CHALMERS
UNIVERSITY OF TECHNOLOGY

LOW TEMPERATURE ORGANIC RANKINE CYCLE FOR A EUROPEAN RO-PAX VESSEL

Applying a method for working fluid selection in the context of European and maritime constraints while maximizing cost performance

Diploma Thesis in Marine Engineering

ROBIN ÖDLUND, CHRISTIAN DER NEDERLANDEN

Department of Marine Engineering
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2015

DIPLOMA THESIS SI-15/165

LOW TEMPERATURE ORGANIC RANKINE CYCLE FOR A EUROPEAN RO-PAX VESSEL

Applying a method for working fluid selection in the context of European and maritime constraints while maximizing cost performance

DER NEDERLANDEN, ÖDLUND

Department of Shipping and Marine Technology
Marine Engineering Program
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2015

LOW TEMPERATURE ORGANIC RANKINE CYCLE FOR A EUROPEAN RO-PAX VESSEL
Applying a method for working fluid selection in the context of European and maritime constraints while maximizing cost performance

LÅGTEMPERATUR ORGANISK RANKINE CYKEL FÖR EN EUROPEEISK RO-PAX FÄRJA
Tillämpning av en metod för urval av värmebärare i förhållande till de maritima- och europeiska avgränsningarna för optimal ekonomisk vinning

Supervisors: Ulrik Larsen and Cecilia Gabrielii
Examiner: Johan Eliasson

Diploma Thesis SI-15/165
Department of Shipping and Marine Technology
Marine Engineering Program

Chalmers University of Technology
SE-412 96 Gothenburg
Telephone +46 31 772 1000

Typeset in LaTeX
Printed by Chalmers
Gothenburg, Sweden 2015

LOW TEMPERATURE ORGANIC RANKINE CYCLE FOR A EUROPEAN RO-PAX VESSEL

Applying a method for working fluid selection in the context of European and maritime constraints while maximizing cost performance

DER NEDERLANDEN, ÖDLUND
Department of Shipping and Marine Technology
Chalmers University of Technology

Abstract

In this study a method for determining a suitable working fluid for a potential ORC installation recovering waste heat from the high temperature cooling water system on board the RoPax vessel Stena Scandinavica is applied. This study is based on an exhaustive investigation into previous research in this area of thermodynamics and engineering. The working fluids are evaluated for their potential output, performance, safety and their economic and environmental performance. These evaluation criterias are matched with this study's intended application on board a european merchant marine vessel. The results of this study found that one such installation is substantially more profitable both from an economical and environmental perspective. The study found that the newer fourth generation working fluids such as R1234ze(E) performed very well in the evaluation and are good candidates for a low temperature ORC plant at sea. One other conclusion drawn by this study is that working fluids with a high GWP value generally out perform the newer less environmentally impactful working fluids. The theory behind and application of this method is also in detail described in this study.

Keywords: Organic Rankine cycle, waste heat recovery, working fluid selection, energy efficiency

Sammanfattning

Den här fallstudien tillämpar en metod för att finna ett lämligt arbetsmedium för att ur en potentiell, lågt tempererad, spillvärmekälla återvinna användbar energi med hjälp av en ORC installation ombord RoPax fartyget Stena Scandinavica. Studien är baserad på insamlingar och analyser av tidigare arbeten och forskning relaterat till områderna kring liknande termodynamik och spillvärmeåtervinning. Lämpliga arbetsmedier utvärderas efter deras potentiella effekt, prestanda, säkerhet och deras ekonomiska och miljömässiga prestanda. Urvalskriterierna passas till det tänkta tillämpningsområdet, ombord RoPax-färjan Stena Scandinavica med tillhörande regelverk och praktiska möjligheter. Studiens resultat visar att en eventuell framtida ORC anläggning kan innebära påtagliga potentiella vinster ur både miljömässig och ekonomisk synvinkel. Resultatet av studien antyder också att någon av de modernare arbetsmedierna, exempelvis R1234ze(E), är goda kandidater för en ORC installation med låga arbetstemperaturer och därmed tillämpbara för återvinning av spillvärmen i marina HT-system. Studien kan även dra slutsatsen att beprövade arbetsmedier med högt GWP-värde generellt presterar bättre än miljövänligare arbetsmedier. Även de termodynamiska teorier och samband som ligger till grund för tillämpningen av den här metoden beskrivs i rapporten.

Nyckelord: organisk rankinecykel, spillvärmeåtervinning, utvärdering värmebärare, energieffektivisering

Acknowledgements

The authors would like to thank supervisors Cecilia Gabrielii and Ulrik Larsen for their tutelage and personal engagement in this study. Your experience in the field and similar research has been of great importance for this work.

