

DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

**APPLICATION OF ORGANIC RANKINE CYCLE
(ORC) SYSTEM TO MARINE VESSELS**



by
Olgun KONUR

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İZMİR

APPLICATION OF ORGANIC RANKINE CYCLE (ORC) SYSTEM TO MARINE VESSELS

**A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
In Partial Fulfillment of the Requirements for the Degree of Doctor of
Philosophy in Marine Transportation Systems Engineering**

**by
Olgun KONUR**

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İZMİR**

Ph.D. THESIS EXAMINATION RESULT FORM

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ABSTRACT

The coexistence of different kinds of waste heat sources on marine vessels with various temperature ranges increases the need for an optimal heat exchanger network (HEN) design for the heat collection process to reduce the unutilized heat that needs to be discharged overboard. The optimal HEN design has not been taken into consideration by using pinch point analysis in previous studies of marine organic Rankine cycle (ORC) systems that utilize different kinds of waste heat sources. The first objective of this thesis is to determine the optimal HEN design for an ORC integrated waste heat recovery system (WHRS) of a reference container ship by utilizing the pinch point analysis to improve the overall energy efficiency. The availability and potential use of cargo oil pump turbine (COPT) based waste heat offer more feasible solutions for ORC utilization to tanker ships. The second objective of the thesis is to introduce a new WHRS design for tanker ships that can utilize both marine diesel engines waste heat during navigation and COPT based waste heat during cargo discharge operations in ports. As a case study, the waste heat sources of a reference oil tanker vessel were identified and the thermodynamic analyses were carried out. The last objective is to attain exergetic sustainability indicators for both cases to assist decision-making processes that consist of decisions of implementing the proposed system design and which performance parameters should be used to achieve a better sustainability performance. The results indicate that all the waste heat cannot be collected by thermal oil, as some of the waste heat needs to be discharged through the sea. The most efficient and most sustainable working fluids among the selected alternatives are found as R1234ze(Z) at higher evaporating pressures than 1.4 MPa for container ship case, and R1233zd(E) for tanker ship case. The COPT based waste heat recovery potential is found as almost 3.5 times higher at full load operation of three COPTs than waste heat recovery potential of the main engine at full load navigation.

Keywords: Marine engineering, waste heat recovery, pinch point analysis, organic Rankine cycle, thermodynamic analysis, sustainability

DENİZ TAŞITLARINDA ORGANİK RANKİNE ÇEVİRİMİ (ORC) SİSTEMİ UYGULAMASI

ÖZ

Deniz taşıtlarında farklı sıcaklık aralığında ve farklı türde atık ısı kaynaklarının bir arada bulunması sebebiyle, atık ısıları toplamada kullanılamayacak ısıyı azaltmak için bir optimum ısı değiştirici ağı (IDA) tasarımı ihtiyacı bulunmaktadır. Deniz taşıtları için Organik Rankine çevrimi (ORÇ) sistemleri konusunda yapılan çalışmalarda, farklı türde atık ısı kaynaklarından yararlanan gemiler için pinç noktası analizi kullanılarak optimal IDA tasarımı oluşturulması konusu dikkate alınmamıştır. Bu tezin ilk amacı, referans olarak alınan bir konteyner gemisinin ORÇ entegreli atık ısı geri kazanım sistemi (AIGS) için optimal IDA tasarımını pinç noktası analizini kullanarak belirlemek ve böylece toplam enerji verimliliğini iyileştirmektir. Tanker gemilerinde kargo yağ pompası türbini (KYPT) bazlı yüksek potansiyelli atık ısının bulunabilmesi, ORÇ kullanımı için avantajlı çözümler sunabilmektedir. Tezin ikinci amacı, tanker gemileri için hem gemi dizel motorlarının seyir sırasındaki atık ısını hem de limanlarda yük boşaltma operasyonları sırasında KYPT bazlı atık ısıyı kullanabilen yeni bir AIGS tasarımı önerisi sunmaktır. Bu amaçla, referans olarak ele alınan bir petrol tankeri gemisinin atık ısı kaynakları tanımlanmış ve termodinamik analizleri yapılmıştır. Çalışmanın son hedefi, her iki örnek çalışma için önerilen sistem tasarımlarının uygulanmasına ilişkin karar verme süreçlerine yardımcı olmak ve daha iyi olan sürdürülebilirlik performanslarını belirtmek amacıyla ekserjetik sürdürülebilirlik göstergelerini ortaya koymaktır. Sonuçlar, tüm atık ısının termal yağ ile toplanamayacağını, bir kısmının denize atılması gerektiğini göstermektedir. Seçilen alternatifler arasında en verimli ve sürdürülebilir çalışma akışkanları konteyner gemisi için 1,4 MPa'dan daha yüksek evaporatör basınçlarında R1234ze(Z), tanker gemisi için R1233zd(E) olarak bulunmuştur. Tam yük işletimde KYPT bazlı atık ısı geri kazanım potansiyelinin, ana makinenin seyir sırasında tam yükteki atık ısı geri kazanım potansiyeline göre yaklaşık 3,5 kat daha yüksek olduğu sonucuna ulaşılmıştır.

Anahtar kelimeler: Gemi makineleri işletme mühendisliği, atık ısı geri kazanımı, pinç noktası analizi, organik Rankine çevrimi, termodinamik analiz, sürdürülebilirlik

CONTENTS

	Page
Ph.D. THESIS EXAMINATION RESULT FORM	ii
ACKNOWLEDGEMENTS	iii
ABSTRACT	iv
ÖZ	v
LIST OF FIGURES	ix
LIST OF TABLES	xii
 CHAPTER ONE - INTRODUCTION	 1
1.1 Scope of the Thesis.....	1
1.2 Objective and Motivation	3
1.3 Thesis Outline.....	5
 CHAPTER TWO - BACKGROUND AND LITERATURE REVIEW OF ORGANIC RANKINE CYCLE SYSTEMS ON MARINE APPLICATIONS....	 7
2.1 Potential Waste Heat Recovery Options for Marine Vessels.....	7
2.1.1 Internal Combustion Engine Based Waste Heat	8
2.1.2 Auxiliary Machinery Based Waste Heat	13
2.1.3 Boiler/Economizer Based Waste Heat	14
2.1.4 LNG Based Waste Heat	16
2.1.5 Cargo Oil Pump Turbine Based Waste Heat	17
2.2 Introduction to Organic Rankine Cycle Systems	19
2.2.1 Organic Rankine Cycle System Theory	20
2.2.2 Organic Rankine Cycle System Components.....	26
2.2.2.1 Evaporator and Condenser.....	27
2.2.2.2 Expander	29
2.2.2.3 Feed Pump	34
2.2.3 Different Working Fluids Utilized in Organic Rankine Cycle Systems..	36
2.2.4 Applications of Organic Rankine Cycle Systems.....	40

2.2.4.1 Geothermal Power Plants.....	41
2.2.4.2 Biomass Power Plants	42
2.2.4.3 Solar Thermal Power.....	43
2.2.4.4 Waste Heat Recovery	45
2.3 Organic Rankine Cycle Systems for Marine Applications	46
2.3.1 Literature Review for Studies on the Organic Rankine Cycle Systems Used in Ships.....	46
2.3.1.1 Organic Rankine Cycle System Configurations for Various Ship Based Waste Heat Sources	47
2.3.1.2 The Integration of Organic Rankine Cycle Systems to Emission Reduction Technologies for Marine Applications	55
2.3.1.3 Prototype Development and Demonstration Projects for Using Organic Rankine Cycle Systems in Ships.....	56
2.3.2 Challenges for Organic Rankine Cycle Applications on Marine Vessels	60
CHAPTER THREE - THERMODYNAMIC MODELING AND DESIGN OF AN ORC INTEGRATED WASTE HEAT RECOVERY SYSTEM OF A MARINE VESSEL	63
3.1 System Description.....	63
3.2 The Optimal Heat Exchanger Network Design with Pinch Point Analysis	65
3.2.1 Overview of Pinch Point Analysis	66
3.2.2 Pinch Point Analysis Rules.....	66
3.2.3 The Optimal Heat Exchanger Network Design of Reference Container Ship with Marine Diesel Engine Based Waste Heat Sources	67
3.3 Energy and Exergy Analyses for ORC Integration to Optimal Heat Exchanger Network Design.....	72
3.4 Energy and Exergy Analyses Results.....	75
3.5 Exergetic Sustainability Assessment for the Proposed System Design	82
3.5.1 Background of Sustainability Assessment.....	83
3.5.2 Exergy Based Sustainability Assessment of Energy Systems.....	86
3.5.2.1 Exergetic Sustainability Indicators.....	87

3.5.2.2 Literature Survey on Exergy Based Sustainability Assessment	89
3.5.3 Exergy Based Sustainability Assessment Results of the Organic Rankine Cycle Integrated Waste Heat Recovery System for the Reference Container Ship	95
3.6 Discussion	99
 CHAPTER FOUR - AN ORC INTEGRATED WASTE HEAT RECOVERY SYSTEM FOR TANKER SHIPS WITH CARGO OIL PUMP TURBINE	101
4.1 System Description.....	101
4.2 Optimal Heat Exchanger Network Design with Pinch Point Analysis for Navigation Mode	103
4.3 Integration of Cargo Oil Pump Turbine Based Waste Heat to Optimal Heat Exchanger Network Design for Discharge Operation Mode	108
4.4 Energy and Exergy Analyses of the Proposed ORC Integrated Waste Heat Recovery System Design	110
4.5 Energy and Exergy Analyses Results for Navigation and Discharge Operation Modes of the Proposed ORC Integrated Waste Heat Recovery System Design	113
4.6 Exergy Based Sustainability Assessment of the Proposed ORC Integrated Waste Heat Recovery System for the Reference Tanker Vessel	120
4.7 Discussion	125
 CHAPTER FIVE - CONCLUSIONS.....	129
 REFERENCES.....	131
 APPENDICES	171
APPENDIX 1: NOMENCLATURE	171
APPENDIX 2: SUBSCRIPTS.....	172
APPENDIX 3: ABBREVIATIONS	173

LIST OF FIGURES

	Page
Figure 2.1 Heat balance obtained from unit fuel burned for MAN B&W engines of (a) 12S90ME-C9.2 (b) 6S60MC-C7 IMO Tier I engine	9
Figure 2.2 A schematic diagram of a typical low-temperature cooling water system onboard.....	13
Figure 2.3 Steam service system of a ship	15
Figure 2.4 Fuel system layout of LNG-powered marine vessels	16
Figure 2.5 International GHG emissions by voyage-based operational phase of ships	18
Figure 2.6 Schematic diagram of a simple ORC system	20
Figure 2.7 (a) Deviation of actual RC from the ideal RC (b) The effect of pump and turbine irreversibilities on the ideal RC	22
Figure 2.8 Schematic diagram of a recuperated ORC system	24
Figure 2.9 Schematic diagram of a reheated ORC system.....	25
Figure 2.10 T-s diagram of (a) recuperated (b) reheated ORC systems	26
Figure 2.11 Heat exchangers types used in ORC systems (a) Gasketed-plate (b) Brazed-plate (c) Plate-and-fin (d) Shell-and-tube (e) Double-pipe	28
Figure 2.12 Application ranges of volumetric expanders compared to turbo-expanders and Tesla turbines	31
Figure 2.13 Types of expander technologies used in ORC systems	33
Figure 2.14 Temperature – entropy (T-s) diagrams for (a) Wet fluid (b) Dry fluids (c) Isentropic fluids	37
Figure 2.15 The trend in the installed ORC plant capacity for several application areas in the period of 1984-2017.....	40
Figure 2.16 Characteristic hot water plant reservoir temperatures and power ranges for heat to power conversion systems.....	41
Figure 2.17 Schematic view of a solar ORC system with parabolic trough solar collectors	44
Figure 3.1 Available waste heat sources of the reference ship (a) that sent to boiler plant and through the atmosphere as exhaust (b) that are cooled at the coolers and rejected to sea by the central cooler	64

Figure 3.2 (a) Composite curve and (b) grand composite curve of the heat sources for the engine at 76% MCR engine load.....	70
Figure 3.3 Optimal HEN design with the PinCH 3.0 software.....	72
Figure 3.4 Validation of the ORC model with the experimental results of a reference study (Unamba et al., 2019) for a 1 kW ORC system operating with R245fa for (a) the net power output and (b) the thermal efficiency	77
Figure 3.5 Layout of the integrated optimal HEN and ORC system for the reference ship	77
Figure 3.6 The effect of evaporator pressure and ORC fluid type on the thermal efficiency of the system.....	79
Figure 3.7 The effect of evaporator pressure and ORC fluid type on the exergy efficiency of the system.....	79
Figure 3.8 Sankey diagrams showing the heat utilization through the proposed HEN to the ORC unit for 100% MCR, 76% MCR, 50% MCR, and 25% MCR main engine loads.....	81
Figure 3.9 Distribution of exergy destruction rates of the components on varying engine loads.....	82
Figure 3.10 DPSIR model for sustainability assessment	85
Figure 3.11 The role of sustainability assessment indicators of energy systems for a planned sustainable energy society	86
Figure 3.12 WER values obtained for different evaporator pressures and working fluids	96
Figure 3.13 (a) EEF (b) EDC values calculated for different evaporator pressures and working fluids	97
Figure 3.14 (a) ESI (b) SI values calculated for different evaporator pressures and working fluids	98
Figure 3.15 IP for different engine loads and working fluids at ORC evaporator pressure of 2400 kPa	98
Figure 4.1 Schematic of a COPT system onboard a large tanker ship.....	102
Figure 4.2 (a) Composite curve (CC) and (b) grand composite curve (GCC) of the heat sources for the engine at 60% MCR engine load	106

Figure 4.3 Optimal HEN design for reference crude oil tanker ship for 60% MCR engine load	107
Figure 4.4 Layout of the proposed optimal HEN design for the ORC integrated WHRS of the reference crude oil tanker ship	109
Figure 4.5 The effect of condenser temperature and ORC fluid type on thermal efficiency of the system.....	114
Figure 4.6 The effect of condenser temperature and ORC fluid type on exergy efficiency of the system.....	114
Figure 4.7 Distribution of exergy destruction rates of the proposed ORC integrated WHRS components on varying engine loads of the reference tanker ship	118
Figure 4.8 Exergy destruction ratios of WHRS components during navigation mode at 60% MCR.....	119
Figure 4.9 Distribution of exergy destruction rates of the proposed ORC integrated WHRS components on cargo discharge operation mode of the reference tanker ship for three different COPT speeds.....	119
Figure 4.10 Exergy destruction ratios of WHRS components during cargo discharge operation mode	120
Figure 4.11 WER values obtained for different ORC condenser temperatures and working fluids	121
Figure 4.12 (a) EEF (b) EDC values calculated for different ORC condenser outlet temperatures and working fluids.....	122
Figure 4.13 (a) ESI (b) SI values calculated for different ORC condenser outlet temperatures and working fluids.....	123
Figure 4.14 IP for different engine loads and working fluids at ORC condenser temperature of 15 °C on navigation mode of the reference tanker ship	124
Figure 4.15 IP for different COPT speeds and working fluids at ORC condenser temperature of 15 °C on cargo discharge operation mode of the reference tanker ship.....	124

LIST OF TABLES

	Page
Table 1.1 Methods for energy-efficient operations of ships	3
Table 2.1 The number of oil tankers and the calculated total CO ₂ emissions emitted from the international oil tanker fleet in years between 2007-2018.....	17
Table 2.2 Comparison of different ORC expander technologies.....	34
Table 2.3 Properties of organic working fluids.....	38
Table 2.4 Review of the studies based on various ship-based waste heat sources utilized for waste heat recovery with ORC systems on marine applications	53
Table 2.5 Reported ORC installation projects to marine vessels.....	59
Table 3.1 Input parameters for waste heat sources and cold streams	68
Table 3.2 Input parameters of the ORC system	75
Table 3.3 Equations used in the model	75
Table 3.4 Comparison of the results of the current study and the reference study....	76
Table 3.5 Properties of the selected organic working fluids.....	78
Table 4.1 Input parameters for waste heat sources and cooling utility for navigation mode of the tanker	104
Table 4.2 Real-time data taken from the vacuum condenser of the reference tanker vessel during cargo discharge operation with three COPTs running to assess the COPT based WHR potential.....	110
Table 4.3 Input parameters of the ORC system of the proposed WHRS for the reference tanker ship.....	112
Table 4.4 Properties of the selected organic working fluids.....	113
Table 4.5 Heat utilization amounts through the proposed HEN design to the ORC system and the net power output generated by the ORC system for 30-100% MCR engine loads (units are indicated in kW)	116
Table 4.6 The amount of utilized COPT based waste heat and net power output generated by the ORC system during cargo discharge operation for varying COPT speeds (units are indicated in kW)	117

CHAPTER ONE

INTRODUCTION

1.1 Scope of the Thesis

Excessive population growth and technological developments in the last century have dramatically increased the energy demand, resulting in the rapid depletion of natural resources and consequently increased environmental pollution. According to the International Energy Agency report titled “Global Energy Review 2021”, global energy demand is set to increase by 4.6% in 2017, underpinning a rise in emissions of almost 5%, or 1500 Mt (IEA, 2021). National and international organizations have begun to introduce stringent emission regulations (e.g. Kyoto Protocol, Paris Agreement, and Emission Control Areas) to suppress this trend.

Ship-based emissions caused by worldwide shipping activities are account for around 2.5% of the global greenhouse gas (GHG) emissions (Smith et al., 2014). Faber et al. (2020) revealed in their Fourth IMO GHG Study that shipping emissions have increased from 977 million tonnes to 1,076 million tonnes (9.6% increase) from 2012 to 2018. Although strict measures were taken in this period for the shipping industry, this result was expected because of the growing worldwide transportation demand. The effect of the COVID-19 pandemic will certainly show a decrease in the trend for 2020 and 2021 but is not foreseen to have significant effects on the long-term projections. The increase in carbon emissions for 2050 is projected for up to 50% of the 2018 values with the use of alternative fuels and the implementation of new environmental-friendly technologies to marine vessels. It shows that the world still needs more breakthrough innovations in this field to reduce the environmental effects of shipping activities.

In the maritime sector, the regulations for the Prevention of Air Pollution from Ships was added to MARPOL Convention in 1997 and entered into force as Annex VI on 19 May 2005 to minimize the airborne emissions from ships that damage the ozone layer and cause negative environmental impacts to nature (IMO, 2020). The International Maritime Organization (IMO) has then adopted two important mandatory

regulations under MARPOL Annex VI related to increase energy efficiency and reduce GHG emissions of international shipping. The first one of the regulations is the EEDI (Energy Efficiency Design Index) that can be applied in the design stage of the ships to be produced, and the other is the SEEMP (Ship Energy Efficiency Management Plan) which can be applied to all ships of above 400 gross registered tonnages (IMO MEPC, 2011). The maritime companies have been obliged to invest in new technological developments, apply energy efficiency improvement measures, monitor, and report the energy efficiency performance of the vessel to meet the requirements under SEEMP and EEDI since the entry into force date on January 1, 2013.

Energy efficiency improvement on a ship can be achieved by different approaches, such as increasing propulsion system efficiency, reducing aerodynamic and hydrodynamic resistances affecting the ship, and improving the engine efficiency using alternative fuels, waste heat recovery, and improved control systems (Wang & Lutsey, 2013). IMO released a guideline (IMO MEPC, 2012), in which a set of energy efficiency improving techniques were suggested as best practices for the fuel-efficient operation of ships to develop a ship energy efficiency management plan for ships (e.g. route optimization, trim optimization, hull maintenance, energy management, fuel type selection, and waste heat recovery). These methods are summarized in Table 1.

The concept of waste heat is defined as the heat emitted to the environment regardless of its temperature and possible usage areas (Goldstick & Thumann, 1986). Ships lose a large amount of energy due to their operations. The main energy losses within this heat dissipation are due to the ship's engine (Suárez & Greig, 2013). Despite the high efficiencies achieved by marine engines, which can reach 50% and above in recent years, a significant amount of heat loss occurs mainly during the release of exhaust gases, and the cooling of scavenge air, lubricating oil and cooling water (MAN Diesel & Turbo, 2014a). Therefore, improving engine efficiency can lead to a significant reduction in a ship's fuel consumption. In addition, the recovery of the waste heat from ships appears to be an effective method that increases the total energy efficiency of ships. In this regard, organic Rankine cycle (ORC) systems, as a waste

heat recovery (WHR) technology, are frequently studied in the literature in recent years and given as one of the promising technologies that can be adapted to the ships easily among the given techniques in Table 1. ORC systems can use the waste heat of a machine to generate mechanical or electrical energy. The generated energy can be used for additional propulsion power and the demand for utility services without additional fuel costs and CO₂ emissions (Singh & Pedersen, 2016).

Table 1.1 Methods for energy-efficient operations of ships (Lloyd's Register, 2012; IMO MEPC, 2012)

Category	Energy Efficiency Improvement Method	Description
Fuel-Efficient Operations	Improved voyage planning	Careful planning and execution of voyages
	Weather routeing	Using commercially available routeing tools
	Just in time	Optimize the speed by connecting an early communication with next port for berth availability
	Speed optimization	To minimize fuel consumption, taking into account the engine manufacturer's optimal settings and arrival times/availability of berths at the port
	Optimized shaft power	Operating the engine at constant shaft RPM
Optimized Ship Handling	Trim optimization	Operating at optimum trim for specified draft and speed to reduce the resistance of the ship
	Optimum ballast	Ballasting for optimum trim and steering conditions
	Optimum propeller and propeller inflow considerations	Possible retrofitting of improved propeller designs and/or inflow modifiers such as fins or ducts to improve efficiency
	Optimum heading control (Autopilots)	Reducing distance sailed 'off track' and minimizing losses caused by rudder corrections through retrofitting efficient autopilot and rudder systems
Hull Maintenance		Advanced coating systems, better management of cleaning intervals and underwater inspection
Propulsion System		Systematic minimization of heat and mechanical loss through electronic controlled engines
Propulsion System Maintenance		Use engine condition monitoring tools and planned maintenance schedules to maintain efficiency
Improved Fleet Management		Better utilization of fleet capacity and promoting the best practice among ships within a company
Improved Cargo Handling		Optimum solutions matched to ship and port requirements for cargo handling
Energy Management		Review of energy usage onboard such as electrical services and HVAC systems
Waste Heat Recovery		Using thermal heat losses on the ship to either generate electricity or additional propulsion
Alternative Fuels		The potential use of emerging alternative fuels
Other Measures		Computer software to calculate fuel consumption, renewable energy technologies and shore power

1.2 Objective and Motivation

In the field of marine engineering, there is a strong motivation to increase the propulsion system efficiency, mainly due to rising fuel prices and the new regulations concerning worldwide shipping. ORC systems have received significant attention to increase the energy efficiency of marine vessels in the recent decade because of their relatively high energy conversion rates and relatively lower initial costs among other WHR technologies. MAN Diesel & Turbo (2014a) indicates that WHRSs with power turbine and steam turbines can provide efficient operation for marine diesel engines of above 25 MW. It is recommended to use ORC systems for WHR for lower power ship propulsion systems. From this point of view, it can be seen that ORC systems in WHR will appeal to a very large market share.

There are several studies mainly focused on the gathering of the waste heat from different sources of marine diesel engines, determination of the WHR potential, selection of the organic working fluid for the ORC system, and the payback time of the ORC system. The optimal heat exchanger network (HEN) design has not been taken into consideration by using pinch point analysis for marine ORC systems that utilize different kinds of waste heat sources. Yang (2016) clearly showed in his review that a gap exists in the literature for utilizing the exhaust gas, cylinder jacket cooling water, scavenge air, and lubricating oil waste heat sources of marine diesel engines altogether in waste heat recovery systems (WHRS). The increased complexity with the co-utilization of waste heat sources makes it harder to propose an optimal HEN design. The first objective of this study is to determine the optimal HEN design for an ORC integrated WHRS of a marine vessel by utilizing the pinch point analysis to improve the overall energy efficiency. Lubricating oil, high temperature (HT) cooling water and scavenge air of the main engine, and the exhaust gas emitted from the boiler plant were identified as the major waste heat sources of a reference ship. A heat collection stream, in which thermal oil is used as the heat transfer fluid that transfers the collected heat to an ORC system, was proposed.

The waste heat sources in the above-mentioned paragraph are strictly dependent on using the main engine of the marine vessels. This situation brings limitations to the utilization of ORC systems on marine vessels during the cargo loading/discharging operations and waiting periods in ports. The unavailability of waste heat sources of large amounts in ports increases the payback period and reduces the economic viability of ORC systems. In many cases for tanker ships, it is usual to utilize cargo oil pump turbines (COPT) for discharging the cargo oil in the ship to the port facility. COPTs are driven by large amounts of steam and the return of this steam is cooled down by seawater to be utilized again in the closed circuit. The second objective of this study is to define this waste heat as a utilizable source to literature and prove that the ORC systems can be more economically viable for tanker ships with the utilization of waste heat from COPTs during cargo discharge operations. A tanker ship is taken as a reference in a case study to identify both main engine based and COPT based waste heat sources. The optimal HEN design for an ORC integrated WHRS is determined for the main engine based waste heats by utilizing the pinch point analysis. Then, the integration of COPT based waste heat to optimal HEN design is carried out to make the ORC integrated WHRS operative in both navigation and discharge operation modes. In addition, the exergetic sustainability analyses are carried out for both cases to assisting decision-making processes that consist of decisions of which performance parameters should be used to achieve a better sustainability performance.

The results of this research will make a significant contribution to the development of new generation ships that are prominent in environmental impact and energy efficiency aspects in compliance with IMO and European Union standards.

1.3 Thesis Outline

In Chapter Two, the background and literature review of ORC systems on marine applications was investigated. The potential waste heat recovery options for marine vessels were represented into five categories as; internal combustion engine waste heat, auxiliary machinery based waste heat, boiler/economizer based waste heat, LNG based waste heat, and cargo oil pump turbine-based waste heat. The working principle

of ORC systems and the system efficiency improvement methods were discussed. Then, the ORC system components were presented. Different types of heat exchangers, turbines, and pumps were reviewed for various operating conditions of ORC applications. The organic working fluids utilized in ORC systems were categorized and the characteristics of good working fluids were summarized. Chapter Two continued with the widespread application areas of ORC systems. Next, a comprehensive literature review was conducted that compiles the studies of ORC system configurations for various ship-based waste heat sources and the integration of ORC systems to emission reduction technologies. The real-time applications of ORC systems to marine vessels and prototype development projects were also investigated and reported in this Chapter. In addition, the challenges for the application of ORC systems on marine vessels were discussed.

Chapter Three includes the thermodynamic modeling and heat exchanger network design of an ORC integrated WHRS of a reference container ship. In this section, the pinch point analysis was described and the optimal heat exchanger network design of the reference vessel was proposed using pinch analysis rules using the real data sets for four different engine loads. Lubricating oil, jacket cooling water, scavenge air, and the exhaust gas of the main engine were identified as the major waste heat sources of the reference ship. The energy and exergy analyses were indicated for nine different working fluids and varying evaporator pressures by parametric analyses. This Chapter includes the background of sustainability assessment, a literature survey on exergetic sustainability assessment and the exergetic sustainability assessment results of the proposed ORC integrated WHRS design.

The fourth chapter includes another case study of an ORC integrated WHRS proposal that is specifically designed for a reference tanker ship equipped with COPTs for cargo discharge operations. The proposed optimal heat exchanger network design was formed by the pinch point analysis using pinch analysis rules. The energy and exergy analyses in addition to an exergy based sustainability assessment of the proposed ORC integrated WHRS design for both navigation mode and discharge operation mode of the crude oil tanker ship were examined.

CHAPTER TWO

BACKGROUND AND LITERATURE REVIEW OF ORGANIC RANKINE CYCLE SYSTEMS ON MARINE APPLICATIONS

2.1 Potential Waste Heat Recovery Options for Marine Vessels

The primary objective of the ship-owners and the rule-makers in the maritime sector has been to increase the energy efficiency of the vessels to reduce the operational costs and environmental effects. The efficiency of a marine vessel can be affected by many factors; such as friction forces acting on the vessel and the propulsion system, using non-optimized propulsion system settings, lack of maintenance, and heat dissipation through the environment. WHR technologies offer a great deal by recovering a considerable amount of the waste energy that has been discharged overboard during the existing operations of the vessels.

The WHR technologies have been rapidly developing in the last few decades because of the unstable fuel prices and the rising trend of increasing energy efficiency and reducing emissions. Thermoelectric generators are an emerging WHR technology even though the working principle has been known since the 19th century. The working principle is based on a set of thermoelectric modules that produce electrical power output using the temperature difference with no moving parts, high reliability, and direct conversion of thermal energy. However, the practical applications of thermoelectric generators are still limited to small-scale applications and space applications because of low efficiency and high initial costs (Champier, 2017). It can be deduced that this technology is in its very early stage for utilization on marine applications but shows a promising performance when they are implemented to heat exchangers or electrical circuits (Zogogianni et al., 2016; Kimball & Wallace, 2019). Steam and power turbines are suggested as reliable WHR technologies since the technological maturity of those technologies have been reached. In addition, steam and power turbines have been widely adopted in industrial applications (Mitsubishi Power, 2020; Siemens Energy, 2020; MAN Energy Solutions, 2020a; Wartsila, 2020). The steam and power turbines are presented in WHR solutions for marine vessels that can

recover around 10% of the delivered heat (ABB, 2020; MAN Diesel & Turbo, 2014a; Mitsubishi Heavy Industries, 2016). However, the steam production for steam turbines requires high ($>400\text{ }^{\circ}\text{C}$) or medium temperature ($100\text{-}400\text{ }^{\circ}\text{C}$) heat sources (Brückner et al., 2015) that still leaves many WHR potential onboard. ORC is pointed as a promising technology for recovering low-grade waste heat (Quoilin et al., 2013; Shi et al., 2018; Mondejar et al., 2018; Chintala et al., 2018). There have some studies considering ORCs to be utilized after the steam/power turbine arrangements for the remaining low-grade waste heat (Durmusoglu et al., 2009; Nielsen et al., 2014). Additionally, ORCs have been preferred in utilizing the waste heat of jacket cooling water of the main engine (Opcon Marine, 2012; Sellers, 2017). However, ORCs have been competitively developing for utilizing all grades of waste heat in marine vessels.

Marine vessels possess many kinds of waste heat sources due to their propulsion and operation needs. Today, the propulsion of marine vessels are mostly carried out by the internal combustion engines that emit harmful emissions from the exhaust and dissipate the waste heat through many sources to the environment. WHR from those heat sources becomes attractive when considering the majority of the size of those waste heats. The internal combustion engine based waste heat sources get the most interest in recent years but there can be found many other waste heat sources onboard, which may vary according to the type of the vessel such as LNG and COPT based waste heat.

2.1.1 Internal Combustion Engine Based Waste Heat

Internal combustion engines have been producing an effect in our lives since the inventions made in the late 19th century. The efficiency of internal combustion engines has been improved significantly with the implementation of technological advancements and retrofits in time (TÜV NORD, 2019). The technological improvements of propulsion engines in the maritime sector step up with the IMO and EU regulations in the last two decades for improving the energy efficiency and achieving zero pollution goal (European Commission, 2019). Based on today's technology, around 50% of unit fuel burned in marine diesel engines can be utilized

for propelling the vessel and the remaining energy flows through the environment as waste heat as presented in the Sankey diagrams in Figure 2.1(a) and Figure 2.1(b).

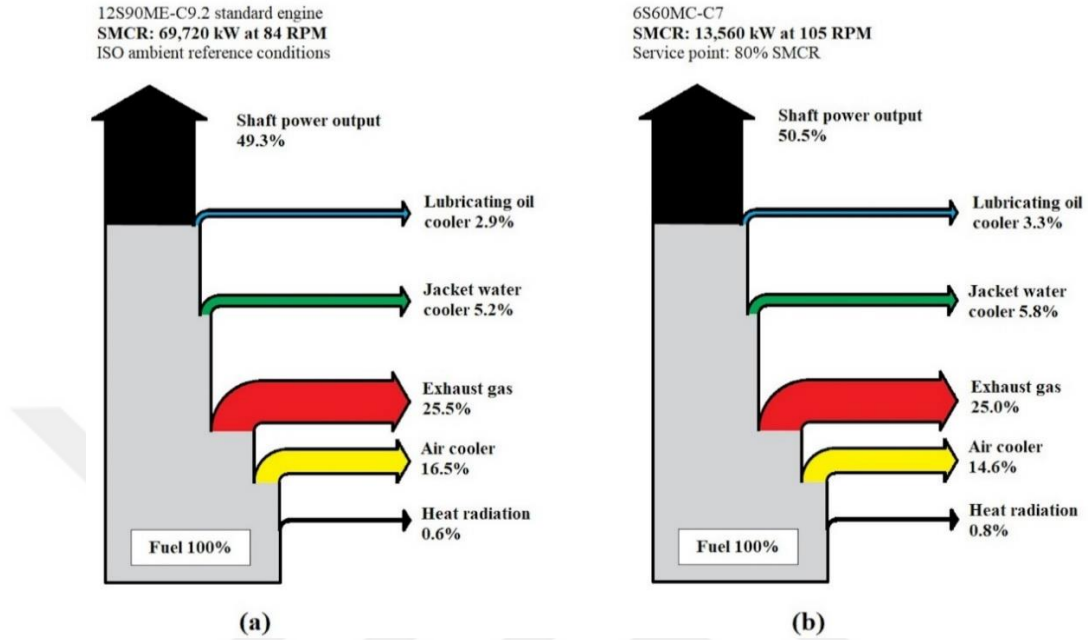


Figure 2.1 Heat balance obtained from unit fuel burned for MAN B&W engines of (a) 12S90ME-C9.2 (b) 6S60MC-C7 IMO Tier I engine (MAN Diesel & Turbo, 2014a; MAN Diesel & Turbo, 2014b)

Among various waste heat sources of marine vessels, exhaust gas is a very attractive waste heat source because of the high availability and high exergy content (Bellolio et al., 2015). The high-temperature exhaust gases are expelled from the cylinders and collected in the exhaust manifold. Marine diesel engines are equipped with turbochargers after the exhaust manifold to supply more scavenge air to the burning chambers. Turbochargers utilize the kinetic energy of the exhaust gases but the drop in the exhaust outlet pressure and the effect of the frictional forces and heat transfers cause an enthalpy loss that reduces the outlet temperature of the exhaust gas flow. The flow is then directed to the funnel to be expelled to the atmosphere directly or after passing through an economizer. The turbine outlet temperature of the exhaust gas is generally utilizable for steam production with economizers. However, the exhaust gas temperature can be increased by adjusting the in-cylinder combustion conditions to increase the WHR potential with steam turbine arrangements (MAN Diesel & Turbo, 2014a). In addition, the exhaust gas bypass method is effectively utilized for reducing NO_x emissions and IMO Tier II compliance of marine diesel engines (MAN Energy

Solutions, 2017). This is carried out by decreasing the turbocharger capacity for efficient low load operations, but higher loads require bypass of the exhaust gas before turbocharger to prevent overspeeding, where the exhaust gas temperature rises and create potentials for further utilization of waste heat with steam and power turbine arrangements. ORC technology can also be adapted for recovering the exhaust gas waste heat on marine vessels as described in the previous section. One of the most important challenges among the use of ORC systems for exhaust gas WHR is the sulphuric acid production at low temperatures of exhaust gas as the acid may corrode the heat exchanger, exhaust piping and specific engine parts. The use of low-sulphur and carbon-free fuels in marine vessels would significantly decrease that risk and increases the potential utilization of ORCs onboard.

Turbochargers and auxiliary blowers provide pressurized air to the cylinders for reducing the specific fuel consumption of an engine with an effective combustion process and increased thermal efficiency. When compared with naturally aspired engines at the same size, the density of the scavenge air is increased with the air compression process, which results in an increased engine power output. The pressurized air provided by turbochargers would be insufficient at low loads because the energy in exhaust gas flow would not be enough to drive the turbocharger at the speed required for necessary air mass flow (Woodyard, 2009). Auxiliary blowers, which are driven by electric motors, are utilized while marine diesel engines are operated at low RPMs, e.g., dead slow or slow. The turbochargers can compress the ambient air up to 10.5 bars with two-stage turbocharging arrangements, which causes a high-temperature difference between the air inlet and outlet of the turbocharger (MAN Energy Solutions, 2020b). The scavenge air temperature is cooled down from around 200 °C to 40-45 °C by scavenge air cooler thereby further increasing the air density and engine efficiency (Sanguri, 2019). The amount of waste heat discharged by the scavenge air cooler is very large considering the massiveness of the scavenge air mass flow rate and the temperature difference as can be seen in Figure 2(a) and Figure 2(b). The utilization from waste heat of the scavenge air is thus proposed in many studies in the field of WHR on marine vessels as a preheating stream for the feed water of the steam service system (Nielsen et al., 2014; Andreasen et al., 2017) or a

hot stream that is utilized for ORC systems along with other waste heat sources (Baldasso et al., 2017; Rech et al., 2017; Fuente et al., 2018; Lion et al., 2019).

Jacket water coolers are used to remove the waste heat dissipated from combustion chambers through the cylinder walls and jacket cooling water. The outlet temperature of jacket water from the cylinders is typically adjusted to 80-85 °C (may slightly vary according to the engine manufacturer's instructions). The jacket water is cooled down to 60-70 °C in the jacket water cooler to be sent back to the engine. Those temperature limits are vital for the normal operation of diesel engines. The cylinder walls, cylinder covers, fuel valves and valve mechanisms may be overheated because of the high temperature of jacket cooling water. A jacket cooling water below the specified limits may cause thermal cracks in the cylinder walls and covers because of the thermal stress caused by the temperature difference between the combustion chamber and cooling water. The jacket cooling water can be classified as a low-grade waste heat source (Yang & Yeh, 2014). As a common application, the waste heat of the jacket water is utilized in freshwater generators (a kind of desalination system) to produce freshwater from seawater in marine vessels. The seawater is evaporated at around 70 °C by creating a vacuum in the evaporating chamber by using ejectors. The excess waste heat of the jacket water is bypassed to the jacket water cooler to be discharged overboard. The waste heat of jacket water can be utilized in ORC applications reliably. The first installations of ORC systems on marine vessels are based on the utilization of only jacket water waste heat. The recovered waste heat is used for supporting the generators for the electric power demand of the vessel. Alternatively, the jacket water waste heat has the potential of preheating the working fluid for more utilization of the medium and high-grade waste heat sources with any WHR options.

The lubricating oil is used to lubricate the moving parts of the engines where it is heated because of the frictional forces and heat transferred from engine parts caused by the combustion. Lubricating oil is pumped from the engine sump tank and then recirculated back after being used. The heat collected through the closed-loop circuit has to be cooled down to specific temperature limits (from 65-75 °C to $\approx$ 45 °C) (CIMAC, 2008; Agarwal, 2020) because the lubricating properties of the oil would

deteriorate if overheated. Overcooling is also a problem for proper lubrication since the oil film will be thicker than design considerations. Lubricating oil coolers are equipped for removing the waste heat collected by the lubricating oil to overboard. As such in the jacket water coolers, the cooling media of the lubricating oil coolers is generally low-temperature cooling water but seawater can also be utilized in some cases, which is particularly determined according to the ship's cooling system. The waste heat of the lubricating oil coolers cannot be utilized for WHR with steam or power turbines because of low operating temperatures. ORC systems would be more applicable for recovering low-temperature waste heat by selecting an appropriate organic working fluid. The WHR potential and the temperature limits of lubricating oil are lower than jacket water. In these circumstances, lubricating oil waste heat can be collected before jacket water waste heat by a heat collection system, which might also collect other types of waste heats onboard, as operating temperature limits of the lubricating oil are lower than the jacket water.

The radiation heat transfer occurs from the heated engine parts, e.g., covers, fuel pumps, fuel injectors, cylinders, crankcase, exhaust valves, exhaust funnel, pipings, coolers, and moving parts. Radiation heat transfer increases the ambient temperature in the engine room but also helps to cool the engine parts. As the waste heat collection process would be challenging and the amount of the waste heat is relatively low, the radiation heat transfer is not considered as a waste heat source for WHR applications onboard yet. However, some promising studies with thermoelectric generators have been published that might potentially be utilized in this field (Zogogianni et al., 2016; Al-Nimr & Alajlouni, 2018; Kimball & Wallace, 2019; Nonthakarn et al., 2019).

The electricity demand of marine vessels is supplied by generators, which are driven by internal combustion engines but still treated as a subject of auxiliary machinery. The potential WHR sources are the same as the propulsion engines as they both operate with the same internal combustion engine working principles. However, the generators might be more compact than the propulsion engine because the generators are supposed to have smaller dimensions, which would also make it harder to place a heat collection system among the WHR sources.

2.1.2 Auxiliary Machinery Based Waste Heat

Auxiliary systems are utilized in marine vessels to supply the auxiliary requirements of the propulsion system and the other ship services, e.g., generators, provision systems, air conditioning, heating services, compressed air systems, and hot water supply. A vast majority of the auxiliary systems need cooling media because of the heat caused by the frictional forces among moving parts or the need for the reuse of the working fluid/gas in a closed-loop system. The cooling media may be low-temperature cooling water that can be collected into a central cooling system that gives a possible low-grade WHR option.

