered as a multi-variable optimization task,

$$\begin{aligned}
&\max P_{SRC,el}\,(T_1, p_6, \dot{m}_{st}) \\
&s.t.,\ T_{hot,i} - T_i \geq \Delta T_{min} \\
&\qquad \dot{m}_{st} \geq 0 \\
&\qquad p_3 < p_5 \\
&\qquad x_2 \geq 0.85 \\
&\qquad p_2 = p_3 = p_C \\
&\qquad p_5 = p_6 = p_1 = p_B
\end{aligned} \tag{7.19}$$

where $T_{hot,i}$ and T_i represent the temperature of a hot stream, which rejects heat to the circulation water of the SRC, which has temperature of T_i through the wall of a heat exchanger at location i of the cycle. p_c and p_b represent the boiler and evaporator pressures.

7.4 Case study

In this section, using pinch analysis method, a theoretical study is carried out to explore the WHR potential of the SRC onboard a case bulker carrier B.Delta 37, briefly introduced in Chapter 1.4.

7.4.1 Case ship operating profiles

The study is conducted under ISO conditions, where seawater temperature is 21°C air temperature is 25°C. The typical operation profile of the case ship is given as cumulative hours of operation on a single operation mode, shown in Table 7.1.

Table 7.1. Waste heat data at different operating modes under ISO conditions. [4]

	Yearly operation hours	Jacket water cooler	Lubrication oil cooler	Scavenge air cooler	AC chiller	Surplus steam dumping condenser	Auxiliary diesel engine	Other LT consumers	Central cooler
Dimensioning case, tropical conditions	-	950	600	1980	350	450	583	166	5662
ISO, 100% ME load	-	930	590	1940	150	10	465	66	4046
ISO, Scantling draught 74,4% ME load	1226,4	760	540	1490	150	10	465	66	3366
ISO, Scantling draught 39,0% ME load	1226,4	520	420	440	150	5	465	66	1951
ISO, Design draught NCR load 67,8% ME load	919,8	720	520	1220	150	5	465	66	3031
ISO, Design draught slow steaming 35,1% ME load	919,8	485	400	330	150	0	465	66	1781
ISO, Ballast draught 63,7% ME load	919,8	690	500	1100	150	0	465	66	2856
ISO, Ballast draught slow steaming 25,0% ME load	919,8	415	360	100	150	0	465	66	1441
ISO, Maneuvering	96	0	0	0	150	0	498	66	776
ISO, Harbour loading	936	0	0	0	150	0	1026	66	616
ISO, Harbour unloading	1176	0	0	0	150	0	1026	66	776
ISO, Harbour stand still	456	0	0	0	150	0	423	66	516

Onboard electricity consumption ranges from 200 kW, when the ship is mooring at the port, to the peak consumption of 900 kW, which is usually reached only when the electric manoeuvring thrusters or cargo machinery is used. The detailed electricity consumption profiles are shown in Table 7.2. Other than electric consumers, there is a large amount of heat consumers onboard the case ship. The total heat demand is about 770 kW at sea and 250 kW at harbour, respectively.

Table 7.2. Electricity consumers onboard the case ship. [4]

Sailing mode	Hours/year	Electricity demand (kW)
Scantling draught	1226.4	470
Scantling draught, slow steaming	1226.4	470
Design draught, NCR load	919.8	470
Design draught, slow steaming	919.8	470
Ballast draught	919.8	470
Ballast draught, slow steaming	919.8	470
Manoeuvring	105.12	670
Harbour loading	919.8	900
Harbour unloading	1156.32	900
Harbour standstill	446.76	294

Table 7.3. Heat consumers onboard the case ship. [4]

High temperature heat demand	Heat demand (sea) (kW)	Heat demand (harbour) (kW)	T_{supply} (°C)	T_{target} (°C)
Fuel heater for engines	21.83	1.75	98.0	145.0
HFO Service tank No.1	6.86	6.86	95.0	95.0
HFO Service tank No.2	4.97	4.97	95.0	95.0
ME Preheating unit	0.00	60.50	85.0	85.0
Separator heaters	14.41	1.15	65.0	98.0
HFO Settling tank No.1	3.84	-6.20	70.0	70.0
HFO Settling tank No.2	0.61	0.61	70.0	70.0
LO separator for ME	20.06	15.60	40.0	95.0
LO separator for AE's	7.83	15.66	40.0	95.0
HFO Storage tank	27.55	27.55	45.0	45.0
FO Overflow tank	0.80	0.80	45.0	45.0
AC and space heating	37.50	37.50	50.0	70.0
System oil tank	0.00	48.15	40.0	40.0
LO renovating tank	5.00	5.00	40.0	40.0
Sludge collecting tank	2.50	2.50	40.0	40.0
Waste oil tank	2.50	2.50	40.0	40.0
Leakage oil	4.46	4.46	40.0	40.0
Bilge water settling tank	15.00	15.00	40.0	40.0
Potable water system	14.58	14.58	25.0	55.0
Bilge holding tank	7.50	0.00	30.0	30.0
Freshwater generator	570	0.00	70	70
Total	767.8	258.9		

The main waste heat sources of the case ship are the main engine (ME) and auxiliary engines (AEs). Each engine has four waste heat sources that are the engine jacket cooler (JC), engine lubrication oil cooler (LOC), high temperature exhaust gases (EG) and the charge air cooler (CAC) as shown in Figure 7.6.

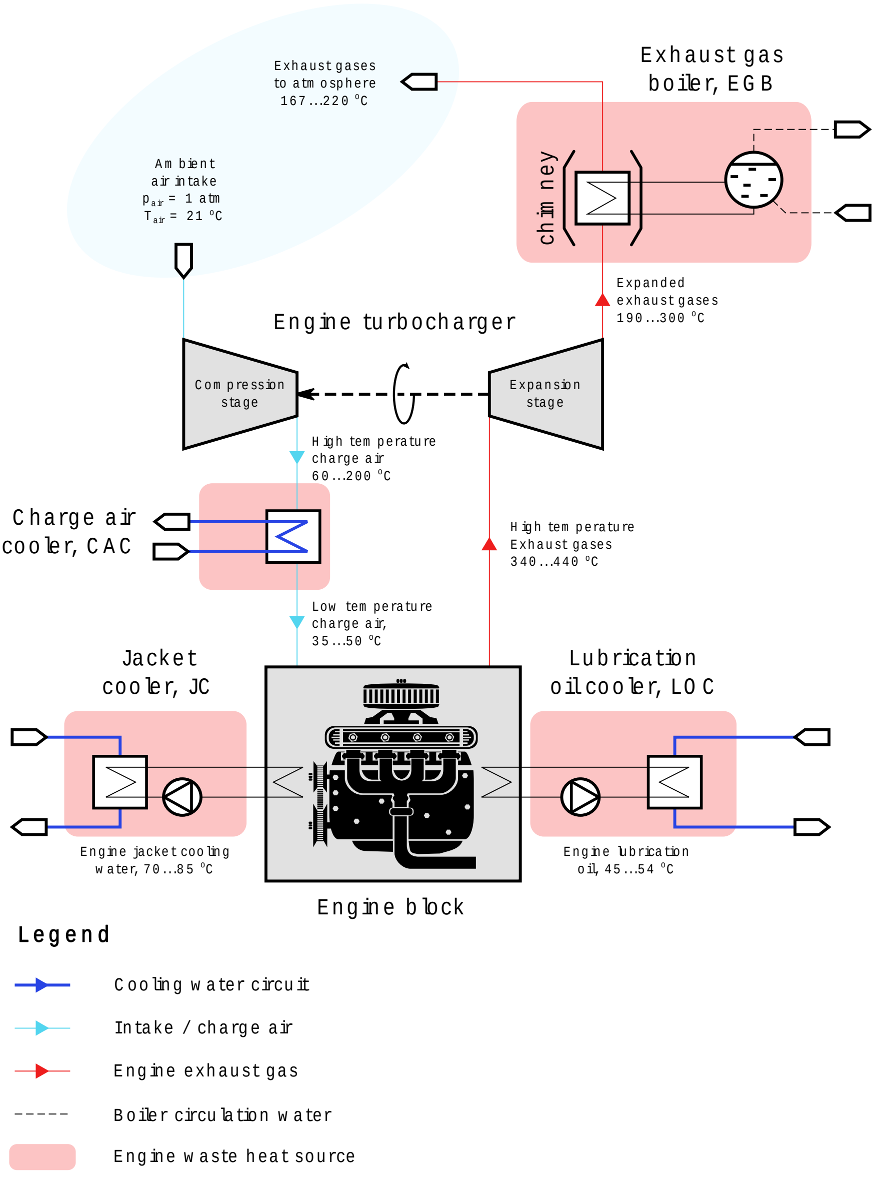

Figure 7.6. Four main engine waste heat sources: Charge air (CA), Jacket cooling (JC), Lubrication oil (LO) and hot exhaust gases (EG). [1]

7.4.2 Ship waste heat GCC with integrated SRC

In this section, we have used pinch analysis to explore the potential of SRC for WHR under nine operation modes of the case ship. Three pinch points (0ºC, 10ºC and 20ºC) were studied separately for each mode. Evidently, pinch point 0ºC in heat exchangers is not possible because it would require infinite amount of heat exchanger surface between hot and cold streams. It is merely used for illustrative purposes to show the ideal MER of the network. The SRC is also drafted (black) under each grand composite curve, as illustrated in Figure 7.5 and the state point value of the corresponding SRC's operation conditions are shown in Figure 7.4.

7.4.2.1 Scantling draught modes

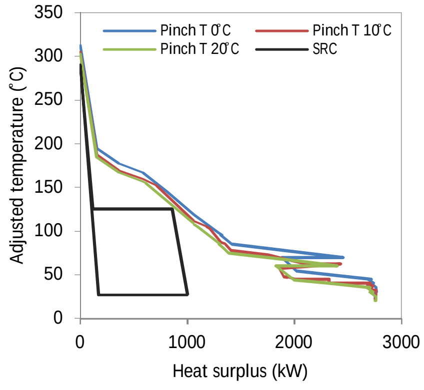

Figure 7.7. GCC – Scantling draught.

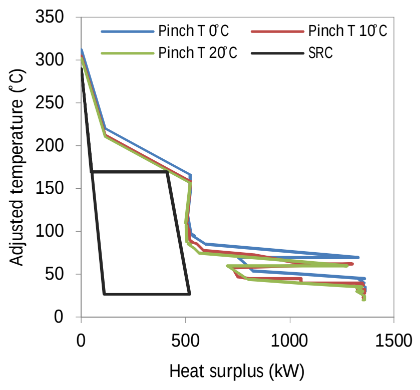

Figure 7.8. GCC – Scantling draught, slow steaming.