The authors would also like to thank the company Stena Lines and project leader Andreas Ericson in particular. Without your engagement and provided data, estimations and practical research would not have been possible for this study.

Also a thanks to David Frisk for providing the authors with this latex Template.

DER NEDERLANDEN, ÖDLUND, Gothenburg, May 2015

Contents

List of Figures

List of Tables

Nomenclature

Short
ALT – Atmospheric Life Time
Cp $[kJ/kg \cdot K]$ – Specific heat capacity constant pressure
DO – Diesel Oil
ESP – Energy Savings Program
eta – Efficiency
GWP – Greenhouse Warming Potential
h $[kJ/kg]$ – Enthalpy
HT $[K]$ – High Temperature
IACS- International Associations of Classification Societies
IMO – International Maritime Organization
LMTD $[K]$ – Logarithmic mean temperature difference
$\dot{m}[kg/s]$ – Mass flow
ODP – Ozone Depleting Potential
ORC – Organic Rankine cycle
PBT – Pay back time
p $[Pa]$ - Pressure
P $[kJ/s]$ – Power
$\dot{Q}[kJ/s]$ – Heat Transfer
$\rho[kg/m^3]$ – Density
RoPax – Roll on Passenger ship
RoRo – Roll on Roll off
S $[kJ/kg \cdot K]$ – Entropy
SFOC $[g/kWh]$ – Specific Fuel Oil Consumption
SP – Expander size
T $[K]$ – Temperature
W $[Nm]$ – Work
WHR – Waste Heat Recovery

Sub script
a – Condensation point
b – Evaporation point
c – Low temperature side
con – Condenser
cw – Cooling water
cw – Cooling water
e – Superheating point
eva – Evaporator
exp – Expander
GEN – Generator
HT – High Temperature
LT – Low temperature
ME – Main Engine
pp – Pinch point
pp1 – Pinch point 1 the point of HT-water when wf hits evaporation point
pp2 – Pinch point 2 the point of HT-water when wf starts superheating
s – Isentropic efficiency corrected
wf – Working fluid
wfp – Working fluid pump
x – Mean of x

1 Introduction

Energy optimization in the marine merchant industry has always been of interest because higher efficiency reduces costs and lower costs means higher profit margins. This is even more important today because of the rising bunker oil prices (Krichene, 2008) and stricter environmental regulations (IMO, 2015)

This study will examine the economic and environmental profit of harnessing one unutilized energy source on board merchant maritime vessels. The main engine cooling water, where about 5% - 7% (Kuiken, 2012) of the total chemical energy in the fuel is rejected and eventually dissipated to the sea. Until recently this waste heat energy have been inaccessible for the purpose of generating electricity due to its low temperature of about 85 degrees Celsius (Kuiken, 2012). It is now possible with modern working fluids and the organic Rankine cycle to generate electricity (Hung et al., 2010) by accessing this previously unutilized waste heat energy to increase overall plant efficiency and lower generator fuel costs.

Stena Lines is currently developing an energy optimization program for their merchant fleet, introduced in the year of 2005 as their energy saving program, ESP (Stena Lines, 2015). This case study is a diploma thesis suggesting further energy optimization on board using organic Rankine cycle (ORC) to recover waste heat from the main engines high temperature cooling water systems.

1.1 Questions

Can waste heat be recovered from the High Temperature (HT) cooling water system utilizing modern working fluids and ORC to convert waste heat into electricity be beneficial on a marine merchant vessel?

The main question is divided into three main objectives:

- This study aims to find suitable working fluid candidates for a potential ORC installation on board Stena Scandinavica,
- find a method to rank those working fluid candidates and
- estimating potential energy saving for one such ORC installation

1.2 Purpose

The main purpose of this case-study is to examine the profit potential in recovering waste heat from the high temperature cooling water system on board a marine merchant vessel utilising the organic Rankine cycle (ORC) to generate electricity.

1.3 Delimitations

The available heat in the HT system varies depending on the type of ship analysed. Therefore this study is delimited to ships with approximately the same available heat. The study will focus on systems with main engines of medium speed four stroke type, which are common in the marine RoRo- and RoPax sector. The propulsion plant of Stena Scandinavica will be the model for this case study.

Possible working fluids are limited with regard to present and upcoming environmental and safety regulations in the marine sector. Operational data from Stena Scandinavica is used.

2 Background

Chapter two presents the main components and working principles of a simple ORC-plant. A brief overview of the working fluid's impact on the plants design and performance is also presented.