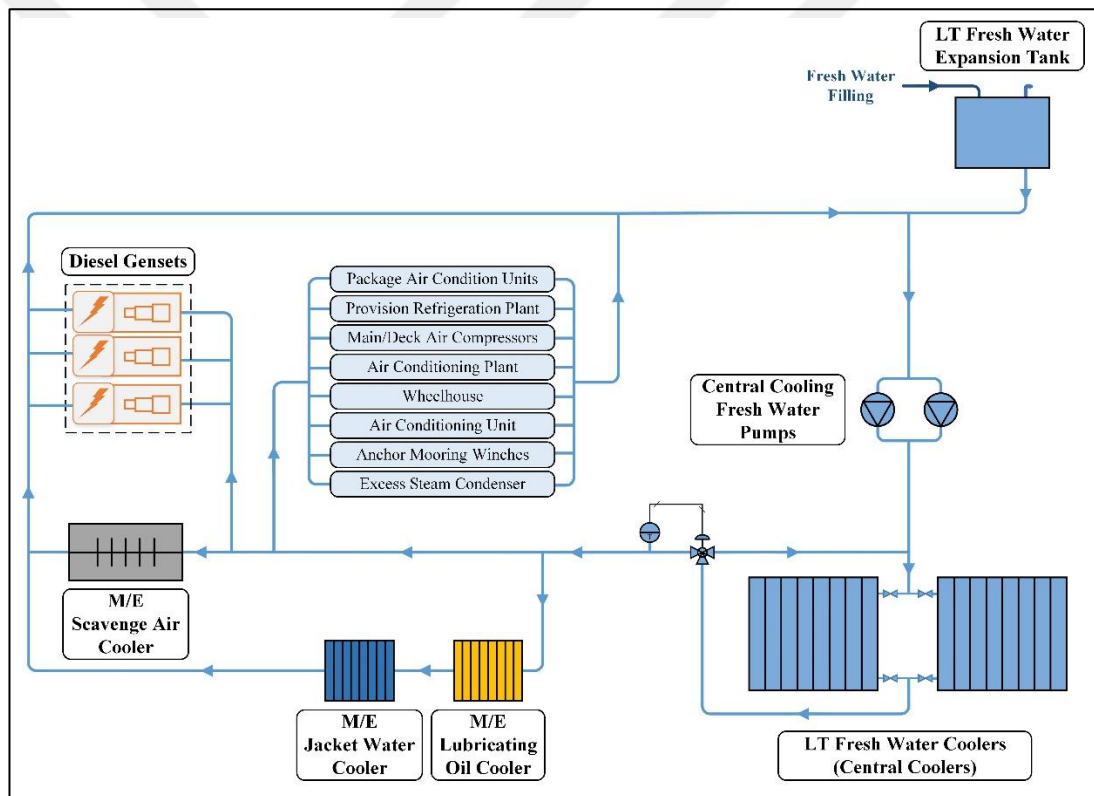


Figure 2.2 A schematic diagram of a typical low-temperature cooling water system onboard

The waste heat delivered from the auxiliary machinery is generally very low when compared with the waste heat from the propulsion system. The low-temperature cooling water in the central cooling circuit branches to cool down the auxiliary machinery in need and some of the auxiliary and propulsion engine coolers that require low-temperature cooling water as illustrated in Figure 2.2. The waste heat collected

with the low-temperature cooling water combines before the central coolers and discharged overboard through seawater. The temperature difference between the inlet and outlet of the low-temperature cooling water is low because of the high mass flow rate supplied to the central cooling circuit. The utilization of ORC might be an option for WHR from the central coolers instead of discharging the waste heat overboard but this operation might require comprehensive heat exchanger network designs and control circuits to increase the inlet temperature to the ORC evaporator.

2.1.3 Boiler/Economizer Based Waste Heat

Exhaust gas economizer systems are very common applications for WHR in ships. An exhaust gas economizer transfers the exhaust gas waste heat of the engine through feedwater by a shell-and-tube type heat exchanger that is placed on the main engine funnel. The feed water in the economizer drum boils and forms pressurized steam, thus transforming the exhaust gas waste heat into useful work by using the produced steam into different sections of the ship's steam service system such as heating the fuel tanks and circuits, adjusting the oil/fuel separation temperature, meeting the heating demand of the bilge separator and hot water supply. However, the heat demand of the ship is relatively small compared to the heat released from the exhaust of the engine (Baldi & Gabriellii, 2015). In this point of view, steam turbines are proposed for WHR in marine vessels but the applications have been limited to large vessels (main engine power of over 25 MW) (MAN Diesel & Turbo, 2014a). The steam production process in marine vessels takes place in a closed-loop circuit with steam Rankine cycle (SRC) working principle that is inefficient below 200-250 °C (Baldi et al., 2015). The exhaust gas outlet temperature of the boiler plant is generally in the range of 180-220 °C (Snow, 2001). The temperatures below these ranges are not desirable because of the low-temperature corrosion effect of the increased sulphuric acid production (Choi & Kim, 2013). It means that only the waste heat with high temperature is utilized for steam production. The remaining exhaust gases contain medium or low-grade waste heat that cannot be utilized for the process, which is potentially suitable for utilization with a suitable WHR technology like ORCs.

Boilers operate with the same working principle as exhaust gas economizers but require a fuel combustion process with burners to provide heat to boil the feed water, unlike economizers. Boilers can be placed stand-alone and/or combined with the economizer (composite) in ships. Composite boilers can provide steam with the economizer section while the main engine is running, and turn into boiler mode during port stays and maneuverings. It applies to both boilers and economizers that the steam or saturated liquid-vapor mixture needs to be cooled down to the saturated liquid phase in a hotwell tank to recirculate the feed water to the steam drum after the steam is used by the end-users of the steam service system. The cooling process is carried out by condensers that remove the waste heat overboard as shown in Figure 2.3. This waste heat can be classified as low-grade waste heat, which can be utilized for the reheat of the feed water after being pumped. Utilization of the feed water waste heat with an ORC system that collects various heat sources in a HEN can be suggested as another WHR option, which still lacks a feasibility study in the literature.

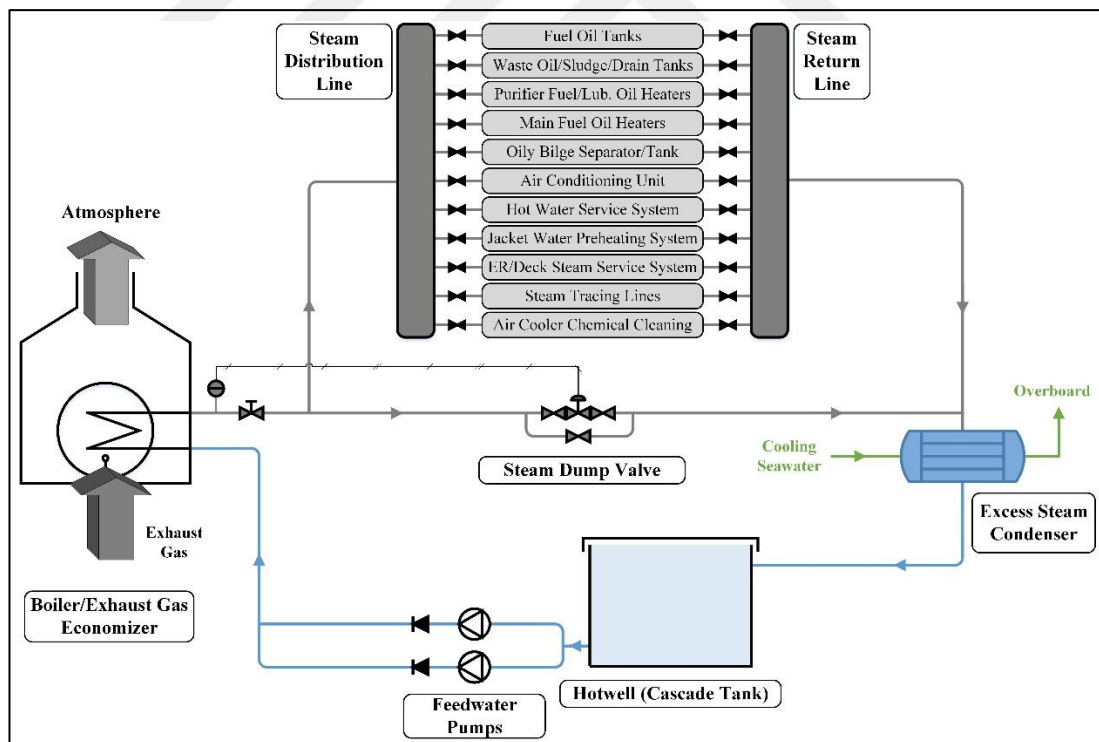


Figure 2.3 Steam service system of a ship

2.1.4 LNG Based Waste Heat

LNG based waste heat has become an interesting and popular subject for LNG-powered marine vessels in recent years. LNG is stored in high-pressure fuel tanks at very low temperatures (e.g., $-138\text{ }^{\circ}\text{C}$ and 5 bar) onboard and needs to be heated up to $27\text{ }^{\circ}\text{C}$ before using as fuel for the engine (Tsougranis & Wu, 2018). This process exposes cryogenic waste energy that is discharged from the vaporizer unit to overboard with seawater or a heating water supply as can be seen in Figure 2.4. The system is equipped with a pressure build-up unit to maintain stable operating pressure in the LNG tank. A heat addition from the main gas heater might also be required to achieve the target fuel supply temperature for the engine according to the ambient conditions of the area that the ship is sailing.

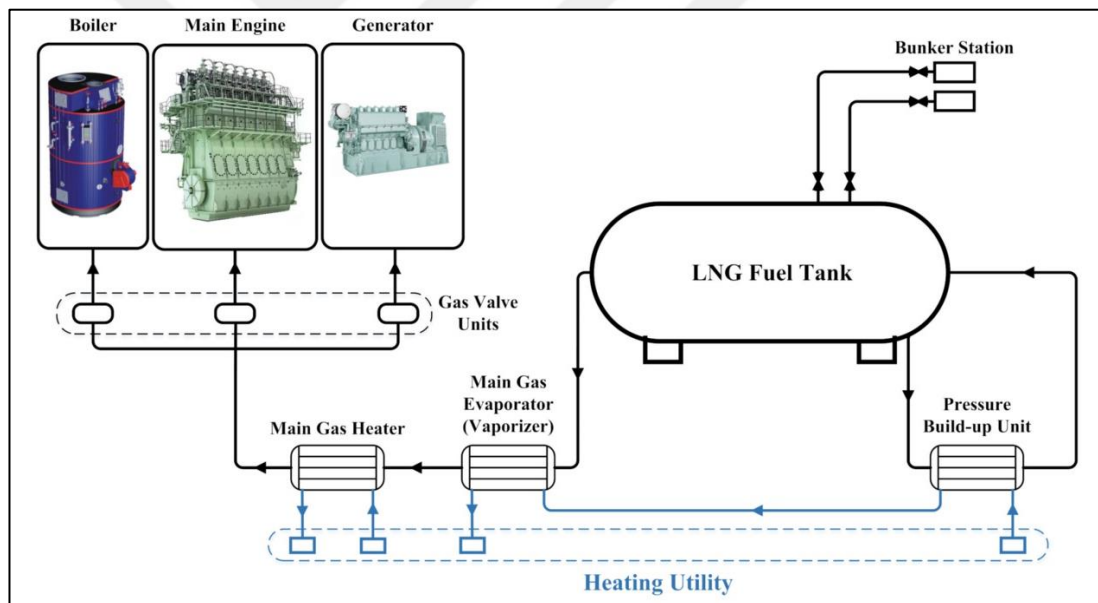


Figure 2.4 Fuel system layout of LNG-powered marine vessels (Adapted from MAN Diesel & Turbo, 2016; Tsougranis & Wu, 2018)

ORC systems can utilize the waste heat of exhaust gases after igniting the LNG at the evaporator and then use the cold waste heat of LNG at the condenser to liquefy the working fluid, that increase the energy efficiency of the ORC system. Although some ORC configurations were proposed for LNG-powered marine vessels (Rech et al., 2017; Tsougranis & Wu, 2018; Bao, 2018; Tian et al., 2020; Tian et al., 2021), there

still have lacks in the literature for experimental studies in this field. In this context, the subject of the utilization of LNG-based waste heat onboard needs more attention by considering that the LNG is among one of the most promising clean fuel alternatives for marine vessels.

2.1.5 Cargo Oil Pump Turbine Based Waste Heat

Thanks to the successful adoption of EEDI and SEEMP regulations and the implementation of newly developed technologies on ships, the overall EEOI (Energy Efficiency Operational Indicator) of oil tankers decreased by 25-26% in 2018 compared with the year 2008. It implies a reduced specific fuel consumption for the total fleet that makes them more efficient. This is supported with that the average main engine load used per voyage is reduced below 50% MCR for oil tankers according to the SEEMP reports provided from the vessels. However, the total deadweight of oil tankers increased by 19.4% between 2012 and 2018 that increased the trend of total international fuel consumption of oil tankers by 9.3% consistently. The increasing trend in the number of oil tankers continues over the years, which in turn brings on an increase in the total installed power (Faber et al., 2020). Considering all these points, the fleet now possesses even larger CO₂ emission potential than it did in 2007 as can be seen in Table 2.1.

Table 2.1 The number of oil tankers and the calculated total CO₂ emissions emitted from the international oil tanker fleet in years between 2007-2018 (Smith et al., 2014; Faber et al., 2020)

Year	2007	2008	2009	2010	2011	2012	2013	2014	2015	2016	2017	2018
Number of oil tankers	7,121	7,216	7,376	7,848	7,593	10,082	11,616	12,193	11,884	12,259	13,819	14,131
Total CO₂ emissions (in million tonnes)	160.1	159.8	138.5	120.1	129.2	141.8	142.3	142.2	151.1	159.5	164.8	161.1

The shipping emissions are mostly based on the operations while the main engine is running. It can be seen from Figure 2.5 that the GHG emission production of oil tankers and chemical tankers are relatively large during the operations while at berth or anchored because these vessels use their cargo pumps during the cargo discharge

operations to the port facility. Chemical tankers mostly use hydraulic pumps that are powered by the generators onboard. Large oil tankers are provided with steam-powered pumps (COPTs) that require very high power demand because it would not be practical to utilize electricity from generators on such scale applications.

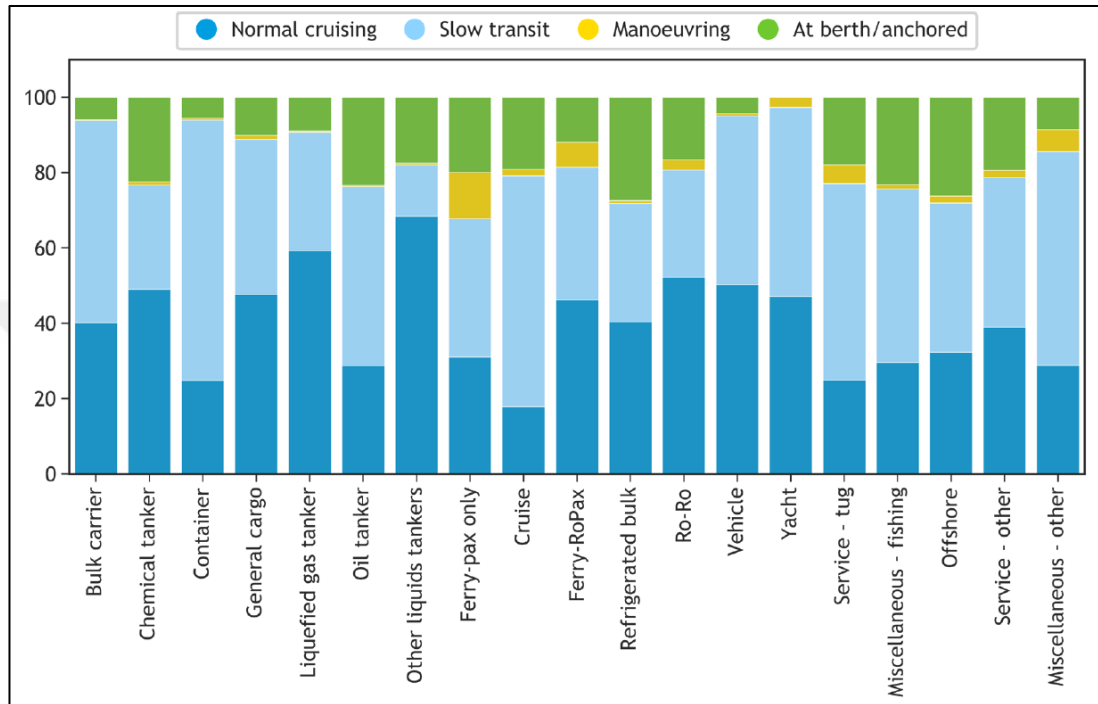


Figure 2.5 International GHG emissions by voyage-based operational phase of ships (Faber et al., 2020)

COPTs are used to discharge the cargo oil from the oil tankers to the port facility effectively. COPTs require high-pressure steam to drive the turbine section. The exhaust steam of the COPT is cooled down to saturated liquid form in a vacuum condenser to be reused in the closed-loop circuit. The cooling media in the vacuum condenser is seawater that is discharged overboard after its cooling duty is accomplished. The waste heat discharged overboard can be recovered during the discharge operations of oil tankers that would make WHRSs more feasible since the WHR will be possible in cases the propulsion engine is not running. The auxiliary boilers onboard are operated by burning fuel to create heat to produce the required steam for COPT systems. The exhaust gas from the boiler plant can be considered as another WHR option, considering the major energy losses from boiler plants typically represent 10-20% of fuel burnt (Cohen & Fritz, 1962; MIURA, 2008; NRCAN, 2018).

2.2 Introduction to Organic Rankine Cycle Systems

The organic Rankine cycle (ORC) technology is used to generate electrical power or mechanical energy by reliably converting the available heat. ORC systems can be utilized in a wide range of applications, e.g., from a few kW to large MW power plants. Although the first applications of ORC systems were experimented with ether and naphtha in the early 19th century and carried on with a few solar ORC engine designs in the early 1900s, the first modern application of ORC plants has waited up until it was built by D'Amelio in 1936. The plant was based on a solar-power-heated monochloroethane working fluid and a single-stage impulse turbine. After the involvement of today's biggest players in the ORC market with their prototypes starting from the 1960s, innovative ORC configuration proposals have been accelerated with the economic incentives and the contributions made by the researchers from other institutions (Bronicki, 2017; Tartière & Astolfi, 2017). Over the last two decades, ORC systems have been crossing through the infancy phases rapidly with the increasing number of studies and gained experiences from the successful adoptions of the ORC systems to geothermal power plants with low-enthalpy reservoirs, solar power plants, sewage/sludge disposal plants, and facilities that produce waste heat at appropriate conditions (Quick et al., 2010).

The ORC operates on the same working principle as the Rankine cycle (RC), which obtains power from a heat source by a closed-loop evaporation-condensation cycle (Cengel & Boles, 2014). The most significant difference between RC and ORC is the working fluid used in the cycle. While water is used as the working fluid in the RC, the ORC utilizes organic working fluids with low evaporation temperatures, e.g., hydrocarbons, hydrofluorocarbons, siloxanes, ethers, or blends of these organic fluids (Bao & Zhao, 2013; Shi et al., 2018). While RC applications can only be utilized with medium or high-temperature heat sources, ORC systems stay one step ahead of RC systems in wide-range applicability because of the possibility to select the best working fluid depending on the properties of the available heat source and the plant size (Tartière & Astolfi, 2017). In addition, ORC systems work more effective in low and medium power applications considering the factors, such as the availability for

simple cycle configurations, no need for a steam drum, operation at lower pressure levels, fewer stress levels at the boiler/evaporator, easier control (Sung & Kim, 2017; Saadon & Islam, 2019). However, it should be noted that the SRC system shows an upward trend and significantly greater performance in thermal efficiency and power generation when the heat source temperature is above 200 °C (Zhang et al., 2016).

2.2.1 Organic Rankine Cycle System Theory

ORC systems are a modified form of well-known SRC systems, where the system layout of both processes can be composed of similar basic components. A simple ORC system is composed of an evaporator, condenser, expander, and compressor as illustrated in Figure 2.6. The cycle can be modified with reheaters and recuperators to preheat the high-pressure working fluid before the ORC evaporator, multiple-loop configurations or combined cycles that can be constituted as cogeneration plants.

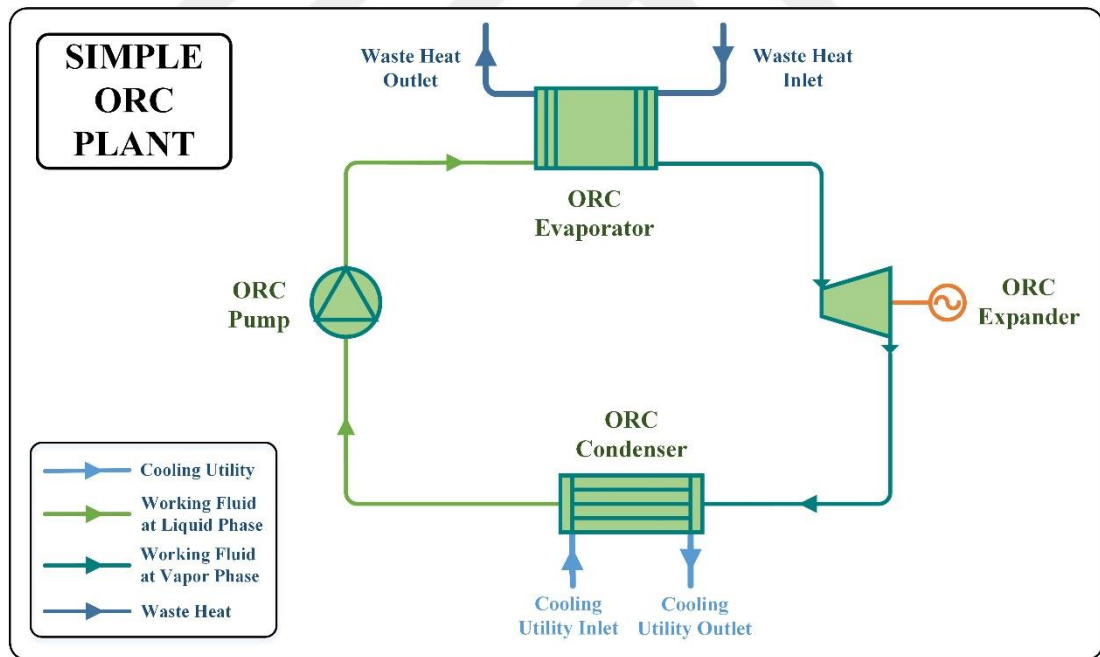


Figure 2.6 Schematic diagram of a simple ORC system

The organic working fluids used in ORCs have lower boiling points and higher vapor pressures than water/steam that allows the operation of the ORCs at lower temperatures than the conventional SRC systems (Chintala et al., 2018). This gives an

advantage to ORC systems in utilizing even very low-temperature heat sources by using organic fluids with low boiling points. Moreover, a higher-pressure difference can be achieved between the inlet and outlet of the expander thereby increasing the thermal efficiency of the system. The ORC systems have several advantages over the SRC system that make them preferable in small to medium-scale WHR applications (Tchanche et al., 2011; Bao & Zhao, 2013):

- Less heat input is required for the evaporation process.
- The evaporation of the working fluid in low pressure and temperature is possible.
- It is not necessary to use multi-stage and expensive turbine designs.
- Superheating is not necessarily required as the expansion process takes place in the vapor region, thus reducing the risk of damaging turbine blades.

The system theory of the ORC works the same as the SRC, in which the only difference takes place in the saturation curve depending on the type of the working fluid. The ideal RC is an idealized thermodynamic cycle in which friction losses and heat losses to the surroundings from all of the components are neglected. The cycle is used to convert heat into mechanical work by following the four ideal processes (Cengel & Boles, 2014) as can be seen in Figure 2.7(b):

- (1 – 2s) Isentropic compression in a pump
- (2s – 3) Constant pressure heat addition in an evaporator/boiler
- (3 – 4s) Isentropic expansion in an expander
- (4s – 1) Constant pressure heat rejection in a condenser

At state (1) of the ideal RC, the working fluid enters the pump in the saturated liquid phase. The pump compresses the working fluid to the operating pressure of the evaporator/boiler and the expander sequentially. The isentropic compression (1 – 2s) is concluded with a slight increase in the temperature of the working fluid due to the decrease in the specific volume thereby carrying up the working fluid to the compressed liquid phase before entering the evaporator/boiler at state (2s). Constant pressure heat addition from the heat source to the working fluid takes place in the

evaporator/boiler section. The water has to be superheated in boilers for SRC systems due to the negative curve of the superheated vapor slope. On the contrary, evaporators are utilized for ORC systems because superheating is not necessarily required for the organic fluids with dry fluid properties owing to the almost vertical temperature-entropy (T - s) curve between the saturated vapor and superheated vapor phases (Saadon & Islam, 2019). The working fluid then enters to isentropic expansion process (3 – 4s) in an expander and produces mechanical work by rotating a shaft coupled to an alternator. The pressure and the temperature of the working fluid drop to the state (4s) before entering the condenser in ideal conditions. The working fluid is condensed to its saturated liquid phase by rejecting heat at constant pressure from the condenser to a cooling utility, e.g., cooling freshwater, seawater, or air. The cycle is completed when the working fluid enters the pump as saturated liquid.

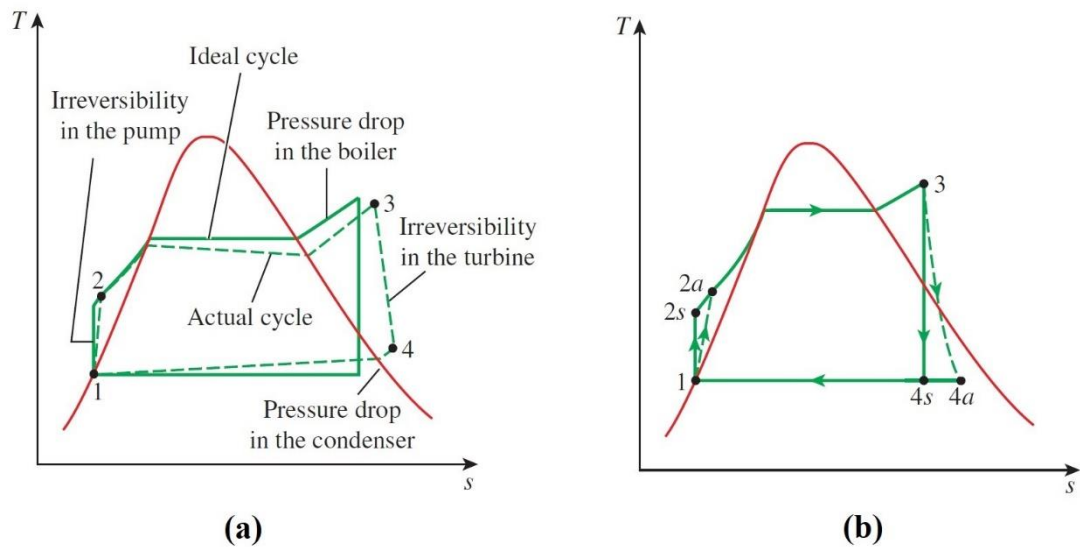


Figure 2.7 (a) Deviation of actual RC from the ideal RC (b) The effect of pump and turbine irreversibilities on the ideal RC (Cengel & Boles, 2014)

As the irreversibilities are quite difficult to avoid in real life, the actual RC differs from the ideal RC as illustrated in Figure 2.7(a). The major sources of the irreversibilities and losses are associated with the flow of working fluid through the closed-loop circuit. The expansion and compression processes of the working fluid in the expander and compressor cause internal irreversibilities, which are concluded with entropy generation. The irreversibilities in the expander have a higher impact than the

pump irreversibilities on the cycle because the pump work is lesser. Fluid friction that acts through the evaporator, condenser and piping between various components are other sources of irreversibility resulting in pressure reductions throughout the system. Pressure losses might be neglected in many applications that do not need precise results for simplicity issues. However, the working fluid needs to be pumped to a higher pressure than the ideal conditions, which requires a larger pump capacity and work input to pump (Cengel & Boles, 2014). The other major source of irreversibility is the heat losses through surroundings as the working fluid flows through components. The cycle efficiency decreases with undesired heat losses, as more heat needs to be supplied in the evaporator to compensate for these losses (Moran et al., 2018).

The basic principles for increasing the net power output and the thermal efficiency of the cycle are based on increasing the average temperature during the heat addition process in the evaporator or decreasing the average temperature during the heat rejection process in the condenser. The working fluid is expected to be in superheated vapor or saturated mixture form in the condenser after passing through the expander. The quality of the working fluid in the condenser is strictly dependent on the operating pressure of the condenser. Lowering the condensing pressure will result in a lower condensing temperature in the state (4) and higher net power output obtained from the cycle, despite the slight increase in heat input demand (Cengel & Boles, 2014).

Superheating the working fluid is another subject that needs careful consideration for ORC systems. The working fluid temperature at the expander inlet can be increased to higher temperatures without increasing the evaporator operating pressure by increasing the heat input. It increases the net power output for the same condensing pressure and the quality of the working fluid at the outlet of the expander. However, the superheating temperature is limited to the inlet temperature of the heat source and the material strength of the system components. Evaporator pressure can also affect the superheating temperature. Increasing the evaporator pressure will increase increasing the net power output by increasing the superheating temperature as a first impression. Considering the maximum allowable superheating temperature of the system (fixed expander inlet temperature), the evaporator pressure needs to be chosen

carefully as it may negatively affect the net power output because of the reduced entropy at the isentropic expansion process if the entropy for the same superheating temperature gets so close to the saturation curve. The best performance of the RC systems can be derived from comprehensive thermodynamic analyses that show the suitable superheating degree for specific working fluids at certain operating pressures (Roy et al., 2011; Cengel & Boles, 2014; Moran et al., 2018; Konur et al., 2020b).

The working fluid at the turbine outlet can be beneficial as a possible preheating source before the evaporator inlet if the turbine outlet temperature is considerably higher than the evaporator inlet temperature. Recuperators (internal heat exchangers) are equipped for this purpose between turbine outlet and condenser as illustrated in Figure 2.8 for transferring the waste heat to the working fluid that is delivered to the recuperator after pressurized by the pump. The recuperation process can yield a higher thermal efficiency for the cycle since the heat loads on the evaporator and the condenser are reduced as shown in the T-s diagram of a recuperated ORC system in Figure 2.10(a). However, recuperators add higher complexity and higher initial costs to the system (Rahbar et al., 2017). It should also be noted that the use of recuperators might not be feasible for applications that have no limit on the heat source temperature (Dai et al., 2009; Li et al., 2011; Maraver et al., 2014; Andreasen et al., 2015).

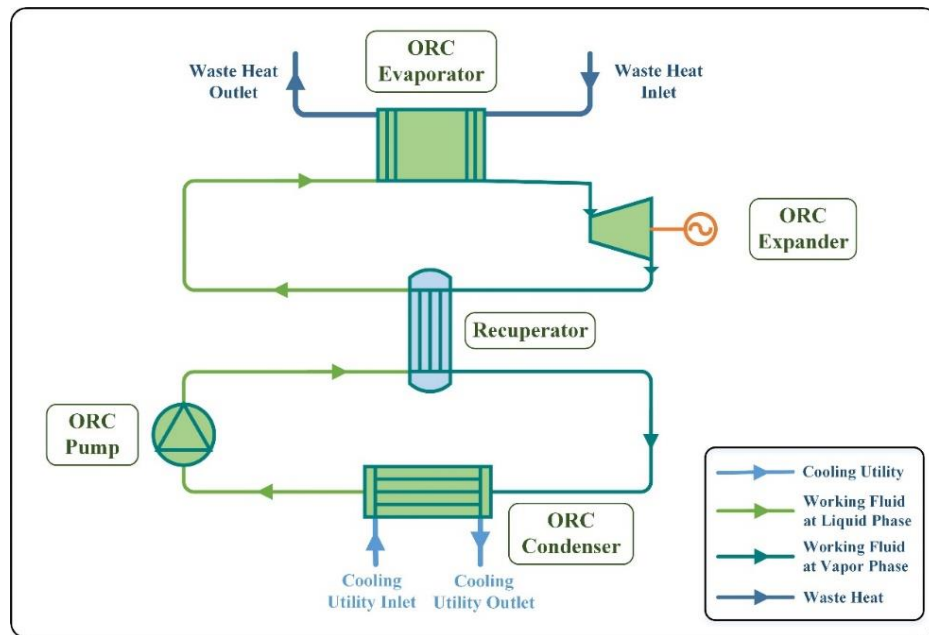


Figure 2.8 Schematic diagram of a recuperated ORC system

Reheating the working fluid between two expansion stages is another option for increasing the energy efficiency of the ORC, where superheating the working fluid is unsafe metallurgically. Figure 2.9 and Figure 2.10(b) show a schematic of a reheated ORC system, in which two expanders work in series, and corresponding T-s diagram of the cycle. The first expander is utilized for high-pressure working fluid and the second is for low-pressure working fluid that is reheated by the evaporator at constant pressure after passing through the high-pressure expander. As the average temperature at which heat is transferred to the working fluid increases, the cycle efficiency improves with the reheating process. In addition, high-temperature operations and condensation of the working fluid during the expansion process is prevented so that the risk of damaging the turbine blades is reduced (Cengel & Boles, 2014; Li, 2016).

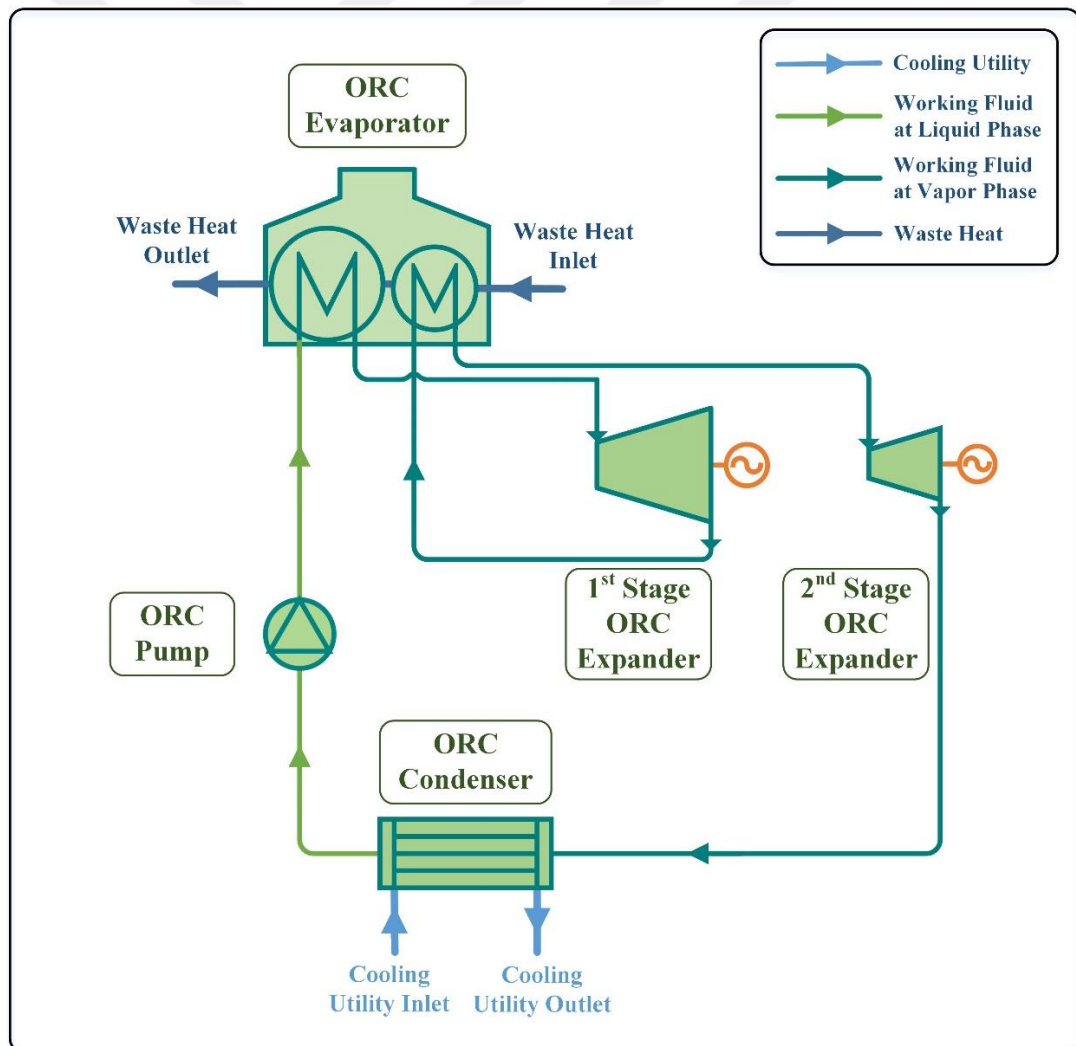


Figure 2.9 Schematic diagram of a reheated ORC system

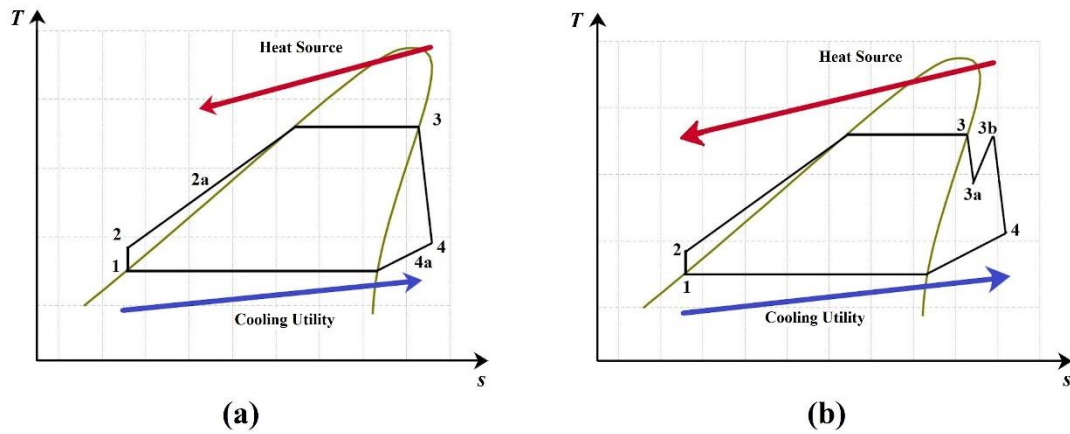


Figure 2.10 T-s diagram of (a) recuperated (b) reheated ORC systems (Adapted from Li, 2016)

2.2.2 Organic Rankine Cycle System Components

The ORC systems consist of an expander to generate work from the expansion of the working fluid, a feed pump that pressurizes the working fluid to the desired pressure level, and a heat exchanger set called evaporator and condenser that collects and removes the heat respectively. The selection of different type and size of these components directly affects the performance of ORC systems. The efficiency of an ORC system mainly depends on the selection of a proper working fluid, the heat extraction ability of the evaporator from the waste heat source, and the performance of the expander under varying pressures, mass flow rates, and heat load ranges (Bao & Zhao, 2013; Li, 2016; Chatzopoulou, 2019; Dumont et al., 2019). The selection of variable-speed pumps throughout the system can give flexibility for adjusting the mass flow rate of the working fluid and the cold utility of the condenser, thus giving controllability to the heat load extracted/rejected or outlet temperatures of the heat exchangers. The types of ORC system components have been further investigated in the following sections. It should be noted that to ensure the proper functioning of the entire system, auxiliary components are also required in addition to the basic components of ORC systems, which can be specified as a reservoir, valve sets, sensors, and an electrical panel that is capable of regulating, distributing, and monitoring the electrical power line (Zywica et al., 2016).

2.2.2.1 Evaporator and Condenser

Heat exchangers are utilized to transfer the thermal energy between two fluids at different temperatures in ORC systems (Kakaç et al., 2020). The required evaporation of the working fluid during the heat addition process to the system and condensation of the working fluid during the heat extraction process from the system make those heat exchangers called evaporator and condenser that can work in a two-phase heat transfer mechanism. The heat transfer process from one fluid to another is substantially affected by the physical properties of the fluids on both sides, e.g., the dynamic viscosity, density, thermal conductivity, and specific heat (Bartlett, 1996). Operating temperature and pressure, mass flow rates, pressure drops, thermal effectiveness, fouling tendency, cost, materials of construction, horizontal or vertical mounting arrangements, are other criteria that should be taken into consideration when selecting the type of evaporator and condenser for the ORC system (Thulukkanam, 2013).

The heat exchangers can be classified as tubular, plate type, extended surface, and regenerative according to their geometry of construction. Among those types of heat exchanger geometries, shell-and-tube, double-pipe, brazed-plate, gasketed-plate, and plate-and-fin type heat exchangers are most commonly preferred in ORC applications, which are illustrated in Figure 2.11 (Mersch, 2019).

Shell-and-tube heat exchangers are advantageous with their flexible and robust design, and easy maintainability. They also provide relatively large ratios of heat transfer area to volume and weight. However, shell-and-tube heat exchanger constructions are heavy and require a large installation area. They may deal with high pressures up to 600 bar, but plate heat exchangers might perform a better economic performance for low pressure and low-temperature operations (Thulukkanam, 2013; Kakaç et al., 2020). Double-pipe heat exchangers are generally used in small-scale ORC applications because the cost per unit surface area is relatively high. They can be preferred due to their simple construction and high-pressure applicability (Hewitt et al., 1994; Chatzopoulou, 2019). Plate heat exchangers can provide high heat transfer performance with a highly turbulent flow regime and close pinch point temperature

differences. Gasketed-plate designs may restrict the use of corrosive fluids and limit the operating pressure and temperature. Brazed-plate heat exchangers overcome these limitations with welded pairs of plates on one or both fluid sides. The disadvantage of the brazed-plate design is the difficulties in maintenance on the welded fluid sides (Shah & Sekulic, 2003). If the size of the heat exchangers gets larger, the temperature difference reduces. A decision should be made accordingly as it makes available of carrying a higher heat load, but the installation costs increases.