Figure 7.7 and Figure 7.8 show the grand composite curves of the case vessel's MER heat exchanger network under the modes of "scantling draught" and "scantling draught, slow steaming". It can be seen that there is no pinch point for either of the operation modes. The peaks in both GCC curves are between adjusted temperatures of about 200ºC and 300ºC, related to the auxiliary engine's exhaust gases. Hence, the SRC's steam can be superheated with the AE's exhaust gases to improve the steam quality at the low-pressure side of the SRC's steam turbine. As shown in Table 7.4, The WHR potential of SRC under "scantling draught" mode is 60% more than when running scantling draught under slow steaming.

Table 7.4. The WHR potential of SRC under two "scantling draught" modes.

Operation mode	$\dot{m}_{ST}$ (kg/h)	p_B (bar)	T_s (ºC)	$P_{SRC,el}$ (kW)	Quality, x_2
Scantling draught	1225	2.32	290	163	99.9%
Scantling draught, slow steaming	641	7.92	290	104	94.7%

7.4.2.2 Design draught modes

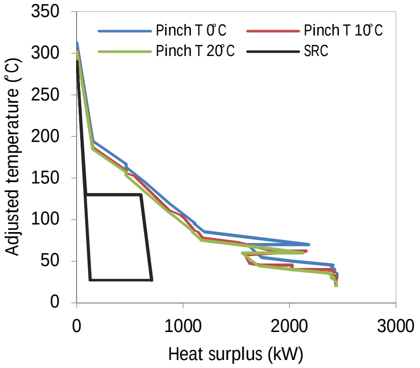

Figure 7.9. GCC – Design draught, NCR load

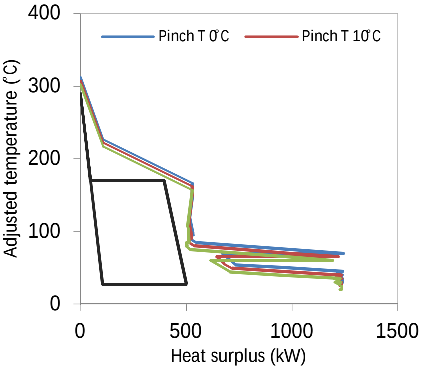

Figure 7.10. GCC – Design draught, slow steaming

The GCCs under the "design draught" modes are similar to those under the "scantling draught" modes, except that the total energy available is much less due to the load difference. It is also noted that the WHR potentials of SRC under the two modes of "design draught" are more or less the same, as shown in Table 7.5.

Table 7.5. The WHR potential of SRC under two "design draught" modes.

Operation mode	$\dot{m}_{ST}$ (kg/h)	p_B (bar)	T_s (°C)	$P_{SRC,el}$ (kW)	Quality, x_2
Design draught, NCR load	858	2.70	290	108	99.3%
Design draught, slow steaming	616	7.92	290	100	94.7%

7.4.2.3 Ballast draught modes

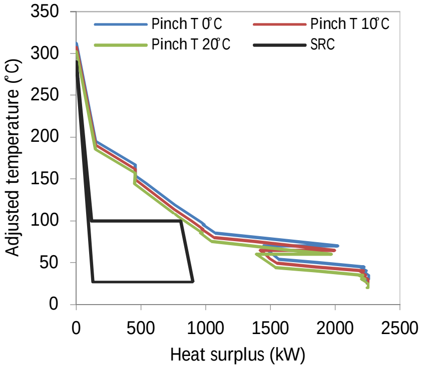

Figure 7.11. GCC – Ballast draught.

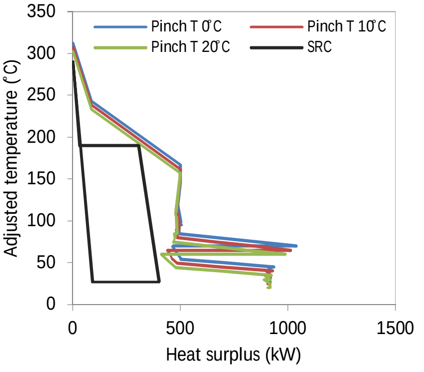

Figure 7.12. GCC – Ballast draught, slow steaming.

Figure 7.11 and Figure 7.12 show the grand composite curves of the case vessel's MER heat exchanger network under the two "Ballast draught" modes. It can be seen that there is still no pinch point for either of the operation modes. In this case, if slow steaming enabled, the WHR potential with SRC under the "ballast draught" mode would be only around two-thirds of the potentials without slow steaming, as shown in Table 7.6.

Table 7.6. The WHR potential of SRC under two "ballast draught" modes.

Operation mode	$\dot{m}_{ST}$ (kg/h)	p_B (bar)	T_s (°C)	$P_{SRC,el}$ (kW)	Quality, x_2
Ballast draught	1 014	1.01	290	123	Superheated
Ballast draught, slow steaming	495	12.55	290	85	92.5%

7.4.2.4 Harbour area operations

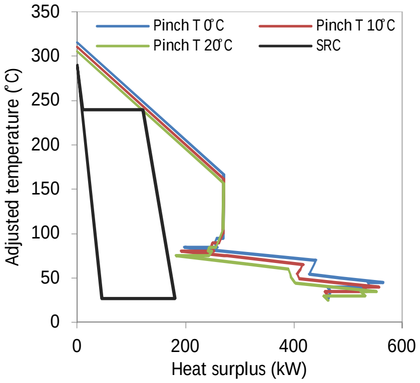

Figure 7.13. GCC – Manoeuvring

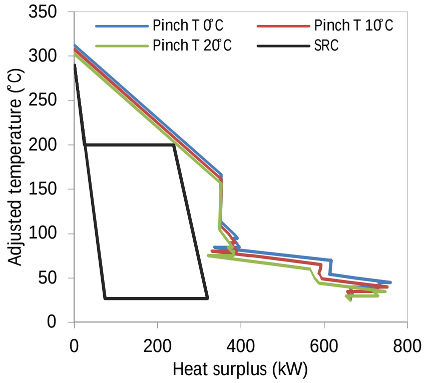

Figure 7.14. GCC – Harbour loading /unloading

Figure 7.13 and Figure 7.14 show the grand composite curves of the MER heat exchanger network when the case ship operates in the harbour areas, including the modes of "manoeuvring" and "harbour loading/unloading". It can be seen that there is still no pinch point for either of the operation modes. In this case, the WHR potential with SRC under the "manoeuvring" mode would be double of the potentials under "harbour loading/unloading" modes, as shown in Table 7.7.

Table 7.7. The WHR potential of SRC when operated in harbour areas.

Operation mode	$\dot{m}_{ST}$ (kg/h)	p_B (bar)	T_s (°C)	$P_{SRC,el}$ (kW)	Quality, x_2
Manoeuvring	228	33.47	290	123	87.0%
Harbour loading/ Harbour unloading	397	15.54	290	64	91.5%

7.4.3 The MER-network waste heat emissions and fuel savings

Based on all the calculations in the section 7.4.2 and the operating profiles listed in Table 7.1, we are able to calculate the current waste heat dissipation to the surroundings and the heat dissipation reduction, with the SRC integrated, of the optimized MER heat exchange network under each operation mode of the case ship. Furthermore, we can calculate the annual waste heat dissipation of the ship if the SRC is integrated into the optimized MER network, which is around 11.8 GWh, as shown in Table 7.8. The annual heat dissipation reduction using SRC would be over 7% of the total waste heat dissipation.

Since there is no pinch point identified during the pinch analysis process under any of the operation modes, no extra heat sources, i.e., the oil-fired boiler, were needed in the optimized MER network. Hence, the optimized MER network, at least with the global pinch points between 0°C and 20°C, would reduce the OFB usage by 100%, which represents annual fuel savings of 11 metric tons. Furthermore, the integrated SRC could reduce the yearly consumption of the external cooling utilities by approximately by 920 MWh.

Table 7.8. Waste heat dissipation of the MER-heat recovery network with or without SRC.

Operation mode	Cumulative annual operation hours (h)	Heat dissipation without SRC (kW)	Heat dissipation reduction by SRC (kW)	Annual heat dissipation with SRC (MWh)
Scantling draught	1226.4	2753	167	3172
Scantling draught, slow steaming	1226.4	1350	109	1522
Design draught, NCR load	919.8	2434	120	2128
Design draught, slow steaming	919.8	1228	105	1033
Ballast draught	919.8	2246	125	1951
Ballast draught slow steaming	919.8	907	91	751
Manoeuvring	105.12	463	49	44
Harbour loading	919.8	664	77	540
Harbour unloading	1156.32	664	77	678
			Total (MWh/year)	11 818

The energy balance of the SRC is shown in Table 7.9. As seen from the table, the annual net electricity production of the SRC, would be nearly 874 MWh. According to the calculations, most energy would be produced during “Scantling draught”, “Design draught, NCR load” and “Ballast draught” representing 27%, 18% and 16% of the total electricity, produced by the SRC.

Table 7.9. Energy balance of the SRC integrated into the MER network.

operation mode	SRC superheating temperature	Boiler pressure (bar)	Heat absorbed by SRC (kW_{th})	Heat rejected by SRC (kW_{th})	Electricity production (kW_{el})	Annual production (MWh_{el})
Scantling draught	290	2.32	1000.00	828.7	162.7	200
Scantling draught, slow steaming	290	7.92	520.00	410.5	104.0	128
Design draught, NCR load	290	2.70	700.00	576.6	117.2	108
Design draught, slow steaming	290	7.92	500.00	394.7	100.0	92
Ballast draught	290	1.01	900.00	770.1	123.4	114
Ballast draught, slow steaming	290	12.55	400.00	310.1	85.4	79
Maneuvering	290	33.47	180.00	134.5	43.2	5
Harbour loading	290	15.55	320.00	246.0	70.3	65
Harbour unloading	290	15.55	325.00	249.9	71.4	83
					Total	870.81

Figure 7.15 shows the fuel savings that are derived from the optimized MER-heat recovery network, obtained by pinch analysis, along with the integrated SRC. As it can be seen from the figure, 94% of the fuel savings would be received from the WHR electricity production of the SRC cycle. The removal of the oil-fired boilers would enable another 6% of the fuel savings. As a whole, the annual fuel saving potential of the case vessel would be 192 metric tons if both the optimized MER-heat exchanger network and the SRC were implemented.

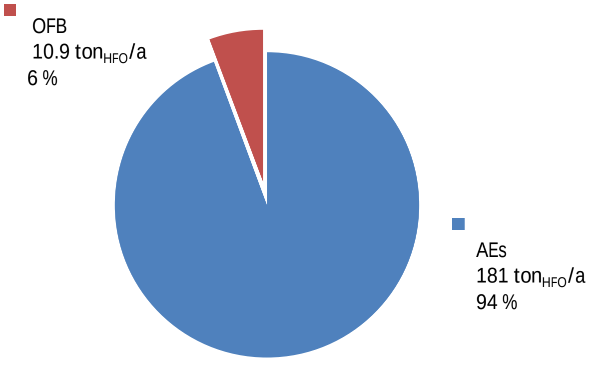

Figure 7.15. Fuel saving per annum by the theoretical process integration.