2.1 ORC technology and applications

The organic Rankine cycle is comparable to the more common Rankine cycle which utilizes steam as its working fluid; instead the organic Rankine cycle utilizes an organic working fluid. By definition an organic fluid is a fluid that is mainly composed out of carbon.

The main components of the basic ideal Rankine cycle and also the ideal organic Rankine cycle are: the evaporator where the heat source exchanges heat energy with the working fluid, causing a phase change from liquid to vapour. The expander, where the vapour is expanded, and where pressure and heat energy is converted to work. The condenser is where the expanded vapour is condensed to the liquid phase. The working fluid pump circulates and pressurises the condensed fluid in the cycle.

The heat source can be a liquid or gas with enough heat energy to overcome the losses incurred by the pump, expander and pressure drop. If the overall losses of a system is greater than the available energy in a heat source the heat source can not be utilized for a ORC. This study focuses on the HT cooling water system for a medium speed four stroke diesel engine, part of a marine propulsion plant as the heat source. For this study the existing LT-system onboard Scandinavica has been chosen. The cooling source is relevant since a realistic cooling temperature is needed for the plant to function satisfactory. The cooling source should also be realistically available to all common marine engine plants.

With ORC plants the heat source will govern the available selection of working fluids. Because of the thermodynamic properties of different working fluids some are better suited for low temperature heat sources, such as the HT cooling water system at around 90 degrees Celsius. Because of this, careful consideration of the working fluid is paramount. This is because the thermodynamic properties of the chosen working fluid will dictate the final design and efficiency of the ORC plant, at the given temperature (Invernizzi, 2013).

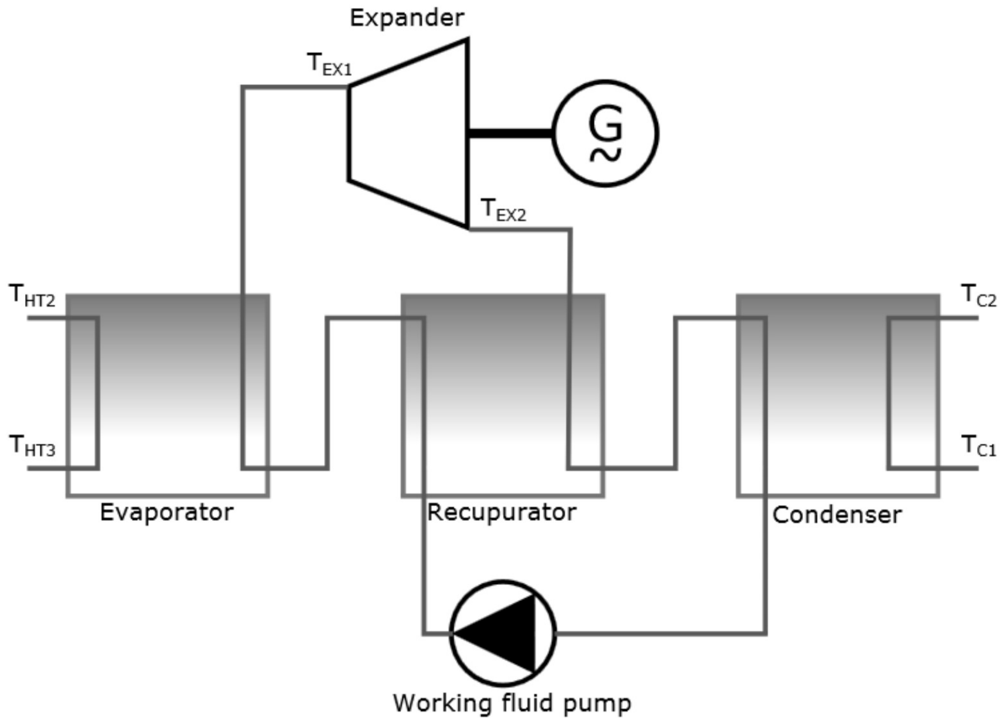

Figure 2.1: Description of a general organic Rankine cycle plant with expander, working fluid pump, condenser, evaporator and recuperator.

The selected working fluid and heat source determines the plant pressure. This is of great importance because higher pressures will increase costs and potential risks. The plant pressure is especially important for marine applications where high pressure vessels will need to be classed and regularly inspected which further increases the life cycle cost of the ORC plant (DNV, 2003).

Most working fluids chemical composition will cause a high greenhouse gas potential index. Consequently environmental regulations will further delimit which working fluids are possible for a marine ORC plant. The requirement for marine closed power cycle instalments differs from industrial plants ashore (Bertrand et al., 2008).