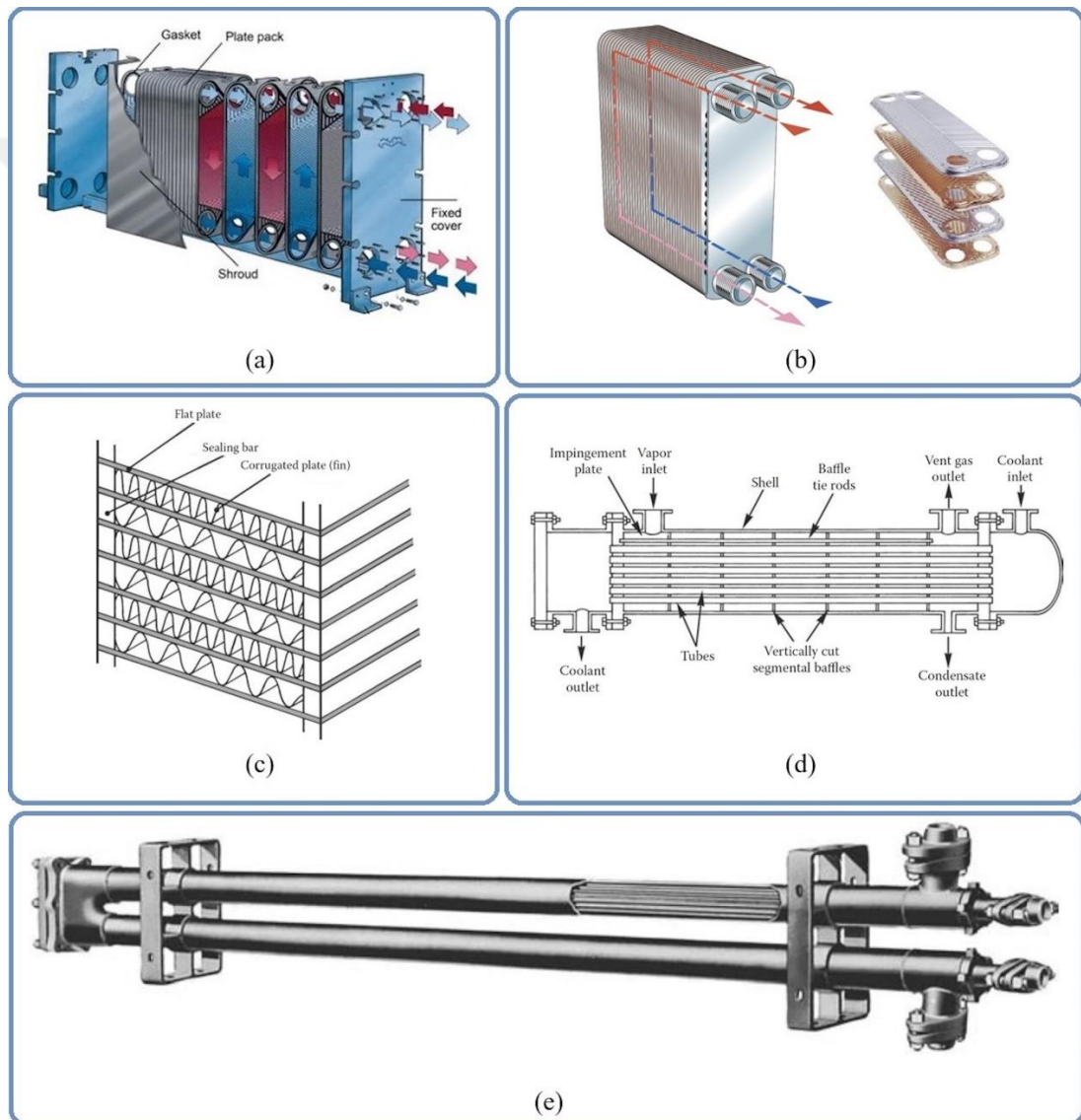


Figure 2.11 Heat exchangers types used in ORC systems (a) Gasketed-plate (b) Brazed-plate (c) Plate-and-fin (d) Shell-and-tube (e) Double-pipe (Alfa Laval; 2004; Kakaç et al., 2012; Alfa Laval, 2020)

2.2.2.2 Expander

The expander is the most expensive and technically advanced component in ORC systems and plays one of the key roles in the overall cycle efficiency of ORC systems. The selection of the type of expander is generally based on the power output range of the ORC system design (Kolasinski, 2020). However, it is also necessary to analyze and compare the performance of different expander technologies for selecting the best practice application as it depends on several parameters, consisting of the operating conditions, type of the working fluid, system compactness, complexity, cost constraints, size and weight, lubrication requirements, sealing, component availability, reliability and maintainability (Saghlataoun et al., 2014; Weiß, 2015; Dumont et al., 2019; Talluri et al., 2020).

The two types of expanders mostly used in ORC systems are volumetric-type (so-called positive displacement type) and velocity-type (so-called turbine, turbo-expander or dynamic-type) expanders. Tesla turbines are also an unconventional type of turbine that have been significantly paying attention in the field of ORCs in recent years. Tesla turbines are bladeless expanders that utilize the viscous forces and adhesiveness of fluids. The turbine design consists of a set of parallel discs that are fixed to a shaft with very tight clearances. The fluid flow is converted to mechanical energy with the momentum exchange due to shear forces by following a spiral path starting from the tangential nozzles located at the rotor periphery to the inner radius of the rotor (Sivaramakrishnaiah et al., 2017; Manfrida et al., 2017; Fiaschi & Talluri, 2019). As Tesla turbines are an emerging technology, the application areas of Tesla turbines are currently limited with tens of kW to micro-scale power ranges. Their simple construction provides many advantages in terms of compatibility to high temperature and high-pressure applications with a wide range of rotational speeds (Dumont et al., 2019).

The working principle of turbo-expanders based on the rotation of turbine blades while the high-pressure working fluid in the gaseous phase expands and transfers its kinetic energy to the turbine blades through its way on the expander. Turbo-expanders

can be categorized as axial and radial turbines following the different flow directions of the working fluid at the turbine inlet in relation to the output shaft. Both turbine types have the advantage of a compact structure with lightweight construction. However, in addition to the efficiency drops in unsteady operations, wet expansion is another major challenge on turbo-expanders because liquid formations may easily damage the turbine blades (Pantano & Capata, 2017; Talluri et al., 2020). Axial turbine blades are not appropriate for operations with low mass flow rates and high-pressure ratios because the leakages from the small tip clearance between the turbine casing and the blades increase and cause a significant drop in the expander efficiency (Alshammari et al., 2018). Therefore, axial turbine arrangements are well-suited to large-scale applications of over 500 kW with a low pressure ratio and high mass flow rate. Unlike axial turbines, radial turbines can effectively handle the required enthalpy drop in a single-stage expansion process at operations with lower mass flow rates and high pressure ratios. They are also advantageous in high-density working fluids because of the robust turbine blade design that can support high loads on the blades. Radial turbines are predominantly preferred in medium-scale applications ranging from 30 to 500 kW power output capacity. The economic viability and high-efficiency values achieved in high power outputs start reducing below this range, since micro-scale (<10 kW) turbo-expander arrangements have not been well-developed yet (Qui et al., 2011; Weiß, 2015; Alshammari et al., 2018; Talluri et al., 2020).

Positive displacement (volumetric) expanders increase the volume of the working fluid while decreasing its pressure in a series of working chambers. The energy in the working fluid is transferred to the rotors during the expansion that induce a rotating movement on the expander shaft. Volumetric expanders are used in micro to medium-scale ORC applications due to their suitability to low power applications with high efficiency and reliability at reasonable costs (Lemort & Legros, 2017; Manolakos et al., 2019). In comparison with the other types of expanders used in ORC systems, the application ranges of volumetric expanders are presented in Figure 2.12 below.

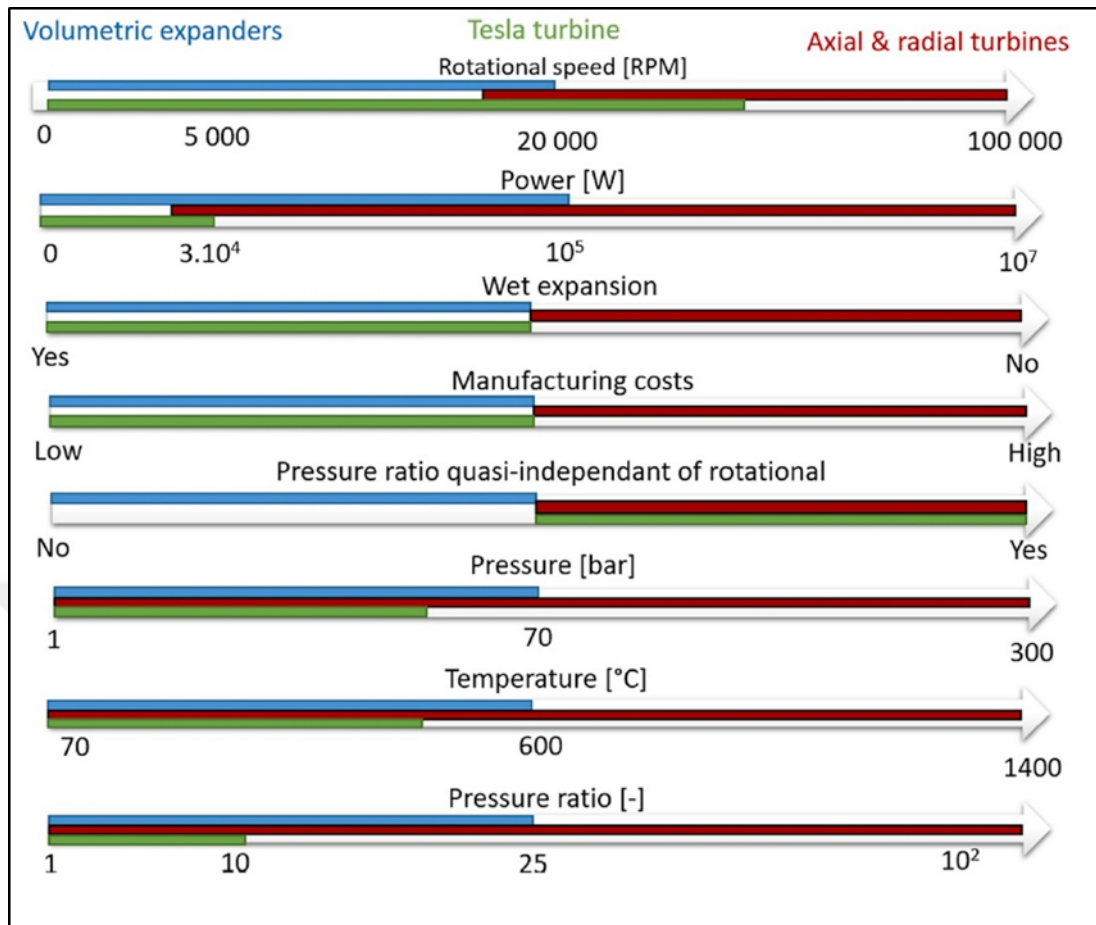


Figure 2.12 Application ranges of volumetric expanders compared to turbo-expanders and Tesla turbines (Talluri et al., 2020)

The most common types of positive displacement machines are scroll, screw, piston, roots, and rotary vane expanders. Scroll expanders expand the working fluid drawn from the center of the machine with an orbiting scroll that moves along with a fixed scroll. They are usually preferred in operations in the range of 0.5-10 kW because of their limited speed and installation cost. The boundary of the pressure ratio is limited with the built-in volume ratio (specific volume ratio of the working fluid between the inlet and outlet of the expander), which would also cause a decrease in the performance by over and under expansion losses during part-load operations. However, they benefit from working with few rotating components that leads to low noise operations and reduced vibration (Emhardt et al., 2018). In addition, scroll expanders can endure wet expansion and reach high efficiencies of up to 80% (Imran et al., 2016) for various operating conditions. Screw expanders are less costly alternatives to turbo-expanders in medium-scale ORC applications (Saghlataoun et al., 2014). They consist of two

matching helical rotors that trap the required amount of working fluid inside and expand it during their synchronized rotational movements. Wet fluids might be more preferable to dry fluids for screw machines because lubrication is necessary for screw expanders unless the working fluid has lubricant properties. They can reliably operate at above 5000 RPM with efficiency levels up to 70% even for micro-scales since the leakage and friction losses are relatively small thanks to their design with very small rotor clearance (Dumont et al., 2017; Alshammari et al., 2018). Compactness and handling wet expansion are also among the major advantages of screw expanders but they are not primarily preferred in low power applications due to the increased manufacturing costs and leakage losses (Qui et al., 2011; Imran et al., 2016; Dumont et al., 2019). Piston expanders work with the basic principles of a reciprocating piston and cylinder with inlet and exhaust valve arrangements, which makes the design complex and costly (Weiß, 2015). The piston expander design is not appropriate for wet expansion but can be assessed for high-temperature and high-pressure applications with pressure ratios up to 14. Despite the high efficiencies achieved up to 70%, they are favored in low-power system designs below 10 kW due to increased vibration and installation costs (Weiß, 2015; Dumont et al., 2019). Besides the traditional piston expanders, rotary piston (Wankel), gerotor and rolling piston expanders are other types of piston expanders that can be utilized in ORC systems (Zywica et al., 2016). Rotary vane technology is commonly used in a wide range of industrial applications with their devices of pumps, compressors and air motors. In rotary vane expander design, the rotor is placed in an asymmetric orientation and the vanes are allowed to move outwards as the rotor rotates. The volume of the trapped working fluid between two vanes is increased while moving through the exit ports and the expansion takes place. Rotary vane expanders are currently underdeveloped but studied in many researches for their ability to handle wet expansion, compactness, relatively cheap design, and low operating speeds (around 3,000 rpm) that can match generator speeds without gearboxes. Their typical pressure ratios are between 5-7 and power outputs range from a few watts to 2.2 kW (Bao & Zhao, 2013; Alshammari et al., 2018; Kolasiński, 2020). Roots expanders have two identical lobes that work in a similar working principle with screw expanders. There have limited applications of roots expanders in the literature. They can be preferred in low pressure and low expansion ratio applications for

rotational speeds up to 20,000 rpm within the power range of 1 to 30 kW (Dumont et al., 2019; Talluri et al., 2020). Geometric designs of the above-mentioned expander technologies used in ORC systems are illustrated in Figure 2.13.

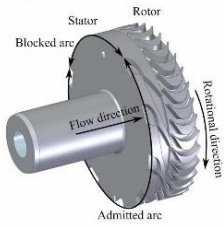
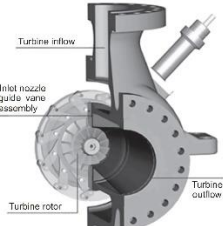
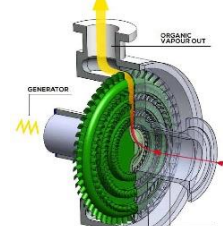
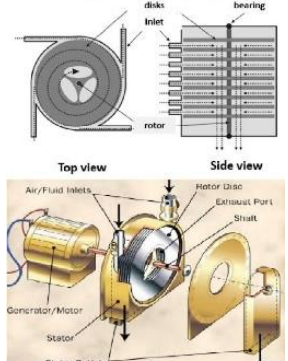
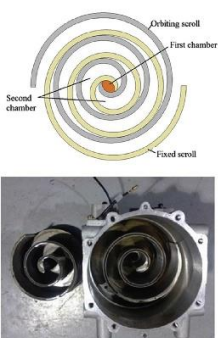
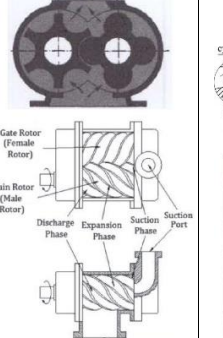
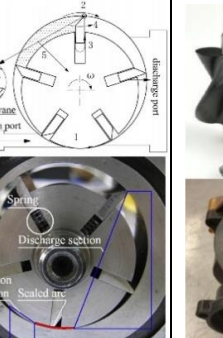
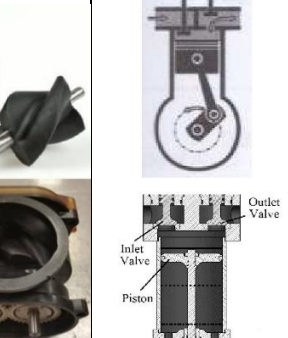
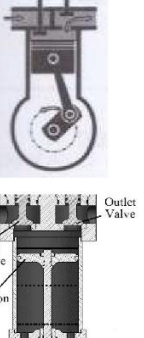
Dynamic expanders (Turbo-expanders)			Unconventional expanders	
(a) Axial turbine	(b) Radial turbine		(c) Tesla turbine	
	Inflow	Outflow		
				
Volumetric (Positive displacement) expanders				
Rotary displacement				Reciprocating
(d) Scroll expander	(e) Screw expander	(f) Rotary vane expander	(g) Roots expander	(h) Piston expander
				

Figure 2.13 Types of expander technologies used in ORC systems ((a) Seume et al., 2017; (b) Valdimarsson, 2014; Franco et al., 2017; (c) Krishnan, 2015; Sivaramakrishnaiah et al., 2017; (d) Wu et al., 2015a; (e) Weiß, 2015; Alshammari et al., 2018; (f) Milewski et al., 2017; (g) Stretch et al., 2016; (h) Weiß, 2015; Li et al., 2016)

Table 2.2 summarizes the properties of different expander technologies that can be utilized for ORC systems, where compactness is defined as the ratio of the nominal shaft power to the total volume of the expansion machine, and flexibility is the ability to work efficiently under part-load conditions.

Table 2.2 Comparison of different ORC expander technologies (Baek et al., 2005; Lampart & Jędrzejewski, 2011; Fiaschi et al., 2012; Bao & Zhao, 2013; Lemort et al., 2013; Biederman et al., 2014; Imran et al., 2016; Macchi & Astolfi, 2017; Pantano & Capata, 2017; Alshammari et al., 2018; Dumont et al., 2017; Emhardt et al., 2018; Dumont et al., 2019; Kolasinski, 2020; Talluri et al., 2020)

	High pressure and temperature operation	High speed operation	Wet expansion capability	Compactness	Flexibility	Power range	Efficiency
Tesla turbine	High range (~60 bar, 500 °C)	Up to 20,000 rpm	Tolerable	Good	Good	0-30 kW	Low (below 50%)
Axial turbine	From low to very high ranges up to 300bar, 1,400 °C	A few rpm - 100,000 rpm	Not compatible	Low	Low	Over 500 kW	Very high (~80 - 90%)
Radial turbine	From low to very high ranges	8,000 - 80,000 rpm	Not compatible	Low	Low	30 to 500 kW	Very high (up to 85%)
Scroll expander	Moderate (up to 40 bar, 250 °C)	Low speeds (1,500 - 6,000 rpm)	Very good	Good	Good	Low (<12 kW)	High (~80%)
Screw expander	Moderate (~16 bar, 190 °C)	High speeds up to 20,000 rpm	Very good	Very good	Very good	1.5 kW to 1,000 kW	Moderate (below 70%)
Piston expander	High range (up to 200 bar, 560 °C)	Up to 8,000 rpm	Tolerable	Moderate	Moderate	10-100 kW	Moderate (below 70%)
Rotary vane expander	High pressure, low temperature (up to 80 bar, ~150 °C)	Low speeds (<3,000 rpm)	Tolerable	Moderate	Moderate	0-2.2 kW	Low (~50%)
Roots expander	Low range (~6 bar)	Up to 20,000 rpm	Very good	Moderate	Low	1-30 kW	Low (below 50%)

2.2.2.3 Feed Pump

The feed pump is the least expensive component in ORC systems (Mersch, 2019) but has a major role in terms of providing controllability and efficiency. According to the heat load and the desired superheating temperature, the variation of the working fluid mass flow rate can be adjusted by variable speed pumps that use inverters to control the rotational speed of the pump. The mass flow rate of the working fluid can be considered as proportional to the rotational speed of positive displacement type pumps but the net positive suction head (NPSH - the difference between the evaporating and condensing pressure) also affects the mass flow rate in centrifugal pumps (Quoilin et al., 2013). For those systems that use conventional constant speed pumps, the control of the mass flow rate can be achieved by using a by-pass line that recirculates the working fluid from the pump discharge to the pump suction line or the

reservoir (Bianchi et al., 2017). However, this operation increases the operational costs and the back-work ratio (BWR – the ratio of the power consumption of the pump by the power generated by the expander) since the same pump power consumption remains for varying loads of the ORC system.

In conventional steam Rankine cycle systems, the pump work is usually neglected as it is too low compared to the work generated from the expander output. In ORC systems, BWR is mostly higher because the ratio between the density of organic working fluids in vapor and liquid phase is lower when compared with water (Bianchi et al., 2017). For that reason, the pump work of feed pumps that operate with organic working fluids cannot be considered negligible and significantly affects the overall energy and exergy efficiency of the ORC cycle (Maraver et al., 2014). Moreover, BWR can be described as a function of the evaporating temperature for various organic working fluids. As the evaporating temperature increases and gets closer to the critical point, BWR also increases. However, it gets lower with the utilization of working fluids that have a higher critical temperature (Quoilin et al., 2013).

Organic fluids work on high vapor pressure that may cause increased NPSH. NPSH should be taken into account at the design stage of ORC systems to avoid the possibility of pump cavitation, which may lead to damages, instabilities, and inefficiencies in the pump and reductions in the flow rate (D'Amico et al., 2018). Pump cavitation may also be increased for micro-scale ORC systems during the initial operation phase of the pump due to unsuccessful NPSH adjustment, which shows itself as serious oscillations in mass flow rate, cycle pressure and temperature (Leontaritis et al., 2015). Some strategies have been described by Quoilin et al. (2013) to reduce the NPSH, e.g., using a pre-feed pump, installing the pump to a lower place than the condenser or reservoir (gravity feeding), adding small fractions of non-condensable gases like nitrogen into the working fluid, and sub-cooling the working fluid before feeding into the pump. Sub-cooling of the condensed working fluid is a method that is usually applied to avoid cavitation but also causes reductions in the thermal efficiency of the system.

Centrifugal pumps are usually utilized in large-scale applications with high mass flow rates and pressure ratios while benefiting from the high efficiency values (~65%) that can be achieved in the range of MWs (Chatzopoulou et al., 2019). Positive displacement pumps, such as rotary-vane, gear, diaphragm, scroll, and reciprocating pumps, are rather preferred in micro to small-scale ORC applications for their ability to handle high pressure rises even at very small mass flow rates and simple control of the flow by almost proportionally adjusting the pump rotational speed (Bianchi et al., 2017). Rotary pumps are used in ORC applications with low mass flow rates and pressure ratios. Reciprocating pumps can be applied to wide-range application areas that even require low mass flow rates at high pressure ratios. The selection criteria include the tightness of the pump, which is due to the high costs and in some cases high flammability, toxicity or environmental impact of organic working fluids. Diaphragm pumps generate a pulsed flow rate and can be selected considering the tightness criteria despite their low efficiencies caused by the fluctuations in pressure and the flow rate (Quoilin et al., 2013). In addition, mixing small fractions of lubricating oil into the working fluid is recommended to increase the sealing effect through the blades of pumps and expanders (Bianchi et al., 2017). It should also be noted that the sealing effect will diminish because of the increased internal leakages at lower revolution speeds, which in turn cause drops in the isentropic efficiency of the pump.

2.2.3 Different Working Fluids Utilized in Organic Rankine Cycle Systems

The working fluids can be categorized depending on the slope of their saturation vapor curves on temperature – entropy (T-s) diagram as wet, isentropic and dry fluids. Wet fluids generally have low molecular mass (e.g., water, ammonia, and ethanol) and draw a negative slope on their saturated vapor curves (Chintala et al., 2018) as depicted in Figure 2.14(a). When the expansion occurs the fluid will more likely be in the wet (two-phase) region because of the negative slope. Wet working fluids are less likely preferred on ORC applications because they would need superheating to avoid liquid droplet impingement in the expander machine during the expansion (Chen et al., 2010). Figure 2.14(b) and (c) show that the enthalpy value increases from the critical

point (where the temperature range reaches the maximum) to the turning point (where the entropy value reaches the maximum) for both dry and isentropic fluids (Zhang et al., 2019). The isentropic working fluids are called for their almost vertical saturated vapor curve after the turning point where the entropy value keep unchanged, e.g., butene, R11, R123, R1234ze, R236ea, and R245fa. However, dry fluids show a positive curve after the turning point. The dry fluids are more likely to drop into the superheated vapor zone after expansion that removes the necessity of superheating the working fluid before the expander (Saadon & Islam, 2019). However, an extra cooling load for the condenser might have left after the expansion process. This waste heat can be utilized in a recuperator, where the waste heat is used to preheat the working fluid before entering the evaporator. Some examples to dry organic working fluids can be given as isopentane, benzene, toluene, HFE7000, MD4M, and R365mfc (Zhang et al., 2019).

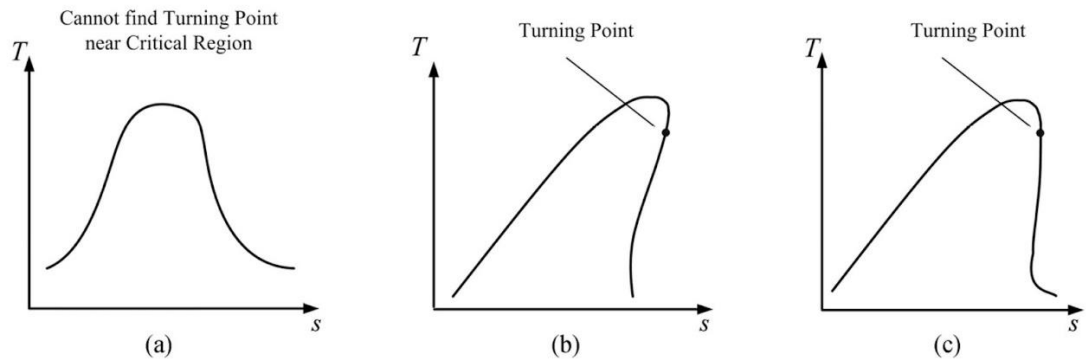


Figure 2.14 Temperature – entropy (T-s) diagrams for (a) Wet fluid (b) Dry fluids (c) Isentropic fluids (Zhang et al., 2019)

The organic working fluid has a critical role in ORC systems where it expands, compresses and transfers the heat. The selection criteria of organic working fluid include operating conditions of the system, thermodynamic performance, environmental impact, thermal stability, safety, availability, and economic viability of the working fluid (Chen et al., 2010; Bao & Zhao, 2013; Larsen et al., 2013; Bellolio et al., 2015; Thimmanoor, 2018). The heat source temperature directly influences the working fluid selection. Organic working fluids with higher critical temperatures allow the system to operate at higher evaporating temperatures within the design pressure limits, which induce an increase in the system efficiency (Loni et al., 2020).

The environmental effects of working fluids have been deeply discussed since the discovery of ozone depletion in the Antarctica region. The Montreal Protocol in 1987 is signed by 197 countries and considered the most successful global environmental action that restricted the use of ozone-depleting working fluids (EPA, 2020). The concept of ozone depletion potential (ODP) has been proposed as a measure of destructive effects of a chemical in removing ozone relative to a similar mass of R-11 (also known as CFC-11 or Freon-11), regarding the ODP value of R-11 as 1.0 (Dincer, 2018). In addition to ODP values, the working fluids are also defined by their global warming potential (GWP) values. GWP of gas stands for calculating the amount of energy emission that 1-ton of gas can carry on compared with the emissions of 1-ton of CO₂ (Vallero, 2019). The ODP and GWP values of working fluids used in ORC systems are important factors as selection criteria because as a normal operation, the working fluid leaks to the atmosphere in small proportions through piping, flanges, valves, and other components on the system design or during maintenance duties.

Table 2.3 Properties of organic working fluids (Radulovic, 2016a; Radulovic, 2016b; Lindeman et al., 2019; Zhang et al., 2019; Refrigerant Report, 2020; Linde Industrial Gases, 2021)

	GWP ^a	ODP ^b	ASHRAE 34 ^c	Critical temperature (°C)	Critical pressure (kPa)	Fluid Type
Ammonia (R717)	0	0	B2	132.4	1133	Wet
R410a	2088	0	A1	71.3	4901	Wet
R152a	124	0	A2	113.3	4520	Wet
R1234ze(Z)	<1	0	A2L	150.1	3533	Isentropic
R1233zd(E)	4.5	0	A2L	165	3573	Isentropic
R245fa	1030	0	A1	154.0	3651	Isentropic
R600	4	0	A1	152.0	3796	Isentropic
R236ea	1200	0	A1	139.3	3429	Isentropic
R236fa	9400	0	A1	124.9	3200	Isentropic
R134a	1430	0	A1	101.0	4059	Isentropic
R1234yf	4	0	A2L	94.7	3382	Isentropic
R365mfc	782	0	Unknown	186.8	3266	Dry
Benzene	Low	0	B2	288.8	4907	Dry
Toluene	Low	0	A3	318.6	4126	Dry
Isobutane	3	0	A3	134.7	3640	Dry
Cyclohexane	Low	0	A3	280.4	4080	Dry

^aGWP: global warming potential, relative to CO₂.

^bODP: ozone depletion potential, relative to R11.

^cASHRAE Standard 34 - Refrigerant safety group classification. 1: nonflammable; 2L: mildly flammable; 2: lower flammability; 3: higher flammability; A: lower toxicity; B: higher toxicity.

ASHRAE Standard 34 assigns the safety classifications of refrigerants that include toxicity and flammability data (ASHRAE, 2019). Table 2.3 shows some examples from wet, isentropic, and dry organic working fluids with GWP, ODP, and safety data. The safety class of the working fluid is an important factor, as the fluid would better not possess any threats to the safety of operators and system. If the fluid indicators show any flammability or toxicity, necessary precautions should be taken. The ORC market is capable of delivering many types of working fluids with good thermodynamic performance like hydrocarbons (so-called “natural refrigerants”, e.g. cyclopropane, isobutane, toluene, benzene, and cis-2-butene) for specific system designs (Larsen et al., 2017) however, safety issues and operation difficulties restrict the use of those types of working fluids on many applications.

To summarize, a good working fluid usually has the following characteristics (Chen et al., 2010; Bao & Zhao, 2013; Larsen et al., 2013; Bellolio et al., 2015; Li, 2016; Chatzopoulou et al., 2018; Thimmanoor, 2018):

- Dry or isentropic fluid characteristic
- High densities in both liquid and vapor phase
- Moderate critical temperature and pressure
- High specific heat and high latent heat
- Moderate evaporating and condensing pressures
- Low cost
- Market availability
- Low GWP and ODP values
- Low toxicity
- Low flammability
- Good material compatibility
- Low corrosiveness
- High fluid stability at operating temperatures

2.2.4 Applications of Organic Rankine Cycle Systems

The ORC technology has a wide range of application areas from micro-scale (a few kW) applications to large-scale (up to multi-megawatts) power plants. While micro-scale applications of ORC are applied to domestic cogeneration plants and experimental setups, medium to large-scale applications are preferred for industrial-level electricity production, e.g., waste heat recovery, geothermal, biomass, and solar power plants. According to the world ORC market overview compiled by Tartière & Astolfi (2017), the ORC market has increased its potential with more than 705 projects and 1,754 ORC units that have a cumulated installed capacity of over 2.7 GW. The applications of ORC systems have accelerated with the successful adoptions on low-temperature geothermal heat sources as previously discussed in Section 2.2. Geothermal power generation has been dominating the ORC market and significantly increasing its potential after 2009 and the entrance of new ORC manufacturers as can be seen in Figure 2.15. The United States, followed by Turkey and New Zealand have the largest share in the installed ORC capacity due to their rich geothermal sources.

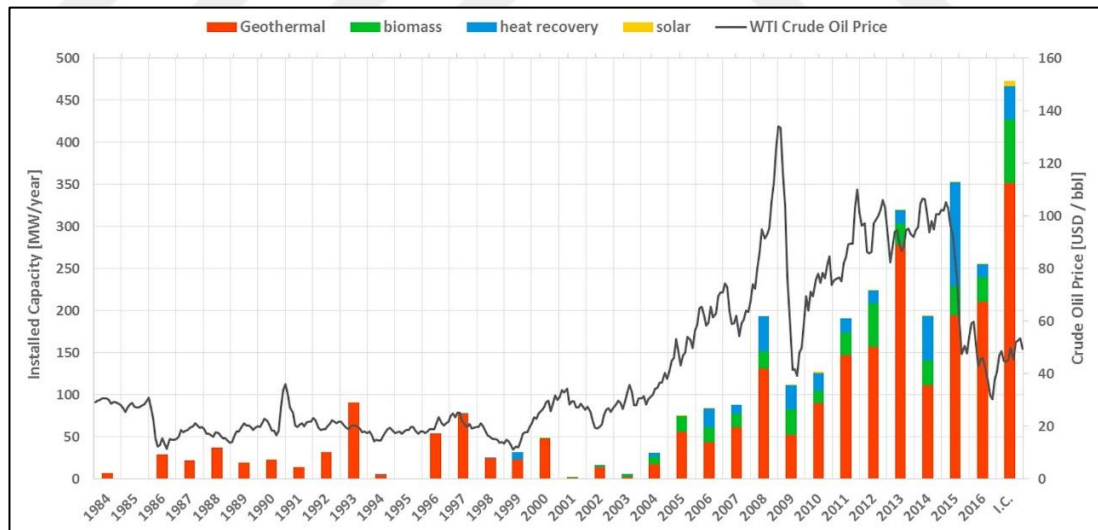


Figure 2.15 The trend in the installed ORC plant capacity for several application areas in the period of 1984-2017 (Tartière & Astolfi, 2017)

Biomass power plants and WHR applications are mostly preferred in the United States, Turkey, Germany, Canada, Italy, and China (Tartière & Astolfi, 2017). The applications areas of ORC systems are further discussed in the following sections.

2.2.4.1 Geothermal Power Plants

The source of geothermal energy is based on the heat stored interior of the earth in which the temperature is over 1000 °C. The typical temperature range of geothermal energy reservoirs is between 50 to 370 °C (Quick et al., 2010). This heat can be directly used for space heating or heating requirements of industrial applications. The geothermic heat can also be transformed into electricity by energy conversion systems; meanwhile, both electricity production and heating can take place in cogeneration plants. Dry and flash steam power plants effectively carry out electricity production for heat sink temperatures of above 220 °C (Liu et al., 2016). Figure 2.16 depicts that ORC and Kalina cycle plants are the most suitable heat to power conversion technologies for heat source temperatures between 80 - 220 °C. Considering the largest availability of geothermal reservoirs on earth is between 100 - 220 °C (Hettiarachchi et al., 2007), ORC systems are playing an important role in geothermal power plants for electricity production (Song et al., 2020).

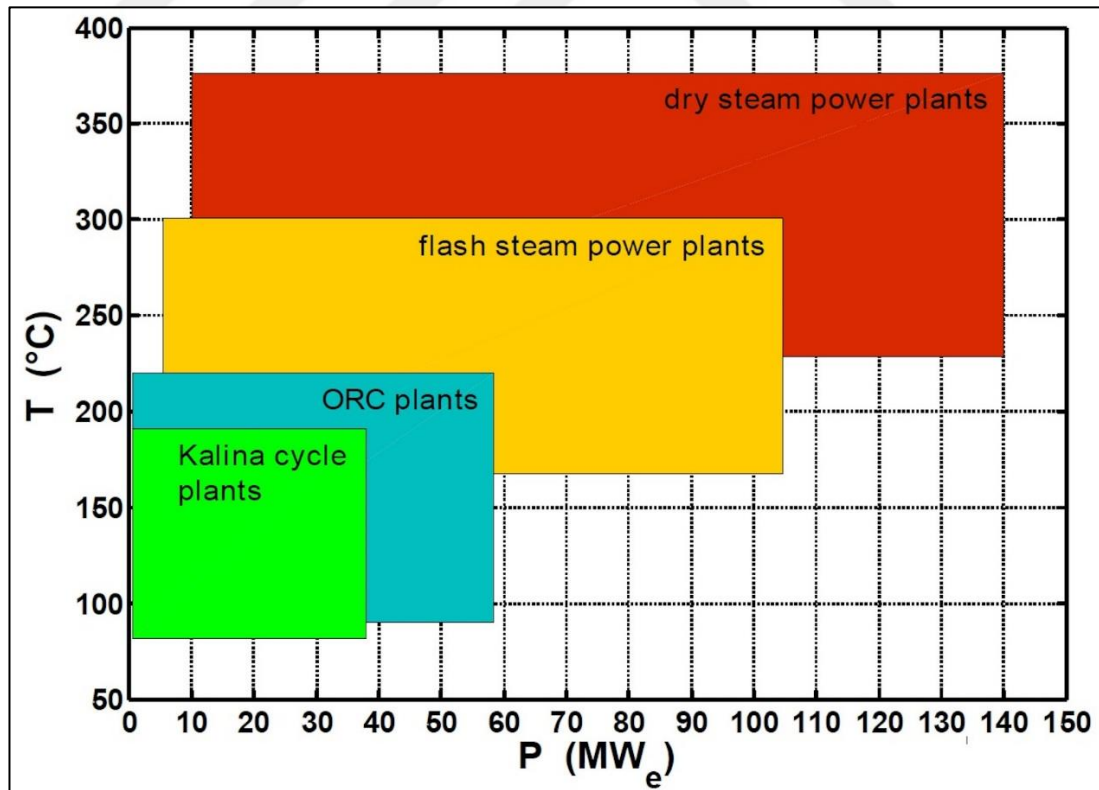


Figure 2.16 Characteristic hot water plant reservoir temperatures and power ranges for heat to power conversion systems (Clauser, 2006)

In geothermal power plants, the hot water is sucked from the production well by a brine pump. Then, the hot water circulates through the evaporator of the energy conversion unit and to the injection well. The power consumption of auxiliary pump units is relatively high (from 30% to 50% of the gross output power) because the brine has to be circulated over very large distances below the ground and with high flow rates. The BWR of the ORC plants reduces with the decreasing evaporation temperature in low-temperature geothermal power plants, resulting in higher auxiliary energy consumption and reduced energy efficiency. Therefore, power generation with geothermal energy is not economically viable for temperatures below 80 °C, where it can only be utilized for distinct heating purposes (Quoilin et al., 2013). In this regard, hybrid-powering options can be assessed for geothermal power integration with solar power (Cakici et al., 2017; Erdogan et al., 2017).

2.2.4.2 Biomass Power Plants

Biomass energy is a promising renewable energy source for heat and power generation, which offers cost-efficient, viable, flexible, and environment-friendly solutions. Amongst other renewable sources, biomass energy has the advantages of wide-range availability, producing enough heat for heating purposes, and providing continuous energy supply from biomass power plants (Wegener et al., 2018). Direct combustion of wood and other organic matter is widely adopted today and has the most advanced technologies for energy production. As long as the moisture content is below 50%, it is possible to burn any type of biomass. Pyrolysis, gasification, and liquefaction processes are emerging technologies that use a series of chemical reactions to produce bio-oils and biogas (Kim et al., 2017; Abbas et al., 2020).

The heat utilization in biomass power plants is an important factor to ensure the economic viability of the plant by achieving high energy conversion efficiency. However, it might limit the installed plant power capacity to a few MW scales due to the difficulties in heat transportation over long distances. Steam RC systems are not competitive in this power range but can be utilized in higher power capacities (Quoilin et al., 2013). In this regard, ORC systems are proposed as a promising technology for

producing electricity and/or heat from the produced heat through biomass combustion in low to medium power ranges.

2.2.4.3 Solar Thermal Power

Solar power has been attracting the world for almost five decades since the oil crisis hit the world heavily. The search for alternative energy sources to fossil fuels increased the investments in solar power, which has proven its existence in the 19th century.

National Renewable Energy Laboratory (NREL) has been tracking the technological advancements on solar cell technologies in “Best Research-Cell Efficiencies Chart” from 1976 to the present (NREL, 2020). The crystalline silicon photovoltaic (PV) cells have gone so far up until now with 27.6% maximum energy conversion efficiency and have the largest market share among other technologies benefiting from the advantage of relatively low production costs. However, multi-junction PV cell technologies increased their energy conversion efficiencies up to 47.1% by using concentrator and solar tracker combinations that direct large amounts of sunlight to a much smaller spot where a solar cell is located. PV cells with concentrated solar power (CSP) technologies mostly require cooling systems due to the substantially increased temperature on the cell surface. In general, PV cell efficiencies decrease with the increased temperature in relation to the temperature coefficient of the cell (Konur et al., 2020a), so cooling systems give benefits to the parabolic dish type CSP systems in particular by both increasing the cell efficiency and transferring the heat to be utilized as a thermal heat source. The solar tower is another type of point-focused CSP systems. A large number of concentrators reflect the sunbeam to a collector on the tower (Quoilin et al., 2013). Parabolic trough solar collectors (PTSC) also use parabolic shaped linear reflectors that concentrate the sunbeam on a linear receiver located at the focus of the parabola (Kalogirou, 2014) and offer low-cost solar thermal powering options (Erdogan et al., 2019). Many other types of solar thermal collector systems have been developed in the past decade consisting of salt-gradient solar ponds, flat plat collectors, evacuated tube collectors, linear Fresnel collectors, and compound parabolic collectors (Orosz & Dickes, 2017).

CSP systems are beneficial for utilizing solar thermal energy. The collected thermal energy needs to be supplied to an energy conversion system to produce electricity from the heat. Steam RC systems have been widely adopted for high-temperature heat utilization. However, the same limitations apply to steam RC systems as in geothermal or biomass power plants. Stirling engines for small-scale plants with high concentration factor and combined cycle for high-temperature heat sources as in solar towers can also be used as alternative power cycles (Giovanelli, 2015). ORC systems are advantageous with their ability to work efficiently on lower temperatures and wide power ranges up to multi-MWs for solar power applications. The schematic diagram in Figure 2.17 can be given as a particular example of a solar ORC system.

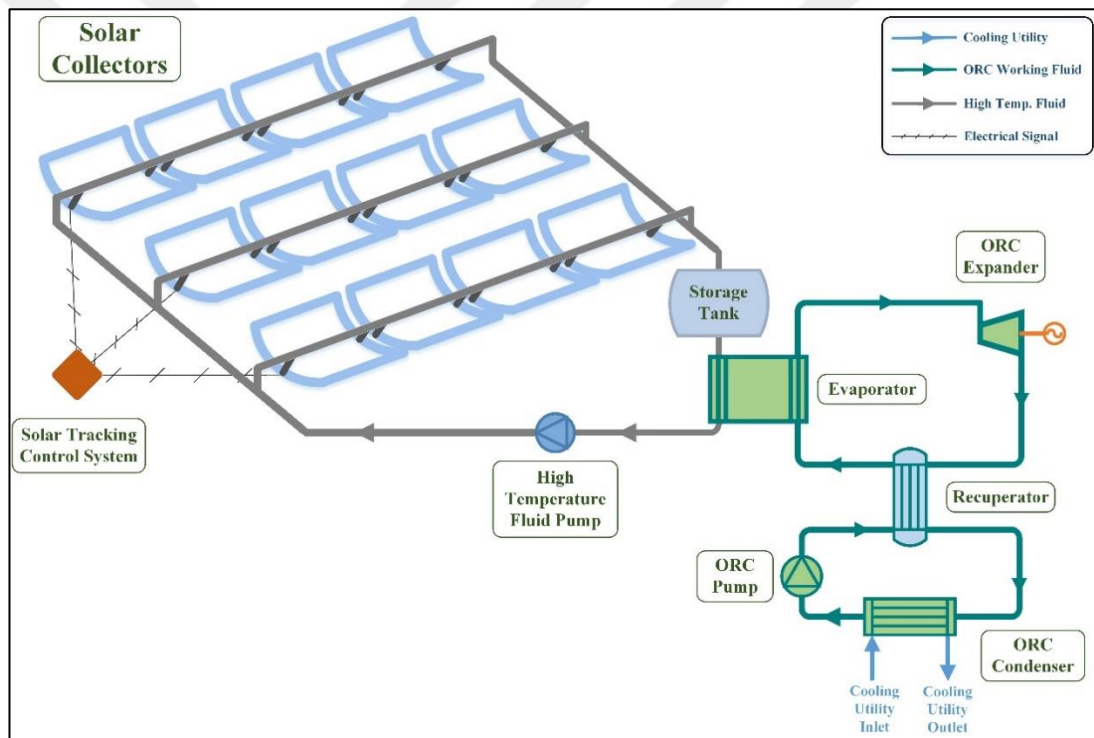


Figure 2.17 Schematic view of a solar ORC system with parabolic trough solar collectors

Solar ORC systems are preferred even in kW scale applications due to their relatively higher efficiencies, higher flexibility, simple operation and lower installation costs compared to other energy conversion systems (Quoilin et al., 2013; Exergy, 2020). They can easily and quickly adapt to load variations and allow efficient part-load operations (Turboden, 2020). The solar ORCs are increasingly considered for grid-connected or off-grid power generation systems, desalination systems, irrigation

systems, cogeneration systems for residential/industrial purposes and hybrid energy utilization systems (Orosz & Dickes, 2017).