7.5 Discussion

According to the study, pinch analysis could potentially help optimize the heat exchange network onboard ships. The potential could get even bigger if extra proper waste heat recovery technologies are integrated into the system.

Overall, the results of this study are in line with other WHR studies done for the same case ship under similar operation conditions. For example, the study presented in Chapter 4, shows that, the WHR potential of integrating an absorption refrigeration system to provide chilling for the case ship would be 41.8 metric tons of fuel savings annually. In in another theoretical study, the annual fuel savings of 158 tons was suggested by changing the layout of the ship heat exchanger network and by integrating an absorption heat pump for producing process heat and a cascaded steam and organic Rankine cycle to lower the demand for auxiliary engines and oil fired boiler.

References

[1] Vanttola J., Kuosa M. and Lahdelma R. Case study of Waste Heat Recovery Potential in a Bulk Carrier Ship. Master's Degree. Aalto University: Energy Systems for Communities.

[2] Lemmon E.W., Huber M.L. and Mclinden M.O. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP. Version 9.1. Gaithersburg: National Institute of Standards and Technology, Standard Reference Data Program.

[3] Arrieta F.R.P. and Lora E.E.S. Influence of ambient temperature on combined-cycle power-plant performance. Applied Energy, 80(3), pp. 261-272.

[4] Elg M., Kuosa M., Tammi K., Mäkipeska P., Kinnunen A. and Rahkola P. Supporting the energy efficient ship design process with energy flow simulations: case energy efficient cooling water system. 4th November 2015 2015, Royal Institution of Naval Architects.

8. Integrated simulation and optimization of ship energy systems

Zou Guangrong
VTT Technical Research Centre of Finland Ltd

Summary: In this chapter, an integrated approach to ship energy system design and operation was thoroughly introduced. Considering variable operating conditions, we proposed a unified multi-level simulation and optimization framework to better integrate holistic optimization and dynamic simulation of ship energy systems already at the early stage of the ship design and operation processes throughout its life cycles. The developed ship energy system simulation and optimization platform adopts both model-based and data-driven approaches to optimize the design and operation of ship energy systems under dynamic operating conditions and uncertainties. Consequently, ship energy systems can be properly dimensioned with most promising energy efficiency technologies for specific ship design, which can help the ship owners and operators for their business-critical decision making on both ship energy system design and operation.

8.1 Introduction

As modern ships are getting bigger and more complex, proper design and operation of ship energy systems are getting more and more important in order to improve energy efficiency and to reduce emission under real operating conditions. Partly due to lack of systematic consideration during the design and operation processes, over dimensioning of ship energy systems and poor ship-level energy efficiency are still very common in the shipping industry. Especially, ship energy systems have been commonly configured at very early design phase (even at the early stage of the concept design phase shown in Figure 8.1 [1]) when very little information and few details are yet available. Large compromises have to be made at the early design phase, and consequently ships are rarely operated at their design points. How to systematically select the proper energy technologies and to properly dimension the energy systems is essentially important but also challenging during ship design processes. The situation gets worse while more and more novel energy saving technologies are being introduced into the shipping industry. Besides, as ships are normally operated under various dynamic conditions, it is hence important to bring the real operating conditions into the ship design process.

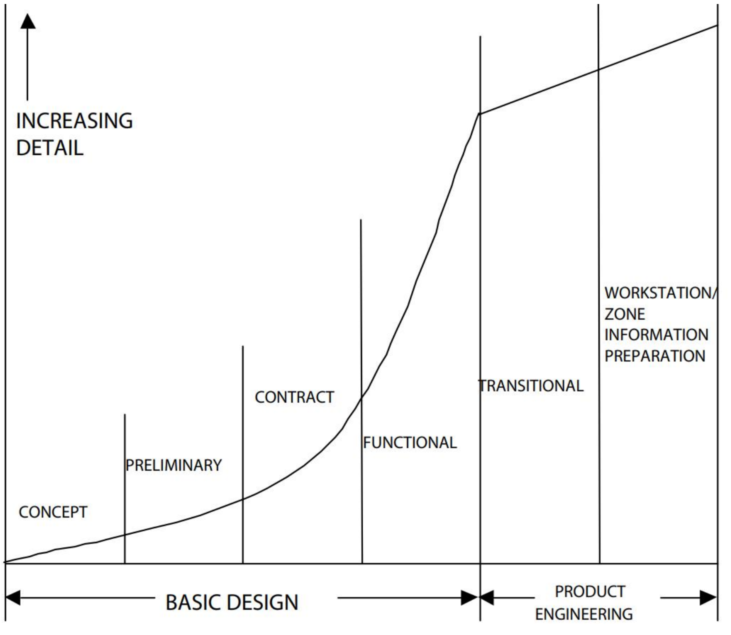

Figure 8.1. Ship design process. [1]

In the reminder of the chapter, we introduce a systematic approach to integrated simulation and optimization of ship energy systems. Thanks to the advancement of modelling technologies, we adopted a model-based simulation and optimization method to holistically represent ship energy systems, which help integrate the real operating conditions into the early-stage design process of ship energy systems. In turn, the developed energy system models can provide further assistance for the operation of ship energy systems.

8.2 The conceptual framework

In order to improve the energy system performances in real life, the energy system design and operation have to be considered more interactively at different phases of their life cycles. Hence, we proposes a 3-level simulation-based approach to interactively design and operate often-complex ship energy systems at different levels, namely ship level, system level and component level, during their life cycles. As shown in Figure 8.2, for energy system design, the systems are to be first configured at the ship level due to very limited information about the system. Once more detailed information about the specific systems is available, the design is to be detailed and further refined with the specified operation profile and/or real operation data measured from the similar designs at the system level. At the component level, the key components of the energy systems can be further detailed and evaluated under real dynamic operating conditions. All the 3-level designs can interactively communicate with each other during the whole design processes to guarantee the designed systems can achieve the desired performances at the plant level. For energy system operation, it can incorporate or run in parallel with the real energy management systems to monitor and manage the energy systems at above-mentioned three levels, which is especially beneficial for complex energy systems, which are often managed through decentralized control systems. The focus is not to design and operate local energy subsystems at their highest efficiency but to design and operate the energy systems at the plant level to achieve the best performances.

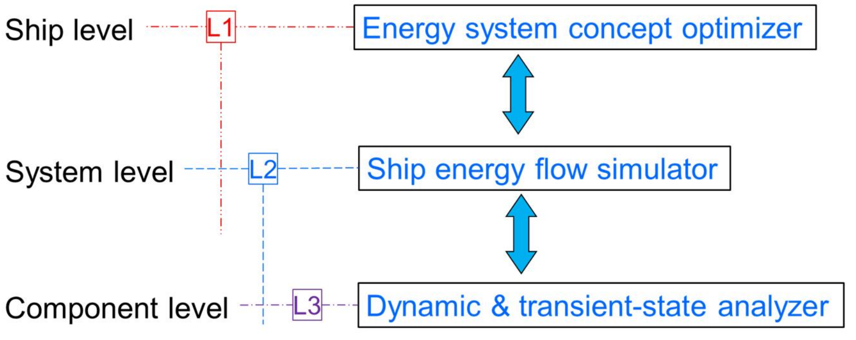

Figure 8.2. 3-level simulation-based approach to ship energy system.

Based on this approach, a unified simulation and optimization framework, namely ship energy system simulation and optimization platform (as illustrated in Figure 8.3), is developed to help better design, retrofit or operate ship energy systems under real operating conditions. The system-level ship energy flow simulation platform is employed, using a multi-domain simulation method, mainly to model and simulate the ship energy systems under real dynamic simulation conditions; while the ship energy system optimization platform is the optimization layer that holistically generate and optimization the design and operation scenarios of the ships energy system in question.

For the operation of ship energy systems, based on the case ship configuration, a ship energy flow simulator is first developed to holistically represent the energy system at a system level. The developed simulator can run in parallel with the real ship energy systems, in real time or offline, to collect the operating data from the ships and, using the optimization platform, to give guidance and suggestions to the ship energy system operation. [2] With the help of machine learning and dynamic simulation and optimization methods, the estimation and forecasting of energy system operating scenarios can be generated based on the real operating conditions and possible forecasting information. The system performances are highly dependent on the information available, data accuracy and the targeted forecasting horizons. In general, acceptable performances can be achieved for the horizons of up to 24 hours. [3–5] Besides, the developed energy flow simulator can be used for energy system condition monitoring, diagnosis and “measuring” critical sports via simulation, called equipment-free soft sensors.

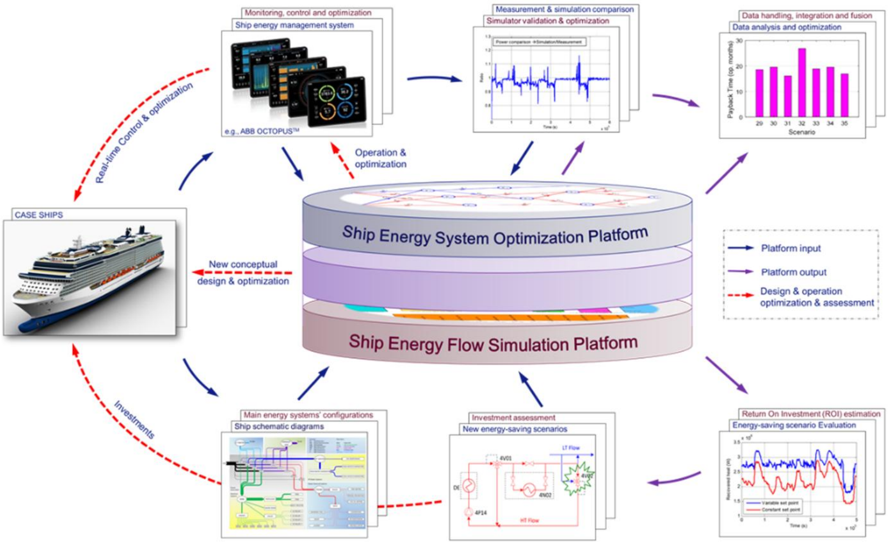

Figure 8.3. The conceptual framework of the ship energy system simulation and optimization platform.

For energy system design of new ships, initial information is first gathered and pre-proceeded, including ship design specification, possible sister ship configuration and measurement data, and preferred energy system options available in the library of the platform. Using the ship energy system simulation and optimization platform, a list of energy system scenarios are generated, evaluated and further optimized through a multi-objective optimization process under specified operating profiles or real dynamic profiles when available. Different scenarios are ordered based on their practical performances with respect to energy efficiency, emission and cost (life cycle cost [LCC], initial capital expenditure [CAPEX] or operating expenditure [OPEX]). An estimation on return on investment (ROI) can be further carried out to help the owners and operators make business decisions.