Previous research have mainly focused on land based applications such as solar and geothermal power generation (Vélez et al., 2012) This marine perspective of this study is further limited in its selection of fluids considering specific marine regulations concerning fluids with high greenhouse gas potential index.

Using ORC to recover waste heat is yet not very common in the marine sector and the HT-system is a quite utilized waste heat source. But installations of the kind that this study evaluates has been done on board other vessels which insinuate that the idea of the plant installation is very much realistic (Siuru Bill, 2013).

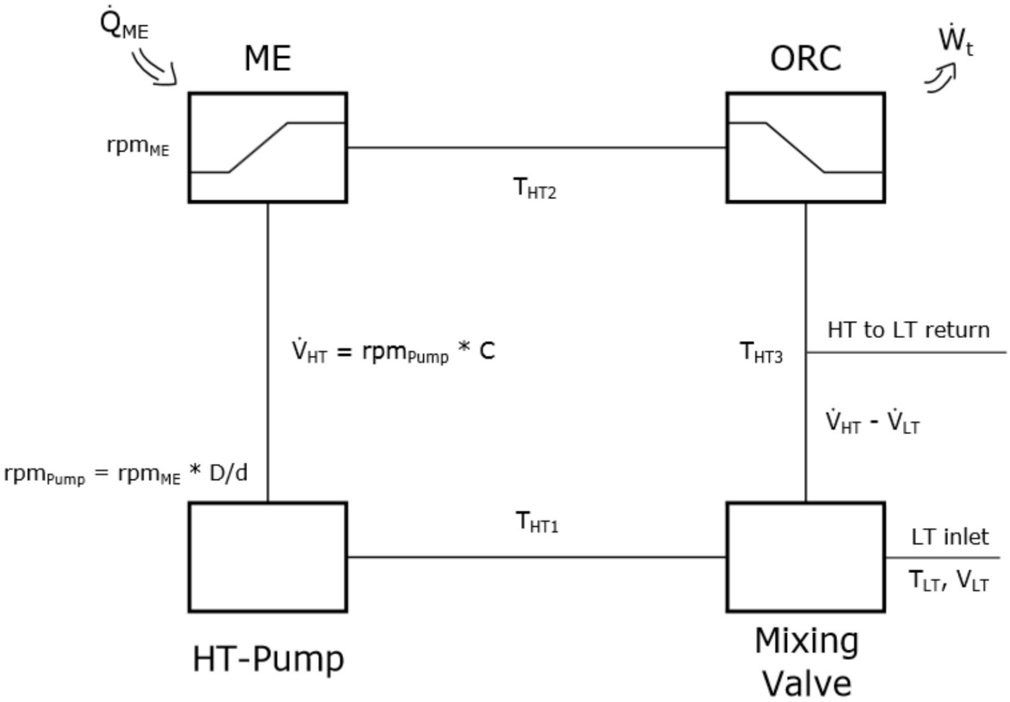

Figure 2.2: This figure depicts a schematic overview of the HT cooling water system on board Stena Scandinavia. This is the system the study evaluates its prospective working fluids against.

2.2 Main Components of an ORC plant

The dimensioning of the ORC plant depends on the working fluids requirements and so forth the working fluid also determines the plants initial cost since different plant components means different purchase prices.

2.2.1 Evaporator and preheater

There are various options for choosing the evaporator and preheater for the ORC plant. Major parameters of the heat exchangers of this study are engine room layout and the total heat exchanging areas. Careful consideration should be done when selecting optimal heat exchanger since its major impact of the total initial cost for the plant is significant (Quoilin et al., 2013).

When installing a WHR system, the WHR must not interfere with the main process of the heat source. This should not be a problem on board Scandinavica since the system already consists of a regulating mixing valve to maintain ideal temperature in the main engine HT system. Recovered heat by the ORC plant is to be considered as free energy for this matter since this energy would otherwise have been rejected to the LT system and ultimately to the sea.

Because of the rate of heat transfers for the different working fluids the size of the evaporator will be different to extract the same amount of heat from the heat source. If we correlate the size of the evaporator, i.e. the area, we get a good approximation of the relative price. Meaning that some working fluids will result in a higher initial investment cost for the evaporator relative to other working fluids for the same amount of heat extracted. For the evaporator to be able to withstand greater pressure, the thickness of the material in the evaporator will have to be increased. Because of this the working fluid able to extract the same amount of heat at a lower pressure would decrease the cost of the evaporator and is therefore preferred (Bertrand et al., 2008).