2.2.4.4 Waste Heat Recovery

The ORC system applications for waste heat recovery aim to increase the energy efficiency of a system and decrease the energy demand. These outcomes are accompanied by improved sustainability performance and reduced specific production costs (Santarossa & Barbon, 2020). The ORC systems are beneficial for recovering the waste heat generated from micro to large-scale applications such as cogeneration power plants (Obi, 2015; Oyewunmi et al., 2017; Morais et al., 2020), marine vessels (Larsen et al., 2013; Baldi & Gabrieli, 2015; Sellers, 2017; Mito et al., 2018; Lion et al., 2019; Kaya et al., 2020), ground vehicles (Boretti, 2012; Rosset et al., 2018), cement (Ustaoglu, 2017; Ahmed et al., 2018; Fierro et al., 2020), steel (Campana et al., 2013; Turboden, 2014; Couvreur et al., 2020), chemical (Yu et al., 2017), oil and gas (Campana et al., 2013; Song et al., 2014), glass (Turboden, 2017; Danieli et al., 2019), ceramic (Peris et al., 2017), paper and pulp (Eyidogan et al., 2016; Lin et al., 2017; Costa et al., 2019), food and beverage processing (Tchanché et al., 2011; Pantaleo et al., 2018) industries. The cold (cryogenic) waste heat from LNG regasification plants (Astolfi et al., 2017; Sun et al., 2018; He et al., 2020) and LNG powered vessels (Tsougranis & Wu, 2018; Tian et al., 2021) are also utilized efficiently by ORC systems if any hot waste heat source is readily available.

The electricity generation from industrial waste heat with ORC systems may provide significant contributions to economic and environmental sustainability. The potential of WHR in European countries has been studied by Campana et al. (2013) for only including cement, steel, glass, oil and gas industries. 2,705 MW of ORC gross power is estimated for the selected industries with 21.6 TWh/year electricity production, which represents almost 2% of the European Industry consumption. The savings of almost 2 billion euros and elimination of 8 million tons of GHG emission should be taken as considerable outcomes of WHR systems. The assessment of waste heat recovery potentials for China (Lu et al., 2016) and North America (Elson et al.,

2015) are further investigated for energy-intensive land-based industries showing more than half of the delivered waste heat is low-grade that requires ORC utilization for increased power outputs.

The ORC technology can utilize waste heat with a wide temperature range from any available heat sources. This makes ORC technology a better choice than traditional steam RC systems for WHR applications that do not possess high-temperature waste heat (Loni et al., 2020). The ORC systems have the advantage of utilization in small-scale WHR applications with simple plant configurations for lower specific costs. However, the investment costs are relatively high and uncompetitive compared to other power generation options like wind, solar, internal combustion engines, etc. (Tocci et al., 2017). The investment cost trend declines with the increased power outputs, which makes ORC systems more preferable in medium-scale power range WHR applications. However, ORC systems for any power range should be reviewed considering the operational continuity of electricity production, space restrictions and economic payback periods because many renewable energy solutions are strictly dependent on the ambient conditions and require large installation areas. In addition, the consideration criteria should include that the waste heat produced from the industrial or domestic processes is better to be reduced to promote a sustainable life.

2.3 Organic Rankine Cycle Systems for Marine Applications

In this section, ORC systems for marine applications were discussed with a comprehensive literature review and the challenges related to ORC system applications on marine vessels.

2.3.1 Literature Review for Studies on the Organic Rankine Cycle Systems Used in Ships

The literature survey shows that there are several papers on the gathering of waste heat from different sources of marine diesel engines. Determination of the WHR potential, selection of the organic working fluid for the ORC systems, different ORC

system configurations and the payback time of the ORC systems have been presented in the papers as mentioned below.

2.3.1.1 Organic Rankine Cycle System Configurations for Various Ship-Based Waste Heat Sources

ORC systems for marine applications have received significant attention in the recent decade. The researchers investigated different ORC configurations by utilizing specific waste heat sources for marine applications. For example, Choi & Kim (2013) analyzed the performance of a dual-loop heat recovery system that comprises an upper trilateral cycle and a lower ORC for the case of a 6,800 twenty-foot equivalent unit (TEU) container ship operating on a China-Korea-U.S. route. Only the waste heat of the main engine exhaust gas is theoretically recovered in the study. The WHRS exhibited a maximum net output of 2,069.8 kW, a maximum system efficiency of 10.93%, and a maximum system exergy efficiency of 58.77%. The results show that an improvement of 2.826% in the propulsion efficiency resulted in about 6.06% reduction in the specific fuel oil consumption.

Nielsen et al. (2014) designed an advanced WHRS by fitting an ORC system to a conventional WHRS. A container ship that has a prime mover of 10,126 kW power was chosen as the case study. The numerical modeling results show that the proposed system is capable of extracting sulphuric acid from the exhaust gas up to 98%, while at the same time increasing the combined cycle thermal efficiency by 2.6%. Wang et al. (2014) proposed and analyzed an ORC system in combination with a desalination plant that is driven with scavenge air waste heat. The results show that 2.61 MW power output can be obtained from the ORC system in addition to more than 200 tons of freshwater production per day from the reference marine vessel having a 12S90ME-C9.2 main engine. Bellolio et al. (2015) proposed an ORC integrated WHRS design using the exhaust gas waste heat of both main engines and diesel generators. They conducted the analyses for four different operating modes of the vessel to assess the WHR potential high load and economic load operations of the engines. The WHR potential from the gensets is also studied on operation mode 2. The results represent

that the total power output generated by the vessel can be increased by the range between 4-5%, which can lead up to a 6% reduction in fuel consumption. In their studies based on exergy analysis, Baldi & Gabrielli (2015) estimated that a fuel saving of 5-15% for a Panamax chemical/product tanker could be achieved with the use of ORC systems while calculating that the total efficiency of diesel-powered ships can be increased to 60%. It is possible to find many theoretical studies in the literature that support these data and that have been carried out for low and high-temperature waste heat sources in ships (Durmusoglu et al., 2009; Larsen et al., 2013; Sprouse & Depric, 2013; Grljušić et al. (2014); Baldi et al., 2015; Pierobon et al., 2015; Soffiato et al., 2015; Lion et al., 2016; Fuente et al., 2017; Koroglu & Sogut, 2017; Mondejar et al., 2017; Akman & Ergin, 2019; Mondejar et al., 2018; Akman & Ergin, 2020; Liu et al., 2020a).

Song et al. (2015) conducted a study of WHR with the ORC technology from a ship's diesel engine having 996 kW power output. They designed two separate ORC, which is optimized and non-optimized using both cooling water, and the main engine exhaust gas as waste heat sources. Non-optimized design is presented by separate two ORC systems that collect the waste heat from the sources. The optimized ORC system is designed using the jacket water as the preheating medium and the exhaust gas as the evaporating waste heat source. Maximum net power output was chosen for selecting the appropriate working fluids and defining the optimum system parameters. The results showed that the optimized design is technically feasible and economically attractive. The net power output from the optimized system is 1.4% lower than of the traditional design however the initial costs can be reduced by 34.5%. Uusitalo et al. (2019) analyzed large cruise ships to recover waste heat of the steam RC process from the condenser with an ORC system integration to recover the lower grade waste heat of the boiler feed water. The simulated power production from the combined cycle reaches up to 950 kW for the steam turbine and 600 kW for the ORC system. The maximum annual electrical energy is estimated at 5300 MW/h that corresponds to about 1.5% of the cruise ships total energy production. Liu et al. (2020a) also proposed a WHRS design that combines steam RC with ORC for jacket water and exhaust gas WHR from a 14-cylinder two-stroke marine diesel engine. A comparison of the

proposed system with a dual pressure ORC system and a steam RC system was carried out to assess the most suitable ORC system design for the reference marine vessel. The results show that the proposed combined WHRS design could improve the thermal efficiency of the engine by 4.42%, while dual pressure ORC and steam RC system could only increase by 3.42 and 2.68%, respectively. Another research conducted by Liu et al. (2020b) introduced the performance analysis of a combined cycle with steam RC (SRC), ORC and absorption refrigeration cycle (ARC) for the same marine diesel engine. The results show that the designed WHRS can produce 7620 kW electricity in addition to 2940 kW cooling energy, which improves the thermal efficiency of the engine by 10.5%. Bounefour et al. (2020) conducted an exergy analysis of various ORC-VCC (vapor compression cycle) for WHR onboard ships. The results indicate that R600 as working fluid has the best performance. ORC with internal heat exchanger and cascaded ORC systems exhibit an average of 0.52% and 2.41% higher COP values in comparison with the simple ORC system under given operating conditions.

Yang & Yeh (2015a) presented the optimal thermodynamic and economical performances of the ORC system to recover the exhaust gas waste heat from a large two-stroke marine diesel engine of a merchant ship. They performed a parametric analysis to find the effect of the inlet and outlet pressure of the turbine, turbine inlet temperature, and the type of ORC working fluid on the performance of the system. The results showed that R1234ze has the highest thermal efficiency among the others. In addition, their proposed system can reduce 76% CO₂ emissions per kilowatt-hour compared with traditional diesel oil feeding. Another study from this research group (Yang & Yeh, 2014), developed thermodynamic analysis and a finite-temperature difference heat transfer method for recovering waste heat from cylinder jacket water of marine engine to determine the thermal efficiency, total heat exchanger area, and exergy destruction of ORC system. In addition, six different working fluids with environmentally friendly to compare the performance of the ORC. The results showed that R600a performed the best performance followed by R1234ze, R1234yf, R245fa, R245ca, and R1233zd, respectively. Yang (2016) investigated the economic performance of the WHRS for a marine diesel engine based on transcritical RC. Six

different working fluids were used in this system. A parametric analysis was conducted to assess the effect of inlet temperature and pressure of the expander on the net power output and thermal efficiency of the system. In addition, an economic optimization study was performed to identify the optimal parameters in the WHRS. The results showed that the optimal levelized energy cost of R236fa is the most suitable one and lower than that of R1234ze, R134a, R152a, R1234yf, and R290. On the other hand, both the net power output and thermal efficiency of the system when R152a is used as the working fluid are larger than those when the other type of fluids are used. In addition, R236fa yields the shortest payback period among the other fluids. The payback period reduction of ORC systems by mixing the working fluids of R236fa, R245fa, R600 and R1234ze in the optimal mass fractions was investigated by Yang (2018) for large marine diesel engines. The study was conducted with a simple ORC system utilizing the exhaust gas waste heat. The results indicate that using mixed working fluids are an effective way to reduce the payback period of the installed ORC system. The payback period of R600/R1234ze (0.5/0.5) is shorter than its alternatives that is calculated between 5.7 and 6.2 years according to the operating conditions. Ma et al. (2018) conducted a study to assess the payback period of an ORC system installation that utilizes the exhaust gas waste heat of a 10,000 TEU container ship. They compared six working fluids of R123, R141b, R245ca, R245fa, n-pentane and isopentane. Since the only waste heat source is exhaust gas, a degree of freedom is given to the evaporating temperature of different fluids for optimization. The shortest payback time is calculated for R141b at 195.3 °C, which is around 1.5 years. Zhu et al. (2018) confirmed the results of Ma et al. (2018)'s study with their thermo-economic analysis based optimization method utilized for a 662 kW diesel engine that produces exhaust gas waste heat at the range of 293 - 387 °C.

As plentiful sea water is readily available for marine vessels, the cooling media for the ORC condenser has not been seen as a problem for years. However, the cryogenic waste heat of LNG fuel attracted many researchers in the field of marine ORC because it has been realized that the cold waste heat of LNG can be utilized for cooling of ORC working fluid in the condenser to very low-temperature ranges. Sung & Kim (2016) introduced a novel dual-loop ORC system design for LNG carriers having propelled

by dual-fuel engines. The system consists of an upper recuperated ORC loop for exhaust gas waste heat and a bottom ORC loop that recovers both jacket water waste heat and the cryogenic waste heat of LNG fuel. The proposed ORC system design can work at low evaporating temperatures as the cryogenic waste heat is used as the cooling medium in the system. The WHR potential from the proposed ORC system is calculated to achieve up to 5.17% more power output than the existing dual-fuel engine output with R125 working fluid. The performance of different ORC configurations for LNG-fueled marine vessels was further investigated for recovering the waste heat of jacket water only and both exhaust gas and jacket water in the study of Baldasso et al. (2017). Six possible ORC configurations have been analyzed regarding the heat sinks with seawater cooling, LNG preheating and using both seawater cooling and LNG preheating. The thermal efficiency of the ORC system boosts with preheating the LNG fuel at an additional ORC condenser in addition to seawater cooling, which is indicated by an increase of up to 1%. The utilization of LNG preheating only is found to be the most efficient ORC configuration but the waste energy of the LNG fuel is not sufficient to be used for both exhaust gas and jacket water WHR. Tsougranis & Wu (2018) proposed an ORC system that is designed explicitly for LNG-powered ships. The proposed system utilizes the waste heat of the main engine exhaust. The cold waste heat of the LNG fuel cools the expanded ORC working fluid, which is then pressurized through a pump to be delivered to the evaporator. The economic feasibility of using 2-stage regenerative ORC against single-stage regenerative ORC was investigated in the study that gave almost the same payback time of around four years. Ng et al. (2019) proposed using free piston linear expanders for marine applications. The research group analyzed an LNG-fueled platform supply vessel for recovering the waste heat of engine exhaust and cryogenic waste heat of LNG fuel. The potential fuel savings of the vessel is indicated as 5.9% with a payback period of 3.1 years. Tian et al. (2020) investigated an ORC configuration for jacket water and exhaust gas WHR. A triple parallel loop ORC system is proposed due to the high temperature differences between the heat sources and the LNG cooled heat sink. The investment cost of the proposed ORC system is reported as 2 million dollars and have a payback period ranged from 8.36 to 8.74 years for 200 kW installed power. Tian et al. (2021) further investigated the performance of a dual parallel ORC system using zeotropic mixtures for exhaust

gas and jacket water WHR from an LNG fueled ship. It is found that the zeotropic mixtures can enhance the system performance but the beneficial effects to the net power output of the ORC system are limited. The maximum energy and exergy efficiencies are calculated as 22.09% and 23.28%, respectively. The economic performance of the ORC system that generates 147 kW power output is found to be 598,000 \$ with a payback time of 4.58 years. Ouyang et al. (2021) proposed a scheme for LNG cold energy utilization on a large marine vessel. They suggested the efficient use of the LNG cold energy can be accomplished by using the LNG as the cooling media of an integrated system of ORC, subzero storage, seawater desalination and air-conditioning systems. The integrated system performance after a genetic algorithm-based optimization is reported as 26.3% thermal efficiency with 210 kW power output having a payback period of 4.27 years.

An ORC system utilizing the waste heat of the scavenge air for marine vessels navigating at the Arctic Sea was studied by Fuente et al. (2017). They suggest that the ORC systems can attract more importance with the increased number of shipping activities from the Arctic Circle due to the low ambient conditions that give the possibility to increase the system thermal efficiency. A container ship having propelled with a 41.1 MW main engine is taken as a reference in the study to determine the ORC performance and the emission reduction potential. The results show that R236fa working fluid has a better thermodynamic performance for the selected parameters but the analyses were carried out using R1233zd(E) working fluid for its very low GWP value. Fuente et al. (2018) also proposed an ORC system consists of a hybrid air-cooled condenser, which benefits from the forward movement of the vessel. Since the ambient temperature is also relatively low, the results showed that the power requirement of the condenser side can be reduced by up to 60% with the proposed supportive cooling system in comparison with the fan cooling.

A review of the studies based on various ship-based waste heat sources is given in Table 2.4 with the utilized ORC configurations and performance parameters of net power output and the thermal efficiency of the ORC system.

Table 2.4 Review of the studies based on various ship-based waste heat sources utilized for waste heat recovery with ORC systems on marine applications

Reference	ORC configuration	Exhaust gas	Scavenge air cooler	Jacket water	Lubricating oil	Boiler steam	Net power output (MW)	Efficiency (%)
Choi & Kim (2013)	Single / dual loop	✓					1.61 - 2.07	8.5 - 10.9
Fuente & Greig (2013)	Regenerative	✓					1.98 - 2.23	18.6 - 22.8
Larsen et al. (2013)	Recuperative	✓					-	13.3 - 24.5
Yang & Yeh (2014)	Recuperative			✓			-	4.1 - 4.5
Nielsen et al. (2014)	Combined cycle with simple ORC	✓	✓				0.90	-
Wang et al. (2014)	Simple		✓				2.61	15.0
Baldi & Gabriellii (2015)	Recuperative	✓	✓	✓	✓		0.48 - 1.8	18.0
Bellolio et al. (2015)	Simple	✓					0.22 - 0.64	-
Kalikatzarakis & Frangopoulos (2015)	Recuperative		✓	✓	✓		1.15 - 2.02	6.8 - 9.6
Song et al. (2015)	Simple	✓		✓			0.09	21.3
Soffiato et al. (2015)	Simple / two-stage / recuperative		✓	✓	✓		0.51 - 0.82	7.7 - 8.4
Yang & Yeh (2015a)	Recuperative	✓					0.08	9.6
Yang & Yeh (2015b)	Simple	✓		✓			0.26 - 0.32	8.8 - 9.4
Yun et al. (2015)	Dual parallel	✓					0.2	11.2
Yukse & Mirmobin (2015)	Simple			✓			0.12	6.2
Sung & Kim (2016)	Dual loop	✓		✓			0.73 - 0.91	12.2 - 20.2
Yang (2016)	Simple	✓	✓	✓	✓		3.16 - 3.40	13.4 - 15.9
Baldasso et al. (2017)	Recuperative	✓		✓			1.68 - 1.78	16.7 - 17.5
Fuente et al. (2017)	Simple		✓				0.55 - 0.85	-
Mondejar et al. (2017)	Recuperative	✓					0.4	20.0
Rech et al. (2017)	Simple / two-stage		✓	✓	✓		0.52	12.6
Ma et al. (2018)	Simple	✓					1.04 - 1.47	8.4 - 12.0
Tsougranis & Wu (2018)	Recuperative and reheated two stage	✓					0.39 - 0.65	23.6 - 35.6
Akman & Ergin (2019)	Simple	✓	✓	✓			0.12 - 0.60	11.9 - 15.5
Lion et al. (2019)	Simple		✓				0.20 - 0.85	-
Nawi et al. (2019)	Simple	✓					0.05	2.3
Uusitalo et al. (2019)	Combined cycle with simple ORC					✓	0.60	11.0
Akman & Ergin (2020)	Simple	✓					0.23	16.8
Bounefour et al. (2020)	Recuperative			✓			0.01	6.2
Liu et al. (2020a)	Combined cycle / dual pressure ORC	✓		✓			3.90 - 7.10	17.2 - 23.5
Liu et al. (2020b)	Combined SRC + ORC + ARC	✓		✓			10.56	22.0
Tian et al. (2020)	Triple parallel	✓		✓			0.20	13.6 - 14.7
Liu et al. (2021b)	TEG-ORC	✓	✓	✓			2.9	24.7
Ouyang et al. (2021)	Simple			✓			0.21	26.3
Tian et al. (2021)	Dual parallel	✓		✓			0.15	22.1
Wang et al. (2021)	Simple	✓	✓	✓			2.3	12.1

Several concept designs have also been proposed in the field of marine ORC systems. Larsen et al. (2017) introduced the concept of a two-stroke engine, which utilizes one of the cylinders as the expander of the ORC fluid. While the remaining eleven cylinders produce power by the injected fuel, the expander cylinder produces power and delivers the torque through the crankshaft of the engine simultaneously with the others. The ORC working fluid collects the heat from the scavenge air at the first stage; then flows through the second stage ORC boiler that is heated up by the exhaust gas of the main engine. The thermodynamic analysis of the concept showed that the expander cylinder could produce almost as much power as one cylinder rotating with the combustion of fuel, which means the power produced by one cylinder is for free. The concept seems feasible theoretically; but still needs much attention with design conditions considering the friction forces and the torsion on the crankshaft because of the different loads produced by the cylinders, maintenance activities, and the lifetime of the engine. Nawi et al. (2019) proposed producing bioethanol onboard for utilizing as the organic working fluid in the ORC system due to its low cost and high availability. The analyses were carried out for a simple ORC system utilizing only the exhaust gas waste heat of a 1 MW marine diesel engine. The thermal efficiency of the ORC system with ethanol as working fluid is found as 2.28%, which is very much less competitive than alternative working fluids. However, it should be noted that it is important to achieve the most sustainable system configuration that eliminates the environmental concerns by using more advanced, clean and renewable working fluids. Wang et al. (2021) proposed a WHRS design that collects the waste heat of scavenge air, jacket water and exhaust gas of a marine diesel engine to be utilized for both hydrogen and electricity production with a solid oxide electrolyzer cell and an ORC system. The integrated system design could recover 44.13% of the engine waste heat with a hydrogen production rate of 0.431 kg/s and power output of 32.4 MW in addition to the ORC system power output of 2.3 MW. Liu et al. (2021a, 2021b, 2021c) introduced the TEG (thermoelectric generator) – ORC concept that utilizes the waste heat of jacket water, scavenge air and exhaust gas of the engine. The hot side of the TEG is placed on the exhaust gas line while the cold side is placed after the ORC pump. The simulation results show that the maximum thermal efficiency of the TEG-ORC system could reach up to 24.68% with the evaporating pressure of 3 MPa.

2.3.1.2 The Integration of Organic Rankine Cycle Systems to Emission Reduction Technologies for Marine Applications

The integration of ORC systems to emission reduction technologies for marine applications has become a prominent subject in recent years. For instance, Baldasso et al. (2019) studied the technical and economic feasibility of implementing an ORC system to a feeder ship operating inside the NO_x emission control areas (NECA). In the study, the ORC implementation to the following NO_x abatement techniques was compared; exhaust gas recirculation (EGR) system, high-pressure selective catalytic reduction (HP SCR) system, low-pressure selective catalytic reduction (LP SCR) system, and Tier II mode operation. The results showed that the EGR-ORC combination has higher economic potential than the other options, as significant reductions at the temperature of the exhaust gas take place during the reactions on SCR systems. Lion et al. (2019) have also supported the idea of using emission reduction technologies combined with ORC systems. They developed a thermodynamic model of two scenarios (Tier II and Tier III configurations) for a tanker propelling with a 13.6 MW main engine and compared them with and without an LP EGR system. The economic analysis in this study revealed that the increase in the specific fuel consumption caused by the EGR system could be compensated with the ORC at specific operating conditions of the ship. The economic profit can reach 5.4% and 5.9% for Tier II and Tier III configurations, respectively.

The studies in this field are limited with the expense coverage of the utilization of both emission reduction technologies and the ORC itself for economic feasibility. More comprehensive studies should be carried out about the utilization of ORC systems with emission reduction technologies due to the changes in WHR potentials on large waste heat sources in cases for different emission reduction technology implementations onboard.

2.3.1.3 Prototype Development and Demonstration Projects for Using Organic Rankine Cycle Systems in Ships

The marine vessels possess too large WHR potentials, thus the required ORC systems to recover those waste heat should also be large like almost MW-scale. The expensiveness of developing industrial-scale ORC systems with marine compliances has limited the marine applications of ORC systems. Since the maturity of the ORC technology has not been reached yet, the demonstration projects to marine vessels and the application of new prototypes lately started to increase with some example installations, which are dominantly utilizing the jacket water and exhaust waste heat.

The first reported reference ORC installation onboard is reported in the study of Öhman & Lundqvist (2013), which is delivered to a newly built Ro-Ro vessel (M/V Figaro) by Opcon Marine in 2012. In addition to a small wet steam turbine arrangement powered by the surplus steam available on the vessel, a 500 kW ORC system, which is driven with R236fa working fluid and utilizing the jacket water waste heat, was installed to the vessel (Opcon Marine, 2012). French ORC equipment manufacturer Enogia installed a 20 kW ORC system on the fishing vessel Orizzonte in 2017. It was estimated a fuel saving potential of 5% after the successful commissioning of the vessel (Motorship, 2020). Calnetix in conjunction with MHI (Mitsubishi Heavy Industries) developed the HydrocurrentTM ORC system and taken the approval for marine applications that recovers the jacket water waste heat of the main engine (Yukse & Mirmobin, 2015). The first application of the system has been installed on the M/V Arnold Maersk container ship in 2016. The field operation test results of the 125 kW ORC system prototype was reported in the study of Sellers (2017). It was observed that a negative relationship was shown between the seawater cooling temperature and ORC power output. In addition, the system running hours were low during short distance travelling between ports.

Kobe Steel completed the sea trials of a binary cycle (ORC) system for ships in 2016, which has received approval from ClassNK (Nippon Kaiji Kyokai) first, and then from Lloyd's Register, DNV (Det Norske Veritas) and ABS (American Bureau

of Shipping), as well. Kobe Steel reported that an engine power output of 7,500 kW generated 125 kW of electricity from exhaust gas waste heat, which is equivalent to about 20-25% of the auxiliary power requirement of the ship (M/V Asahi Miru) (Kobelco, 2017). The system is also installed on a capesize bulk carrier owned by Shunzan Kaiun Co., Ltd. in 2020. The scavenge air waste heat is utilized in the vessel by a 100 kW ORC system (Kobelco, 2020).

The Swedish clean technology company Climeon has marketed their patented award-winning low-temperature Heat Power System solution, which reduces the evaporating pressure of water below 4 bars by creating a vacuum at the condenser with reported thermal efficiency levels of up to 14% (Cision, 2015). A 150 kW module is installed to M/S Viking Grace for WHR from the main engine jacket water in 2015. It is reported that the system will save 700,000 kWh of electricity and 200 tonnes of fuel per year (Viking Line, 2014). Another Climeon WHRS installation to the M/S Viking Glory and onboard tests of the system has already been completed, which is estimated to generate up to 40% of the electricity required for the passenger functions and reduce the CO₂ emissions by about 4,000 tonnes per year. The ship is also equipped with a WHRS that recovers the cold energy of LNG fuel to be utilized for refrigeration appliances and cold rooms (Viking Line, 2019). Virgin Voyages also showed interest in Climeon's WHRS solution. A 900 kW module incorporated with six Climeon HP 150 units utilizing the exhaust gas and jacket water waste heat has been tested and approved on the Virgin Voyages' new building cruise ferry M/S Scarlet Lady in April 2020 (Climeon, 2021a). Virgin Voyages reported three more sister cruise ferries to be built until 2023 that will be equipped with Climeon's WHRS solution. One of the reported vessels M/S Valiant Lady has completed the sea trials and expected to be in service in summer 2021 (Climeon, 2021b). The third ship M/S Resilient Lady is due to set sail in Q2 2022, while the fourth unnamed vessel will make her maiden voyage in Q4 2023 (Virgin, 2021). The Norwegian coastal line Havila Voyages AS also gives an order of the Climeon's Heat Power system weighing an order value of around 1.85 million dollars for four new building sister cruise ferries constructed at Tersan shipyard. The deliveries will be made during 2021 and 2022 however, the WHRS capacities have not yet been reported (Climeon, 2020).

The ORC solution of Orcan Energy AG has been approved for ships by Lloyd's Register in 2018. The new compact efficiency PACK designs have been well-introduced to the market according to the indications of the company's increasing number of orders from shipping companies (15 units sold in 2017, 43 units sold till 2020) (Orcan Energy AG, 2017; Orcan Energy AG, 2020d). The first marine installation of the product is retrofitted to an inland pusher boat Veerle, owned by the Dutch shipping company Ruijven. The waste heat from the jacket water and the exhaust gas is recovered by three efficiency PACKs onboard. The waste heat utilization strategy is not yet clearly expressed in the field reports as the vessel might effectively benefit from electrical energy production or directing the mechanical energy to the engine's crankshaft (Orcan Energy AG, 2018). In 2019, an efficiency PACK is installed on the 5,000 DWT inland waterway barge M/V Maranta, owned by the Dutch shipping company Maranta Shipping. The waste heat of the exhaust gas and jacket cooling water is converted into mechanical energy, which is fed back to the starboard engine's crankshaft. More than a 5% reduction in fuel consumption and thus 36 tonnes of diesel fuel per year is reported to be saved (Orcan Energy AG, 2019). The LNG-fueled 70-meters long passenger ferries M/S Willem Barentsz and M/S Willem de Vlamingh owned by Rederij Doeksen shipping company were equipped with twin efficiency PACKs to the vessels that charge the battery packs on the vessel to power the bow thrusters during maneuverings. The system plays a particularly important role to reduce the ship-based emissions in ports by a reported amount of 318 tons CO₂, and 260 tons of fuel per year (Orcan Energy AG, 2020b). Swedish tanker operator Donsötank provided Orcan Energy's efficiency PACK ORC system to their newly built sister oil and chemical tankers M/T Pacifico and M/T Prospero in 2021 (Orcan Energy AG, 2020c). The ORC system is established for the WHR from the jacket water during navigation and from the thermal oil system during discharge operations to provide well over 80 kW additional net electrical power output to the ship's grid. Green Jade, which will be one of the largest offshore installation vessels in the world after the scheduled deployment in Q4 2022, is agreed to supply eight efficiency PACKs having an electrical power output of 500 kW onboard. The waste heat of jacket water (as preheater) and exhaust gas (as superheater) is recovered from four dual-fuel main engines by the equipped two PACKs for each engine (Orcan Energy AG, 2020a).

Table 2.5 Reported ORC installation projects to marine vessels

Vessel	Year	Vessel Type	Cargo Capacity	ORC Capacity	ORC Maker	Fuel Savings	Payback Time	Reference
Reported Completed Projects								
Figaro	2012	Ro-Ro	31,143 DWT	500 kW	Opcon Marine	4 - 6%	3 years	Opcon Marine (2012)
Viking Grace	2015	Cruise Ferry	57,565 GT	150 kW	Climeon	Up to 5%	-	Viking Line (2014)
Arnold Maersk	2016	Container	109,000 DWT	125 kW	Calnetix & MHI	10 - 15%	5 years	Haglund et al. (2017); Sellers (2017)
Asahi Maru	2017	Bulk Carrier	85,009 DWT	125 kW	Kobe Steel	3%	-	Kobelco (2017)
Orizzonte	2017	Fishing Vessel	-	20 kW	Enogia	5%	-	Motorship (2020)
Veerle	2018	Pusher Boat	250 GT	3 efficiency PACKs	Orcan Energy	5%	2 - 4 years	Orcan Energy AG (2018)
Scarlet Lady	2019	Cruise Ferry	110,000 GT	900 kW	Climeon	-	-	Climeon (2021a)
Maranta	2019	Pusher Barge	5,000 DWT	1 efficiency PACK	Orcan Energy	5%	2 - 4 years	Orcan Energy AG (2019)
Shunzan Kaiun KK	2020	Bulk Carrier	215,000 DWT	100 kW	Kobe Steel	3%	-	Kobelco (2020); MOL (2020)
Willem Barentsz	2020	Passenger Ferry	3,744 GT	154 kW	Orcan Energy	6 - 9%	2 - 4 years	Orcan Energy AG (2020b)
Willem de Vlamingh	2020	Passenger Ferry	3,744 GT	154 kW	Orcan Energy	6 - 9%	2 - 4 years	Orcan Energy AG (2020b)
Pacifico	2021	Product Tanker	22,000 DWT	80 kW	Orcan Energy	6 - 9%	2 - 4 years	Orcan Energy AG (2020c)
Prospero	2021	Product Tanker	22,000 DWT	80 kW	Orcan Energy	6 - 9%	2 - 4 years	Orcan Energy AG (2020c)
Viking Glory	2021	Cruise Ferry	63,813 GT	Up to 40% of passenger functions	Climeon	Up to 5%	-	Viking Line (2019)
Reported Upcoming Projects								
Valiant Lady	2021	Cruise Ferry	110,000 GT	900 kW	Climeon	-	-	Climeon (2021b)
Resilient Lady	2021	Cruise Ferry	110,000 GT	900 kW	Climeon	-	-	Climeon (2021b)
Havila Castor	2021	Cruise Ferry	15,471 GT	-	Climeon	-	-	Climeon (2020)
Havila Capella	2021	Cruise Ferry	15,471 GT	-	Climeon	-	-	Climeon (2020)
Havila Polaris	2022	Cruise Ferry	15,471 GT	-	Climeon	-	-	Climeon (2020)
Havila Pollux	2022	Cruise Ferry	15,471 GT	-	Climeon	-	-	Climeon (2020)
Green Jade	2022	Offshore Installation	-	500 kW	Orcan Energy	-	-	Orcan Energy AG (2020a)
Unnamed	2023	Cruise Ferry	110,000 GT	900 kW	Climeon	-	-	Climeon (2021b)

Several researchers working in the field of marine ORC have also studied experimental setups and prototypes that are specifically designed for marine vessels. Haglind et al. (2017) published the report of the project “PilotORC” that enabled evaluating the technical and economic feasibility of onboard ORC systems used in vessels. The main aim of the project is to utilize the waste heat of the jacket water cooling system of the main engine using a 125 kW ORC unit. Onboard experimental tests were carried out on M/V Arnold Maersk container ship to validate their numerical models. In addition, three case scenarios were proposed for evaluating the waste heat onboard which are; (1) retrofitting of ORC system to utilize the jacket water and the lubricating oil waste heat, (2) retrofitting of ORC unit to utilize the heat from the steam service system, (3) utilizing the heat from the steam service system on a new-building ship. The economic analysis showed that the proposed ORC systems have payback periods varying between 4 to 9 years according to the average fuel price. Pallis et al. (2021) developed and tested a 5 kW marine ORC prototype with R134a working fluid for jacket water WHR. A fully automated control system and the operation strategy were developed to ensure stable operation and robust response to load changes. A techno-economic analysis is carried out by upscaling the power output of the ORC prototype to 164.8 kW_e, which is sized according to the jacket water WHR potential of a real ship. The analysis results based on the seawater temperature indicated a payback period between 10.35 to 12.09 years in the North Sea and the Mediterranean, respectively.

2.3.2 Challenges for Organic Rankine Cycle Applications on Marine Vessels

The current regulations and the given emission reduction and fuel saving goals to ships increased the attraction to achieve more sustainable shipping activities. The first action of the marine sector has been on reducing the emission production caused by the marine vessels. The large WHR potential from marine vessels has indeed created much attention among researchers, ORC manufacturers and shipping companies, however the WHR with ORC systems still possesses some challenges for the widespread utilization on marine vessels. The challenges related to advanced ORC architectures are well identified by Lecompte et al. (2015). The authors highlighted the

knowledge gaps where additional development is required, and a general lack of experimental validation exists. Some modifications to simple RC configuration would result in a requirement for extra components, high-pressure operation and stability issues for working fluids, availability of high pressure and efficient components, high temperature differences between fluids and increased installation costs. In addition, limited knowledge and practical experience about the zeotropic mixtures and supercritical fluids discourages the practical applications of advanced ORC systems. For those particular reasons, it can be seen from Section 2.3.1.3 that the commercial marine applications of ORC are mostly designed with basic ORC architecture.

One of the main challenges of ORC systems is identified as the working fluid selection (Ziviani et al., 2014). The optimal efficiency of an ORC system is achieved by using a proper working fluid that best suits the operating temperature and the pressure levels of the system. This is decided by comprehensive analyses that have been carried out with many working fluid alternatives. As the new generation organic working fluid types continuously emerge with lower GWP, ODP, toxicity and flammability properties, the working fluid selection challenge seems to grow bigger by also considering the high costs of the existing working fluids (Mondejar et al., 2018). The utilization of hydrocarbons are not seen as feasible candidates for marine application since IMO and SOLAS regulations state that the flash point of a fluid may not be lower than 60 °C. This eliminates a significant amount of working fluids showing high thermodynamic performance (Larsen et al., 2013). Moreover, high-pressure operations of ORC systems increase the risks on the expanders caused by sealing issues, lubrication requirements, and friction losses. Another weakness of ORC systems is the low conversion efficiency of around 10-20%. However, it should be noted that the efficiency levels of ORC systems are very competitive among the other alternatives for low-grade WHR (David et al., 2011). Tartière et al. (2017) stated that long-term payback periods of ORC systems increases the financial risks for companies, where they intend to give the priority to invest in the alternatives that are closer to the company's core business. High initial costs of ORC systems, limited capital budgets, insufficient incentives, lack of awareness and lack of recognition of environmental benefits are seen as other barriers to ORC systems (Hood et al., 2017).

The ORC systems are utilized for the low-grade WHR on marine vessels however, the different temperature ranges among the waste heat sources create a complexity of designing a proper heat exchanger network for collecting all those waste heat. Although the theoretical studies propose advanced heat collection systems for recovering the waste heat of exhaust gas, scavenge air, jacket water and lubricating oil, the experimental and commercial applications are limited with the utilization of two waste heat sources, one for preheating and the other for evaporating). Variable engine load operations of ships during the voyage also create a need for advanced control systems that would adjust the amount and temperature of the heat transfer fluids for optimal WHR on off-design conditions.

One of the most important bottlenecks for the utilization of exhaust gas waste heat has been sulphuric acid production at low temperatures of exhaust gas (below the dew point of sulphuric acid). The sulphuric acid in liquid form may corrode and puncture the heat exchangers, exhaust piping and specific engine parts. The use of low-sulphur and carbon-free fuels in marine vessels would significantly decrease that risk and increases the potential utilization of ORCs onboard. In addition, a heat exchanger placement to a retrofitted exhaust funnel by-pass line to collect the exhaust waste heat would solve the problem as the sulphuric acid formations can be collected from below the heat exchanger likewise in sulphur scrubber units.

The size of the ORC systems has much importance recently due to its commercialization has been started for a few years. A wide range of sizes of ORC systems can be produced with the available technology however, the investment costs would also increase for the ORC manufacturer to create a large product range. The recent trend of ORC sizes followed by the marine type ORC manufacturers is to build a compact ORC system in tens or hundreds of kW, considering the drawbacks of ship owners. If more power output than one ORC capacity is required, they simply couple a series of compact ORC systems and create a module on the heat collection line (Orcan Energy AG, 2018; Climeon, 2020). The space required to place the ORC systems will have major importance in case the willingness to implement higher ORC capacities, which is strictly limited on ships' engine room.

CHAPTER THREE

THERMODYNAMIC MODELING AND DESIGN OF AN ORC INTEGRATED WASTE HEAT RECOVERY SYSTEM OF A MARINE VESSEL

3.1 System Description

In this chapter, pinch point analysis was utilized to determine the optimal HEN design of an ORC integrated WHRS for the desired temperature ranges and mass flow rates of a marine vessel utilizing all these heat sources to improve the overall energy efficiency of the vessel. As a case study, the major waste heat sources of a reference container ship were identified. The design parameters for the pinch point analysis were chosen according to the 76% MCR (max. continuous rating) considering the ship type and the engine load that the ship would use most on navigation. An ORC system was coupled with the HEN, to the ends of the thermal oil stream, to transfer the collected waste heat to the organic working fluid. Energy and exergy analyses were performed to indicate the performance of the system using the real data sets of the reference ship for the engine loads of 25, 50, 76, and 100% MCR. A parametric study was conducted to assess the most suitable ORC working fluid and the evaporator inlet pressure of the ORC working fluid for 76% MCR by calculating the thermal efficiency and the exergy efficiency of the ORC system.

A container ship of 1,221 TEU capacity was chosen as a reference to propose the optimal HEN design and integrate this design with an ORC system through theoretical studies. MAN 7S50 MC-C engine that has a rated MCR power of 11,060 kW at 127 RPM was taken as the prime mover to propel the ship. In the existing system design, the exhaust gases produced by the main engine of the vessel are delivered to the economizer section of the boiler plant as a waste heat source illustrated in Figure 3.1(a). Some of the waste heat is recovered for steam production, which is utilized to supply the heating demand of the ship. The steam is condensed while passing through heat exchangers by the end-users of the steam service system. The condensed water is collected in the hotwell tank and then pumped back to the economizer drum to

complete the cycle. The remaining heat of exhaust is emitted into the atmosphere as low-grade waste heat in the range of 180-220 °C (Snow, 2001). The temperatures below these ranges are not considered desirable because of the increased sulphuric acid production (Choi & Kim, 2013). Hence, a considerable amount of waste heat remains in the exhaust gases, which is expelled into the atmosphere. The WHR potential of the remaining exhaust gas is relatively high, but steam RC applications do not efficiently operate below this temperature range, where ORC systems perform better in utilizing low-grade waste heat (MAN Diesel & Turbo, 2014a; Mitsubishi Heavy Industries, 2016).