For the retrofit of ship energy systems, a ship energy flow simulator first developed as a baseline model for further evaluation. Then, based on the retrofitting specification and preferences, a similar optimization procedure, as for the energy system design, is followed to select the best scenario for retrofit, with respect to different criteria from the owners. [6]

8.3 Ship energy system design/operation concept optimizer

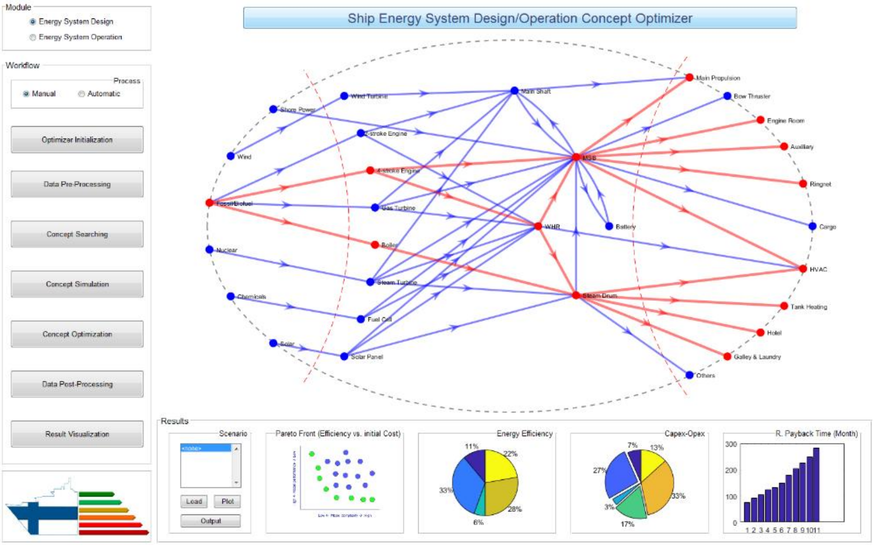

Figure 8.4. A top-level screenshot of the ship energy system design and operation concept optimizer.

The above-mentioned ship energy system simulation and optimization platform is further implemented in Matlab environment due to its superior capability of integrate optimization and dynamic modelling and simulation of engineering systems within one environment and a large collection of different engineering and optimization functions and toolboxes. It is also easy to implement further customized development for specific purposes.

Figure 8.4 shows the main graphical user interface (GUI) of the developed ship energy system design and operation concept optimizer, which is divided into four main sections, Module, Workflow, Results and a main graphical display window. In the module section, the user is able to select one of the two working modes of the optimizer, depending on the main purpose of the task, for energy system design or for energy system operation. The Workflow section gives the step-to-step instruction to the whole working process, which is shown also in Figure 8.5. It is also possible to automatically execute the whole process once necessary initial information and specification is given.

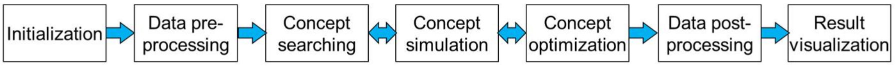

Figure 8.5. The main workflow of the ship energy system design and operation concept optimizer.

After the whole simulation and optimization process, the results are available to be visualized in the Results section and in the main graphical display window, either for a single energy system concept or for the comparison of multiple concepts. The main information is about the configuration of the selected energy system concepts and their performances mainly related to energy efficiency, efficiency, investment (CAPEX, OPEX, LCC and ROI). Figure 8.6 shows some typical results that can be generated by the energy system optimizer. In the shown example, it compares the different energy system scenarios with respect to energy efficiency, initial cost, relative payback time, LCC and dynamic performances.

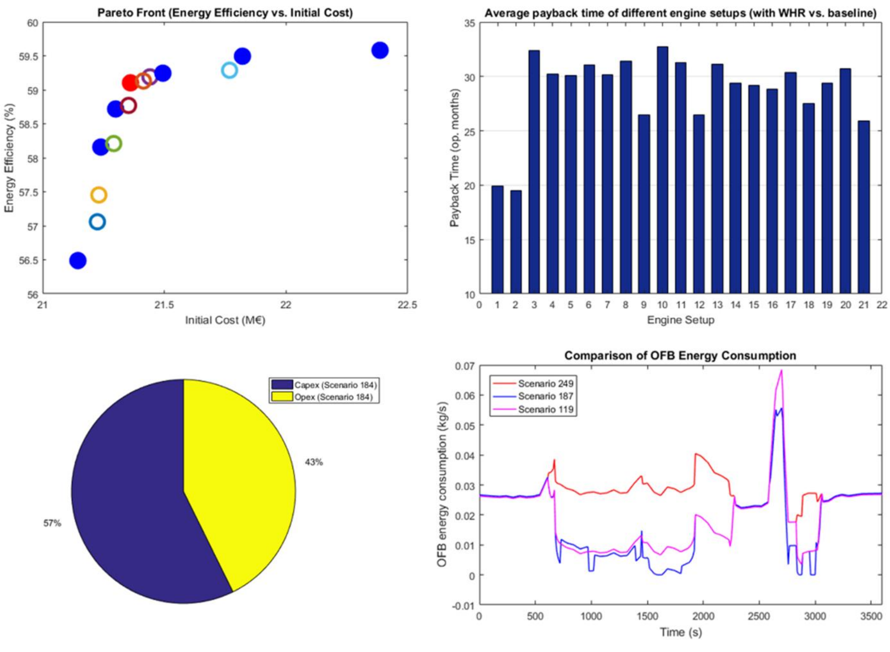

Figure 8.6. Typical results extracted from the ship energy system design and operation concept optimizer.

8.4 Remarks

The proposed multi-level ship energy system simulation and optimization platform is general in nature and hence can be easily applied to nearly all energy-related systems, such as road transportation, district energy systems, power grids, etc. Combining holistic optimization methods with dynamic simulation of the energy systems in question, it provides some unique value especially to address energy systems operated under variable load conditions and uncertainties.

Acknowledgement

The main idea of the work was conceived during a research visit to the Industrial Process and Energy Systems Engineering (IPESE) group at École Polytechnique Fédérale de Lausanne (EPFL), Switzerland. The author wants to express his deep appreciation to Prof. François Maréchal and the whole IPESE group for the great hospitality and fruitful discussions.

References

[1] Lamb T. Ship Design and Construction, Volumes 1-2. Society of Naval Architects and Marine Engineers (SNAME). Available at: http://app.knovel.com/hotlink/toc/id:kpSDCV0001/ship-design-construction /ship-design-construction

[2] Zou G, Elg M, Kinnunen A, Kovanen P, Tammi K, Tervo K. Modeling ship energy flow with multi-domain simulation, Proc. CIMAC 2013

[3] Zou G, Tammi K. How to predict energy flows? EFEU Tomorrow's Energy Efficiency Solutions Seminar. September 14, 2015, Helsinki, Finland

[4] Zou G, Tammi K, etc. Energy flow estimation and forecasting for efficient energy system operation. CLIC seminar on tomorrow's energy solutions, October 31, 2016, Helsinki, Finland

[5] Zou G, Tammi K. Integrated design and operation of energy systems under dynamic conditions and uncertainties. CLIC seminar on tomorrow's energy solutions, October 31, 2016, Helsinki, Finland

[6] Zou G, Kinnunen A, Tervo K, Orivuori J, Vänskä K, Tammi K. Evaluate Ship Energy Saving Scenarios Using Multi-Domain Energy Flow Simulation, Proc. Compit 2014

Part III
Summary of industrial projects

9. Simulation-based design of energy efficient ships[1]

Mia Elg, Päivi Haikkola, Panu Mäkipeska and Ossi Mettälä
Deltamarin Ltd

9.1 Introduction

Energy and Environmental Efficiency (EEE) has been one of the key focus areas for Deltamarin for numerous years. The “EEE” ship design process includes several dimensions, such as practical design tools for modelling the energy consumption and a consistent follow-up method of the performance indicators. In general, requirements for a successful energy saving concept are:

- Good understanding of ship energy systems from a function point of view,
- Adequate design and modelling tools,
- Understanding of the state of the art technologies applicable to the project,
- A skilled team with possibility to commit to further investigations and possibly testing facility.

For a successful project, it is important to do the right things at the right time. The concept design phase is crucial, since the most important choices regarding ship energy efficiency are made during this phase, but the challenge is often the limited time and rough level of information available for the decision-making. Therefore, the primary focus of Deltamarin during the SET project has been on both strengthening the technical knowledge of various technologies and developing the design methods for ship energy efficiency analysis in the concept design phase.

During the SET project, Deltamarin has participated actively in studying various new and interesting technologies with potential suitability for the marine industry. For instance, the following technologies were studied together with other project partners or by meeting various product makers outside the primary project team:

- Natural circulation boilers
- Thermal storage
- Organic Rankine Cycle

[1] The chapter is excerpted and revised from the final report of the Deltamarin’s company project in the SET consortium.

- Compression Heat Pump for heating purposes
- Absorption Cooling Process
- Efficient ship Auxiliary Cooling processes
- Onboard DC distribution
- Electrical energy storage
- Fuel cells (various types)

Several workshops have been arranged with equipment makers and researchers during the project. Furthermore, the knowledge has been transferred in-house by strengthening the existing technology database but also by developing the internal project working practices, where the new information is distributed in projects at an early phase. Another example of applying the various technologies in Deltamarin's work is to create concept project platforms where the new technologies can be showcased. Figure 9.1 shows a ro-pax ship platform, named DeltaChallenger, which included, for example, efficient cargo loading and unloading processes, state of the art air conditioning system, Flettner rotor wind turbines combined with an efficient dual fuel machinery with compact pod propulsion system.

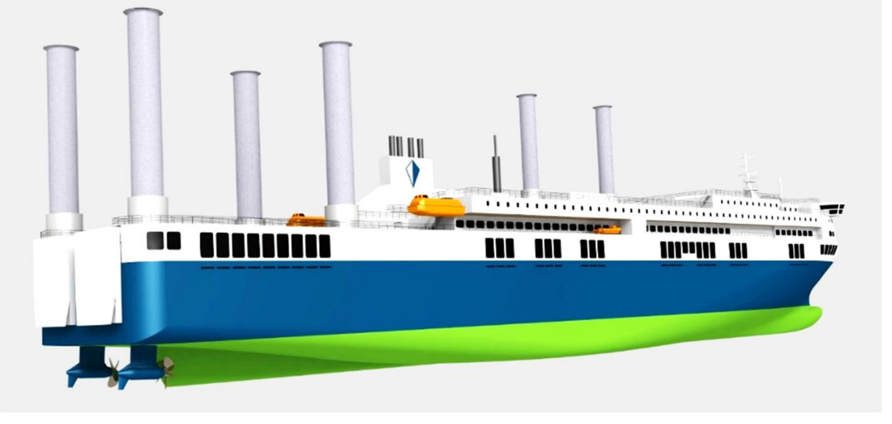

Figure 9.1. DeltaChallenger – a ro-pax ship concept for demonstrating various new technologies with superior performance.