2.2.2 Expander

The selection of expander influence both the capital cost and the return of investment for a plant design. The factors that determine these concerns are the expanders size which correlate to its cost and its efficiency which influence the plants power production i.e. its earnings potential. Because of this great care in needed to be taken when considering the balance of investment to power production.

Various options for the expander application are available and selecting the proper expander depends on the expected system power output and the working fluid used. Previous research has shown that for high capacity plants, fifty to five hundred kilowatts and good post-expansion vapour quality, a radial inflow turbine to be a good candidate because of its high isentropic expansion efficiency. The radial inflow turbine is not suitable for small plants due to its high capital cost (Bao and Zhao, 2013).

For low capacity plants, one to ten kilowatts, the scroll expander is a good candidate. Because of its relatively low rotational speed which reduces the expander's complexity with regard to the manufacturing tolerances, the capital cost is reduced compared to the radial inflow turbine.

Between the high cost radial inflow expander and the low cost scroll expander is the screw expander. The screw expander is suitable for plants in the capacity range between fifteen to two hundred kilowatts. The main disadvantages of the screw expander are the relatively more difficult manufacturing and sealing.

The main advantage of the scroll and the screw expanders is that relatively wet expansion is possible. When plant limitations don't allow for enough superheating of the working fluid the radial inflow turbine may be damaged when the working gas has a high liquid to gas ratio at the end of expansion (Bao and Zhao, 2013).

2.2.3 Condenser

According to the first law of thermodynamics, as the plants efficiency increases the need by the condenser to reject heat decreases. This means that a smaller area of the condenser is needed for rejecting the heat if the plant is highly efficient. An example: For two plants, one have a high efficiency and the other have a lower efficiency. The available heat, evaporation and condensation temperature are equal for both plants. The plant with the lower efficiency would require a greater area to be able to reject the higher part of the available heat that was not converted into useful work. This means that the more efficient plant is able to reject its, relative to the less efficient plant's, unutilized heat through a relatively smaller heat exchanger.

2.2.4 Working fluid pump

The working fluid pumps function is to induce the mass flow rate in the system. To do this work the pump requires power which will have to be deducted from the produced power of the plant. Because of this the isentropic efficiency of the working fluid pump and the required mass flow of the system are of great importance. Particular care is of great importance when selecting matching pump and working fluid. For a given heat source which you want to extract the maximum amount of useful work the working fluid which is able to extract that amount of work at the lowest mass flow will have the lowest loss to pump work and thereby increase the plants overall efficiency. Because of the low required mass flow of the optimal working fluid candidates: its working fluid pump will also be the most cost efficient one in terms of capital investment. Pump power rating correlates with its capital cost.

In virtue of the diaphragm pumps innate tightness due to its design it has so far been the preferred pump for commercial ORC plants. This is because fluid loss to leaking is both economically and environmentally costly (Quoilin et al., 2013).

2.3 Various working fluids

Working fluids can be subdivided into three main categories: isentropic, dry and wet. The main difference between these categories is the vapour state after expansion. This is visualized by the temperature-entropy diagram (T-s). The fluids saturation line's slope after the critical point is positive for dry working fluids, negative for wet working fluids and rectilinear for isentropic working fluids.

2.3.1 Characteristically isentropic working fluid

When the working fluid is expanded along the saturation line of an isentropic working fluid such as R11, the change in entropy is zero. This is ideal and results in a high conversion of the vapour's potential energy to kinetic energy id est. high efficiency as defined by the second law of thermodynamics.

2.3.2 Characteristically dry working fluid

A dry working fluid as mentioned has a retrograde saturation curve after the critical point this means that during expansion the vapour will expand in to the super-heated regain. At the end of expansion the working fluid is still in a vapour state and needs to be condensed in to a liquid state in order to be pumped. This condensation removes heat from the system that for an isentropic fluid could otherwise have been converted to kinetic energy id est. work by the expander.

If the working fluid is dry some of the residual heat from the super heated gas after the expander can be recovered by a recuperator. The recuperator preheats the working fluid before the evaporator.

2.3.3 Characteristically wet working fluid

For working fluids like water, methanol etc., which are counted for as wet working fluids, a need for superheating the media is required before entering the expander. Unless the media is superheated, there is a risk for moisture in the expander. This is why wet working fluids are not suitable for low temperature ORC with a low power output (Bertrand et al., 2008).

2.4 Theory

The theoretical model for organic Rankine cycle is described and defined in the following sections.