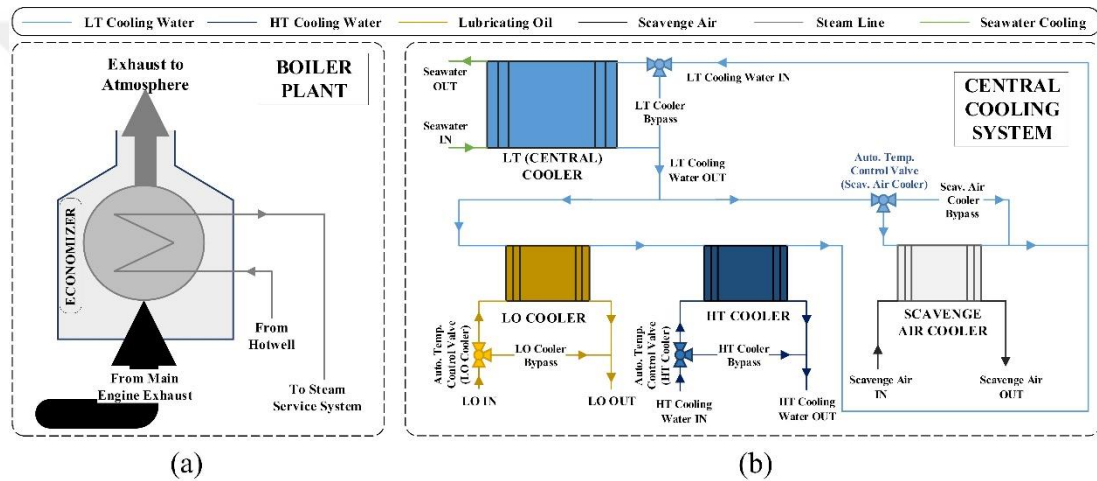


Figure 3.1 Available waste heat sources of the reference ship (a) that sent to boiler plant and through the atmosphere as exhaust (b) that are cooled at the coolers and rejected to sea by the central cooler

The removal of the waste heat generated by the cooling systems of the main engine of the reference ship is carried out by the central cooling system as illustrated in Figure 3.1(b). The cooling utility of each cooler is the low temperature (LT) cooling water as shown in this figure. This water is then cooled in the central cooler by sea water (SW). The temperatures of hot fluid outlets are adjusted with the operation of the automatic temperature control valves that control the flow rates of the streams. The lubricating oil (LO) circulates through the main engine's moving parts (e.g. bearings, pistons, connecting rods, and camshaft) and heats up throughout the way. The LO must be kept in a specific temperature range by the LO cooler to have favorable thermo-physical properties of the oil. The engine's cylinders can reach excessive temperatures that might melt down the surrounded metal. High temperature (HT) cooling water, which

is generally called jacket water, circulates the cylinder liners to remove the excessive heat in the HT cooler until the allowable temperature limits given by the engine manufacturers are satisfied. The scavenge air is the pressurized fresh air that the engine needs for efficient combustion. The scavenge air is pressurized by the exhaust gas turbocharger, which makes the air excessively hot. The excessively hot air should be cooled down before entering the combustion chamber to increase its density. There are also limitations for the outlet temperature of the scavenge air, as too low air temperatures may reduce the thermal efficiency of the engine.

The exhaust gas leaving the economizer and LO, HT cooling water, and scavenge air streams are the main waste heat sources of the reference ship. In this ship, the low-grade waste heat from the coolers is transferred to LT cooling water for the disposal of the unutilized waste heat overboard. In addition, the exhaust gas is emitted into the atmosphere with a large amount of low-grade waste heat.

3.2 The Optimal Heat Exchanger Network Design with Pinch Point Analysis

The quantity, temperature differences, and varying magnitudes of those waste heat make it difficult to determine the optimal layout for WHR. The pinch point analysis is beneficial in this situation that is used to determine an optimal heat exchanger network design for the given target temperatures, pressures and mass flow rates. The analysis will optimize the waste heat utilization from the system in addition to a reduction in the usage of cooling and heating utilities. In sections 3.2.1 and 3.2.2, an overview of pinch point analysis and general rules to apply pinch point analysis have been presented. Then, the central cooling and the exhaust gas systems of the reference ship were hypothetically modified, and an optimal HEN was designed by the pinch point analysis. The integration of the HEN with an ORC system was then examined by the energy and exergy analyses (Section 3.3 and 3.4).

3.2.1 Overview of Pinch Point Analysis

Pinch analysis is an effective method for systematically analyzing the heat exchange processes by utilizing the first and second laws of thermodynamics principles and used to optimize the HEN to provide effective heat transfer between streamlines (Linnhoff & Flower, 1978; Winterborne & Turan, 2015). This method was used in different industrial applications such as thermal process design of a hospital complex (Herrera et al., 2003), design of a diesel hydrotreating unit (Bandyopadhyay et al., 2019), analysis of the HEN in a ventilation system of a building (Misevičiūtė et al., 2018), optimization of waste heat recovery power generation system for a cement plant (Han et al., 2018), crude oil distillation plant optimization (Yan et al., 2019), and integration of hot oil with combined heat and power (CHP) system (Shahidian et al., 2019).

The pinch analysis methodology is based on yielding minimum external utility requirements to the HEN by better process heat integration and thus eliminating financial losses. The benefits of using pinch point analysis include a holistic optimization of the system design, introducing system designs with minimum investment and energy costs, preventing false integration of heat streams, calculation of energy-saving and operating cost reduction potentials for the available streams on a system design, and the ability to connect an entire industry zone energetically with an optimal solution (PinCH, 2021).

3.2.2 Pinch Point Analysis Rules

Pinch analysis has its own specific rules, and the following rules were complied within this study by using PinCH 3.0 software:

- All heat exchangers are accomplished in counter-flow heat exchangers.
- The outer walls of the components are thermally insulated.
- All changes in kinetic and potential energy are negligible.
- The average specific heat of each stream of matter is constant.

- The enthalpy of each stream depends only on temperature.
- The changes in the pressure between the inlet and the outlet streams are neglected.

In the pinch point analysis, firstly, a minimum temperature difference ($\Delta T_{\min}$) is decided according to the process and stream properties. The system should be divided from below and above the pinch point as follows. Above the pinch point, the number of cold streamlines has to be more than that of hot streamlines (if the number of streams is not satisfied, the streams should be split until the number of streams are equalized); the heat capacity flow rate of the cold streams have to be higher than that of hot streams, and only hot utilities have to be used. For below the pinch point, the rules apply vice-versa. In addition, heat transfer across the pinch point is not allowed. The Tick-off Rule should be applied during the stream matching, which can be defined as once two streams are matched for heat transfer, the maximum possible amount of heat should be exchanged (Linnhoff & Flower, 1978; Stoecker, 1980; Kemp, 2011).

3.2.3 The Optimal Heat Exchanger Network Design of Reference Container Ship with Marine Diesel Engine Based Waste Heat Sources

The pinch point analysis was done for the 76% MCR parameters given in Table 3.1 using the PinCH 3.0 software. The exhaust, scavenge air, HT cooling water, and LO streams were selected as the hot streams. LT cooling water was replaced with thermal oil for collecting the available heat of the hot streams. Considering the thermal oil as the cold stream enabled fluid temperatures above 100 °C to keep safer and less complex operation (Guercio & Bini, 2017; Dong & Nguyen, 2019). The thermal oil collects the waste heat of the heat sources, and cools them as well, by circulating through the coolers. The pinch point analysis is required to separate the heat loads below and above the pinch points. In addition, all the waste heat could not be collected as some needed to be discharged from the system by the cooling utilities. The cooling utilities were used to remove the unavailable heat, and to decrease the temperature of the hot streams to the desired levels as well.

Table 3.1 Input parameters for waste heat sources and cold streams

ENGINE LOADS	INPUT PARAMETERS	EXHAUST STREAM	SCAVENGE AIR STREAM	HT STREAM	LO STREAM	THERMAL OIL STREAM	COOLING UTILITY
100 % MCR ENGINE LOAD	Mass flow rate (kg/s)	28.08	27.51	26.46	56.32	52.13	30.23
	Inlet temperature (°C)	201	187	80	54	45	25
	Target outlet temperature (°C)	80	45	65	45	125	32
	Average specific heat (kJ/kg·K)	1.047	1.017	4.191	2.014	2.180	4.205
	Pressure (kPa)*	101.4	370	370	310	420	440
	Substance	6.52% CO ₂ 2.94% H ₂ O 74.49% N ₂ 15.87% O ₂	Humid Air	Water	Unused Engine Oil	Thermia D	Water
76 % MCR ENGINE LOAD	Mass flow rate (kg/s)	23.24	22.81	19.64	42.26	35.63	25.03
	Inlet temperature (°C)	196	152	80	53	45	25
	Target outlet temperature (°C)	80	35	68	44	125	32
	Average specific heat (kJ/kg·K)	1.045	1.014	4.193	2.000	2.180	4.205
	Pressure (kPa)*	100.78	293	370	310	420	440
	Substance	5.95% CO ₂ 2.74% H ₂ O 74.62% N ₂ 16.50% O ₂	Humid Air	Water	Unused Engine Oil	Thermia D	Water
50 % MCR ENGINE LOAD	Mass flow rate (kg/s)	16.30	16.01	13.22	28.17	19.76	24.22
	Inlet temperature (°C)	191	110	81	52	45	25
	Target outlet temperature (°C)	80	34	70	44	125	32
	Average specific heat (kJ/kg·K)	1.045	1.011	4.193	2.008	2.180	4.205
	Pressure (kPa)*	100.39	203	370	310	420	440
	Substance	5.66% CO ₂ 2.67% H ₂ O 74.67% N ₂ 16.82% O ₂	Humid Air	Water	Unused Engine Oil	Thermia D	Water
25 % MCR ENGINE LOAD	Mass flow rate (kg/s)	9.29	9.19	6.61	14.09	8.31	16.82
	Inlet temperature (°C)	186	66	81	52	45	25
	Target outlet temperature (°C)	80	30	71	44	125	32
	Average specific heat (kJ/kg·K)	1.043	1.006	4.194	2.008	2.180	4.205
	Pressure (kPa)*	100.1	140	370	310	420	440
	Substance	5.23% CO ₂ 2.49% H ₂ O 74.81% N ₂ 17.32% O ₂	Humid Air	Water	Unused Engine Oil	Thermia D	Water

*The indicated values are absolute pressures

It is possible to investigate the potential of the energy recovery for the given operating parameters based on the composite curve (CC) and grand composite curve (GCC). The CC was formed by integrating the hot streams as shown in the red line, and cold streams, as shown in the blue line in the heat flow vs. temperature line graph as shown in Figure 3.2(a). The position at which the vertical distance between the hot and cold composite curves is minimum is defined as the Pinch Point. Importantly, this minimum temperature difference between the hot and cold pinch temperatures can be changed by shifting the Cold Composite Curve to the left or right, which is so-called supertargeting. Thereby, the optimal pinch point can be determined. As the pinch point decreases, the investment cost of the heat exchangers increases because of the need for extra heat transfer area, accordingly.

The hot and cold composite curves are then shifted along the temperature (T) axis, by shifting the hot composite curve down and the cold composite curve up by half of the $\Delta T_{\min}$. The vertical axis of the GCC shows the shifted temperatures (T^*), which is a scale of the process interval temperature as given in Equations. 3.1-3.2. The heat flow differences between the shifted composite curves constitute the GCC, which is used to investigate the potential for optimization with respect to the hot and cold utilities. In this study, the supertargeting function is used to eliminate the need for heat utility addition to the system that corresponds to the $\Delta T_{\min}$ of 2.5 °C.

$$T^* = T - (\Delta T_{\min} / 2) \quad (\text{for above the pinch}) \quad (3.1)$$

$$T^* = T + (\Delta T_{\min} / 2) \quad (\text{for below the pinch}) \quad (3.2)$$

The grey-colored area in Figure 3.2(b) shows the maximum recoverable heat of the system, where neither external heating nor external cooling is necessary. The pinch point analysis showed that no heat utility addition is necessary for the specified target temperatures as can be seen in Figure 3.2(b). The cooling utility is only required for the temperatures below the cold pinch as shown in both Figures 3.2(a) and 3.2(b). The pinch temperatures are the closest points that the hot and cold streams approach to each other. The pinch temperatures for the proposed HEN were found as 50.5 °C for the cold pinch, and 53 °C for the hot pinch as can be seen in Figure 3.2(a).

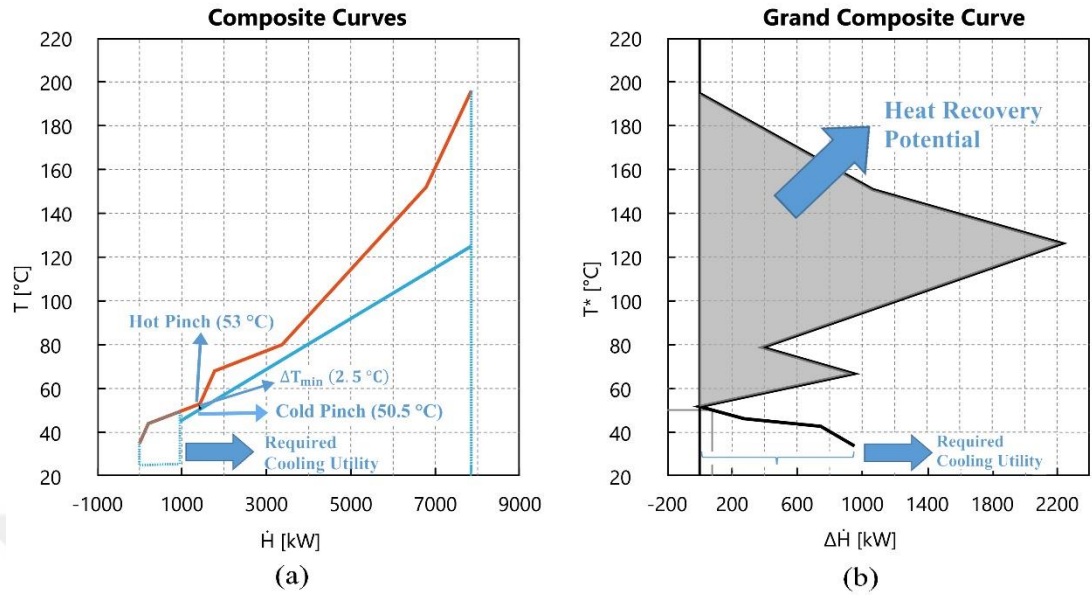


Figure 3.2 (a) Composite curve and (b) grand composite curve of the heat sources for the engine at 76% MCR engine load

The PinCH 3.0 software calculates the maximum heat recovery potential with energy balance equations. The optimal HEN is designed using the pinch rules that would recover the heat as close to the maximum value as possible. This is accomplished by integrating the heat streams of the process to achieve their target temperatures with minimum hot or cold utility usage. If the rules are not applied, the system will need extra utility and most probably extra heat exchangers to restore the energy balance. It should be noted that the proposed HEN design is only optimal for the specified heat loads and target temperatures of the streams.

Figure 3.3 shows the optimal HEN design for the reference ship using the 76% MCR engine load data shown in Table 3.1. After applying the pinch analysis rules on the streams, the scavenge air cooler was divided into three stages; which are named HT air cooler, LT air cooler, and final air cooler. The scavenge air of the engine first cools while passing through the HT air cooler, and then enters the LT air cooler. The final air cooler has to bring the air temperature to the specified target value of the scavenge air inlet of the engine. The heat removed from the final air cooler is transferred to the cooling utility, which is then discharged overboard. The LO cooler,

which is defined as the cooler that collects the available waste heat, is divided into two sections. The other section is defined as the secondary LO cooler, which assures the target outlet temperature of the LO stream is reached. HT cooler does not require any additional cooling utility as the whole waste heat could be transferred to thermal oil at all engine loads. The outlet temperature of the exhaust stream is far above the pinch temperature, which allows using only one heat exchanger on the optimal design according to the pinch rules for utilizing the exhaust gas stream. The cooling utilities are only utilized for points below the pinch point temperature of LO and scavenge air heat streams. All the scavenge air heat below the pinch point temperature were found non-utilizable for WHR.

The thermal oil, which is the only cold stream in the system, starts the circulation from the LO cooler first, as it has the lowest temperature range among the available waste heat sources. The thermal oil then collects the low-temperature waste heat of the scavenge air, which is cooled up to just above the pinch point. The waste heat of the HT stream is also fully recovered in the HT cooler with the same thermal oil stream. The cooler outlet temperatures of the exhaust stream and the high temperature scavenge air stream are very close to each other. The thermal oil stream is therefore split into two streams with different heat capacity rates that are adjusted by the ratio between the two waste heat sources.

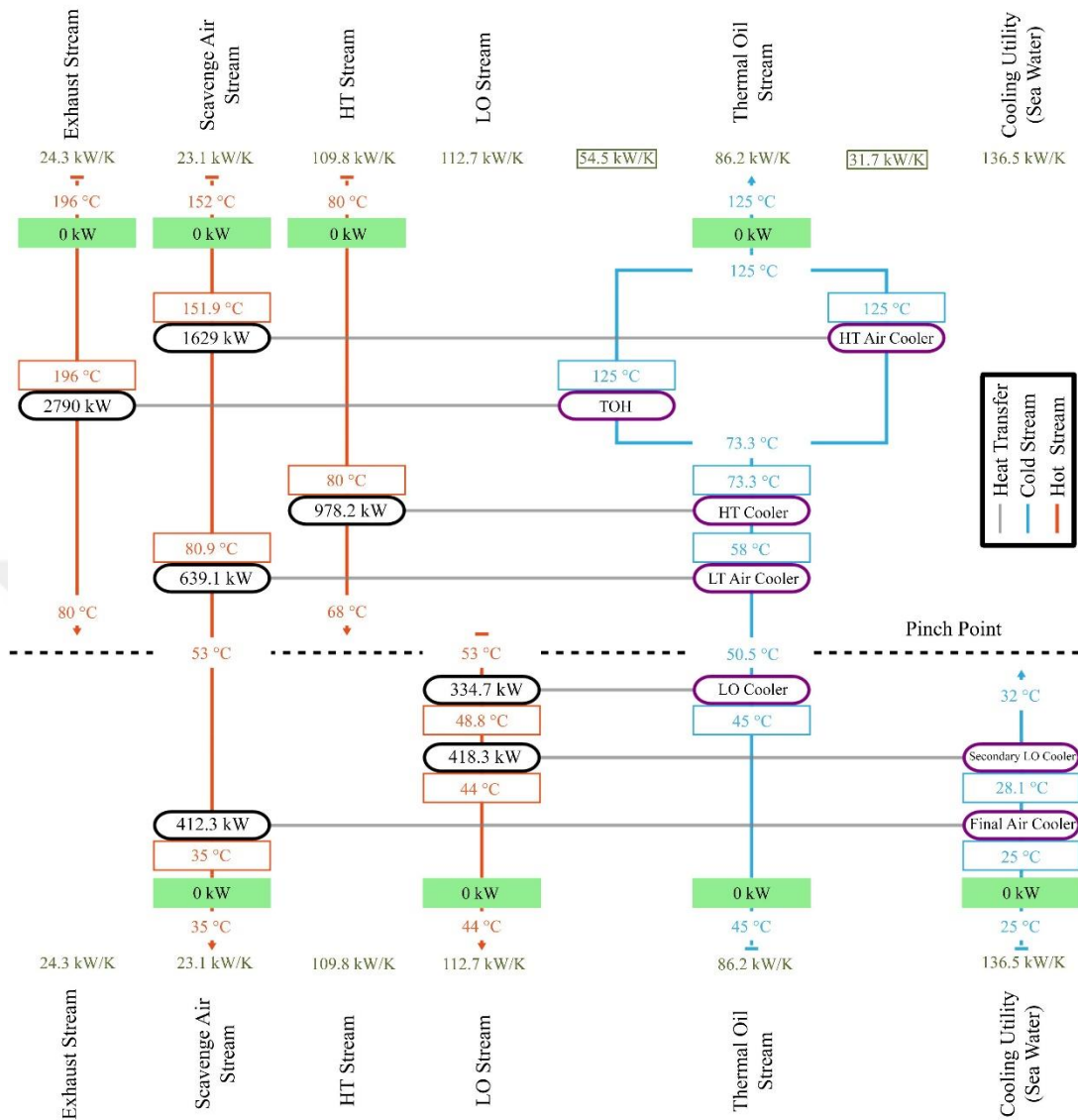


Figure 3.3 Optimal HEN design with the PinCH 3.0 software

3.3 Energy and Exergy Analyses for ORC Integration to Optimal Heat Exchanger Network Design

The pinch point analysis, as given in Section 3.2.3, provided an optimum heat exchanger network design for 76% MCR. In this proposed design, the unutilized waste heat is discharged overboard; whereas the thermal oil stream collects the remaining waste heat. The collected waste heat can be recovered by integrating the thermal oil stream into the ORC. The collected heat is transferred to the ORC working fluid through the evaporator. The heated working fluid enters the turbine in the gaseous

phase to produce power. After passing the turbine, the working fluid enters the condenser, which is cooled to the liquid phase here by the SW cooling system. A pump pressurizes the working fluid and the cycle is completed after feeding it back to the evaporator. The HEN utilizes the thermal oil as the primary cooling media of the system; thus the temperature of thermal oil at the outlet of the evaporator should be kept at a minimum. A simple ORC system without the use of reheaters or recuperators was selected taking into account the maximum cooling ability of the thermal oil. It is essential to assure that the target temperatures are achieved as the streams maintain the operation of vital parts of the engine. In this section, the steady-state energy and exergy balance equations for the input parameters of waste heat sources at 100, 76, 50, and 25% MCR engine loads are described. Design considerations for the ORC system are also given to analyze the net power outputs that can be produced by the ORC system for the given engine loads.

The assumptions for the overall system are presented as follows:

- Changes in kinetic and potential energy and exergy are ignored.
- Both the ORC and the WHR systems run at steady-state conditions.
- The exit of the condenser is in the saturated liquid state.
- Pressure losses across coolers and pipes are neglected.
- All of the pumps placed in the system are considered variable speed pumps that can change the mass flow rates easily on varying engine loads.
- The thermal oil outlet temperature of each cooler is assumed constant on varying engine loads. As there is only one pump for the thermal oil line, mass flow rates of thermal oil inlets through coolers are adjusted by automatic temperature control valves, changing a certain amount of thermal oil to the bypass lines, if necessary.

The mass flow rates of the thermal oil streams are calculated using the energy balances around the control volumes enclosing the thermal oil stream and the waste heat sources. The parameters given in Table 3.1 were taken from the sea trial test records of the reference ship for 100, 76, 50, and 25% MCR engine loads, accordingly.

Table 3.1 indicates the input parameters for calculating the waste heat potential of the major waste heat sources of the reference ship. The effect of the varying loads on the proposed optimal HEN system was also investigated by the energy analyses in the thermodynamic model.

The ORC system is analyzed separately from the optimal HEN; because the pinch point analysis is not able to simulate the pressure variation of the ORC turbine and the ORC pump. Another reason is that the thermal oil switches from the cold stream to the hot stream while passing through the ORC evaporator. Hence, the thermodynamic model of the ORC section was integrated into the optimal HEN design using the Engineering Equation Solver (EES) software (Klein, 2018).

System parameters of the ORC system are given in Table 3.2. The general energy and exergy balance equations to evaluate the performance of the power generation system with optimized HEN are given in Table 3.3 as Equations 3.3-3.7 and the analyses were performed using the EES software. It should be noted that the heat loss to the surroundings from heat exchangers was taken as 1% considering the similar studies in the literature (Srinivasan et al., 2010; Mamun & Biswas, 2012; Song et al., 2015; Nawi et al., 2019) and given in Table 3.2. The ORC system operates in the subcritical region. The outlet temperature of the ORC evaporator is taken 1 °C above the saturation temperature of the working fluid at the ORC evaporator outlet pressure to ensure that the ORC working fluid enters the turbine at the gas phase. The selections of the ORC evaporator outlet pressure and the working fluid type are described in Section 3.4 in accordance with the Figure 3.6. The ORC condenser outlet pressure is taken as the saturation pressure of the working fluid at the target ORC condenser outlet temperature, as given in Table 3.2.

Table 3.2 Input parameters of the ORC system

Parameters	Value	Unit
The pressure of the working fluid entering the ORC evaporator	1000-3000	kPa
The minimum pinch point temperature difference in the ORC evaporator	3	°C
The minimum pinch point temperature difference in the ORC condenser	7	°C
The target temperature of ORC working fluid in the ORC condenser outlet	32	°C
The temperature of cooling water entering the ORC condenser	25	°C
The isentropic efficiency of the ORC pump	0.80	[-]
The isentropic efficiency of the ORC turbine	0.85	[-]
The heat loss from the heat exchangers	0.01	[-]

Table 3.3 Equations used in the model (Moran et al., 2018)

Steady-state Energy Balance	$0 = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_{in} \cdot (h_{in} + \frac{1}{2} V_{in}^2 + g \cdot z_{in}) - \sum_e \dot{m}_{out} \cdot (h_{out} + \frac{1}{2} V_{out}^2 + g \cdot z_{out}) \quad (3.3)$	
Steady-state Exergy Balance	$0 = \sum_j \left(1 - \frac{T_0}{T_j}\right) \cdot \dot{Q}_j - \dot{W}_{cv} + \sum_i \dot{m}_{in} \cdot e_{f,in} - \sum_e \dot{m}_{out} \cdot e_{f,out} - \dot{E}_d \quad (3.4)$	
Steady-state Flow Exergy	$e_f = h - h_0 - T_0 (s - s_0) + \frac{V^2}{2} + g \cdot z \quad (3.5)$	
Thermal Efficiency	$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{ORC, evap}} \quad (3.6)$	
Exergy Efficiency	$\varepsilon = \frac{\dot{W}_{net}}{\Delta \dot{E}_{xORC, evap}} \quad (3.7)$	

3.4 Energy and Exergy Analyses Results

In this section, the layout of the proposed integrated HEN design is formed according to the results of the pinch point analysis. The effect of the evaporator pressure and the working fluid type on the performance of the ORC system is assessed using the steady-state thermodynamic model presented in Section 3.3. In addition, the energy flow from the proposed HEN to the ORC system is shown in Sankey diagrams. The exergy destruction rates of the components of the integrated system with the varying engine loads are also found.

After applying pinch point analysis and proposing the HEN design layout, the integrated ORC model is developed. For this purpose, firstly, an ORC model is

developed and validated with experimental data. The accuracy of the results of the model cannot be directly verified since there are no similar test beds available at the size of the proposed system. In addition, specific operational parameters of the proposed system make it difficult to validate the model through studies found in the literature. Therefore, the thermodynamic model of the ORC system is validated by using the input parameters from the experimental studies for lower capacity ORC systems. These parameters include evaporator pressures (T_{evap}), condenser outlet temperatures (T_{cond}), and mass flow rates of working fluid ($\dot{m}_{\text{wf}}$). Table 3.4 shows the comparison of the thermal efficiency (η_{th}) and the turbine power output ($\dot{W}_{\text{turbine}}$) results of the presented model with the validated study of Yang & Yeh (2014) for isobutane as the working fluid. In addition, for the working fluid as R245fa, the results of this study are compared with the experimental results of a 1 kW ORC system from the study of Unamba et al. (2019). Results were given at the heat source temperatures of 100, 120 and 140 °C in their study. In this comparison, the performance parameters of η_{th} and the net power output ($\dot{W}_{\text{net}}$) are analyzed for the given parameters of T_{evap} , P_{evap} , T_{cond} , P_{cond} , and $\dot{m}_{\text{wf}}$ for the three heat source temperatures as shown in Figure 3.4. The maximum deviation of 5.56% can be observed between the measured (Unamba et al., 2019) and calculated data that might be caused by the assumptions of constant pump and turbine efficiencies. As a result, the numerical solutions obtained from the ORC simulation in this study show consistency with the results obtained from the studies carried out by Yang & Yeh (2014), and Unamba et al. (2019) for similar input parameters.

Table 3.4 Comparison of the results of the current study and the reference study (Yang & Yeh, 2014)

Study	Working Fluid	Input Parameters						Output Parameters	
		T_{evap} (°C)	T_{cond} (°C)	$\dot{m}_{\text{wf}}$ (kg/s)	P_{evap} (kPa)	P_{cond} (kPa)	$\dot{Q}_{\text{evap}}$ (kW)	η_{th} (%)	$\dot{W}_{\text{turbine}}$ (kW)
Reference study	Isobutane	87.15	25	3.61	1552.5	350.7	1449.8	11.63	180.27
Current study	Isobutane	87.15	25	3.61	1550	350	1457	11.73	180.7
Relative error (%)	-	-	-	-	0.16	0.20	0.49	0.85	0.24

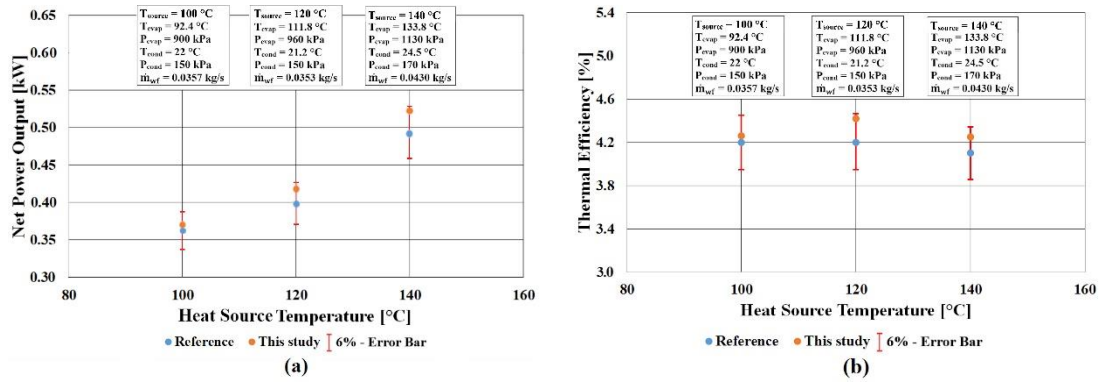


Figure 3.4 Validation of the ORC model with the experimental results of a reference study (Unamba et al., 2019) for a 1 kW ORC system operating with R245fa for (a) the net power output and (b) the thermal efficiency

The layout of the proposed integrated HEN design of the reference ship's heat collection system and ORC system is shown in Figure 3.5.

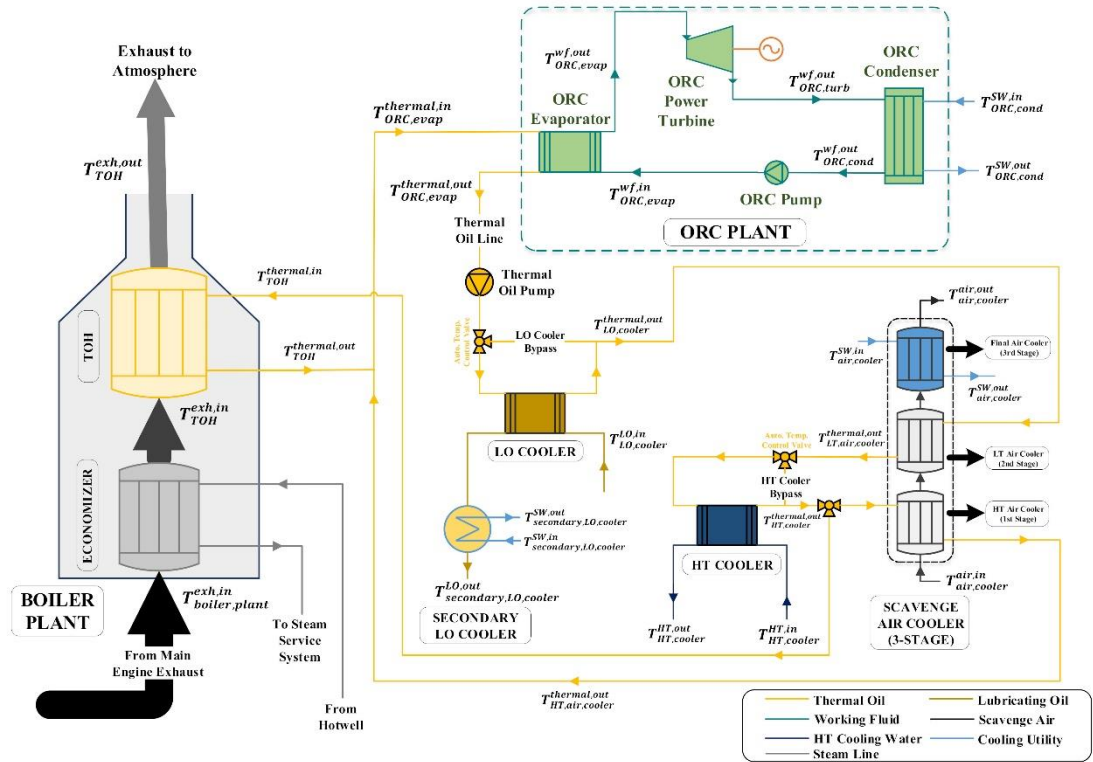


Figure 3.5 Layout of the integrated optimal HEN and ORC system for the reference ship

Because the steam demand of the ship remains, the economizer section of the boiler plant also remains in the proposed design. A heat exchanger, which is the thermal oil

heater (TOH) in the current study, was retrofitted to the top end of the boiler plant to collect the remaining heat from the economizer. More precisely, the exhaust gases flow through the economizer first for satisfying the steam demand of the ship; then enters the TOH section of the boiler plant to collect the available waste heat of the exhaust at the proposed design as can be seen in Figure 3.5.

A parametric study was conducted for nine different organic working fluids that possess relatively low ODP and GWP, as given in Table 3.5. The thermal efficiency and the exergy efficiency of the ORC system were investigated according to the inlet pressure of the working fluid in the ORC evaporator varying between 1000 kPa and 3000 kPa. The parametric study includes the effect of evaporator pressure and ORC working fluid types on the performance of the system, as shown in Figures 3.6 and 3.7. The performance parameters were chosen as the thermal efficiency and exergy efficiency of the ORC system. The maximum and minimum values of the evaporator pressure were taken as 1000 kPa and 3000 kPa, respectively. The results of the parametric studies show that ORC working fluid type has a significant effect on the performance of the ORC as shown in Figures 3.6 and 3.7. The exergy efficiency curves of the ORC system for varying ORC evaporator pressure, as shown in Figure 3.7, show a similar pattern with the thermal efficiency curves. These two figures indicate that the highest thermal and exergy efficiencies can be provided by R1234ze(Z) for the evaporator pressures higher than 1400 kPa amongst the selected organic working fluids; however, R245fa and R600 might be considered as alternatives to R1234ze(Z) because of their lower flammability standard.

Table 3.5 Properties of the selected organic working fluids (Refrigerant Report, 2020; Linde Industrial Gases, 2021)

	R1234ze(Z)	R245fa	R600	R236ea	Isobutane	R236fa	R152a	R134a	R1234yf
GWP ^a	<1	1030	4	1200	3	9400	124	1430	4
ODP ^b	0	0	0	0	0	0	0	0	0
ASHRAE 34 ^c	A2L	A1	A1	A1	A3	A1	A2	A1	A2L
Critical Temp. (°C)	150.1	154.0	152.0	139.3	134.7	124.9	113.3	101	94.7
Critical Press. (kPa)	3533	3651	3796	3429	3640	3200	4520	4059	3382

^aGWP: global warming potential, relative to CO₂.

^bODP: ozone depletion potential, relative to R11.

^cASHRAE Standard 34 - Refrigerant safety group classification. 1: Non-flammable; 2L: Mildly flammable; 2: Lower flammability; 3: Higher flammability; A: Lower toxicity; B: Higher toxicity.

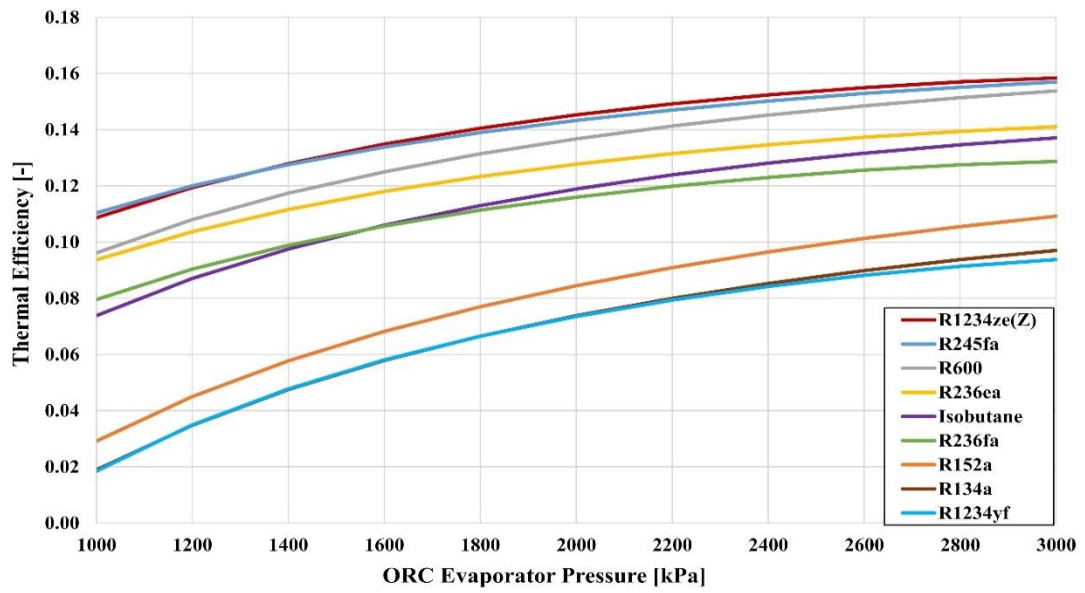


Figure 3.6 The effect of evaporator pressure and ORC fluid type on the thermal efficiency of the system

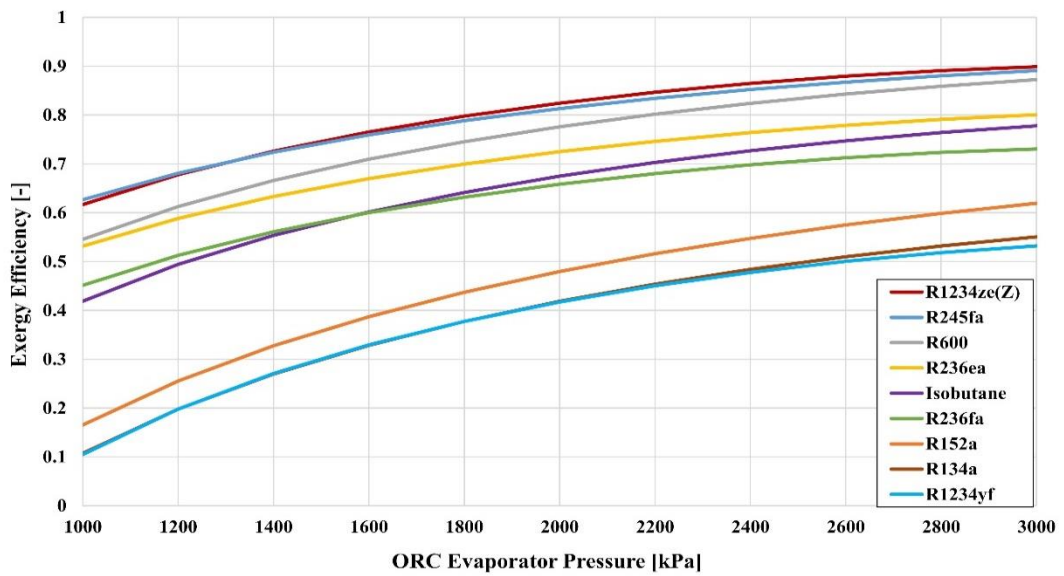


Figure 3.7 The effect of evaporator pressure and ORC fluid type on the exergy efficiency of the system

The remaining parts of the study were carried out with R1234ze(Z) as the working fluid. The lower flammability level and the lower GWP value played an essential role in choosing this working fluid for a safer and environmentally friendly operation. The energy and exergy analyses of the integrated ORC system were carried out using the ORC evaporator pressure of 2400 kPa, as there was no significant performance improvement above 2400 kPa, as can be seen in the Figures 3.6 and 3.7. The performance of the ORC system was simulated using the steady-state model for the

heat loads generated by 25, 50, 76, and 100% MCR engine loads of the reference ship. The mass flow rates and the target temperatures of the hot streams remained the same as the sea trial test records. The compensation of the system balance for varying load conditions was driven by the thermal oil stream with the help of the variable speed thermal oil pump and by-pass lines located before coolers.

Figure 3.8 shows the distribution of the waste heat produced by the main engine of the reference ship through the HEN and to the ORC system for four different engine loads. Unavailable heat was discharged overboard from the LO at the secondary LO cooler and the scavenge air at the final air cooler by the cooling utility, which is defined as SW in this case. The unavailable waste heat was found as 23.71, 16.56, 13.17, and 7.81% of the total waste heat produced by the heat sources for 25, 50, 76, and 100% MCR, respectively. The HT air cooler did not utilize any waste heat at 25% MCR because of the low scavenge air temperature at the current engine load. The exhaust gas has the largest waste heat potential to be used in the ORC system; but considering HT air cooler and LT air cooler as a combined heat exchanger system, the scavenge air also has great waste heat potential that is very close to the heat supplied by the exhaust gas at the engine loads over 76% MCR. In any case, scavenge air can be described as the second-largest waste heat source on the reference ship, which is followed by HT cooling water as the third and the LO as the last among the specified waste heat sources. Non-utilizable heat of the ORC was also discharged to the sea through the ORC condenser. The yellow box in the middle stands for the heat exchange process from the thermal oil to ORC working fluid through the ORC evaporator. The net power outputs generated by the proposed ORC integrated WHRS were calculated as 227.9, 541.9, 970.8, and 1430 kW for 25, 50, 76, and 100% MCR, which correspond to 8.24, 9.80, 11.55, and 12.93% of the net power output produced by the engine, respectively. The thermal and the exergy efficiency of the ORC system was calculated as 15.24%, and 86.47%, respectively. The model of the generator engine set used in the reference ship is 6L23/30(H) of MAN B&W that can produce 780 kW on full load. The values show that all of the electricity demand of the ship can be provided by the WHRS for the engine load above 50% MCR, considering one generator is utilized on around 50-60% MCR engine load during navigation.