Nevertheless, the majority of the development effort during the project was put into developing energy flow simulation tools suitable for conceptual design work. Energy flow simulation tools are useful especially because continuous measurement data from key ship processes is currently available for almost any ship type. Operational data is in the key role in making accurate energy balance calculations, and it is always required when starting a new concept project or making energy efficiency studies onboard an existing vessel. Traditionally, any operational data received from a client has to be filtered and carefully analysed before applying it in the energy balance calculation. The current tools developed for the energy flow analysis are able to utilize the operational data directly.

Deltamarin was one of the main founders of the SET project consortium, since SET has a history in another development project, where Deltamarin, VTT and ABB developed together a ship energy flow modelling and simulation platform. The focus of development for Deltamarin in the SET project was specifically to generate versatile tools and modelling practices that could be directly applied in the everyday project work. This goal was achieved during the project, and the simulation platform developed in-house is actively used currently.

9.2 Example results

During the SET project, the design tools and methods were developed for various ship types by studying several interesting technologies to be applied for the case ships. Deltamarin studied the improvement potential of the different technologies, focusing especially on bulk carrier, cruise ship and ro-pax ship types.

9.2.1 Energy storage

Energy storages have considerable potential for improving the energy efficiency in ships. During the SET project, two master's theses have been started regarding this topic. The first one, finished during the project, was focusing on studying fuel savings in a passenger ship with battery capacity of various sizes for storing electricity. As a result, a simulation tool for dimensioning the energy storage at the concept design phase was generated during the project. Once utilizing the tool for various ship types, the received results seemed to correlate well with reference results received in the project from possible battery system integrators. The ongoing thesis work focuses on heat batteries and generating modelling tools for the ship heat systems. The heat systems include main components and various processes for ship waste heat production and utilization. Figure 9.2 illustrates the improved ship fuel efficiency on a passenger ship equipped with hybrid machinery.

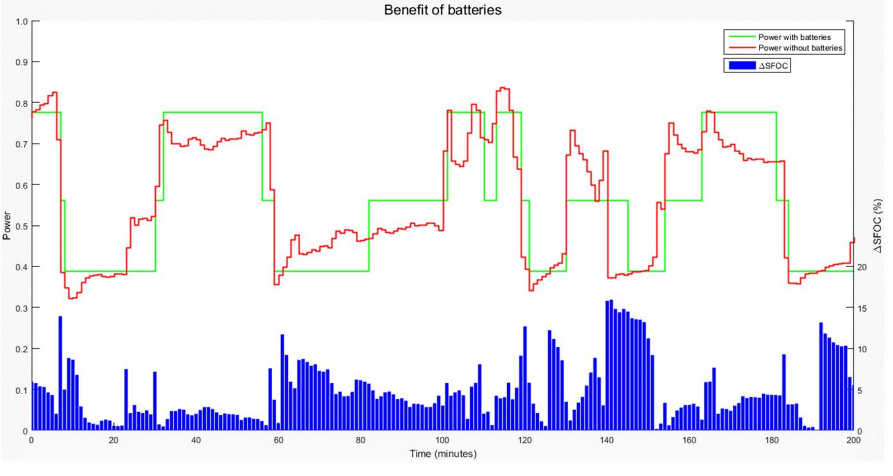

Figure 9.2. Fuel saving potential (presented as ΔSFOC %) of integrating a battery system in a ship machinery, which allows operating the engines at favourable loads.

9.2.2 Efficient auxiliary cooling systems

For bulk carriers, the focus of the project development was mainly on studying the general heat process efficiency and developing efficient auxiliary cooling systems. The reason was that the estimated fuel saving potential of an intelligent auxiliary cooling system could be as much as 4% of the yearly fuel consumption for a cargo ship such as the B.Delta37 bulk carrier that was utilized as the development platform.

Model was taken from modern district heat networks when improving the control of the auxiliary cooling circuit in bulk carriers. The results of the modelling are also presented in the two conference papers during the project period. Even though a bulk carrier was utilized as a platform, the results and the model developed can be utilized for any other ship type. Figure 9.3 illustrates the model utilized for evaluating the auxiliary system electricity consumption.

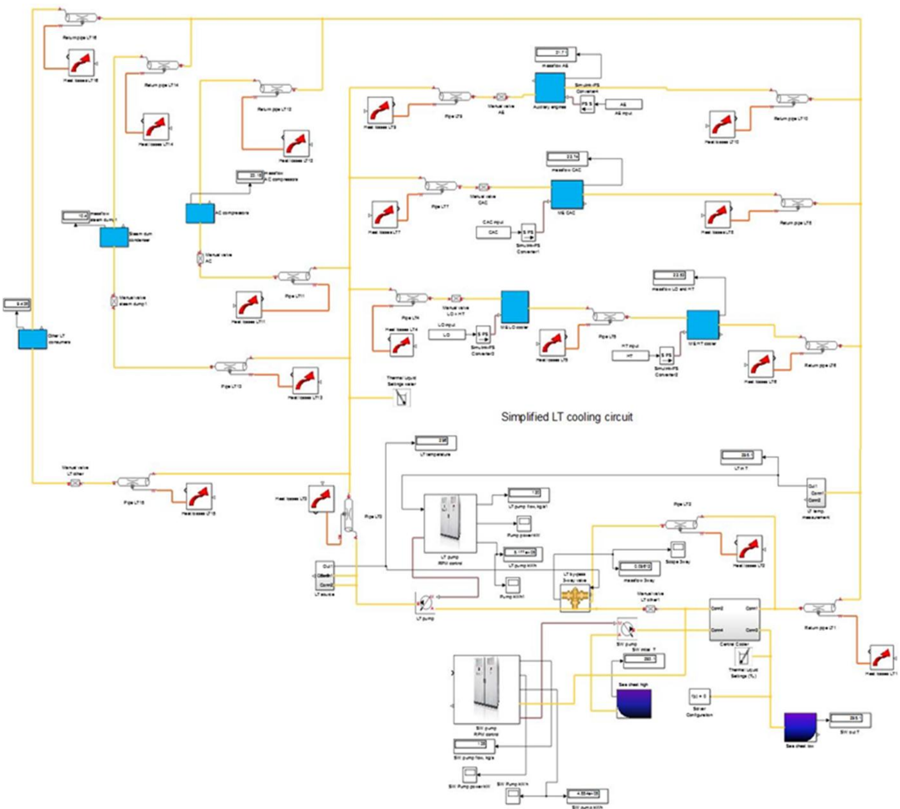

Figure 9.3. Ship auxiliary cooling system model.

9.2.3 Waste heat recovery

The latest development efforts at Deltamarin with the simulation models are strongly focused on the waste heat recovery systems. The energy saving potential of different waste heat recovery technologies, available today in the market, could possibly reach 4–6% even if the primary machinery of ships already consists of the most efficient diesel- or dual-fuel engines in the markets. Reaching this goal or even higher saving potential requires, however, taking a holistic view of all heat processes in the ship. Figure 9.4 presents an example of the energy flow distribution in a passenger ship, as a result of energy flow simulations over a one-day operation cycle of the case ship.

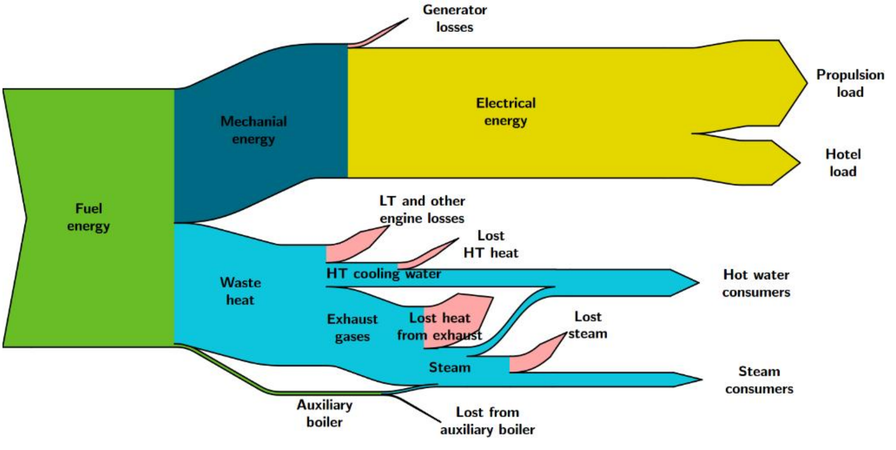

Figure 9.4. Example result of Deltamarin's simulation tool: Sankey diagram of the fuel energy distribution in a case ship, based on the ship's actual operation profile.

Improvement suggestions for an existing ship energy system require a fundamental understanding of various waste heat recovery technologies and their energy "potential" in the various ship heat flows. The thermodynamic definition of heat process efficiency can be quantified by exergy or entropy analysis on ship heat flows. An example of performing such an analysis is presented in a CIMAC congress 2016 conference paper, written in cooperation between Deltamarin, VTT and Aalto University. Even though the current work at Deltamarin focuses mostly on passenger vessels, the case ship chosen for the publication was the B.Delta 37 bulk carrier ship.

Exergy analysis of the ship heat flows helped to identify the magnitude of the improvement potential in a very simple way. Figure 9.5 presents the magnitude of the main waste heat flows in the case ship, and Figure 9.6 illustrates the corresponding total exergy potential in the case ship heat flows. In practice, this implies that there would be a theoretical potential for converting a part of the waste heat to shaft work or electricity, for example. Figure 9.7 presents the exergy destruction rate for each component in the analysis. In the example case the largest cause for inefficiencies was that the charge air cooler heat was not utilized but it was cooled to LT-water. Also, fresh water production with evaporator is a considerable source of exergy destruction and also the heat for this process could be utilized more efficiently.

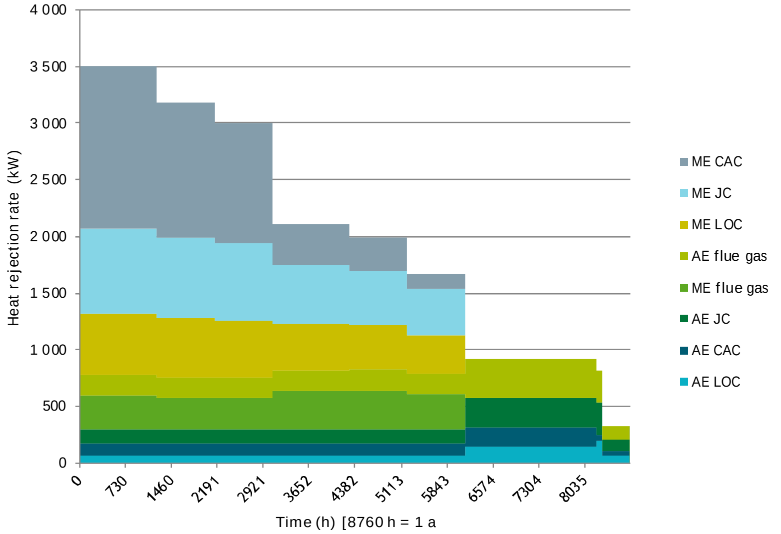

Figure 9.5. Case ship heat rejection rate from the engines to various cooling waters or ship steam system arranged according to yearly operation hours and heat quantity.