2.4.1 The ideal organic Rankine cycle

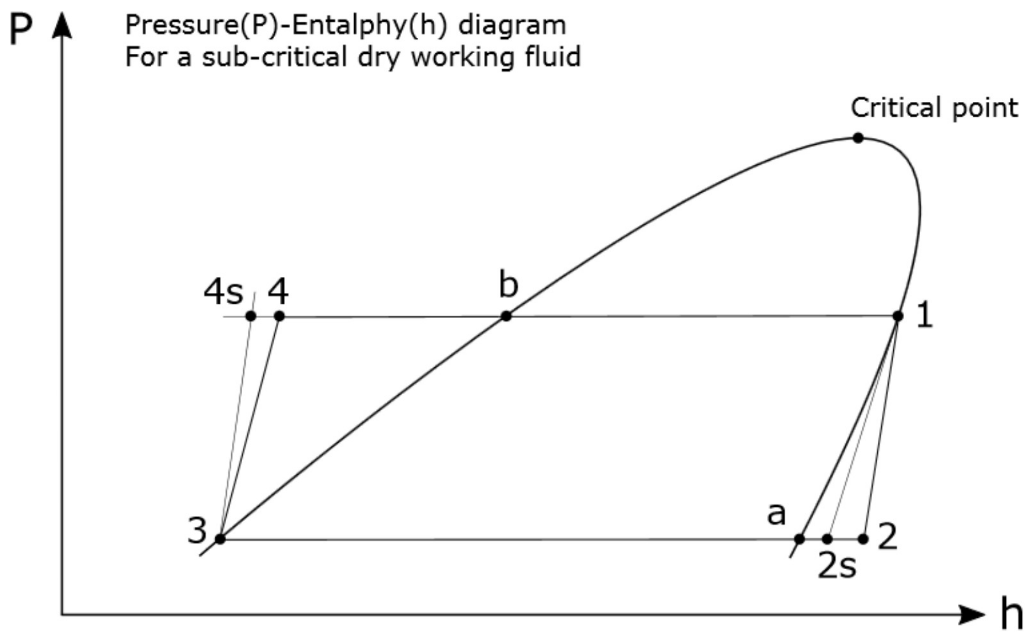

Figure 2.3: Schematic diagram of the interesting points in the organic Rankine cycle, using a dry working fluid in this example.

Table 2.1: State changes in relation to figure 3.

Working principles of the ORC

State	Process	Components
$1 \rightarrow 2s$	Isentropic expansion	Ideal Expander
$1 \rightarrow 2$	Real expansion	Expander
$2 \rightarrow 3$	Isobar heat flow out	Condenser
$3 \rightarrow 4s$	Isentropic compression	Ideal working fluid pump
$3 \rightarrow 4$	Real compression	working fluid pump
$4 \rightarrow 1$	Isobar heat flow in	Evaporator
a	Condensation point	Condenser
b	Evaporation point	Evaporator

2.4.2 Component analysis

This section defines each of the ORC components thermodynamic models.

Expander

Work produced by the expander.

$$\dot{W}_{exp} = \dot{m}_{wf} \cdot (h_1 - h_2) = \dot{m}_{wf} \cdot \eta_{exp} \cdot (h_1 - h_{2s}) \tag{2.1}$$

Power produced by work

$$P_{net} = \dot{W}_{exp} \cdot \eta_{GEN} - \dot{W}_{wfp} \tag{2.2}$$

Condenser

Heat rejected by the condenser

$$\dot{Q}_{con} = \dot{m}_{wf} \cdot (h_2 - h_3) = \dot{m}_{cw} \cdot (h_{c2} - h_{c1}) = \dot{m}_{cw} \cdot \overline{C_{pLT}} \cdot (T_{c2} - T_{c1}) \tag{2.3}$$

Evaporator

Heat to the evaporator

$$\dot{Q}_{ME} = \dot{m}_{ME} \cdot \overline{C_{pHT}} \cdot (T_{HT2} - T_{HT3}) = \dot{m}_{wf} \cdot (h_1 - h_4) \tag{2.4}$$

The evaporation of the working fluid may be divided into three stages:

$$\dot{Q}_{ME} = \dot{Q}_{ME4-b} + \dot{Q}_{MEb-e} + \dot{Q}_{e-1} \tag{2.5}$$

Whereas $\dot{Q}_{ME4-b}$ is the preheating (ph) stage, $\dot{Q}_{MEb-e}$ is the evaporation stage (e), and $\dot{Q}_{e-1}$ is the superheating (sh) stage.