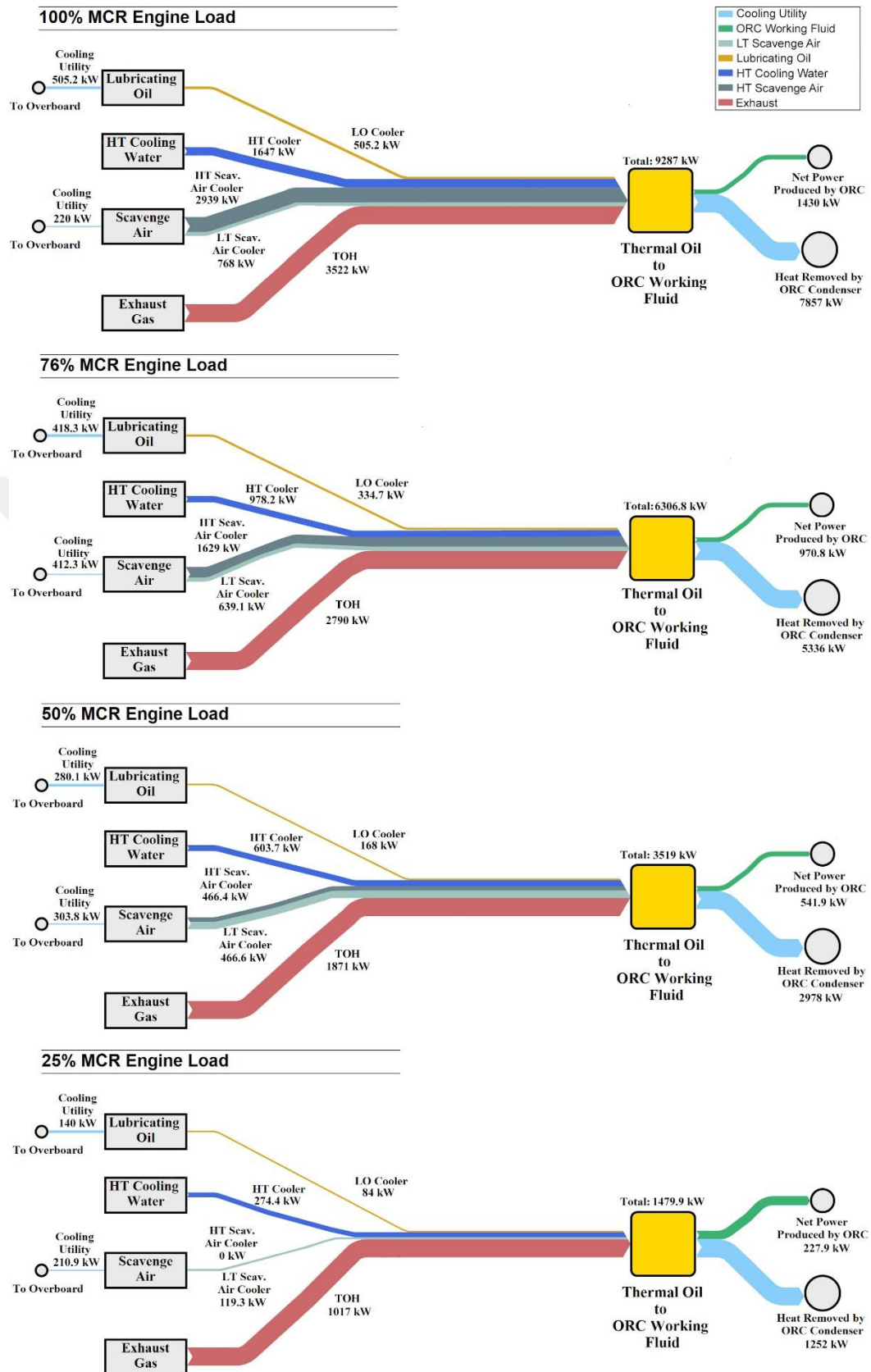


Figure 3.8 Sankey diagrams showing the heat utilization through the proposed HEN to the ORC unit for 100% MCR, 76% MCR, 50% MCR, and 25% MCR main engine loads

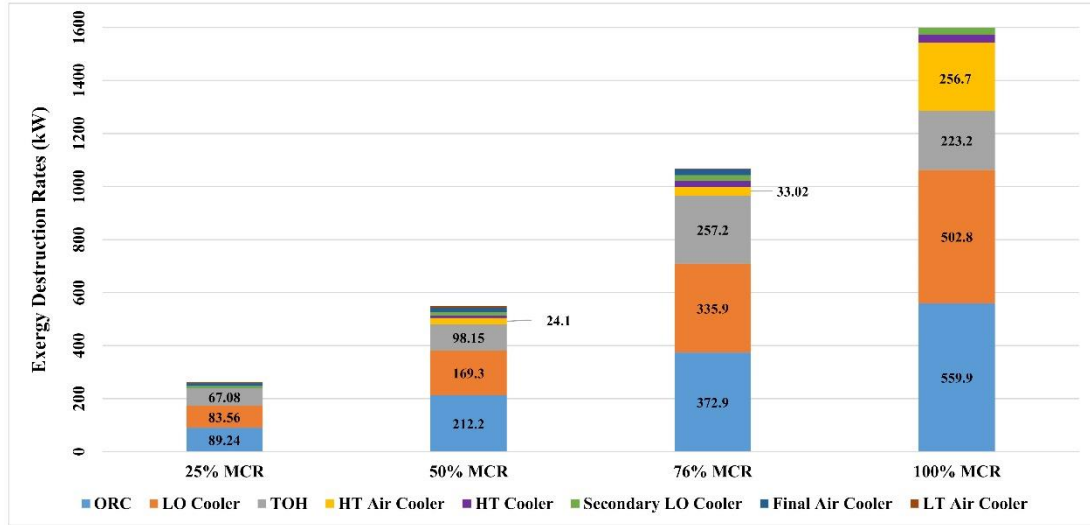


Figure 3.9 Distribution of exergy destruction rates of the components on varying engine loads

Exergy destruction rates of the system components were demonstrated in Figure 3.9. The ORC system, with its all components, has the highest exergy destruction rate for all loads. The exergy destruction rate of the ORC for 76% MCR engine load was found as 372.9 kW (34.94% of the total system). LO cooler and the TOH have exergy destruction rates of 335.9 kW and 257.2 kW (31.47% and 24.10% of the total system), respectively, for the same engine load. The HT air cooler has possessed low exergy destruction ratios until 76% MCR but got higher values at 100% MCR because of the exponentially increased scavenge air inlet temperature of the main engine. The energy analysis calculations of the HEN revealed that the temperature and the mass flow rate of the thermal oil entering the TOH are slightly higher in the 76% MCR case than 100% MCR, causing a minor decrease in the exergy destruction ratio of the TOH at 100% MCR.

3.5 Exergetic Sustainability Assessment for the Proposed System Design

Sustainability assessment plays an important role in energy systems. The sustainability assessment tools on energy systems are beneficial at justifying an energy policy by discussing the effects of different sources of energy, determining the impact of waste energy and exergy on the environment, suggesting measures to reduce emissions, and encouraging the application of more environmentally benign solutions

(Chowdhury et al., 2020a). In this section, the basics of sustainability assessment and the framework that should be shaped within are introduced. The sustainability assessment framework leads to sustainability indicators that are used to assist policy makers and decision-making processes in the first place. The literature review shows that the energy systems are suited to sustainability assessment with exergy based assessment tools and indicators from an environmental point of view. In the present case, the exergy analyses results are used to derive exergetic sustainability indicators, which shows the sustainability improvement potentials with the utilization of the proposed ORC integrated WHRS design.

3.5.1 Background of Sustainability Assessment

A sustainable system refers to a system that does not cause permanent damage to the environment and does not consume excessive resources while performing an efficient operation for a long enough service period and leaving a clean environment and enough resources to posterity to meet their needs of natural resources (Dincer & Zamfirescu, 2018). Sustainable development of a system consists of challenges to achieve no or few negative environmental and social impacts goals, which require continuous and sustained improvement of the system. The sustainability of a system needs to be measured and assessed via sustainability assessment tools, thereby the system can be deduced as sustainable or indicate sustainable development (Hacatoglu et al., 2015).

Sustainability assessment tools are categorized by Ness et al. (2007) into three areas that are indicators or indices, product-related assessment tools, and integrated assessment tools. Indicators are simple measures that can quantitatively represent the economic, social, and/or environmental state of a specified region. The sustainability trends are continuously tracked by calculating or measuring those indicators, which are utilized for short-term and future projections decision-making with a retrospective point of view. The product-related assessment tools include life cycle assessment, life cycle costing, material flow analysis, and product energy analysis. These tools identify inefficiencies and sustainability based risks in relation to the product design and

production processes. The integrated assessment tools are utilized for supporting decisions related to a policy or a project for managing complex problems from a prospective point of view. These tools might be used for a variety of applications in addition to sustainability assessment issues, e.g., system dynamics, risk analysis, cost-benefit analysis, vulnerability analysis, and environmental impact assessment (Dincer & Zamfirescu, 2018). The sustainability assessment tools have been adopted in many applications worldwide to track the environmental, economic, and social impacts of specific cases. These sustainability assessment case studies include electricity and heat production from solar energy, biofuel, hydrogen production, household energy needs, hydroelectric energy, waste heat recovery, and other energy sources; agriculture, food, forestry and other land use; transportation and automobile sectors; construction and building sectors; industrial sectors of ceramic, marble, solvents, wood, plastic, wind turbines, biorefinery, and chemicals (Visentin et al., 2020).

The sustainability assessment tools and relevant evaluations are further processed in a sustainability assessment framework to develop a strategy for the environmental impact of projects (Srinivasan et al., 2011). Drivers-pressures-states-impacts-responses (DPSIR) model has been a widely adopted framework, which acts as an adaptive management tool and describes the interactions between society and the environment (Kristensen, 2004; Walter, 2013). In this model, the casual links start with the drivers that indicate the developments in society, the economy, and the environment. The driving forces exert pressure on the current state of sustainability, which leads to impacts resulting from changes in the environmental quality in any direction. Society plays an important role in this model, where they can give responses to this change to reverse the impacts. In addition, the responses of society can directly act on the drivers, pressures, or state of sustainability with feedback loops, as illustrated in Figure 3.10 (Gabrielsen & Bosh, 2003; Dincer & Zamfirescu, 2018).

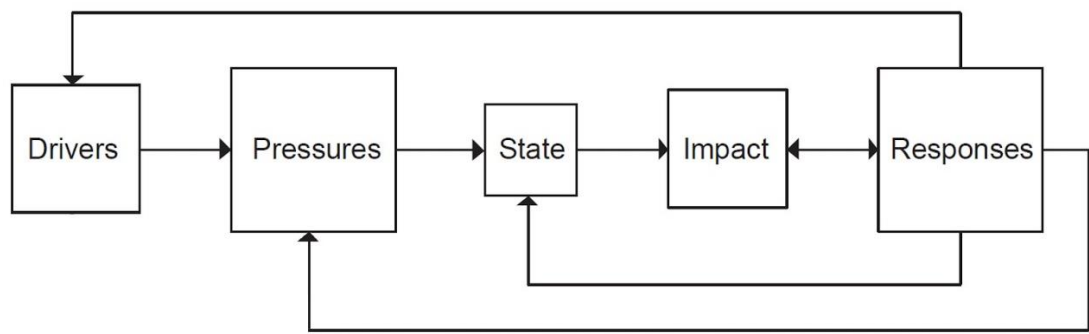


Figure 3.10 DPSIR model for sustainability assessment (Adapted from Dincer & Zamfirescu, 2018)

Sustainability indicators are characterized by their duties in the DPSIR model. From an energy system perspective, those indicators shape the direction of sustainable development strategies with the help of assessment of and advancement in technology. Fossil fuel-dependent economy in present and the developing technological advancements create social and environmental driving forces that cause a change in the current state of the sustainability of energy systems. The policy and decision-making process can be supported with the sustainability assessment indicators that are measured for the current state and technologically advanced state causing another pressure branch for sustainable development. The response should be capable of achieving a more sustainable society for the future. In some cases, this goal can be a driving pressure that creates feedbacks through policy-making for the need for more advanced technologies to create more sustainable solutions for the future. The role of sustainability indicators of energy systems for a planned sustainable energy society is introduced by Dincer & Zamfirescu (2018) and modified as illustrated in Figure 3.11.

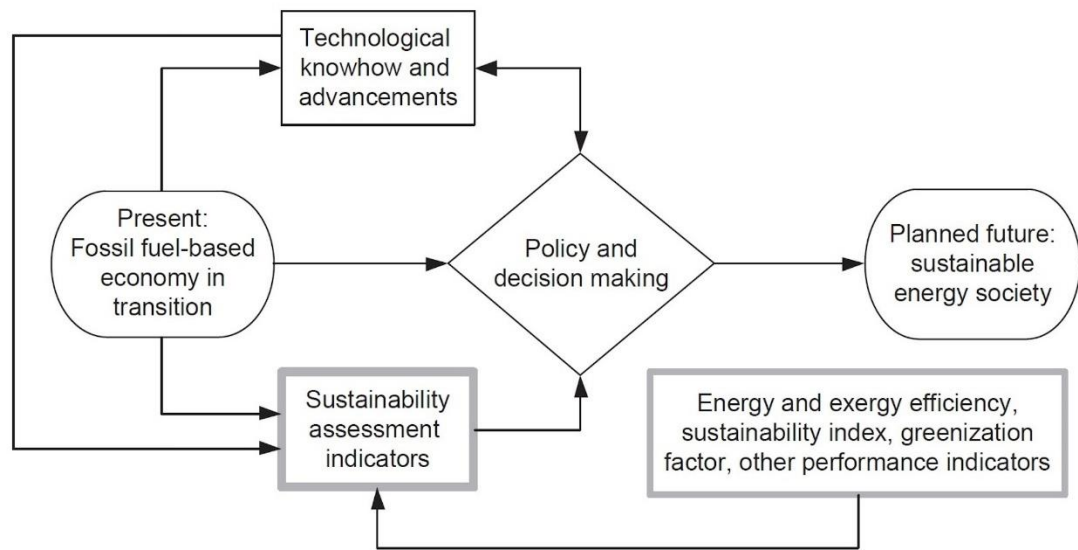


Figure 3.11 The role of sustainability assessment indicators of energy systems for a planned sustainable energy society (Adapted from Dincer & Zamfirescu, 2018)

An energy system consists of resource and process enabler inputs, and in return produces actual process outputs and wastes. Process outputs can be defined as the combination of utilizable exergy and recyclable streams like recoverable heat, emissions, or utilities (Linke et al., 2014). The sustainability of a system can be determined by measuring the wastes produced by the process. The wastes of an energy system generally show themselves in the form of pollutants emissions, waste energy, or material wastes. The energy analyses measure the waste energy according to the given inputs to the system. However, a direct measure of energy efficiency will not cover the effects of heat losses on the environment. In this context, exergy analyses become more reliable in the sustainability assessment of energy systems because the exergy of a system covers environmental factors as a reference.

3.5.2 Exergy Based Sustainability Assessment of Energy Systems

Energy analysis, instead of using the first law of thermodynamics, can also be carried out by using different types of energy measures, such as exergy and emergy. Both these forms of analyses are more advanced since they consider both the quality and the quantity of energy (Rosen & Dincer, 2001). The exergy of a system is the maximal amount of mechanical work that can be extracted (Wall, 1977). In terms of

evaluating the efficiency, loss, and performance of a system, exergy analysis is more useful than energy analysis (Caliskan et al., 2011). Exergy analysis offers a basis for sustainability assessment because exergy is a measure of abatement of the system subjected to the analysis from the environment. Exergy analysis is of major importance in the assessment of sustainability because the exergy-based efficiency of systems and processes represent a true measure of imperfections. It also indicates the possible ways to improve the energy systems and to design better ones. Destruction of exergy must be reduced as much as possible. Assessment of the exergy destruction offers the opportunity to quantify the environmental impact and the sustainability of any energy system (Dincer & Zamfirescu, 2018). Ness et al. (2007) categorize exergy as one of the emerging methods for sustainability assessment. Also, the exergy technique has been applied to different engineering fields thereby bringing understanding to the actual losses involved in energy conversion processes, sustainability level of energy systems and material interaction with the environment (Rosen et al., 2008; Midilli et al., 2012; Aydin, 2013; Gingerich & Mauter, 2015; Turan & Aydin, 2016; Abam et al., 2017; Abam et al., 2018a).

3.5.2.1 Exergetic Sustainability Indicators

The exergy-based parameters for a system or equipment reveal how their operating conditions and system characteristics affect sustainability (Balli & Hepbasli, 2014; Turan et al., 2014). Some examples of exergetic sustainability indicators can be given as exergy efficiency, waste exergy ratio, recoverable exergy ratio, exergy destruction factor (depletion number), environmental effect factor, exergetic sustainability index, sustainability index, exergy utilization factor, lack of productivity, relative irreversibility, and exergo-emission indicator. These indicators are used to quantify the sustainability of a system or equipment. In addition, the quantified data assist decision-making processes for further increase in the efficiency of the system or the equipment and more environmentally benign and sustainable solutions (Turan & Aydin, 2016; Maniali & Silveira, 2015).

The most commonly used exergetic sustainability indicators in the literature are given in Equations 3.8-3.17. Equations 3.15-3.17 are utilized for the assessment of energy systems where fuel depletion takes place on the operation, e.g., power plants, diesel engines, and gas turbines (Aydin, 2013; Turan & Aydin, 2016; Tsougranis & Wu, 2018; Chowdhury et al., 2020b; Jankowski & Borsukiewicz, 2020):

$$\text{Exergy efficiency} = \frac{\text{Total useful exergy output}}{\text{Total exergy inlet}} \quad (3.8)$$

$$\text{Waste exergy ratio, WER} = \frac{\text{Total waste exergy out}}{\text{Total exergy inlet}} = \frac{\text{Total exergy destruction} + \text{Total exergy loss}}{\text{Total exergy inlet}} \quad (3.9)$$

$$\text{Environmental effect factor, EEF} = \frac{\text{WER}}{\text{Exergy efficiency}} \quad (3.10)$$

$$\text{Sustainability index, SI} = \frac{1}{1 - \text{Exergy efficiency}} \quad (3.11)$$

$$\text{Exergetic sustainability index, ESI} = \frac{1}{\text{EEF}} \quad (3.12)$$

$$\text{Environmental destruction coefficient, EDC} = \frac{1}{\text{Exergy efficiency}} \quad (3.13)$$

$$\text{Improvement potential, IP} = (1 - \text{Exergy efficiency}) \cdot (\text{Exergy inlet} - \text{Exergy output}) \quad (3.14)$$

$$\text{Exergy destruction factor (Depletion number, DN)} = \frac{\text{Exergy destruction}}{\text{Total exergy inlet}} \quad (3.15)$$

$$\text{Environmental destruction index, EDI} = \text{DN} \cdot (\text{Unusable}) \text{ WER} \cdot \text{EDC} \quad (3.16)$$

$$\text{Environmental benign index, EBI} = \frac{1}{\text{EDI}} \quad (3.17)$$

3.5.2.2 Literature Survey on Exergy Based Sustainability Assessment

The focus of exergy-based sustainability assessment studies in the early stages was gathered around new technological options for small-scale home applications. The studies for industrial applications were then rapidly increased as the method reveals comparative data for the existing/developing technological options and further improvement potentials of their components. Balta et al. (2010) investigated the performance of four energy options for building HVAC systems, which are ground heat pump, condensing boiler, conventional boiler, and solar collector systems. They aimed to assess the most sustainable system among those four systems by comparing the SI values for all cases at varying reference outdoor temperatures of -10 to 15 °C, and a target indoor temperature of 20 °C. The results indicate that the most sustainable system among those four is the solar collector heating system for the case study of the reference building. Total exergy efficiencies of the heat pump, condensing boiler, conventional boiler, and solar collector for the reference outdoor temperature of 0 °C are 3.66, 3.31, 2.91, and 12.64% with the sustainable index values of 1.039, 1.034, 1.030, and 1.144, respectively. The results also showed that the solar collectors perform a better sustainability graphic for colder ambient temperatures that can reach above 1.40 at -10 °C. Caliskan et al. (2011) conducted a sustainability assessment study of a solar-ground based heat pump with thermal energy storage for a case study of a 120-m² house. The study aimed to obtain the sustainability aspects of the system for varying reference temperatures of 0 to 25 °C. It was indicated that the exergy efficiency, as well as the sustainability index, is inversely proportional to the reference temperature. Therefore, the obtained SI rate values were bigger at 0 °C (1.6946) than 25 °C (1.6193) reference temperatures.

Caliskan & Hepbasli (2011) carried out an exergy-based sustainability assessment to a four-stroke diesel engine for the running up conditions with diesel oil no.2 and two different kinds of biodiesels (soy methyl ester and high-oleic methyl ester). It was found that using biodiesels shows a better sustainability performance than those of diesel no.2. Using the 0 °C reference temperature, the obtained average SI rates were 1.6015, 1.6116, 1.6097 for diesel no.2, soy methyl ester and high-oleic methyl ester

fuels, respectively. Wu et al. (2015b) investigated the renewability and sustainability of household biogas utilization in rural areas of China by using a cosmic exergy based assessment approach that aggregates natural resources, economic inputs and environmental contamination on a common basis. The results indicate that the integrated biogas system nearly doubles the ecological benefits based on a 20-years of lifespan scenario when compared to the conventional production system. Maniali & Silveira (2015) compared the sustainability of different energy options for rural area electrification. They select eleven indicators among economic, environmental, social, technical and institutional pillars of sustainability. The method for evaluating the sustainability indicators was mainly based on normalizing the data for those indicators found in the literature. Ten technologies among the most suitable options for rural areas were investigated for 2005, 2010 and 2015, respectively. The results enabled the investigation of the examined technologies from different indicator perspectives using a radar diagram. The sustainability performance was further investigated by calculating energy technology sustainability indexes for those technologies. It was shown that diesel generators were still a good option but placed 5th among these technologies. Biomass and micro/pico hydroelectric power have resulted as more sustainable options than others for rural area electrification. It has also resulted that the emerging technologies, such as solar and wind energy options, possessed lower sustainability performances but have been increasing their sustainability performance for the years from 2005 to 2015. It would be an interesting study to assess the performance of those technologies considering the new developments through 2020.

Energy, exergy and exergoeconomic analyses with sustainability and environmental analyses have been a hot topic in aviation for different aircraft engines (Balli & Hepbasli, 2014; Aydin et al., 2015; Turan & Aydin, 2016; Şöhret et al., 2016; Ekici et al., 2016; Baklacioglu et al., 2018; Söğüt, 2018; Sahu et al., 2018; Aygun & Turan, 2020; Yuksel et al., 2020). The analyses results of these studies are utilizable as a guide for the design and improvement efforts of the aircraft engines. The major losses from the parts of the aircraft engines were obtained by the energy and exergy analyses. The environmental effect and the sustainability of the engine were assessed using the environmental effect factor and exergetic sustainability index. Waste exergy

ratio, recoverable exergy ratio and exergy destruction factor were other metrics that were taken into consideration while indicating the sustainability of the aircraft engines. Sustainability analysis of the aircraft engines was investigated in a wider perspective in some of these studies with the calculation of the sustainable cost index (SCI), which is the ratio of unit exergy cost of products to the sustainability index. It is mentioned that the methods for lowering the SCI value are sought by investigating the effect of parameters such as; system's product cost flow, exergy efficiency, SI, fuel cost rate, total capital investments (Sahu et al., 2018).

There can be found several studies that were adopted the sustainability index method for the sustainability assessment of cogeneration and tri-generation power plants for various areas. Ozcan & Dincer (2014) analyzed a tri-generation system consisting of parabolic trough solar collectors, an ORC, an absorption chiller and a thermal energy storage system. They obtained that the sustainability of the system improves with higher solar irradiation and lower ambient temperature. The results also showed that the plant performance is strongly dependent on the solar collector design and operational parameters of mass flow rate, pressure ratios and specified target temperatures of the tri-generation system components. Acar & Dincer (2014) utilized the exergetic sustainability index method to investigate the further development potential of the components of zero-emission power plant for coproduction of electricity and methanol. The analyses were carried out for the water electrolysis unit, methanol synthesis unit, overall power plant, at which the index results were found as 1.43, 1.7, 2.43, respectively. They found that the ambient temperature does not have a significant effect on the exergetic performance of the system. They also suggested using turbines driven by combustion gases to meet the work requirements of the electrolysis unit and recover the waste heat to increase the exergy efficiency of the total system by 3%. Caglayan & Caliskan (2018) proposed a cogeneration system for the ceramic industry that can both produce electrical power through a gas turbine and also provide waste heat to be utilized in drying processes. The sustainability index values of cogeneration plant components provided an improvement opportunity by analyzing the system model with varying dead state temperatures. The results showed that the sustainability of the overall cogeneration plant is directly proportional to the

ambient temperature, which scores better at 30 °C with the sustainability index of 1.4282. Nami & Anvari-Moghaddam (2020) investigated tri-generation systems for waste heat recovery from cement plants in exergo-sustainability and economic aspects. Recuperative ORC and steam Rankine cycle systems were considered as power generation options for the proposed combined cooling, heating and power (CCHP) system designs. As the cooling demand of the plant seasonally varies, the two system configurations were compared according to the cooling demand. The results of the study showed that the ORC-based CCHP system has better sustainability performance, but has a higher payback period than the Rankine-based CCHP system.

Stougie (2014) proposed a method called “Total Cumulative Exergy Loss (TCExL)” that takes into account all exergy losses of a system during its lifecycle. The method is used to assess the exergetic sustainability of a system by calculating internal exergy losses including the abatement exergy for the emissions of NO_x, SO₂, CO₂, and phosphate. The reference abatement exergy values were taken from the literature that has been carried out with the abatement technologies of the years between 1997-2011, which were only able to remove a certain amount of the emissions (from 80 to 99%). As the abatement technologies improve, the reference values would need to be updated. The exergy losses caused by the land use is based on the exergy that becomes unavailable to the ecosystem because of the occupied land area for the system. The exergy loss caused by land use is calculated by the multiplication of average solar irradiation dropped to the occupied area with the efficiency of capturing solar energy via photosynthesis and the amount of exergy per amount of sunlight. The study also investigated a case study of power generation with a coal-fired power plant and the integration of ORC and LNG evaporation systems for utilizing the waste heat of the power plant. The sustainability assessment included economic, environmental and social aspects and a comparison of the results for varying TCExL has been carried out. The results showed that exergetic losses of the system inversely affect all of the sustainability scores, but have a much higher impact on the economic indicators. The method was also utilized in Stougie et al. (2018) for the sustainability assessment of power generation from biomass with environmental aspect. Bioethanol, anaerobic digestion and supercritical water gasification systems were assessed using the

calculated exergetic sustainability and ReCiPe scores. The results of the environmental and exergetic sustainability assessments showed that the bioethanol system can be preferred among the other options, but the difference in the scores was not large.

ORC systems are one of the widely adopted technology for WHR. The system can be integrated into many applications, such as cogeneration, tri-generation and stand-alone plants as an electrical power generation option. Because the ORC is still an emerging technology, studies for ORC system utilization have been increasing in the last few years. Baral et al. (2015) studied the thermoeconomic performance of solar organic Rankine cycle (SORC) systems. They investigated the sustainability development through SORC systems by comparing the saved petroleum for electricity generation with petroleum-based plants. Exergy based sustainability was also investigated in this study to assess the most sustainable system configuration for different solar source temperatures of the SORC system, which is concluded as higher solar source temperatures give better sustainability index results. Ataei et al. (2015) investigated the thermodynamic performance and exergetic sustainability of simple, regenerative, recuperative, and combined ORC systems with R113, RC318, isopentane and n-hexane working fluids. They concluded that combined regenerative and recuperative ORC system with n-hexane working fluid showed the best performance among the examined configurations. The results also showed that the sustainability index of the ORC system slightly increases for reducing ambient temperatures. A research group from Nigeria published a set of environmental and exergetic sustainability assessment studies of various ORC configurations for varying operating system parameters (Abam et al., 2017; Abam et al., 2018a; Abam et al., 2018b; Abam et al., 2018c). These studies mainly focused on the exergy efficiency and exergy destruction rates of ORC system components at varying evaporator pressures, turbine inlet temperatures and heat source temperatures. The research group also investigated the performance of different organic working fluids for varying operational parameters. They also presented comparable results for exergetic sustainability assessment of the ORC systems by calculating the waste exergy ratios, environmental effect factors, exergetic sustainability indexes for different ORC working fluid and operational parameters.

The sustainability of different ORC system configurations for marine vessels was also investigated from exergetic, economic and environmental point of views. Kalikatzarakis & Frangopoulos (2015) carried out a multi-criteria evaluation of different ORC system configurations that are designed for a large marine vessel and using the waste heat of the main diesel engine's lubricating oil, jacket cooling water and charge air cooler as the heat source. Economic criteria (initial investment, net present value, dynamic payback period, and internal return rate), environmental criteria (CO_2 , NO_x , and SO_x) and technical criteria (power output, total energy efficiency, and total exergy efficiency) were evaluated to find out the optimized system design parameters by operational and design optimizations for achieving a better sustainability performance with the ORC system. Tsougranis & Wu (2018) studied the feasibility of power generation with ORC systems by using the thermal and cryogenic waste heat energy of LNG powered marine vessels. The proposed ORC system is a simple ORC design consisting of an evaporator using the waste heat exhaust gas emitted from the main engine and a condenser that utilizes the cold waste heat of LNG fuel as the cooling utility before it is supplied to the main engine. Energy, exergy, economic and sustainability assessment analyses were carried out for single-stage and two-stage regenerative ORC system configurations utilizing organic working fluids with low melting points. They calculated a payback period of 4 years for both configurations. The results showed that ethylene (C_2H_4) working fluid performs better for condensing pressures above 1.1 bar. The sustainability assessment results for C_2H_4 working fluid indicate a 12.1% higher sustainability index value for two-stage regenerative ORC configuration in comparison with single-stage regenerative ORC configuration. Jankowski & Borsukiewicz (2020) defined another exergetic sustainability indicator, which they called the exergy utilization index (XUI). The index is used to indicate how much of the heat source exergy is utilized. As the heat source outlet temperature gets closer to the ambient temperature, the XUI is maximized, which is also resulted in a minimization of the exergy waste to the environment. A parametric study was carried out in their study to reveal the relationship between XUI and other exergy based indicators for an ORC system utilizing geothermal energy. Higher values of XUI increased the exergy-based

indicators as expected. However, the payback period of the system is minimized between 60-80% XUI values for the optimal operating conditions of the ORC system.

The literature review shows that the exergy analyses and exergetic sustainability indicators have been successfully implemented as a tool to assess the sustainability of energy systems. In the following section, an exergy based sustainability assessment is carried by using the exergy analyses results of the ORC integrated WHRS of the reference ship in Section 3.4. This study aims at indicating comparable quantified data with the assessment results that show sustainability improvement potentials if the proposed design is applied. It is expected that the quantified sustainability assessment results will help decision-making processes of policy makers and ship owners to apply ORC systems to marine vessels for more energy-efficient and environment-friendly operations.

3.5.3 Exergy Based Sustainability Assessment Results of the Organic Rankine Cycle Integrated Waste Heat Recovery System for the Reference Container Ship

In this section, exergy based sustainability assessment of the proposed ORC integrated WHRS for the reference container ship is carried out using Equations 3.8-3.14. It is aimed to provide exergetic sustainability indicators for assisting decision-making processes of ship owners, engine and ORC system manufacturers that consist of decisions of implementing the proposed system design and which performance parameters should be used to achieve a better sustainability performance.

As one of the sustainability indicators, the exergy efficiency of the proposed ORC system design has been calculated in Section 3.3 and the results have been given in Figure 3.7 for varying ORC evaporator pressures and different types of ORC working fluids. Waste exergy ratio (WER) is defined as the ratio of total waste exergy to the total exergy inlet of the system. Total waste exergy is found as the sum of total exergy destruction from the ORC system and heat losses to the surroundings as in Equation 3.9. Total exergy inlet to the ORC system is the exergy inlet from the ORC evaporator to the ORC system. The effect of ORC evaporator pressure on WER for nine different

working fluids are investigated and the results are presented in Figure 3.12. It can be deduced that WER is inversely related to the exergy efficiency of the ORC system. The maximum value of WER is obtained as 89.5% for R1234yf at 1000 kPa, which shows the worst performance following the results obtained from the energy and exergy analyses. The WER is minimum for R1234ze(Z) at 3 MPa with the value of 10.6%, but R245fa starts to perform better for ORC evaporator pressures less than 1600 kPa.

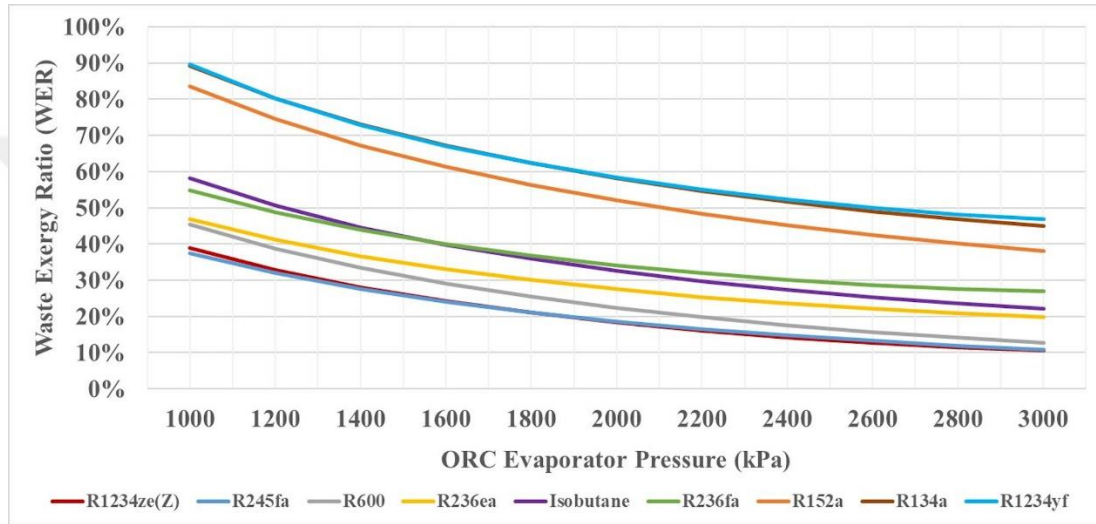


Figure 3.12 WER values obtained for different evaporator pressures and working fluids

Environmental effect factor (EEF) indicates whether the energy conversion system gives harms to the environment. EEF of a system is expected to increase with increased waste exergy output and reduced useful exergy output. EEF values for the proposed ORC integrated WHRS are calculated using Equation 3.10. The environmental destruction coefficient (EDC) is inversely proportional to the exergy efficiency of the system as can be seen in Equation 3.13. EDC shows a similar pattern with the EEF as they both affected by the exergy efficiency. In the sustainability perspective, EEF should be as close as to 0 (zero) that indicates the waste exergy of the system is minimized however, EDC values can vary between 1 to infinity and a system with EDC of closer to 1 will result in more sustainable operation. The EEF and EDC values calculated for the proposed system design are depicted in Figure 3.13 (a) and (b). R1234ze(Z) and R245fa working fluids show good environmental performance in this design with EEF values of 0.167 and 0.174 at the selected operating pressure of 2400

kPa, respectively. EDC values are calculated as 1.666 and 1.173 at the same conditions. The environmental effects can be further reduced with the increased evaporator pressure.

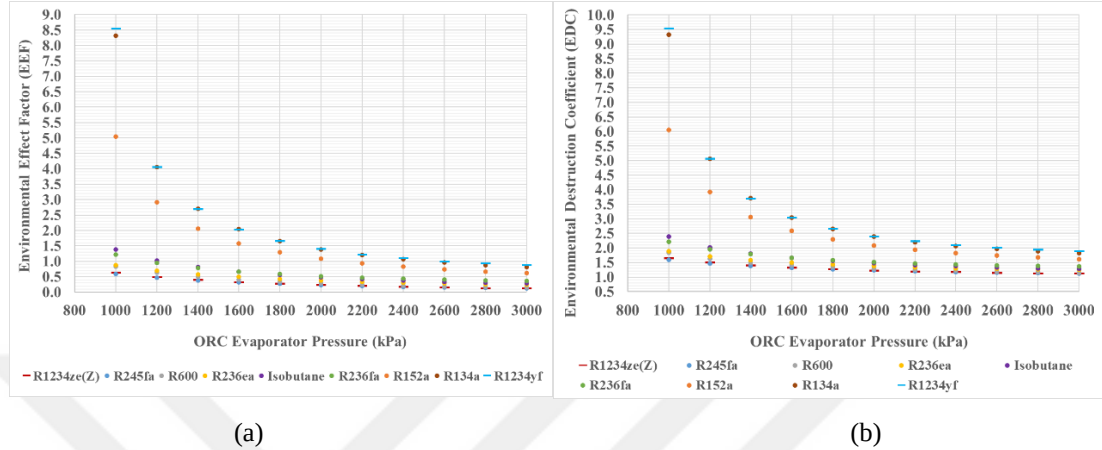


Figure 3.13 (a) EEF (b) EDC values calculated for different evaporator pressures and working fluids

The exergetic sustainability index (ESI) and Sustainability index (SI) are among the most used exergy based sustainability indicators that indicate the degree of sustainability. ESI and SI work in a similar correlation as between EEF and EDC. ESI is the inverse of EEF as given in Equation 3.12; thereby directly affected by the exergy efficiency of the system. ESI varies between 0 and infinity, while SI values are between 1 and infinity according to Equation 3.11. As the system efficiency gets higher, lower waste exergy ratios and environmental effects as a result higher SI and ESI are achieved. The calculated ESI and SI values are given in Figure 3.14 (a) and (b), respectively. The maximum ESI and SI values are obtained from R1234ze(Z) at 3000 kPa at the value of 8.435 and 9.435. For the selected operating pressure of 2400 kPa and working fluid of R245fa, the calculations show ESI and SI values of 5.764 and 6.764.

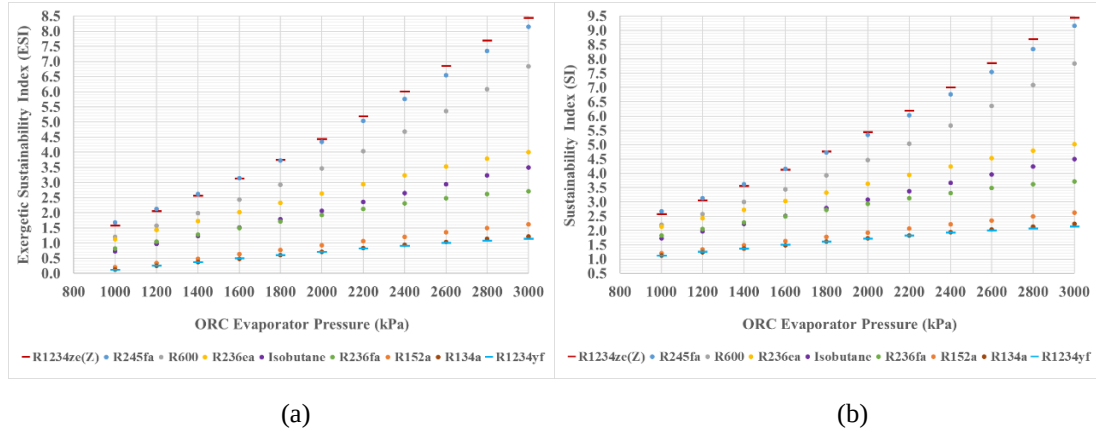


Figure 3.14 (a) ESI (b) SI values calculated for different evaporator pressures and working fluids

Improvement potential (IP) shows how much more work could have been generated by the given exergy input to the system. IP indicator depends on the exergy efficiency with the exergy inputs and outputs as can be seen in Equation 3.14. The difference between the exergy inlet and output is constant at each engine load category for varying working fluid types because the heat flow to the ORC evaporator is limited by the target temperatures of the thermal oil side. It means that the higher exergy efficiency values lead to lower IPs as illustrated in Figure 3.15. As the engine load increases, the exergy input to the ORC system also increases and leaves more improvement potentials for each working fluid types.

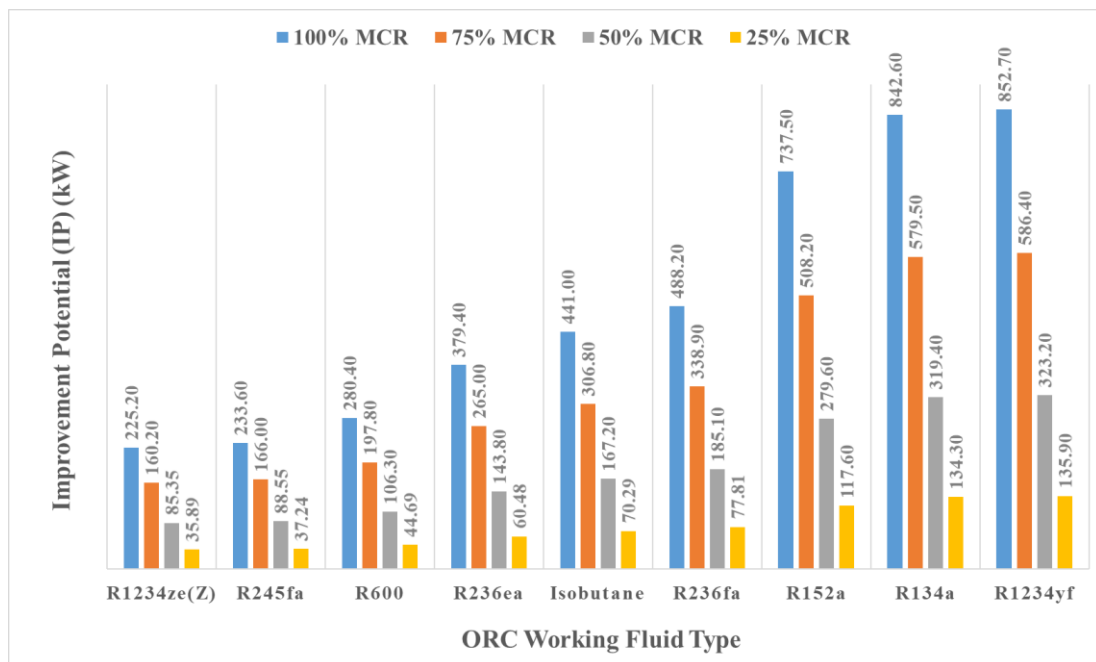


Figure 3.15 IP for different engine loads and working fluids at ORC evaporator pressure of 2400 kPa

3.6 Discussion

The summary of major results derived from the subjects of this chapter are as follows:

- The optimum HEN design can be obtained by dividing the scavenge air cooler into three stages considering the pinch analysis rules because the scavenge air has a large temperature range between 152 °C to 35 °C (for 76% MCR). The WHR of the scavenge air has been obtained above the pinch point, and the remaining heat below the pinch point was discharged overboard.
- All the waste heat cannot be collected by thermal oil, as some of the waste heat needs to be discharged through the sea by cooling utilities. The unavailable waste heat was found as 23.71, 16.56, 13.17, and 7.81% of the total waste heat produced by the heat sources for 25, 50, 76, and 100% MCR, respectively. As the heat load above the pinch point increases, the percentage of the utilizable waste heat also increases; because the cold utilities are only required below the pinch. For those ships that mostly operate on 25% MCR and 50% MCR engine loads, a new HEN design optimization should be carried out to minimize the heat loss through overboard.
- Among the ORC fluid types, it was obtained that R1234ze(Z) and R245fa yield the best performance for the selected target temperatures and working pressures. For example, the thermal efficiency of the ORC system with R1234ze(Z) was found as 15.24% at 2400 kPa, showing a slight difference from R245fa (15.02%) as can be seen in Figure 3.6. R600, R236ea, Isobutane, and R236fa yielded the thermal efficiency of 14.52, 13.46, 12.81, and 12.30%, respectively. It should be also noted that R245fa yielded better performance for systems that has evaporator pressure of 1400 kPa or less. Since the evaporator pressure of 2400 kPa was used in the analyses, R1234ze(Z) working fluid was chosen by taking into consideration the low flammability level and GWP value for a safer and more environmentally friendly operation.