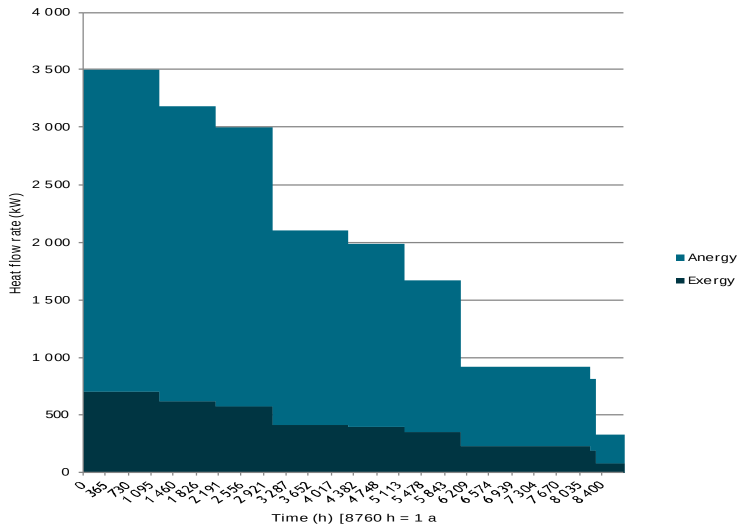

Figure 9.6. Exergy rate in the case ship waste heat flows.

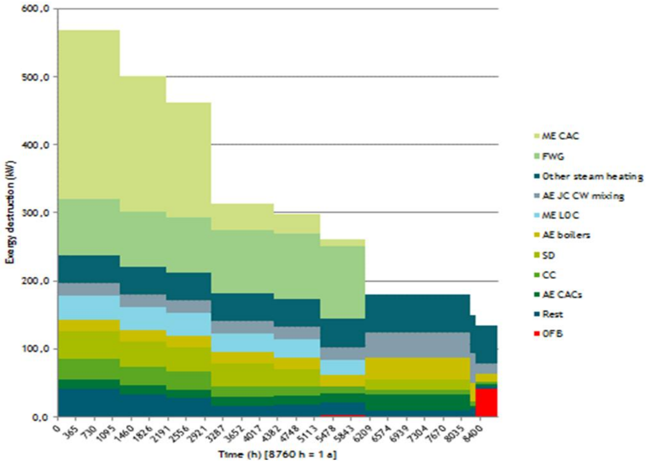

Figure 9.7. Exergy destruction rate due to the various consumers.

The results of the case example led to applying certain waste heat recovery technologies for the case ship, such as organic Rankine cycle, resulting in yearly fuel savings up to 4%.

The presented analysis method is simple and efficient and can provide valuable decision support already in the early phases of ship design process. Once the development potential is identified, various technologies can be applied in the developed ship energy flow simulator and the fuel savings can be quantified. It is also possible to examine only certain operational situations with the model. The latest release of the energy flow simulator also visualizes the main results, as shown in Figure 9.8.

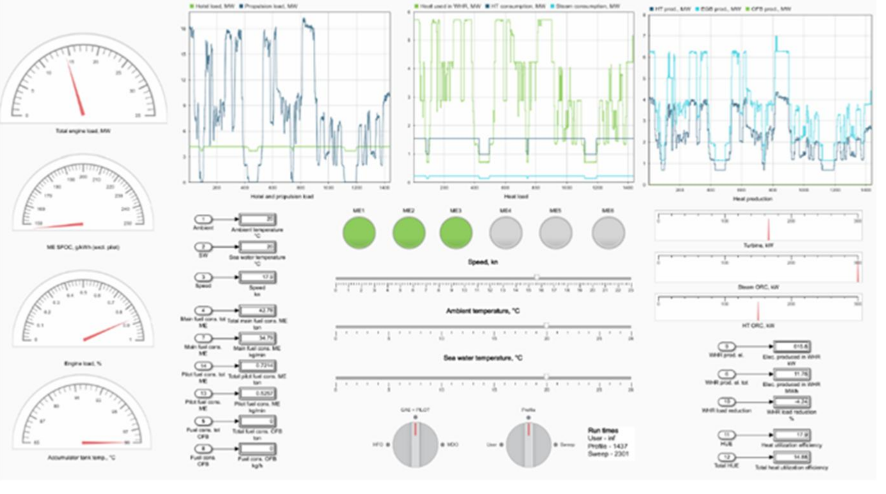

Figure 9.8. Screen shot of the “dashboard” of the energy flow simulator.

9.3 Remarks

Overall, the SET project has provided Deltamarin much more than an opportunity to develop the design tools together with the research partners and to enforce the technology database. Good examples are the excellent yearly seminars held both in 2015 and 2016. The events gathered together experts from various companies from the Finnish marine industry, as well as several ship owner representatives from abroad. The seminars presented an opportunity to showcase the development results and also to strengthen the network between the various stakeholders in Deltamarin's projects. The most obvious indicator of the SET being a successful project for Deltamarin is that the cooperation with the project members has continued even after the active phase of the SET project. The results of the project are also actively applied in Deltamarin's projects and the tools are continuously developed. The results of the latest development will be published, in addition to Deltamarin's own publications, in a doctoral thesis that has been started during the SET project.

References

[1] Mettälä O. Electric energy storage system dimensioning for large hybrid-powered vessels Master's thesis, School of Engineering, Department of Mechanical Engineering, Aalto University, 2016

[2] Elg M., Kuosa M., Lampinen M., Lahdelma R., Mäkipeska P., Raita J., Zou G., Tammi K. Advanced auxiliary cooling system for energy efficient ships. EEMODS conference 15.-17.9. 2015, Helsinki

[3] Elg M., Kuosa M., Tammi K., Mäkipeska P., Kinnunen A. and Rahkola P. Supporting the energy efficient ship design process with energy flow simulations: case energy efficient cooling water system. RINA Energy Efficient Ships conference, 4th November 2015, Rotterdam

[4] Elg M., Vanttola J., Kuosa M., Tammi K., Salmi W., Isomoisio H., Kinnunen A., Zou G., Lahdelma R. Energy saving technologies and new analysis methods in cargo ship machinery design, CIMAC congress 6.-10.6.2016, Helsinki

10. Simulation-based marine business innovations

Kalevi Tervo
ABB Oy Marine and Ports

10.1 Introduction

In the field of ship energy efficiency the “low hanging fruits” have more or less been picked already. For example, in the high-end ship segments, the new-build ships begin to have variable frequency drives (VFDs) installed in the most consuming and cooling systems, such as seawater cooling and engine room ventilation. In such systems, the payback time is rather simple to calculate as the pumps and fans in question are in the processes which transfer the waste heat to sea, to air or supply the diesels with fresh air. The pump and fan power for meeting the cooling/fresh water demand is directly dependent on the external operating conditions such as ambient temperature, seawater temperature and the engine load. The comparison to the conventional systems without VFDs is straightforward.

When it comes to more complex systems, where different domains of energies are interacting with each other and systems have strong nonlinear interconnections and feedback loops, the efficiency improvements due to the technology improvements are not that easily justifiable for a human since the effects might not be intuitive for human comprehension. For example, installing VFDs for all pumps and fans in a ship HVAC cooling system and tuning those to perform locally as well as possible might lead to zero or only slight improvement on the total fuel efficiency. Making improvements in systems connected up- and downstream in the energy conversion and transfer chain to other systems requires system level understanding of the problem. Otherwise, the result might be that in one part of the system, the VFD control decreases the pump/fan power and in some other part, another local control loop increases the power by the same amount. To improve the system level understanding and to develop and optimize the solutions globally, not only locally, overall system level simulation tools are required.

A major challenge in designing complex energy efficiency improvements is in correct design and dimensioning of the systems. When designing the ship, the customers and end users assume some operating profile for the ship, which eventually determines the design and dimensioning of the overall energy systems. However, in many cases, the assumption of the operation profile is incorrect or inaccurate, and thus the systems chosen for the ship might not be designed to work well in the actual operation, which is different from the assumed operation.

In order to improve the development of new ship concepts for different vessel segments, to facilitate the business by new optimization and design tools, and development of new technologies, simulation-based approach is proposed. In Simulation-Based Marine Business Innovations (SIMBA) project, the ship energy flow simulation tool is taken to the next level to become part of ABB Marine design and business processes. The simulation-based design concept ABB proposes in the SIMBA-project overcomes the challenge by utilizing the extensive ship operation database, which constantly gathers in the ABB OCTOPUS cloud database, as well as advanced ship energy flow simulation tools developed in the SIMBA and SET projects. By integrating the simulation tools with the existing software business, the new solutions can be dimensioned and tailored to meet the exact operational requirements of the customer. Thus, typical over-dimensioning of the systems is avoided, resulting in lower costs and better long-term performance.

10.2 Main objectives

10.2.1 System level design and optimization tools development

In the SIMBA project one of the main aims was to develop simulation-based design and optimization tools for specific energy efficiency improvement technologies, such as energy storage, HVAC VSD control, shaft generators and Onboard DC Grid systems. These simulation tools enable rapid feasibility studies of the benefits of the technologies. The actual measured operation profiles can be used as inputs to the simulation so that the fuel saving calculations would be as accurate as possible. In addition to the expected fuel savings, one can study the sensitivity of the solution with respect to the uncertainty in the operation profile and model parameters.

10.2.2 Development of new concepts

The second main objective of the project was to develop new conceptual solutions to improve the energy efficiency of ships. In this objective the ship is approached as a major system of systems without paying too much attention to the specific subsystems. The aim is to find ideas for system level improvements.

10.2.3 Technology, control and optimization innovations

The third main objective of the project was to develop new technology, control and optimization solutions for specific applications within a ship.

10.3 Example results

10.3.1 Robust method for customer-specific optimal solution design

During the project, a new method for using the Energy Flow Simulator for robust model-based design was developed. The method enables us to find solutions which minimize the customer Return of Investment (ROI) time so that the solution is not sensitive to the uncertainty in the operation profile.