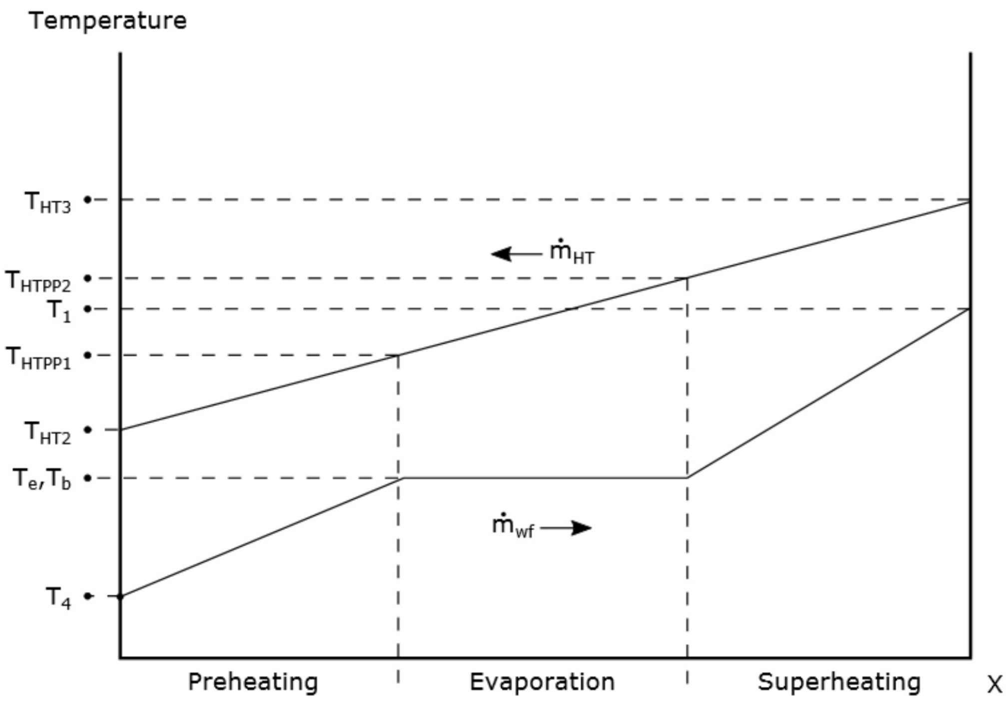

Figure 2.4: A model of the evaporator.

Working fluid pump

Work required by the working fluid pump; η_{wfp} being the isentropic efficiency

$$\dot{W}_{wfp} = \dot{m}_{wf}(h_4 - h_3) = \frac{\dot{m}_{wf}(h_{4s} - h_3)}{\eta_{wfp}} \tag{2.6}$$

Net produced effect

$$P_{net} = \dot{W}_{exp} - (\frac{\dot{W}_{wfp}}{\eta_{wfp}}) \tag{2.7}$$

2.4.3 Working fluid analysis

Careful consideration when selecting a working fluid is essential for optimal plant design and dimensions. The working fluid?s specific heat capacity influences the plants mass flow and what temperatures the plant needs to heat and cool the fluid to gain the thermal work desired in the plant. Specific heat capacity is not constant through the phase changes.

The specific heat capacity each end of the evaporator for the working fluid may be defined as:

$$c_{p4} = \frac{(c_{p1}T_1 - q_{1-4})}{T_4} \tag{2.8}$$

$$c_{p1} = \frac{(q_{1-4} + c_{p4}T_4)}{T_1} \tag{2.9}$$

As found by (Bao and Zhao, 2013) that and increase of the expander outlet working fluid density will lead to a smaller size expander. This is shown by the following relationship which relates the size SP to the volym flow rate and the isentropic enthalpy change over the expander.

$$SP = \frac{\sqrt{\dot{m}_{out}/\rho_{out}}}{(\Delta h)^{1/4}} \tag{2.10}$$

2.5 Cycle analysis

According to the first law of thermodynamics, the total conversion of heat to work is for an ideal ORC be expressed as

$$\dot{Q}_{ME} + \dot{W}_{wfp} = \dot{W}_{exp} + \dot{Q}_{cw} \tag{2.11}$$

Where to heat flow into the ORC is $\dot{Q}_{ME}$

$$\dot{Q}_{ME} = \dot{W}_{exp} - \dot{W}_{wfp} + \dot{Q}_{cw} = \dot{m}_{ME} \cdot \overline{C_{pHT}} \cdot \Delta T_{HT} \tag{2.12}$$

The generators electrical power out put P_{net}

$$P_{net} = \dot{W}_{exp} \cdot \eta_{GEN} - \dot{W}_{wfp} \cdot \eta_{wfp} = (\dot{Q}_{ME} - \dot{Q}_{cw} + \dot{W}_{wfp}) \cdot \eta_{wfp} \cdot \eta_{GEN} - \dot{W}_{wfp} \cdot \eta_{wfp} \tag{2.13}$$

2.6 Working fluid economics

The required area for the heat exchangers a variable influenced by the selection of the working fluid. This is because of the thermal conductivity of the working fluid is inversely proportional to the area of the heat exchanger as stated by Fourier's law. Also the area of the heat exchanger is proportional to the costs of purchasing the heat exchangers e.g. the evaporator and condenser.