- The ORC system, with its all components, has the highest exergy destruction rate for all loads. The exergy destruction rate of the ORC for 76% MCR engine load was found as 372.9 kW (34.94% of the total system). LO cooler and the TOH have exergy destruction rates of 335.9 kW and 257.2 kW (31.47% and 24.10% of the total system), respectively, for the same engine load. The HT air cooler has possessed low exergy destruction ratios until 76% MCR but got higher values at 100% MCR because of the exponentially increased scavenge air inlet temperature of the main engine. The temperature and the mass flow rate of the thermal oil entering the TOH are calculated slightly higher in the 76% MCR case than 100% MCR, causing a minor decrease in the exergy destruction ratio of the TOH at 100% MCR.
- The net power outputs generated by the proposed integrated ORC and WHRS were calculated as 227.9, 541.9, 970.8, and 1430 kW, which correspond to 8.24, 9.80, 11.55, and 12.93% of the net power output produced by the engine for 25, 50, 76, and 100% MCR, respectively. The values show that all of the electricity demand of the ship can be provided by the WHRS for the engine load above 50% MCR. The extra energy production can be stored by the batteries to be utilized when the main engine stopped for loading/unloading, bunkering processes, etc.
- The sustainability of the ORC integrated WHRS design is assessed using exergy based sustainability indicators of WER, EEF, EDC, ESI, SI, and IP. The minimum WER is obtained from R1234ze(Z) at 3 MPa with the value of 10.6%. R1234ze(Z) and R245fa working fluids show good environmental performance for the proposed system design with EEF values of 0.167 and 0.174 at the selected operating pressure of 2400 kPa. The maximum ESI values are obtained from R1234ze(Z) and R245fa at 3000 kPa at the value of 8.435 and 8.154, respectively. The ESI value R245fa at 2400 kPa is calculated as 5.764. As the engine load gets higher, more IP is left out from the system. In addition, IP reduces with the increased exergy efficiency as expected.

CHAPTER FOUR

AN ORC INTEGRATED WASTE HEAT RECOVERY SYSTEM FOR TANKER SHIPS WITH CARGO OIL PUMP TURBINE

4.1 System Description

The availability and potential use of COPT based waste heat has been mentioned in Section 2.1.5. In this chapter, it was aimed to introduce a new WHRS design for tanker ships that can utilize both marine propulsion diesel engines during navigation and COPT based waste heat during cargo discharge operations in ports. As a case study, the waste heat sources of a reference oil tanker vessel were identified. A crude oil tanker ship with 116,000 DWT capacity was chosen as a reference to propose the optimal HEN design with an integrated ORC system, reveal the WHR potential for both on navigation mode, and discharge operation mode. The reference vessel is equipped with an MAN 6S60ME-C main engine that has a specified MCR power of 13,560 kW at 105 RPM for propulsion. The pinch point analysis was utilized to establish the optimal HEN design for navigation mode using the main engine based waste heat sources (Section 4.2). Then, the integration of COPT based waste heat source to the new WHRS design for utilization on discharge operation mode was introduced (Section 4.3). The energy and exergy analyses (Section 4.4) and exergy based sustainability assessment (Section 4.5.1 and 4.5.2) of the proposed ORC integrated WHRS design for both navigation mode and discharge operation mode of the crude oil tanker ship were examined.

A COPT requires high-pressure steam to drive the steam turbine that is connected to a pump with a shaft and gear mechanism. When the steam turbine runs, the pump will start to discharge the cargo oil to the port facility. The high-pressure steam is generated in auxiliary boilers that are heated by an external source of fuel. In a conventional COPT system, boiler feed water is supplied to the system from a cascade (hotwell) tank via feed water pump(s). The water is transformed into high-pressure steam in the auxiliary boiler and directed to the COPTs on cargo holds through the high-pressure steam line. If high-speed discharge operation is required, the second

auxiliary boiler might also be operated for producing higher mass flow rates of steam. The steam dump valve adjusts the rotational speed of the COPTs by controlling the amount of steam bypass to the vacuum condenser. The exhaust of COPTs contains steam or water/vapor mixture that should be cooled down to its liquid form to be recirculated by the feed water pumps. This heat exchange process takes place in a vacuum condenser that is utilized in the system to generate backpressure with the help of condensate water pumps for improving turbine performance and recollect the exhaust steam (DongHwa Entec, 2020). The waste heat of the COPTs is discharged overboard with seawater through the vacuum condenser, which is supplied with constant speed cooling seawater pumps. The schematic of a conventional COPT system is illustrated in Figure 4.1.

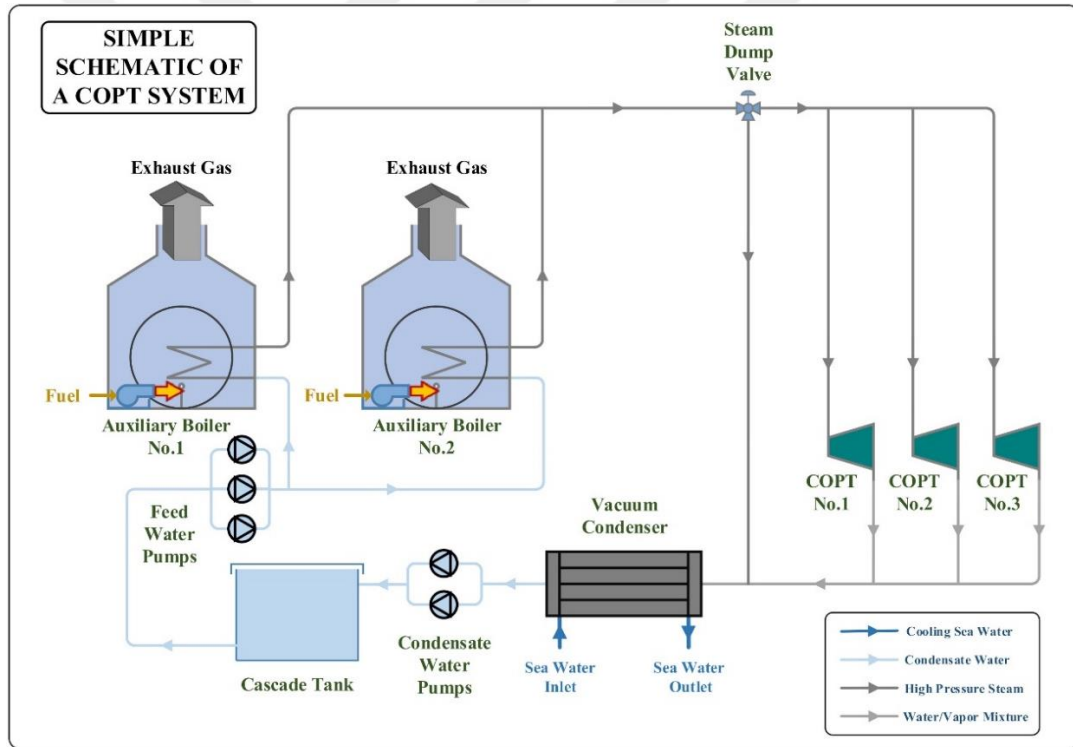


Figure 4.1 Schematic of a COPT system onboard a large tanker ship

The novel idea in the proposed ORC integrated WHRS for the reference tanker ship is that the system collects the COPT based waste heat and keep the power generation in ports during the cargo discharge operations using the ORC system onboard. During cargo discharge operations in ports, the main engine is stopped causing no heat generation from the main engine. When the cargo discharge operation

started, the proposed heat collection system bypasses the main engine based waste heat sources using a three-way control valve and directs the thermal oil to the vacuum condenser. The proposed system design does not require any modifications to the existing COPT system design. The only difference is the cooling media in the vacuum condenser, which is replaced with thermal oil instead of sea water. It should be noted that the COPT based waste heat is only available during discharge operations from the ship because the cargo loading to the ship is carried out by the pump stations on the port. The WHR potential from the exhaust gases of the auxiliary boilers is also feasible to be recovered by the ORC system. However, this waste heat source is not taken into consideration in this study due to the lack of required data on the reference vessel.

4.2 Optimal Heat Exchanger Network Design with Pinch Point Analysis for Navigation Mode

Tanker ships are designed with large block coefficients that mean a ship construction of a very large box-like shape. Ship resistances largely affect this type of construction thus large oil tankers are generally operated at lower service speeds than container ships. The optimal HEN for the navigation mode of the ship is designed using pinch point analysis and the design parameters were chosen according to 60% MCR engine load, which can be accepted as an economic engine load for large oil tankers. The data sets for the main engine based waste heat sources given in Table 4.1 were provided from MAN CEAS Wizard (2021) for an MAN 6S60ME-C engine with 13,560 kW MCR power. The input data except for 60% MCR engine loads were also given in Table 4.1 to be utilized for energy and exergy analyses. A detailed overview and the rules of pinch point analysis has been described in Section 3.2 and Konur et al. (2020b). The major waste heat sources of the reference tanker vessel are the same as the container ship case for navigation mode as given in Figure 3.1 (exhaust gas, scavenge air cooler, jacket cooling water (JW), main lubricating oil). The pinch point analysis is used to optimize the waste heat utilization from the system with minimum cooling and heating utility usage (no heating utility usage takes place in this case) during the navigation phases of the vessel. The central cooling and the exhaust gas systems of the reference tanker ship is hypothetically modified. A closed-loop thermal

oil stream was designed using the pinch point analysis to collect the waste heat and transfer it to the organic working fluid of an ORC system through the ORC evaporator.

Table 4.1 Input parameters for waste heat sources and cooling utility for navigation mode of the tanker

ENGINE LOADS	INPUT PARAMETERS	EXHAUST STREAM	SCAVENGE AIR STREAM	JW STREAM	LO STREAM	THERMAL OIL STREAM	COOLING UTILITY
FOR ALL ENGINE LOADS	Substance	Humid Air	Humid Air	Water	Unused Engine Oil	Thermia D	Water
	Target outlet temperature (°C)	90	21	65	44	120	17
	Average specific heat (kJ/kg·K)	1.089	-	4.191	2.006	2.166	4.195
100 % MCR ENGINE LOAD	Mass flow rate (kg/s)	30.3	29.7	23.56	44.41	-	-
	Inlet temperature (°C)	218	181	81	55	45	10
	Heat dissipated (kW)	4,224	4,830	1,580	980	-	-
	Pressure (kPa)*	-	509	390	530	420	330
90 % MCR ENGINE LOAD	Mass flow rate (kg/s)	28.2	27.7	21.92	43.05	-	-
	Inlet temperature (°C)	205	168	81	55	45	10
	Heat dissipated (kW)	3,532	4,130	1,470	950	-	-
	Pressure (kPa)*	-	472	390	530	420	330
80 % MCR ENGINE LOAD	Mass flow rate (kg/s)	26.0	25.5	20.13	41.69	-	-
	Inlet temperature (°C)	199	156	81	55	45	10
	Heat dissipated (kW)	3,086	3,490	1,350	920	-	-
	Pressure (kPa)*	-	439	390	530	420	330
70 % MCR ENGINE LOAD	Mass flow rate (kg/s)	23.5	23.1	18.34	39.42	-	-
	Inlet temperature (°C)	198	142	81	55	45	10
	Heat dissipated (kW)	2,764	2,840	1,230	870	-	-
	Pressure (kPa)*	-	405	390	530	420	330
60 % MCR ENGINE LOAD	Mass flow rate (kg/s)	20.8	20.4	16.55	37.16	33.09	40.05
	Inlet temperature (°C)	202	125	81	55	45	10
	Heat dissipated (kW)	2,537	2,160	1,110	820	-	-
	Pressure (kPa)*	-	364	390	530	420	330
50 % MCR ENGINE LOAD	Mass flow rate (kg/s)	17.8	17.5	14.91	34.44	-	-
	Inlet temperature (°C)	211	106	81	55	45	10
	Heat dissipated (kW)	2,345	1,510	1,000	760	-	-
	Pressure (kPa)*	-	323	390	530	420	330
40 % MCR ENGINE LOAD	Mass flow rate (kg/s)	14.5	14.3	13.12	31.27	-	-
	Inlet temperature (°C)	227	87	81	55	45	10
	Heat dissipated (kW)	2,163	960	880	690	-	-
	Pressure (kPa)*	-	289	390	530	420	330
30 % MCR ENGINE LOAD	Mass flow rate (kg/s)	12.6	12.4	11.33	27.19	-	-
	Inlet temperature (°C)	200	69	81	55	45	10
	Heat dissipated (kW)	1,509	600	760	600	-	-
	Pressure (kPa)*	-	259	390	530	420	330

*The indicated values are absolute pressures

The next step of pinch point analysis is to investigate the optimal waste heat recovery potential for the given operating parameters of 60% MCR engine load based on the CC and GCC. The CC was formed by integrating hot streams of waste heat sources in a temperature vs. heat flow line graph. The only cold stream in the system is defined as thermal oil as can be seen from the blue line in Figure 4.2(a). The pinch point, which is the minimum vertical distance between the hot and cold composite curves ($\Delta T_{\min}$), can be changed by shifting the cold composite curve to the left or right (supertargetting) for fine-tuning the WHR potential and adjusting the heating or cooling utility requirements that help to achieve the system's optimal WHR goal. As the pinch point decreases, WHR potential increases. However, the investment costs of heat exchangers will be higher due to the increased heat transfer area requirement.

Analysis results show the optimal HEN can be achieved by a 23 kW heating utility addition, thus an additional small heat exchanger is required in the system. Supertargetting was utilized for this case to eliminate the heating utility supply and heat exchanger addition requirement to the system. In addition, it would not be feasible to add heat exchangers for such small amounts of heat on ships that could operate at variable heat loads. The cold composite curve is shifted to the left and $\Delta T_{\min}$ is changed from 5.9 to 5.4 °C, thereby no heat utility addition is necessary for the specified target temperatures. The final CC of the pinch analysis is given in Figure 4.2(a). The pinch temperatures for the proposed HEN were found as 49.6 °C for the cold pinch, and 55 °C for the hot pinch. According to the final CC, the PinCH 3.0 software prints the GCC as given in Figure 4.2(b). GCC is formed by heat flow differences between the hot and cold composite curves on the same temperature axis. The area in grey shows the maximum WHR potential of the HEN for the given operating parameters. The amount of heating or cooling utility requirement and further optimization potentials concerning the heating and cooling utilities can be examined from this graph.

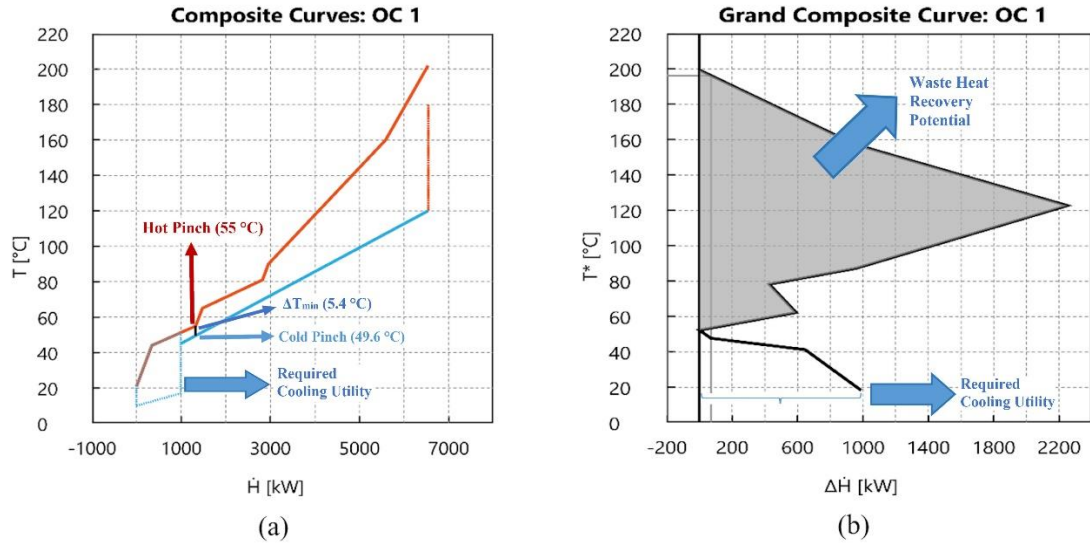


Figure 4.2 (a) Composite curve (CC) and (b) grand composite curve (GCC) of the heat sources for the engine at 60% MCR engine load

PinCH 3.0 software calculates the maximum WHR potential by using energy balance equations. The pinch rules are applied for the design phase of the optimal HEN of the tanker vessel for navigation mode to be able to recover the available waste heat as close as possible to the maximum potential. This is accomplished by integrating the hot and cold streams at the right temperature range with an optimal size heat exchanger. The pinch rules are beneficial in designing a cost-efficient HEN by ensuring that the system does not need extra heating or cooling utility and eliminate excessive heat exchanger utilization to restore the energy balance of the WHRS.

Figure 4.3 shows the optimal HEN design of the reference crude oil tanker ship for 60% MCR engine load. The PinCH 3.0 software generated the heat streams with their target temperatures and heat capacity rates ($\dot{m} \cdot c_p$) on a below the cold pinch/above the hot pinch template. The thermal oil stream is then branched above the pinch point and the heat exchangers were added to the system following the pinch rules. The thermal oil first collects the lowest degree waste heat of the system from the LO stream with the HT LO cooler. A majority of the heat of the LO stream has to be discharged overboard as unutilized waste heat from the Final LO Cooler to achieve the target temperatures of the system. The thermal oil stream is then branched to collect the waste heat at similar temperature ranges. The heat capacity rates of the branches are adjusted

by the amount of waste heat that has placed on the branches. The first branch completely recovers the waste heat of jacket cooling water (JW). The exhaust waste heat is collected into two stages. The first branch collects around two-third of the exhaust waste heat in TOH LT after passing through JW Cooler. The second branch starts its heat collection process with the scavenge air waste heat above the pinch point on HT Air Cooler. The scavenge air waste heat below the pinch point cannot be collected in the system, so the unutilized waste heat is discharged overboard from the Final Air Cooler. The second thermal oil branch continues the heat collection with high-temperature exhaust gas in TOH HT after the HT Air Cooler.

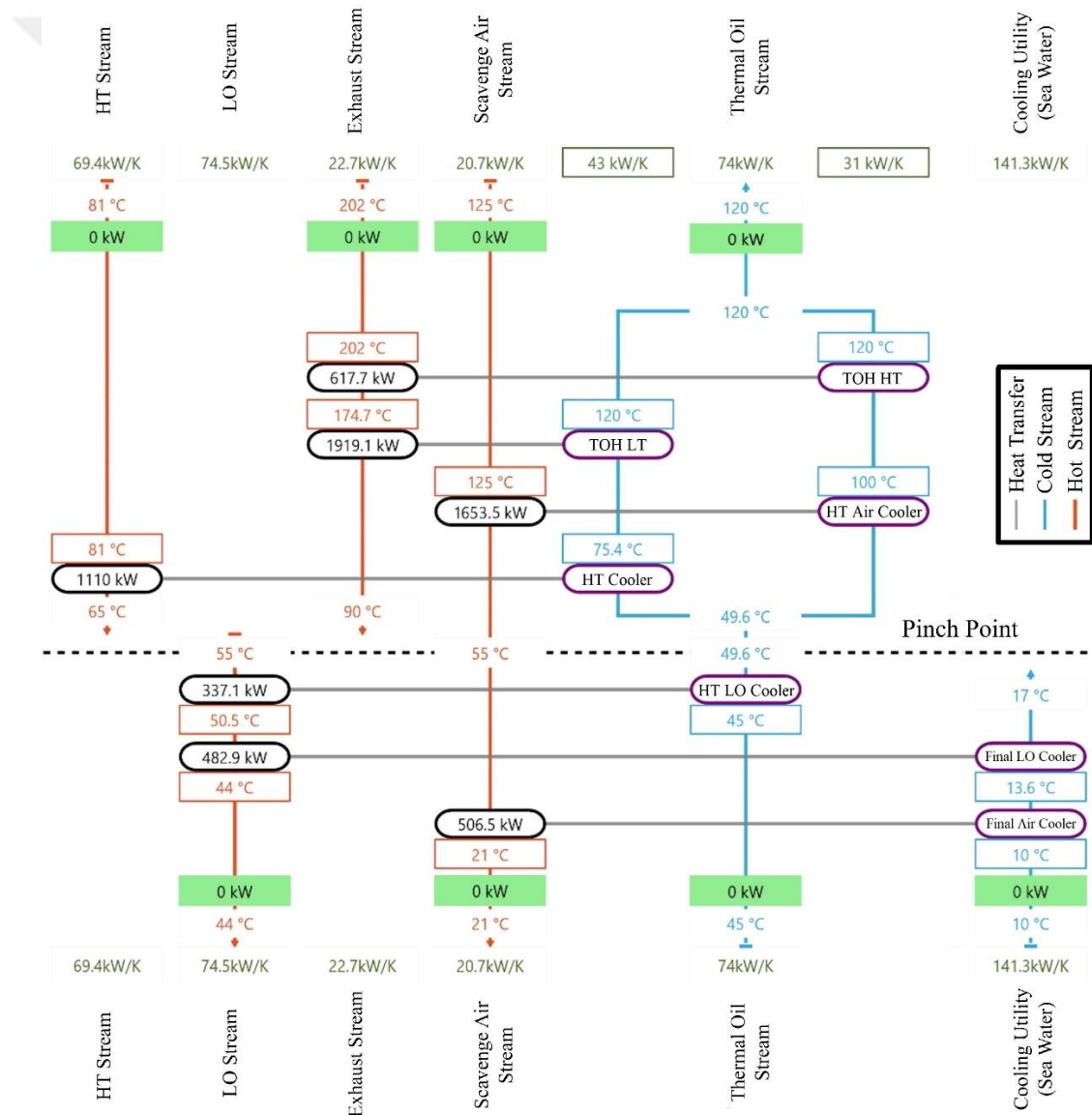


Figure 4.3 Optimal HEN design for reference crude oil tanker ship for 60% MCR engine load

Pinch point analysis is favorably utilized in industrial sectors that have process streams with steady heat flows. The proposed HEN design in Figure 4.3 is only optimal for the given operating parameters for 60% MCR engine load. Considering this information, pinch point analyses should be applied for all other engine loads to conclude that if the proposed HEN design is optimal for all varying engine loads too. The authors applied the pinch point analysis for the remaining engine loads of 30, 40, 50, 70, 80, 90, and 100% MCR. The results show that the optimal HEN design of 60% MCR is also optimal for the HENs established with 30, 40 and 50% MCR engine load input parameters. The optimal HEN designs of 70, 80, 90 and 100% MCR engine loads have all different HEN designs than the optimal HEN design in Figure 4.3. Therefore, high-load operation of the vessel will result in an increased unutilized waste heat ratio with the proposed HEN design than the optimal HEN of 70, 80, 90 and 100% MCR.

4.3 Integration of Cargo Oil Pump Turbine Based Waste Heat to Optimal Heat Exchanger Network Design for Discharge Operation Mode

After finding the optimal HEN design for the navigation mode of the reference tanker ship, the integration of the COPT based waste heat is carried out to be utilized for power generation through the ORC system. As the main engine based waste heat sources are no longer available during discharge operation, a thermal oil bypass line is implemented to the HEN design that directs the thermal oil line to the vacuum condenser of the COPT system. The direction of the thermal oil is controlled by a three-way changeover valve, which is placed after the thermal oil pump. The valve position should be changed to the vacuum condenser side for both cargo oil discharge operation with COPTs and WHR as the cooling media of the vacuum condenser is replaced with thermal oil. In addition to the inlet of the vacuum condenser, the outlet is also directed to the thermal oil line with a changeover valve. The valve is positioned just before the ORC evaporator to be able to bypass the main engine based waste heat sources completely. In any case that the ORC system shows malfunctions, a seawater-cooled thermal oil cooler is also recommended for emergency situations as a backup. The schematic of the proposed optimal HEN design for the ORC integrated WHRS of the reference crude oil tanker ship is illustrated in Figure 4.4.

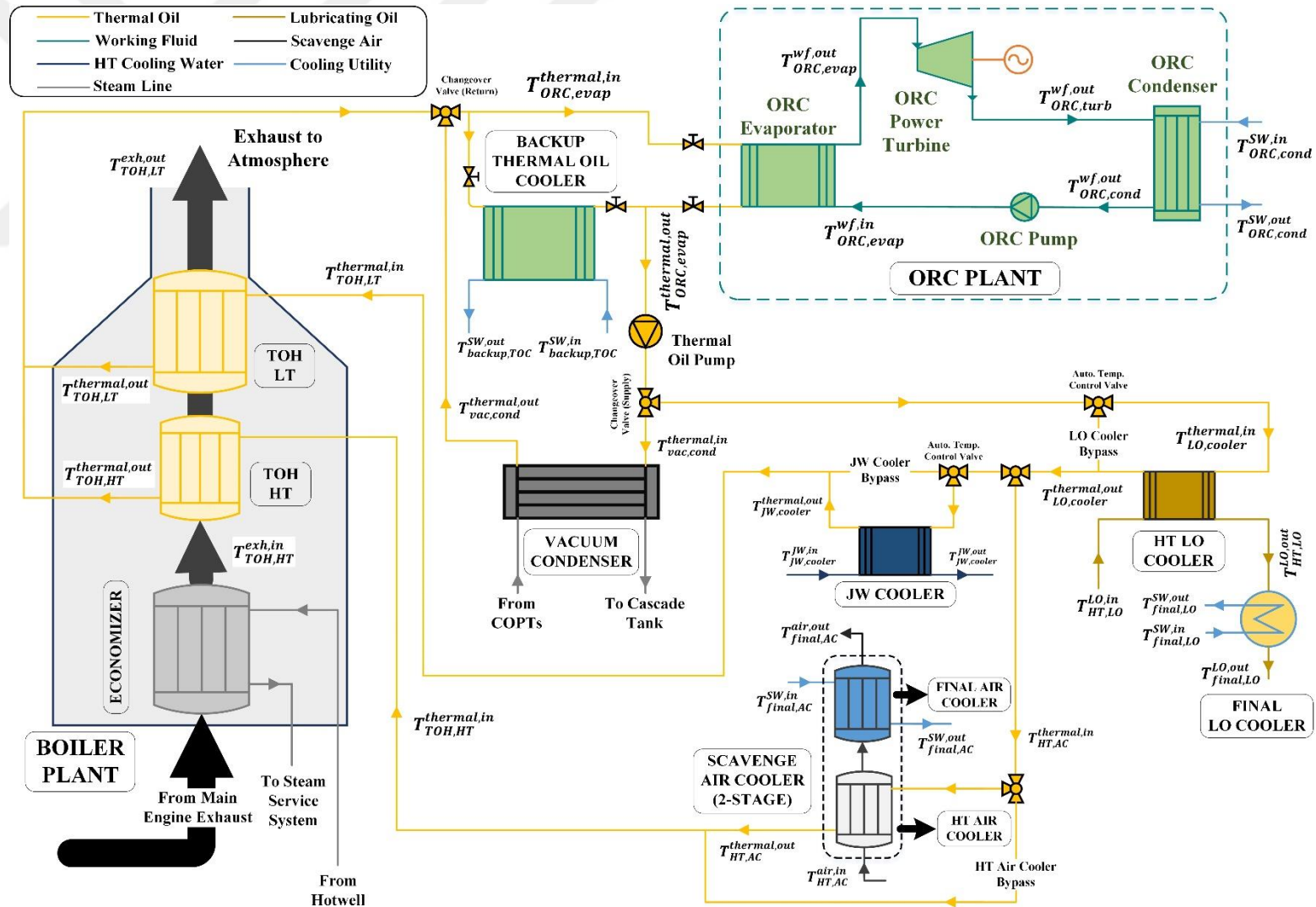


Figure 4.4 Layout of the proposed optimal HEN design for the ORC integrated WHRS of the reference crude oil tanker ship

4.4 Energy and Exergy Analyses of the Proposed ORC Integrated Waste Heat Recovery System Design

In this section, the steady-state energy and exergy balance equations for the input parameters of waste heat sources at 30, 40, 50, 60, 70, 80, 90 and 100% MCR engine loads are described. Design considerations for the ORC system are also given to analyze the net power outputs that can be produced by the ORC system for the given engine loads. The data sets for the main engine based waste heat sources were provided from MAN CEAS Wizard (2021) for the engine loads of 30, 40, 50, 60, 70, 80, 90, and 100% MCR as given in Table 4.1. For discharge operation mode, the required data to assess the WHR potential from COPTs were taken during a real-time cargo discharge operation of the reference vessel in Port of Trieste on December 17, 2019, as given in Table 4.2. Because the system is not equipped with pressure gauges or sensors on the steam side of the vacuum condenser, the energy analyses are carried out by using the available temperature, pressure and mass flow rate data taken from the cooling seawater side of the vacuum condenser, which also infer to the waste heat discharged to overboard in the existing system design. The volumetric flow rate of the cooling sea water is the same for all COPT speeds as the vacuum condenser cooling sea water pump is a constant speed type. Cargo discharge operations might be carried out at reduced mass flow rates than the available pumping capacity according to the communication among the port authority, the ship and the cargo owner. Thus, the data is taken for 735, 1045 and 1200 RPM running conditions while all three COPTs were in operation to acquire comparable COPT based WHR potential of the reference tanker vessel at different operating conditions.

Table 4.2 Real-time data taken from the vacuum condenser of the reference tanker vessel during cargo discharge operation with three COPTs running to assess the COPT based WHR potential

COPTs Average Speed (RPM)	Volumetric Flow Rate of the Cooling Sea Water (m ³ /h)	Vacuum Condenser Sea Water Inlet Temperature (°C)	Vacuum Condenser Sea Water Outlet Temperature (°C)	Calculated Amount of Discharged Waste Heat (kW)
1200	1240 (at 1 bar)	12	36	34,486
1045	1240 (at 1 bar)	12	32	28,757
735	1240 (at 1 bar)	12	28	23,021

The pinch point analysis provided the amount of utilized and unutilized waste heat that is discharged overboard on navigation mode through the optimal HEN design for 60% MCR engine load. The steady-state energy balance equations were used to investigate the heat distribution through the proposed HEN system design on navigation mode for varying engine loads of 30, 40, 50, 70, 80, 90 and 100% MCR. The mass flow rates of the thermal oil streams are calculated using the energy balances around the control volumes enclosing the thermal oil stream and the waste heat sources. The waste heat is collected with the thermal oil stream to be recovered as electricity by an ORC system. It has to be assured that the target outlet temperatures of the hot streams of the HEN design given in Table 4.1 are achieved because the streams maintain the operation of vital parts of the engine. As the thermal oil is utilized as the primary cooling media in the proposed HEN design, the thermal oil temperature at the evaporator outlet should be kept at minimum. Therefore, a simple ORC system without the use of reheaters or recuperators was selected taking into account the maximum cooling ability of the thermal oil. The collected waste heat by thermal oil is transferred to ORC working fluid through the ORC evaporator. ORC working fluid in the liquid phase vaporizes in the ORC evaporator and enters the ORC turbine in the gaseous phase at high pressure. The turbine uses the kinetic energy of the ORC working fluid to produce power and delivers it to the ORC condenser. The condenser cools down the ORC working fluid to its liquid phase by cooling with sea water. Then, the ORC pump sucks the working fluid in the liquid phase and pressurizes to feed it back to the evaporator.

The ORC system is integrated into the optimal HEN design by using the validated thermodynamic model given in Section 3.4 and Konur et al. (2020b). The model is utilized to evaluate the performance of the energy conversion system with the optimum HEN design for various engine loads. Equations used in the thermodynamic model for energy and exergy analyses are also given in Table 3.3. The assumptions for the overall thermodynamic system model are presented as follows:

- Changes in kinetic and potential energy and exergy are ignored.
- Both the ORC and the WHR systems run at steady-state conditions.

- The exit of the condenser is in the saturated liquid state.
- Pressure losses across coolers and pipes are neglected.
- All of the pumps placed in the system are considered variable speed pumps that can change the mass flow rates easily on varying engine loads.
- The thermal oil outlet temperature of each cooler is assumed constant on varying engine loads. As there is only one pump for the thermal oil line, mass flow rates of thermal oil inlets through the HT cooler and HT LO cooler are adjusted by automatic temperature control valves, changing a certain amount of thermal oil to the bypass lines if required.
- The vacuum condenser is capable of transferring the same amount of COPT based waste heat with the calculated amount of discharged waste heat in Table 4.2 considering the inlet and outlet temperatures of the thermal oil is higher than sea water temperature.
- The inlet temperature of the steam entering the vacuum condenser is in an acceptable range for heat exchange between fluids, which is higher than the target temperature of the thermal oil stream (120 °C).

The input parameters of the ORC system is given in Table 4.3. The ORC system operates in the subcritical region. The outlet temperature of the working fluid from the ORC evaporator is taken 7 °C above the saturation temperature of the working fluid at the ORC evaporator outlet pressure to ensure that the ORC working fluid enters the turbine at the gaseous phase. ORC condenser outlet pressure is taken as the saturation pressure of the working fluid at the target ORC condenser outlet temperature.

Table 4.3 Input parameters of the ORC system of the proposed WHRS for the reference tanker ship

Parameters	Value	Unit
The pressure of the working fluid entering the ORC evaporator	1200	kPa
The target temperature of ORC working fluid in the ORC condenser outlet	15 - 35	°C
The temperature of sea water entering the ORC condenser on navigation	10	°C
The temperature of sea water entering the ORC condenser at port	12	°C
The isentropic efficiency of the ORC pump	0.80	[-]
The isentropic efficiency of the ORC turbine	0.85	[-]
The heat loss from the heat exchangers	0.01	[-]

4.5 Energy and Exergy Analyses Results for Navigation and Discharge Operation Modes of the Proposed ORC Integrated Waste Heat Recovery System Design

In this section, energy and exergy analyses were performed to indicate the performance of the proposed ORC integrated WHRS design for the navigation and discharge operation modes. A parametric study was conducted for both navigation and discharge operation modes to assess the most suitable ORC working fluid among the given ten environment friendly alternatives in Table 4.4. It has been concluded in the case study on Chapter 3 that as the ORC evaporator pressure increases, the system efficiency and power output from the turbine also increases. In this section, the same validated steady-state thermodynamic model with the one used for the container ship case in Chapter 3 is utilized. The ORC evaporator pressure is selected as 1200 kPa to ensure that the target temperature of the working fluid in the ORC evaporator outlet is always lower than the thermal oil inlet temperature on the other side of the heat exchanger. The target temperature of the ORC working fluid in the ORC condenser outlet is studied in the range of 15 and 35 °C to assess the effect of the condenser temperature on the ORC system performance. The performance parameters were chosen as the thermal efficiency and the exergy efficiency of the ORC system. The results showing the effect of ORC condenser outlet temperature and the ORC working fluid type on the performance of the ORC system were given in Figure 4.5 and 4.6.

Table 4.4 Properties of the selected organic working fluids (Lindeman, 2019; Refrigerant Report, 2020; Linde Industrial Gases, 2021)

	R1233zd(E)	R1234ze(Z)	R245fa	R600	R236ea	Isobutane	R236fa	R152a	R134a	R1234yf
GWP ^a	4.5	<1	1030	4	1200	3	9400	124	1430	4
ODP ^b	0	0	0	0	0	0	0	0	0	0
ASHRAE 34 ^c	A2L	A2L	A1	A1	A1	A3	A1	A2	A1	A2L
Critical Temp. (°C)	165	150.1	154.0	152.0	139.3	134.7	124.9	113.3	101	94.7
Critical Press. (kPa)	3573	3533	3651	3796	3429	3640	3200	4520	4059	3382

^aGWP: global warming potential, relative to CO₂.

^bODP: ozone depletion potential, relative to R11.

^cASHRAE Standard 34 - Refrigerant safety group classification. 1: Non-flammable; 2L: Mildly flammable; 2: Lower flammability; 3: Higher flammability; A: Lower toxicity; B: Higher toxicity.

As the target temperatures of the ORC system remain the same, the performance parameters shown in Figure 4.5 and 4.6 are valid for both navigation and discharge operation modes. The results of the parametric study show that R1233zd(E) possess the highest thermal and exergy efficiencies for the given input parameters. It can be seen from the figures that increased refrigerant outlet temperature from the condenser negatively affects the ORC performance. The difference in efficiencies between 15 and 35 °C condenser temperatures reach more than 3% for all working fluids, thus it should be chosen as low as possible according to the cooling seawater temperature.

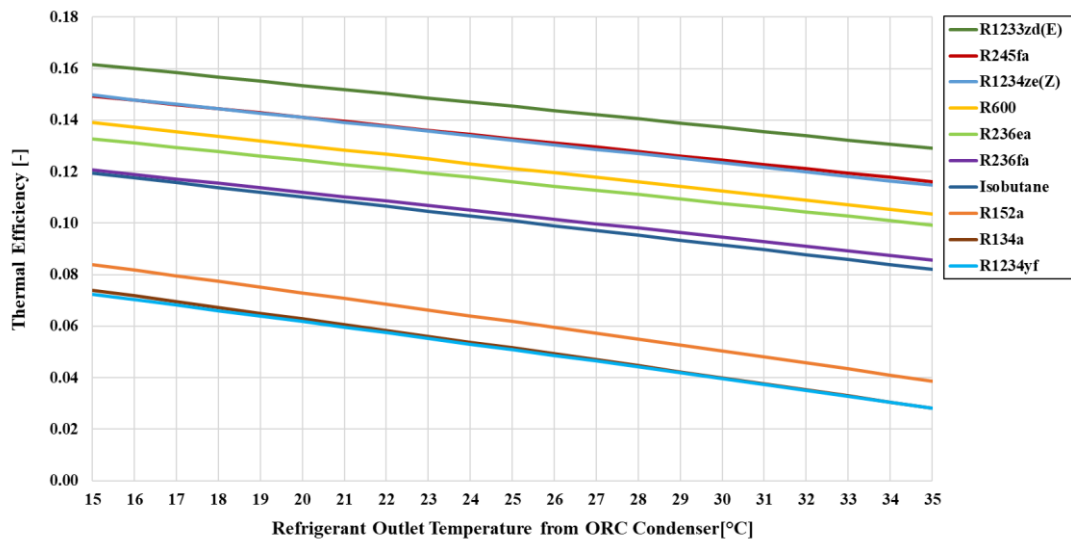


Figure 4.5 The effect of condenser temperature and ORC fluid type on thermal efficiency of the system

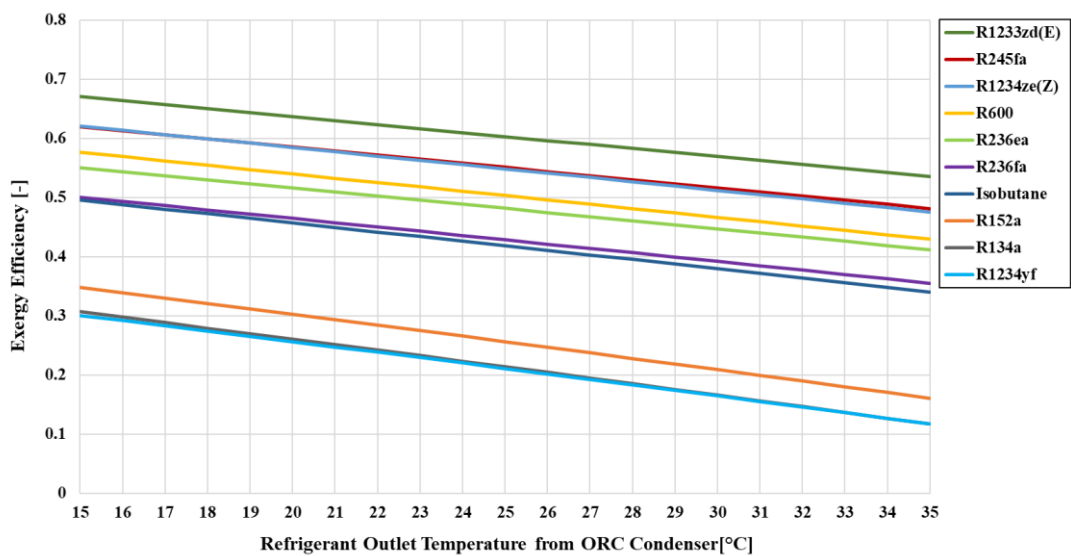


Figure 4.6 The effect of condenser temperature and ORC fluid type on exergy efficiency of the system

According to the performance analysis for the given target temperatures of the ORC system, R1233zd(E) working fluid was selected for the rest of the analyses in this study. It will also allow an environmentally friendly operation of the WHR system with its very low GWP and ODP values. However, still, leakages should be avoided and necessary safety precautions should be taken for operating the ORC system with R1233zd(E) as this working fluid possesses slight flammability. The ambient temperature was cold and the cooling seawater temperature was low enough. Thus, the ORC condenser outlet temperature is selected as 15 °C, where the condenser shows a maximum pinch point temperature difference of 5 °C on navigation mode and 3 °C on the discharge operation mode.