In contrast to traditional model-based design originally dealing with control systems, the proposed approach is specifically for choosing the best components and their dimensions for ship energy systems. As one example for shaft generator application shown in Figure 10.1, the simulation was done for two different scenarios of the operation profiles. Both of the operation profiles were from real operation of a similar ship operating in similar routes as the ship under study. The bar graph on left shows the estimated savings potential for the different scenarios for different Shaft Generator Motor (SGM) sizes. The graph on right shows the ROI times for the different SGM sizes assuming certain scenario to be realized. The result illustrates that in this case the ROI times of the smallest size SGMs are very sensitive to the operation profile guess. If the decision is done according to Scenario 2 but the Scenario 1 is the actual operation profile, the ROI time will be about doubled in comparison to choosing a large or medium size SGM.

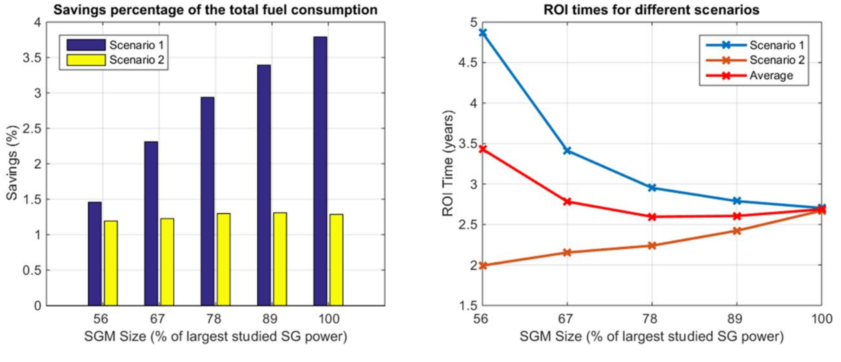

Figure 10.1. Saving potential in total fuel consumption with different sizes of SGM (left). ROI times for the extreme scenarios as well as the average ROI time (right).

10.3.2 Energy flow simulation tools

10.3.2.1 HVAC systems

In the SIMBA project, a new multi-domain simulation model for ship HVAC systems was developed. The tool can be used to study the improvement potential in HVAC system energy efficiency upgrades. An example subsystem of the HVAC simulation model is shown in Figure 10.2.

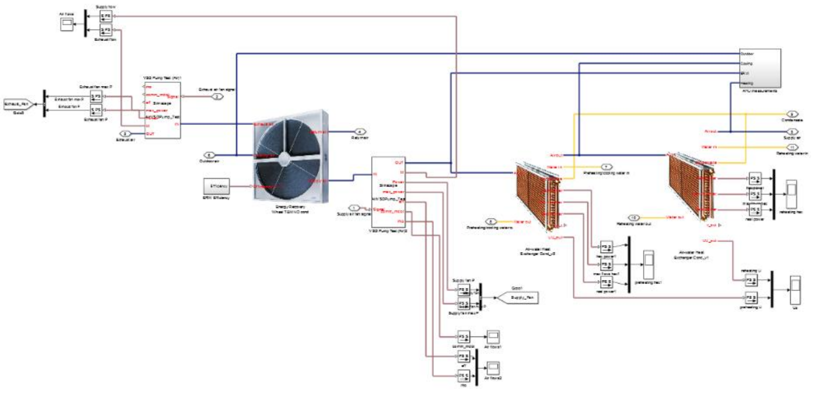

Figure 10.2. Example subsystem of Air Handling Unit (AHU) simulation block.

10.3.2.2 Energy storage systems

In the SIMBA project, a simulation model and tool for studying the improvement potential in energy storage systems in ships was developed. The system takes into account the energy storage key parameters (including the maximum power and nominal energy) and utilizes advanced control algorithms to calculate the optimal use of the energy storage and provide a good estimate on how beneficial the energy storage solution is in specific application with specific operation profile. An example of a simulation model with energy storage is shown in Figure 10.3. An example of the cumulative fuel savings in a specific diesel-electric cruise ship application with one type of operation profile is shown in Figure 10.4.

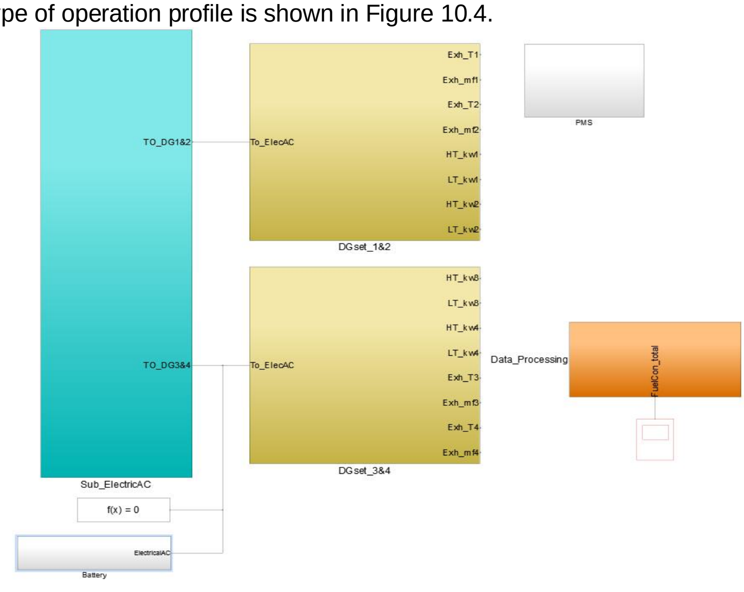

Figure 10.3. Simulation model for a typical diesel electric plant with energy storage.

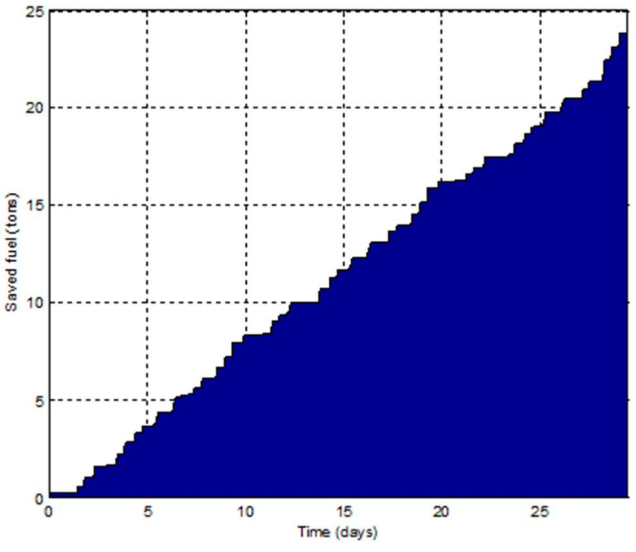

Figure 10.4. Example fuel savings results for a diesel electric cruise ship with energy storage.

10.3.2.3 Combined shaft generator and waste heat recovery systems

In diesel-mechanical vessels, significant amount of fuel savings can be achieved by utilizing exhaust gas waste heat recovery systems together with Power Take-out (PTO) – Power Take-in (PTI) shaft generator solutions. In SIMBA project, the work done in previous projects was extended and a new model with new power management system logic was created for shaft generator solutions. An example simulation model of a diesel mechanical ship with shaft generator and an exhaust gas waste heat recovery system is shown in Figure 10.5.

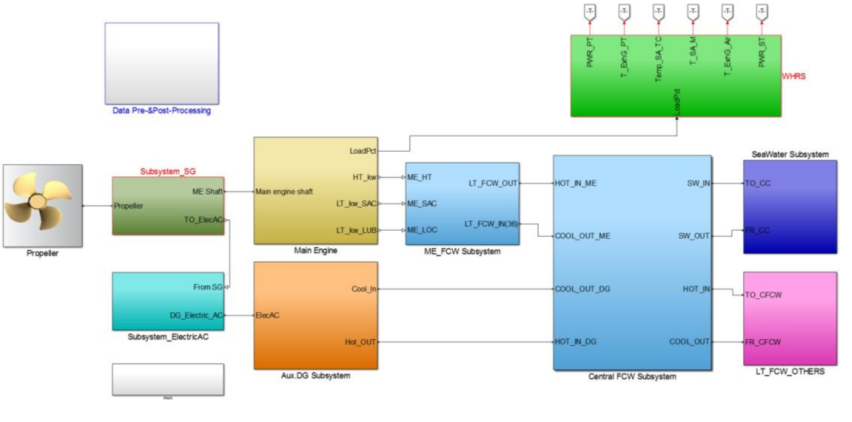

Figure 10.5. Example simulation model of a diesel mechanical container vessel with Shaft Generator Motor (SGM) and Waste Heat Recovery System (WHRS).

11. Energy saving solutions to exhaust gas boiler systems

Pasi Aaltonen
Alfa Laval Aalborg Oy

11.1 Introduction

Alfa Laval Aalborg Oy is located in Rauma, Finland and is part of the Swedish stock listed company Alfa Laval Ab. The main products of the company are exhaust gas heat recovery systems to diesel power plants and boiler systems to marine applications. Main customers in the marine sector are cruise vessel shipyards worldwide and shipyards in Finland and Russia.

Alfa Laval Aalborg is well-known for new innovative solutions. As an example, in early 2000's Alfa Laval Aalborg launched new exhaust gas boiler type to diesel power plant applications. The exhaust gas boiler type, based on natural circulation principle, increased tremendously the reliability and safety of the whole power plant and became very soon the only desired solution by the customers. Figure 11.1 shows an illustration of the natural circulation exhaust gas boiler system.

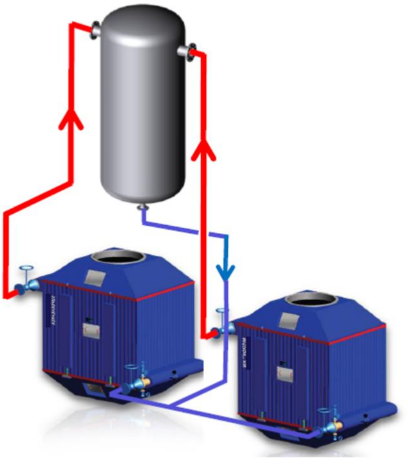

Figure 11.1. An illustration of the natural circulation exhaust gas boiler system.

Some years later, Alfa Laval Aalborg started the development of marine boiler systems. The focus was mainly to improve safety, energy efficiency and reliability of the marine boiler systems. Unfortunately, the global downswing in shipbuilding confused the whole market for several years. The main interest was to survive, not to develop new solutions. When the business started to recover, we also put more effort to the development.

Our development ideas are focusing on the improvement of safety and energy efficiency of the ships. In addition to financing from Tekes, participation in the SET project offered us an excellent platform to introduce our ideas to several parties involved in marine industry: ship-owners, shipyards, classification societies, engineering offices, educational and research institutes etc.

11.2 Main results

The main result of our development work is the natural circulation exhaust gas boiler system, which renews completely the marine boiler concept. Compared to traditional forced circulation system, it offers only benefits. The natural circulation system eliminates the risk of soot fires in the exhaust gas boilers and offers more redundancy, resulting in a huge improvement to the safety of vessels. Figure 11.2 shows a schematic view of the natural circulation boiler system.