The performance of the ORC system was simulated using the steady-state thermodynamic model for the heat loads generated by 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads of the reference tanker ship. The mass flow rates and the target temperatures of the hot streams were entered into the model from the datasets derived from MAN CEAS Wizard (2021) as given in Table 4.1. The energy balance of the system at varying load conditions for the given mass flow rates and target temperatures of the hot streams were compensated by the thermal oil stream with the help of a variable speed thermal oil pump and by-pass lines located before coolers.

Table 4.5 shows the amount of utilized and unutilized waste heat through the proposed HEN design for eight different engine loads during the HEN design is in navigation mode. Net power outputs generated by the ORC system is also given in this table. The exhaust gas is the major waste heat source in the system at all loads. High-temperature exhaust gas is collected from TOH HT, but the majority of the exhaust gas waste heat is utilized in TOH LT to achieve optimal heat recovery from the system. The waste heat of the HT air cooler is the second major waste heat source for the engine loads of above 60% MCR. WHR potential of scavenge air significantly reduces at low loads due to low scavenge air temperatures and reduced mass flow rates. In addition, some of the waste heat of scavenge air needs to be discharged overboard because the target outlet temperature of the scavenge air is relatively low. The pinch point analysis shows all of the waste heat of the jacket cooling water can be utilized in

the system. The amount of collected waste heat from the JW cooler is in the range between 752.3 kW and 1,564 kW for 30-100% MCR engine loads. HT LO cooler collects the utilizable waste heat of main engine lubricating oil. 40.9% of main engine lubricating oil based waste heat is found utilizable by the ORC integrated WHRS as the remaining heat should be discharged overboard to achieve optimal WHR from the heat collection system. The ratio of unutilized waste heat to the total waste heat produced by the heat sources is found as 22.65, 19.32, 18.88, 18.09, 17.16, 16.21, 15.16, 13.95% for 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads, respectively. This is predominantly caused by the increase in operational temperatures in higher engine loads, which allows more heat recovery with the increased thermal oil mass flow rate. The net power outputs generated from the ORC system correspond to 10.48, 11.07, 10.66, 10.58, 10.67, 10.85, 11.14, and 11.70% of the shaft power generated by the main engine for 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads, respectively.

Table 4.5 Heat utilization amounts through the proposed HEN design to the ORC system and the net power output generated by the ORC system for 30-100% MCR engine loads (units are indicated in kW)

	30%	40%	50%	60%	70%	80%	90%	100%
HT LO Cooler	242.8	279.2	307.5	331.8	352.0	372.2	384.4	396.5
JW Cooler	752.3	871.1	990.0	1,099	1,218	1,337	1,455	1,564
HT Air Cooler	173.6	458.0	894.3	1,433	2,019	2,591	3,152	3,774
TOH LT	1,255	1,488	1,714	1,930	2,173	2,420	2,680	2,937
TOH HT	239.4	653.2	608.1	581.3	563.0	635.3	816.0	1,244
Unutilized Waste Heat through Utilities	772.0	889.5	1,040	1,175	1,297	1,409	1,503	1,591
Total Waste Heat Transferred to ORC	2,636	3,713	4,469	5,321	6,262	7,282	8,408	9,817
Net Power Output from ORC System	426.3	600.3	722.6	860.5	1,013	1,177	1,359	1,587

During the discharge operation mode, only the COPT based waste heat is collected by the thermal oil stream. The target temperature of the steam leaving the vacuum condenser is also higher than the thermal oil inlet temperature to the vacuum condenser, so all the waste heat can be collected from the vacuum condenser with a proper heat exchanger design. The amount of waste heat utilized from the vacuum condenser and the net power output generated by the ORC system during cargo discharge operation is calculated and given in Table 4.6 for 735, 1045, and 1200 RPM

COPT speed, while all three COPTs are in operation. The net power outputs from the ORC system shows that the reference tanker ship has a larger WHR potential during the cargo discharge operation than during navigation. The generated electricity is far larger than the ship's electricity demand. In this case, the excessive amount of generated electricity can be evaluated by storing the electrical energy to assist the propulsion with a shaft motor during navigation or can be sold out to the port facility if the sides settle with the terms. The shaft motor option might be more favorable in this case because the ship will also generate an excessive amount of electricity from the ORC system during navigation, which can be directly used in the shaft motor to assist the propulsion and reduce the fuel consumption of the vessel. The hydrogen or ammonia production to be used as fuel to the engine can be considered as alternatives to the storage of electricity to batteries.

Table 4.6 The amount of utilized COPT based waste heat and net power output generated by the ORC system during cargo discharge operation for varying COPT speeds (units are indicated in kW)

COPT Speed	735 RPM	1045 RPM	1200 RPM
Waste Heat Collected from the COPT System	22,791	28,469	34,141
Net Power Output from ORC	3,685	4,604	5,521

Exergy destruction rates of the proposed ORC integrated WHRS components are given in Figure 4.7 for 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads of the reference tanker vessel. The exergy destruction rate of the HT LO cooler is the highest in the system except for 100% MCR engine load. The ORC system has the second-highest exergy destruction rate, which increases at a higher rate than HT LO cooler for increasing engine loads. The exergy destruction rates for 60% MCR engine load (design load for the proposed HEN) is found as 320.20 kW (33.2% of the total system) for HT LO cooler, 230.30 kW (23.8% of the total system) for the ORC system, 152.90 kW (15.8% of the total system) for JW cooler, 71.18 kW (7.4% of the total system) for TOH LT, and 55.58 kW (5.8% of the total system) for final air cooler. The remaining components comprise an exergy destruction rate of 14% of the total system as can be seen from Figure 4.8. HT air cooler shows low exergy destruction rates at low loads but higher values are obtained for over 70% MCR due to increased scavenge air inlet temperatures, which is caused by the increased speed of the turbocharger.

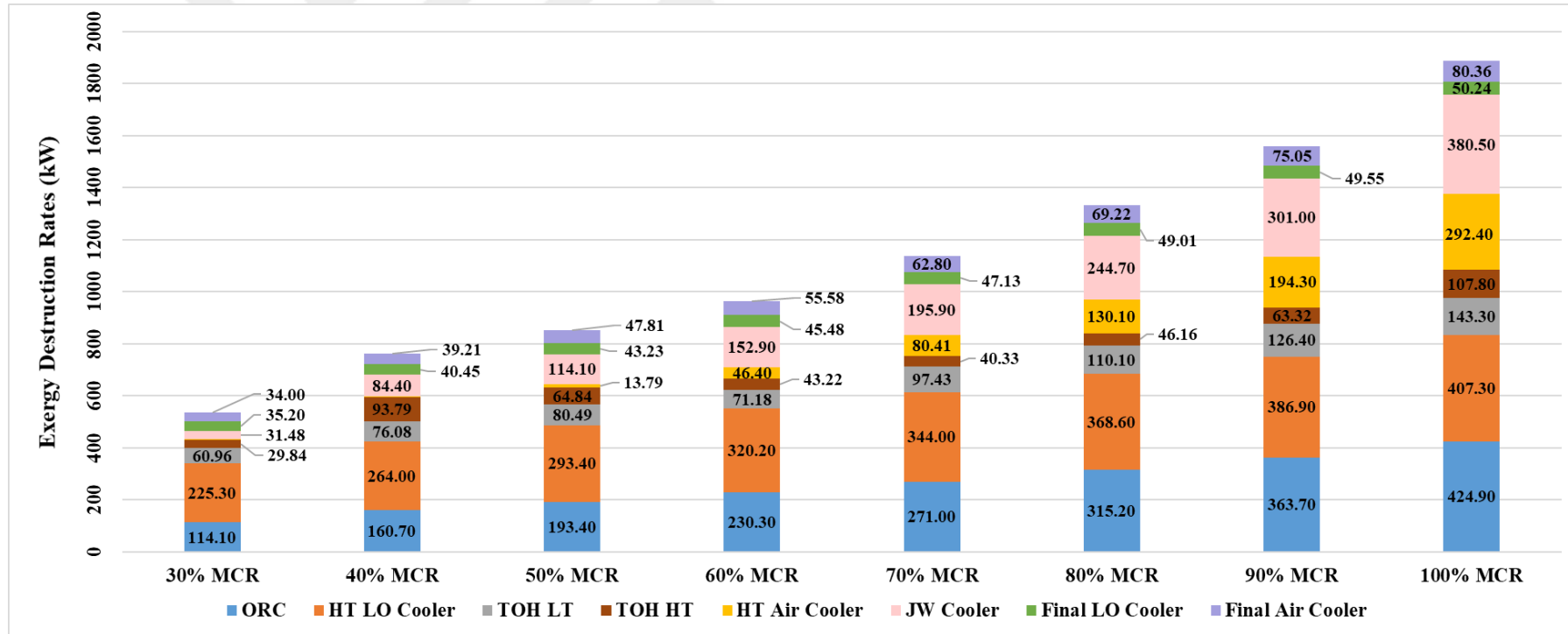


Figure 4.7 Distribution of exergy destruction rates of the proposed ORC integrated WHRS components on varying engine loads of the reference tanker ship

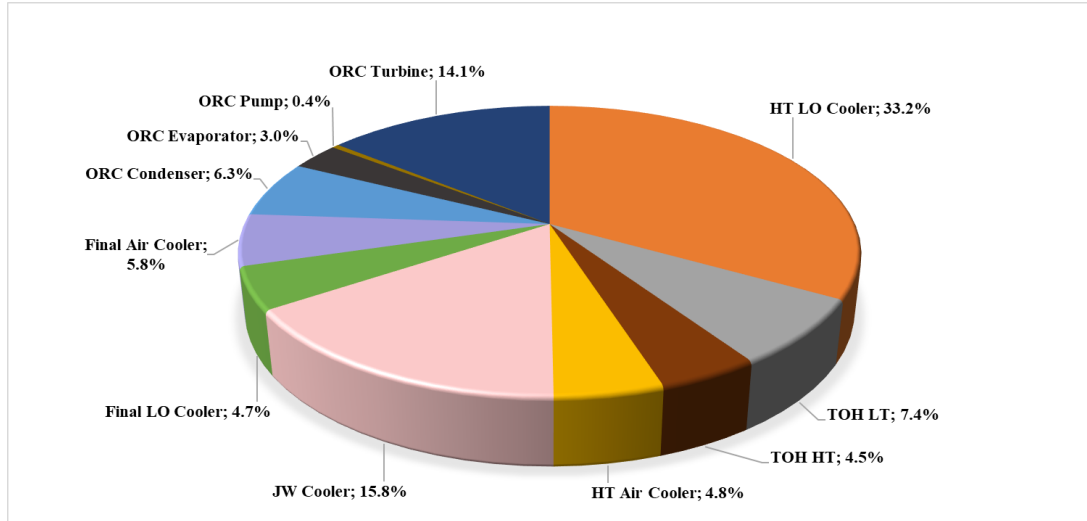


Figure 4.8 Exergy destruction ratios of WHRS components during navigation mode at 60% MCR

Exergy destruction rates of the proposed ORC integrated WHRS components on cargo discharge operation mode of the reference tanker ship are illustrated in Figure 4.9 for all three COPTs are running at 735, 1045 and 1200 RPM turbine speeds. The exergy destruction rate of vacuum condenser is higher than ORC evaporator's because the ORC working fluid temperature at the condenser outlet is selected as close as possible to the seawater temperature (15 °C), which in turn reduces the unutilized exergy at the ORC evaporator and ORC condenser.

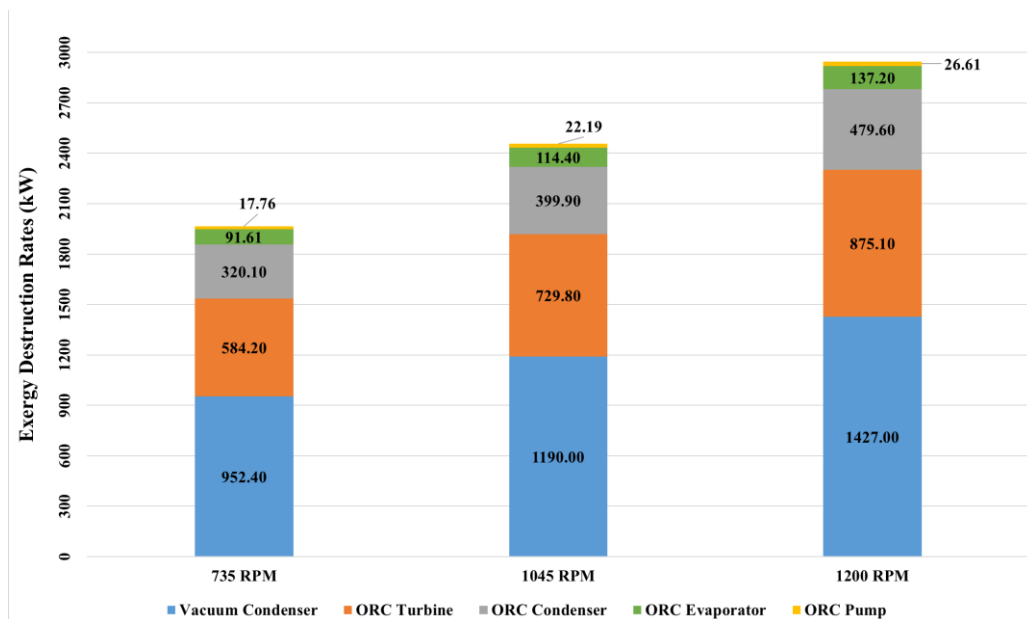


Figure 4.9 Distribution of exergy destruction rates of the proposed ORC integrated WHRS components on cargo discharge operation mode of the reference tanker ship for three different COPT speeds

Exergy destruction ratios of the proposed WHRS components are given in the pie chart in Figure 4.10. It is shown that 48.4% of total system exergy is destroyed in the vacuum condenser. The second highest exergy destruction takes place on the ORC turbine with 29.7% of the total system. The remaining ORC system components comprise 21.9% of the total exergy destruction of the system, which are the least important source of exergy destruction for the cargo discharge operation mode of the proposed ORC integrated WHRS system.

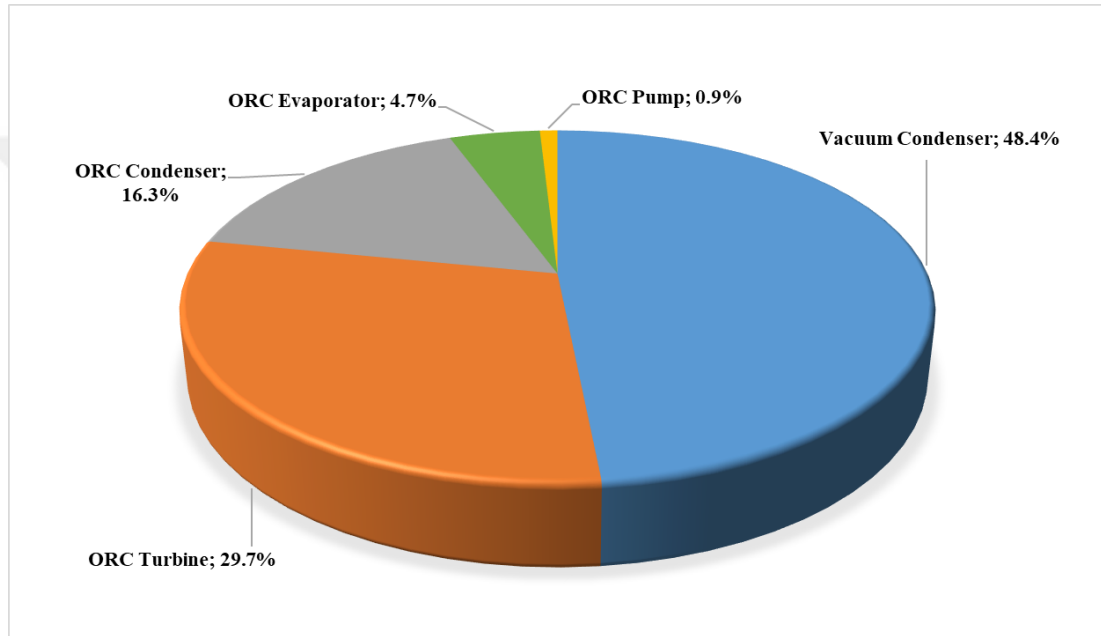


Figure 4.10 Exergy destruction ratios of WHRS components during cargo discharge operation mode

4.6 Exergy Based Sustainability Assessment of the Proposed ORC Integrated Waste Heat Recovery System for the Reference Tanker Vessel

In this section, exergy based sustainability assessments of the proposed ORC integrated WHRS for the reference tanker ship for both navigation and cargo discharge mode operations are carried out by evaluating the exergy-based sustainability indicators in Equations 3.9-3.14 as given in Section 3.5.2.1. The exergy efficiency of the proposed ORC integrated WHRS design is calculated and the results for the varying ORC condenser inlet temperatures and different types of ORC working fluids have been given in Figure 4.6. As the sustainability indicators of WER, EEF, EDC, ESI, and SI are a function of exergy efficiency, the obtained results in Figure 4.11 –

4.13 are given for both navigation and discharge operation modes. However, the IP indicator is investigated separately for navigation and discharge operation modes due to the different exergy inlet and outlet values on varying operating conditions. It is aimed to provide comparable quantified data with the assessment results that show sustainability improvement potentials if the proposed ORC integrated WHRS design is applied. It is expected that the quantified sustainability assessment results will help decision-making processes of policy makers and ship owners to apply ORC systems to marine vessels for more energy-efficient and environmentally friendly operations.

The WER is calculated as in Equation 3.9 by dividing the total waste exergy by the total exergy inlet of the ORC system. The effect of ORC condenser temperature on WER for ten different ORC working fluids is investigated and the obtained WER values are presented in Figure 4.11. The maximum value of WER is obtained as 88.2% for R1234yf at 35 °C condenser temperature, which shows the worst performance following the results obtained from the energy and exergy analyses. The WER is minimum for R1233zd(E) at 15 °C with the value of 32.6%, which performs much better than the closest competitors of R245fa and R1234ze(Z).

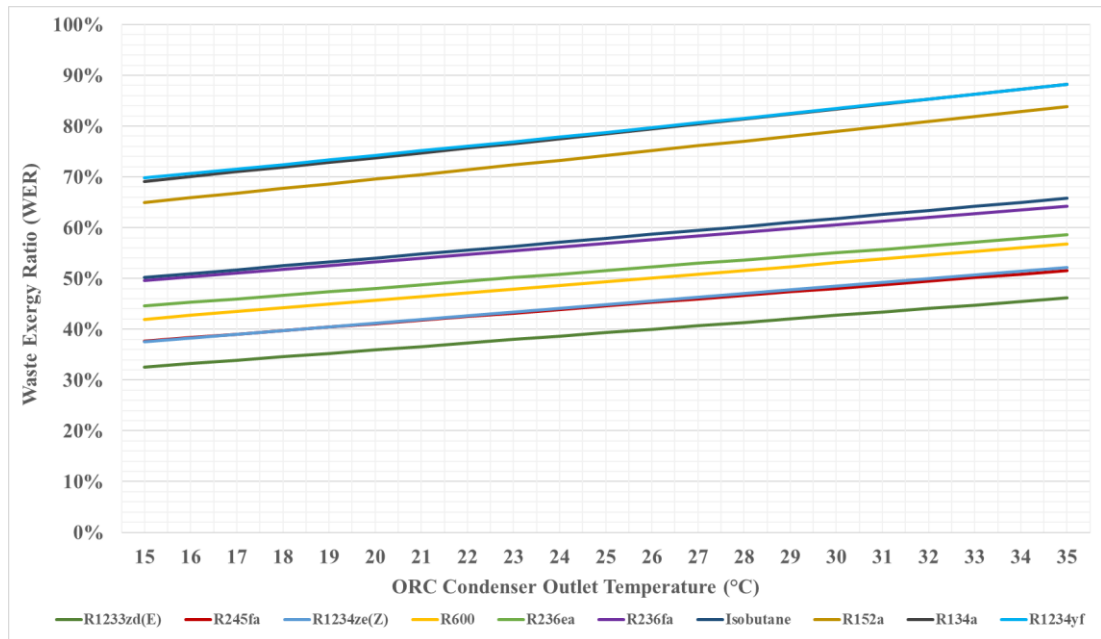


Figure 4.11 WER values obtained for different ORC condenser temperatures and working fluids

EEF and EDC sustainability indicators are normalized values that are calculated for indicating how much the energy conversion system gives harm to the environment. Both these indicators are inversely proportional to the exergy efficiency and show a similar pattern in the sustainability analyses as previously mentioned in Section 3.5.3. The EEF and EDC values are calculated using Equations 3.10 and 3.13 and the results are given in Figure 4.12 (a) and (b), respectively. The EEF and EDC values are less for R1233zd(E) working fluid at all condenser temperatures as it shows better exergetic performance for the proposed ORC integrated WHRS design. The EEF values for R1233zd(E), R245fa, and R1234ze(Z) at 15 °C are calculated as 0.483, 0.604 and 0.601, respectively. It is shown that the environmental effects of the WHRS can be minimized by the reduced condenser outlet temperature, which is directly affected by the cooling seawater temperature and the pinch point temperature difference of the heat exchangers.

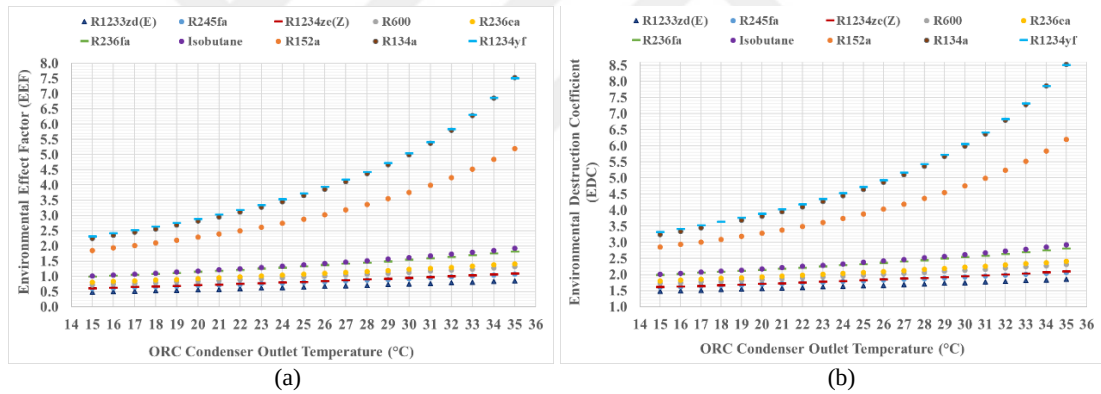


Figure 4.12 (a) EEF (b) EDC values calculated for different ORC condenser outlet temperatures and working fluids

ESI and SI give normalized values that indicate the degree of sustainability of an energy system. SI values are the inverse of EDC, and the ESI and SI are directly affected by the exergy efficiency of the energy system as can be depicted from Equation 3.11 and 3.12. The calculated ESI and SI values for the proposed ORC integrated WHRS of the reference tanker vessel are illustrated in Figure 4.13 (a) and (b). The best ESI and SI values are obtained as 2.07 and 3.07 for R1233zd(E) working fluid at 15 °C condenser outlet temperature. The sustainability performance can be reduced to ESI and SI values of 1.13 and 0.13 if R1234yf working fluid is operated at

the condenser temperature of 35 °C in the ORC system. R245fa and R1234ze(Z) have very similar ESI and SI values for varying ORC condenser outlet temperatures, but R245fa shows slightly higher sustainability performance at higher condenser temperatures that should be considered for operating the ORC system at routes having high seawater temperatures.

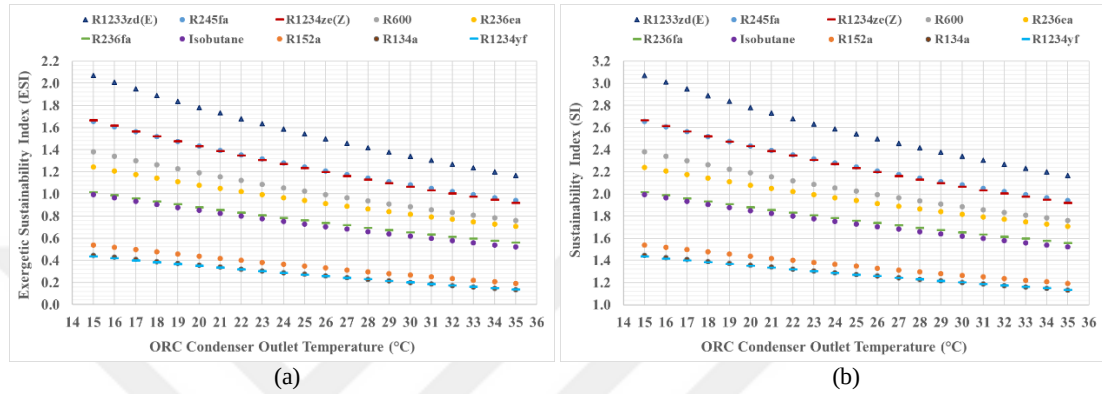


Figure 4.13 (a) ESI (b) SI values calculated for different ORC condenser outlet temperatures and working fluids

IP is a sustainability indicator that shows how much more work could have been generated by the given exergy input to the energy conversion system and can be calculated by Equation 3.14. IP values are affected from the exergy efficiency and the exergy inlet to the energy conversion system. As the engine load or COPT speed increases, the exergy input to the ORC system also increases and leaves more improvement potentials for each working fluid types. The calculated IP values for different working fluid types varying engine loads of the reference tanker ship at the selected optimal condenser outlet temperature of 15 °C are given in Figure 4.14. The higher exergy efficiency values lead to lower IPs as the exergy input to the ORC system is constant at each engine load category for varying working fluid types because the heat flow to the ORC evaporator is limited by the target temperatures of the thermal oil side. IP values for ten different working fluids and varying COPT speeds during cargo discharge operation mode of the reference tanker ship are given in Figure 4.15. The IP values of the ORC integrated WHRS for cargo discharge operation mode are much higher than the navigation mode of the system. This is caused by the much higher thermal oil mass flow rate and the exergy input to the ORC

evaporator for the same exergy efficiency values at each working fluid type when compared with the navigation mode of the vessel.

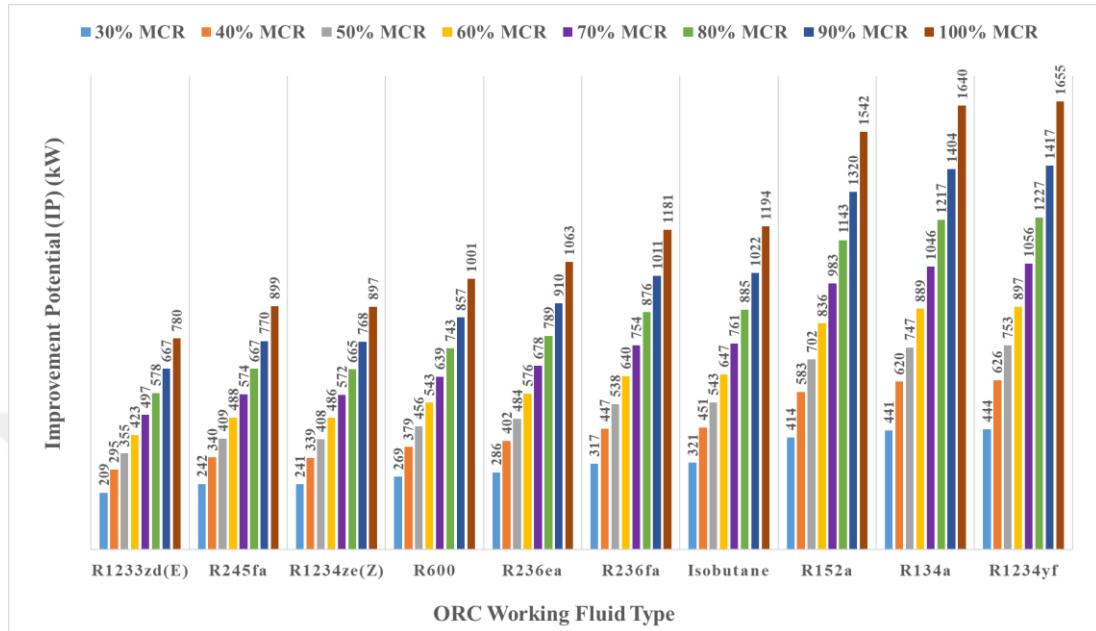


Figure 4.14 IP for different engine loads and working fluids at ORC condenser temperature of 15 °C on navigation mode of the reference tanker ship

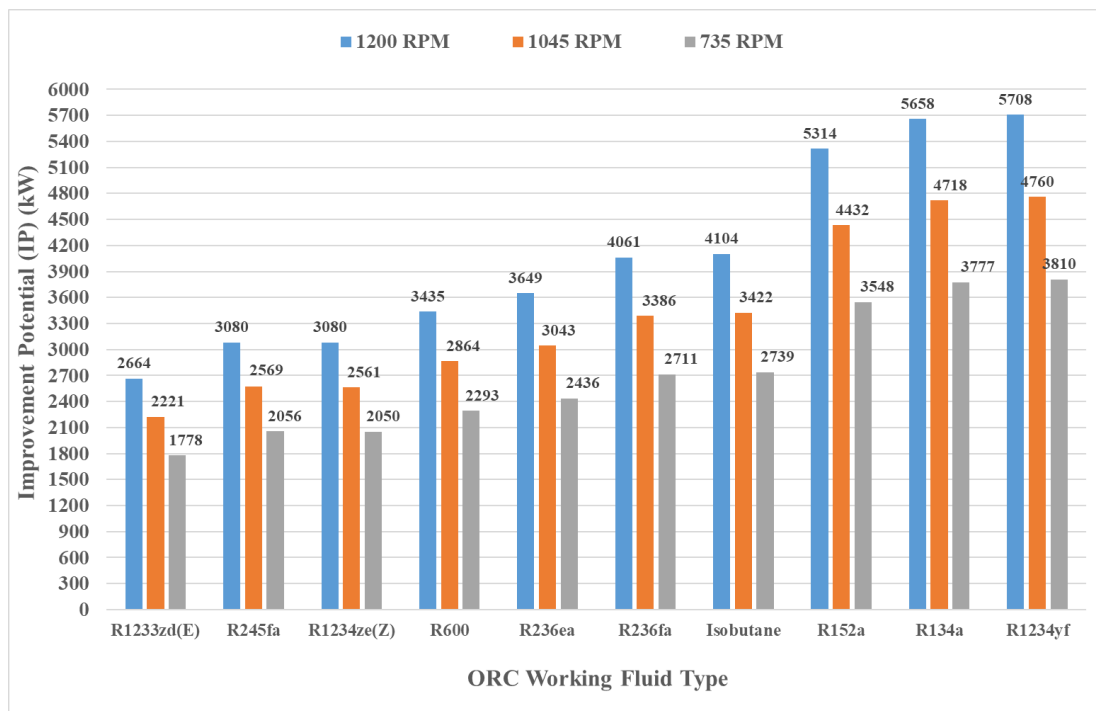


Figure 4.15 IP for different COPT speeds and working fluids at ORC condenser temperature of 15 °C on cargo discharge operation mode of the reference tanker ship

4.7 Discussion

The summary of the major results derived from this chapter are as follows:

- The optimal HEN design of 60% MCR is also optimal for the HENs established for 30, 40 and 50% MCR engine loads. The optimal HEN designs of 70, 80, 90 and 100% MCR engine loads have all different HEN designs than the proposed HEN design. Therefore, the high-load operation of the reference tanker vessel will result in an increased unutilized waste heat ratio with the proposed optimal HEN design than the optimal HEN designs of 70, 80, 90 and 100% MCR engine loads.
- The optimum HEN design is obtained with a simpler scavenge air cooler design than the container ship case, which is divided into two-stage instead of three-stage heat exchanger series considering the pinch analysis rules for the given input parameters of the reference tanker ship at 60% MCR engine load.
- The scavenge air waste heat below the pinch point cannot be collected in the system, so the unutilized waste heat is discharged overboard from the Final Air Cooler. A majority of the heat of the LO stream has to be discharged overboard as unutilized waste heat from the Final LO Cooler to achieve the target temperatures of the system.
- All the waste heat cannot be collected by thermal oil during the navigation mode of the reference crude oil tanker ship, as some of the waste heat needs to be discharged through the sea by cooling utilities. The unavailable waste heat was found as 22.7, 19.3, 18.9, 18.1, 17.2, 16.2, 15.2, and 13.9% of the total waste heat produced by the heat sources for 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads, respectively. As the engine load increases, the heat load above the pinch point also increases. The unutilized waste heat ratio decreases with the increased engine load because the cold utilities are only required below the pinch point temperature range.

- R1233zd(E) yields the best performance among the ten environmentally-friendly organic working fluid alternatives for the selected operating ORC system parameters and the target temperatures of the proposed HEN design of the reference tanker ship for both navigation and cargo discharge operation modes. R1233zd(E) working fluid shows great prospect with very low GWP and ODP values but slightly mildly flammable ASHRAE34 level may require extra precautions for a safe operation.
- The reduced ORC condenser outlet temperature increases the system efficiency of the ORC. The best energy conversion efficiency is achieved at the ORC condenser outlet temperature of 15 °C for R1233zd(E) working fluid with the value of 16.2%. The system efficiency reduces up to 12.9% at the condenser outlet temperature of 35 °C for 1200 kPa evaporator pressure, which is a measure to take into consideration for the ship routes and climate conditions where the cooling seawater temperature is above 12 °C.
- The net power outputs generated by the proposed ORC integrated WHRS is calculated as 426.3, 600.3, 722.6, 860.5, 1013, 1177, 1359, and 1587 kW at the navigation mode for 30, 40, 50, 60, 70, 80, 90, and 100% MCR engine loads. The net power outputs are calculated as 3685, 4604, and 5521 kW for 735, 1045, and 1200 RPM at cargo discharge operation mode. The values show that all of the electricity demand of the ship can be provided by the WHRS for the engine load of above 50% MCR, considering one diesel generator is utilized on around 50-60% engine load during navigation. The net power outputs from the ORC system shows that the reference tanker ship has a larger WHR potential during the cargo discharge operation than during navigation. The generated electricity during the discharge operation mode is far larger than the ship's electricity demand. In this case, the excessive amount of generated electricity can be evaluated by storing the electrical energy to assist the propulsion with a shaft motor during navigation or can be sold out to the port facility if the sides settle with the terms. The shaft motor option might be more favorable in this case because the ship will also generate an excessive amount of electricity from the

ORC system during navigation, which can be directly used in the shaft motor to assist the propulsion and reduce the fuel consumption of the vessel. The hydrogen or ammonia production to be used as fuel to the engine can be considered as alternatives to the storage of electricity to batteries.

- The exergy destruction rate of the HT LO cooler is highest in the system at the navigation mode except for 100% MCR engine load. The ORC system has the second-highest exergy destruction rate, which increases at a higher rate than HT LO cooler for increasing engine loads. The exergy destruction rates for 60% MCR engine load (design load for the proposed HEN) is found as 320.20 kW (33.2% of the total system) for HT LO cooler, 230.30 kW (23.8% of the total system) for the ORC system, 152.90 kW (15.8% of the total system) for JW cooler, 71.18 kW (7.4% of the total system) for TOH LT, and 55.58 kW (5.8% of the total system) for final air cooler. The remaining components comprise an exergy destruction rate of 14% of the total system. HT air cooler shows low exergy destruction rates at low loads but higher values are obtained for over 70% MCR due to the increased scavenge air inlet temperatures, which is caused by the increased speed of the turbocharger.
- The exergy destruction rate of vacuum condenser is higher than ORC evaporator's at the cargo discharge operation mode because the ORC working fluid temperature at the condenser outlet is selected as close as possible to the seawater temperature (15 °C), which in turn reduces the unutilized exergy at the ORC evaporator and ORC condenser. 48.4% of total system exergy is destroyed in the vacuum condenser. The second highest exergy destruction takes place on the ORC turbine with 29.7% of the total system. The remaining ORC system components comprise 21.9% of the total exergy destruction of the system, which are the least important source of exergy destruction for the cargo discharge operation mode of the proposed ORC integrated WHRS system.
- The sustainability of the proposed ORC integrated WHRS design for the reference crude oil tanker ship is assessed using the sustainability indicators of

WER, EEF, EDC, ESI, SI, and IP. The minimum WER value is obtained from R1233zd(E) at the refrigerant condenser outlet temperature of 15 °C by 32.6%, which performs much better than its closest competitors of R245fa and R1234ze(Z). EEF value for the proposed system design at the selected operating parameters with R1233zd(E) working fluid is calculated as 0.48. The maximum ESI of 2.07 is obtained by R1233zd(E). R245fa and R1234ze(Z) working fluids follow R1233zd(E) with the ESI value of 1.66. The IP values of the ORC integrated WHRS for cargo discharge operation mode are much higher than the navigation mode of the system. This is caused by the much higher thermal oil mass flow rate and the exergy input to the ORC evaporator for the same exergy efficiency values at each working fluid type when compared with the navigation mode of the vessel.

CHAPTER FIVE

CONCLUSIONS

In this study, the potential waste heat recovery options for the marine vessel were identified. ORC system components were compiled for various ORC applications. A literature review was carried out among the researches for various ship-based waste heat sources and integration of ORC systems to emission reduction technologies. The prototype development and demonstration projects for using ORC systems in ships have been comprehensively reviewed. In addition, the challenges for the application of ORC systems on marine vessels have been discussed.

As one of the objectives of this study, the optimal HEN design for an ORC integrated WHRS of a reference 1221 TEU container ship was proposed by utilizing the pinch point analysis to improve the overall energy efficiency of the vessel. Lubricating oil, jacket cooling water, scavenge air, and the exhaust gas of the main engine were identified as the major waste heat sources of a reference ship. The waste heat recovery potential and the potential electricity generation from a simple ORC system were calculated according to the target parameters, which were determined according to the operating conditions of the main engine and the temperature of the cooling seawater. In addition, the exergy analysis and the sustainability of the proposed ORC system are investigated by using exergetic sustainability indicators. The analyses were also carried out to select the best performance working fluid among different environmentally friendly alternatives and a proper evaporating pressure by using parametric analyses.

As another objective of the study, a new waste heat source is defined as a utilizable source to the literature, which is the COPT based waste heat. The COPT based waste heat can be utilized from tanker vessels (if implemented) during discharge operations from ship to port facility, which would be expected to increase the uptake time of the ORC system due to the waste heat availability and reduce the payback time of the investment. In this regard, a 116,000 DWT tanker ship was taken as a reference vessel to calculate the waste heat recovery potential from the COPT based waste heat during

the discharge operation of the vessel, and from the main engine based waste heat during navigation mode of the ORC system. A new heat exchanger network design for optimal waste heat recovery was proposed for the navigation mode of the ORC integrated WHRS. The energy and exergy analyses were applied for both navigation and discharge operation modes. The parametric studies were carried out for varying refrigerant outlet temperatures at the condenser and ten different environmentally-friendly organic working fluids. In addition, the sustainability assessment is carried out using the exergetic sustainability indicators to indicate the sustainability level of the system with normalized indicators.

It has been concluded that the ships possess great WHR potential with ORC systems, which is mostly larger than the power requirement of the auxiliary systems. It was found that up to 12.93% of the net power output produced by the main engine could be recovered with the optimal HEN design for the ORC system. The utilization of COPT based waste heat on tanker ships will make this potential even larger. The utilization strategy of all those excessive power generated by COPT based waste heat may be stored as chemical energy in battery packs to assist the propulsion with a shaft motor during navigation, or selling out to the port facility if the sides settle with the terms. The shaft motor option might be more favorable in this case because the ship will also generate an excessive amount of electricity from the ORC system during navigation, which can be directly used in the shaft motor to assist the propulsion and reduce the fuel consumption of the vessel. The hydrogen or ammonia production with the excess electrical energy to be used as fuel to the engine can also be considered as alternatives to the storage of electricity to batteries.

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APPENDICES

APPENDIX 1: NOMENCLATURE

$\dot{E}_d$	exergy destruction rate (kW)
e_f	specific flow exergy (kJ/kg)
g	gravitational acceleration (m/s ²)
h	enthalpy (kJ/kg)
$\dot{H}$	heat flow (kW)
$\Delta\dot{H}$	net heat flow (kW)
$\dot{m}$	mass flow rate (kg/s)
P	Pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
T	temperature (°C)
T^*	shifted temperature (°C)
$\Delta T_{\min}$	minimum temperature difference between hot and cold streams
s	entropy
W	power
z	elevation (m)
η	efficiency (-)

APPENDIX 2: SUBSCRIPTS

0	dead state, environmental
cond	condenser
cv	control volume
e	electrical
evap	evaporator
f	flow
in	inlet
out	outlet
th	thermal
vac	vacuum
wf	working fluid

APPENDIX 3: ABBREVIATIONS

ARC	absorption refrigeration cycle
BWR	back-work ratio
CC	composite curve
CCHP	combined cooling, heating and power
CHP	combined heat and power
COPT	cargo oil pump turbine
CSP	concentrated solar power
DN	depletion number
DPSIR	drivers-pressures-states-impacts-responses
DWT	deadweight tonnage
EBI	Environmental benign index
EDC	environmental destruction coefficient
EDI	environmental destruction index
EEDI	energy efficiency design index
EEF	environmental effect factor
EEOI	energy efficiency operational index
EES	Engineering Equation Solver
EGR	exhaust gas reduction
ESI	exergetic sustainability index
GCC	grand composite curve
GHG	greenhouse gas
GT	gross tonnage
GWP	global warming potential
HEN	heat exchanger network
HP SCR	high pressure selective catalytic reduction
HT	high temperature
HVAC	heating, ventilating and air conditioning
IMO	international maritime organization
IP	improvement potential
JW	jacket water

LNG	liquid natural gas
LO	lubricating oil
LP SCR	low pressure selective catalytic reduction
LT	low temperature
MCR	maximum continuous rating
M/E	main engine
NECA	NO _x emission control areas
NPSH	net positive suction head
ODP	ozone depletion potential
ORC	organic Rankine cycle
PTSC	parabolic trough solar collector
PV	photovoltaic
RC	Rankine cycle
SCI	Sustainable cost index
SEEMP	ship energy efficiency management plan
SI	sustainability index
SRC	steam Rankine cycle
SW	sea water
TCE _{xL}	total cumulative exergy loss
TEG	thermoelectric generator
TEU	twenty foot equivalent unit
TOH	thermal oil heater
VCC	vapor compression cycle
WER	waste exergy ratio
WHR	waste heat recovery
WHRS	waste heat recovery system
XUI	exergy utilization index