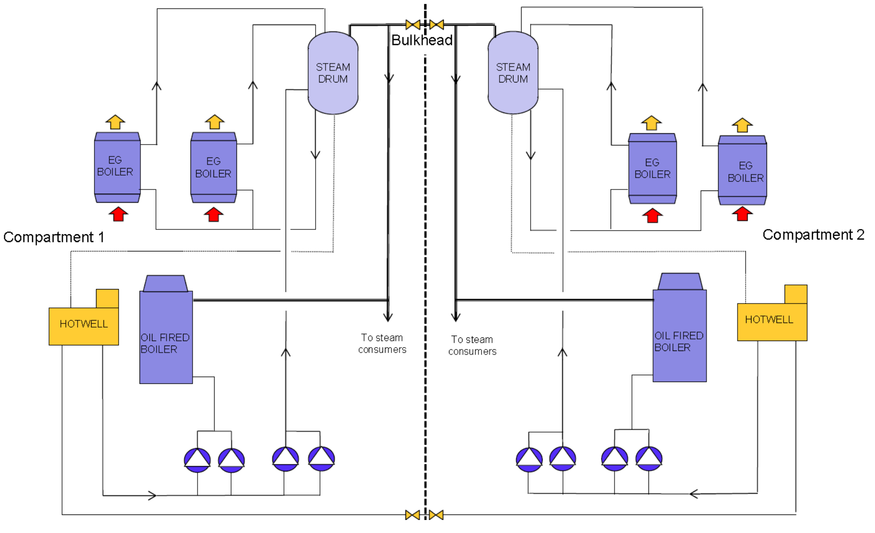

Figure 11.2. A schematic diagram of the natural circulation boiler system.

This system has also considerable impact on the fuel economy of vessels: no electricity consumption of the circulation water pumps and minimized energy loss caused by natural draft from exhaust gas boiler behind stopped engines. Because there are no circulation pumps, which traditionally have been one of the most troublesome equipment due to cavitation, the new system also has a considerable impact to maintenance and service costs.

The payback time of most of the new developments for better fuel economy is less than one year. To help our customers to evaluate the cost-effectiveness of different energy saving means, Alfa Laval Aalborg developed an energy saving calculation tool. This web-based tool includes all necessary technical and commercial information to calculate the payback time based on given input, such as operation profile, size of the engines and boilers, etc. This tool is available on request.

12. Simulation-based conceptual design of ice-going vessels

Tommi Heikkilä
Aker Arctic Technology Inc.

12.1 Company introduction

Aker Arctic Technology Inc. is an independent arctic R&D, engineering, design and consulting company. The company has been engaged in ice research with its own ice model basin for decades and been involved in numerous projects wherever freezing waters are found. The most advanced and innovative ship designs, such as the double-acting and oblique ship concepts, originate from Aker Arctic.

Our experienced, highly qualified and innovative personnel are able to provide our customers with specialised arctic know-how, as well as innovative, cost-effective and reliable solutions for Arctic development projects. The company operates a special test facility in Helsinki – the only privately owned ice model testing facility in the world. This combined with our extensive experience and the world’s largest Arctic reference for icebreakers means we are uniquely placed to bring the benefits of our solutions to our customers.

A portfolio of ice-going ship designs is also available for ship-owners and shipyards. Our past references include 60% of the entire world's icebreakers, many Arctic or Antarctic research vessels, and a large variety of cargo vessels and concepts for offshore structures. We have performed hundreds of scale model and full-size tests. We have produced hundreds of reports and studies of operating and transporting goods from the Arctic environment in various challenging locations.

12.2 Main objectives

Ice going vessels are not generally thought to be very energy efficient among the public. To some extent, this image is based on the traditional icebreakers that are utilized only during winters and having large installed engine power. However, energy efficient design has been a vital part of icebreaker design for a long time due to the simple fact of trying to reduce installed engine and auxiliary system power levels, which brings benefits in size and cost of main equipment. During recent years, the increasingly stringent regulations have also added pressure for implementing energy saving features for new icebreakers and, to some extent, for older ships in order to improve fuel economy.

For Aker Arctic, implementing energy efficient solutions for various ship systems has been previously evaluated mainly using excel based calculations. With the addition of several solutions to many different ship systems, the complexity of the overall effects to energy efficiency at the ship level becomes difficult to cover with basic excel tools. Thus, more advanced simulation methods need to fill in the gaps and enhance the system level knowledge to ship level so that the interactions between different equipment can be assessed in more detail.

Main objectives of the SET project for Aker Arctic were to increase our knowledge on currently available technologies for energy efficiency improvement through the public research project and to further develop our simulation tool in order to get better understanding on the ship level impacts of adopting these new technologies. Additionally, there was a target to gather operational information from already delivered ships regarding electric power consumption of specific equipment for various operational modes, in order to gather information and evaluate it against the calculated consumption figures.

12.3 Main results

In the early phase of the SET project, some state of art studies were done in addition to those belonging to the public research project scope. These studies provided ideas for the development of additional models, which could be introduced to the simulation tool in the future and thus increase the number of applicable solutions for our simulator.

The simulation tool for energy efficiency calculation, which has been partially developed during the SET project, is based on Matlab Simulink and Simscape models. Mainly Simulink is used to model larger components, such as engines, boilers and propulsors, whereas Simscape is used for auxiliary systems. The value of utilizing Simscape for modelling certain auxiliary systems is based on the physical elements, which bring additional realism especially when combining different fluid systems into one common ship scale assembly and most importantly enable faster modelling than by using pure Simulink models.

During the project, a simulation model of machinery systems based on our icebreaker case vessel POLARIS was created. Figure 12.1 gives an illustration of the developed simulation model of the POLARIS icebreaker energy systems. This model includes the main components such as engines, generators, propulsion units and boilers. Additionally the auxiliary systems related to these main components were modelled. The simulation results have been compared with databank of information derived from similar vessels. One of the main objectives for the SET project was to get full-scale data of the POLARIS, however unfortunately this objective could not be achieved due to the delay of ship delivery and thus we could not make real time comparison simulation.

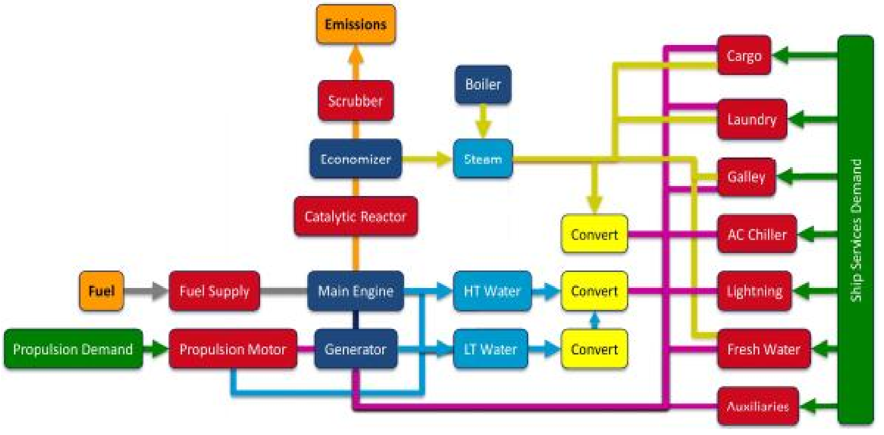

Figure 12.1. An illustration of the developed simulation model of the POLARIS icebreaker energy systems.

The information gathered from already delivered vessels regarding electrical power consumption created valuable data. Based on the gathered actual operation load levels for various ship types, the calculation tables can be adjusted accordingly. Due to this databank, the future designs can get more accurate indication of the required power levels can be used in the future designs and hence increases the level of energy efficiency of the designs.

During the project, our company has gathered useful information about available energy saving technologies and increased overall awareness on the subject. The direct results of the project can be mostly seen in the increased model library for the simulation tool and knowledge of the required information for conducting real customer projects with the created tool. Additional benefits have been gathered from the public research side for increasing the efficiency of modelling and performance of the simulation tool itself.

Series title and number

VTT Technology 306

Title	**Ship energy efficiency technologies – now and the future**
Author(s)	Zou Guangrong (Editor)
Abstract	Ship energy efficiency has recently become essential and crucial to the global shipping industries due to strict regulations, increasing operation cost and fierce competition. New technologies and innovative solutions have been widely applied to ship designing, building, retrofitting and operation processes in the last decades, which, however, rarely result in expected energy efficiency improvements in practice due to lack of system-level consideration under real operating conditions. How to assure the new technologies and solutions to achieve the desired performances at a ship level is still a big challenge. This book showcases some R&D practices and methodologies as a joint effort to address the challenge by a Finnish consortium consisting of four renowned Finnish companies and two top-level Finnish research institutions in the marine industries. Following a forward-looking (now – next step – future) approach, the special focuses of the book are on 1) reviewing and investigating the potential of novel energy efficiency technologies currently adopted or available in the marine industries, 2) how to use energy saving technologies to improve ship energy efficiency, especially for the case ships, and 3) to seek new innovative methods for better design and operate the next-generation ship energy systems. The holistic analysis and optimization methods target not only at instant improvements of ship energy efficiency but more importantly at long-term potential for optimal energy efficient ship solutions. As the final report of the R&D consortium, this book is a collection of some selected work of the projects, which aims to demonstrate the potential of different technologies and methods for improving ship energy efficiency and to offer a sneak peek at the future trend towards smart and green shipping.
ISBN, ISSN, URN	ISBN 978-951-38-8567-0 (Soft back ed.) ISBN 978-951-38-8566-3 (URL: http://www.vttresearch.com/impact/publications) ISSN-L 2242-1211 ISSN 2242-1211 (Print) ISSN 2242-122X (Online) http://urn.fi/URN:ISBN:978-951-38-8566-3
Date	August 2017
Language	English
Pages	143 p.
Name of the project	SET - Ship Energy Efficiency Technologies
Commissioned by	Tekes - the Finnish Funding Agency for Innovation
Keywords	Ship energy systems, energy efficiency, energy saving technologies, modelling and simulation, optimization, process integration
Publisher	VTT Technical Research Centre of Finland Ltd P.O. Box 1000, FI-02044 VTT, Finland, Tel. 020 722 111

Ship energy efficiency technologies – now and the future

ISBN 978-951-38-8567-0 (Soft back ed.)
ISBN 978-951-38-8566-3 (URL: http://www.vttresearch.com/impact/publications)
ISSN-L 2242-1211
ISSN 2242-1211 (Print)
ISSN 2242-122X (Online)
http://urn.fi/URN:ISBN:978-951-38-8566-3

VTT TECHNOLOGY 306 Ship energy efficiency technologies – ...