Some of the potential working fluids have toxic, corrosive and flammable properties. When handling these fluids special care is required, this means more expensive equipment and training (Bertrand et al., 2008).

As noted by (Wong et al., 2013) the expander can make up to about half of the total component costs for an ORC installation. This means that special care when considering an expander should be taken. It has also been found that for small output plants, less than 100kW, a twin screw expander to be a good alternative. (Leibowitz et al., 2006)

Other costs can be divided into three subcategories: the cost of the components, maintenance costs and installation costs. Costs for refilling the system are neglected since the system is meant to be closed. Further on some of these costs are to be considered independent of the working fluid, e.g. technician courses, wiring schemes, drawings to name a few. These costs will not specifically be evaluated of the working fluids performance.

2.7 Environmental regards

In 1974 the discovery was made that chlorofluorocarbons are stable enough to reach Earth's ozone layer in the stratosphere and ultimately destroy Earths protection against UV rays. Since then a number of international and national agreements and installations have been found to prevent this from happening. Consequently many working fluids that were working great in different heat plants have been phased out and are so yet today. Global Warming is another environmental issue that needs to be considered when selecting working fluid (Bertrand et al., 2008) (DiPippo, 2012).

3 Method

3.1 Introduction

This study is focused on one case in particular, the RoPax ferry Stena Scandinavia. The study analyzes 9000 points of operations data representative of a fifty-eighth day period trafficking Gothenburg - Kiel. This data is used to evaluate various working fluids with possible profit potential as the main selection criteria.

3.1.1 Theoretical framework

To determine the dimensions of the components in an ORC cycle for a specific working fluid, e.g. heat exchanger area, thickness etc. can be very difficult to estimate. In the case of estimating the required area for the heat exchangers, the area is dependent on the thermal conductivity of the working fluid as described by the overall heat-transfer coefficient in Fourier's law. Most of the working fluids investigated in this study, their thermal conductivity which must be determined practically for an accurate estimation (DiPippo, 2012), are undetermined. This study will use the relationship between the working fluids expander outlet density and expander size calculated in CoolProp (Bell et al., 2014) as an indication of the ORC-plants dimensions. The higher expander outlet density of working fluid and same amount of heat recovered means smaller dimensions required for the plants expander, and also relative cost of the plant.

The work delivered by the plant in CoolProp will be used to estimate what reduction in the auxiliary engine's bunker consumption is possible with this ORC plant. The study also uses this to indicate a possible budget for installation of the plant considering the kWh earned compared to current price of marine Diesel oil.

Due to difficulties finding heat data for Stena Scandinavia's main engines, a similar main engine has been assumed to produce a proportional amount of available heat at equal load to the ones on Stena Scandinavica. The available heat is also assumed to at least linear for loads lower than the reference. Stena Scandinavica is equipped with 4 x MAN-BW 9L40/54 main engines. Due to the age of the engines appropriate data needed for this study was difficult to find so a similar engine, Wärtsilä 9LDF50, have been used to estimate ORC output.

First a model is defined utilizing the EOS in CoolProp for a fluid to solve for the points in the cycle. Boundary conditions for cycle max and min pressure is defined and cycle high and low temperature. A combination for each combination of sub cooling and superheating temperatures combined with each working fluid available in CoolProp's fluid library is composed. The model is then iterated over for each combination of sub cooling, superheating and working fluid to yield the net work and the required working fluid mass flow for each combination. This result is sorted from highest net work per working fluid mass flow and the top working fluid candidate solutions is fathered manually evaluated for the conditions imposed by shipping in Europe.

For the top working fluid candidates the effective operation time of the plant and the parent system heat available is evaluated as a function of the main engine power output data samples from Stena Scandinavia. This result is then used with the cost per kWh to produce power with the auxiliary machinery. Combining the savings from reducing the auxiliary machinery load, with a desired pay back time, the maximum capital cost for the plant can be calculated.

3.1.2 Data from Stena Scandinavica

Data presented in this section was collected either on site or given by Stena Lines.

Produced energy of the ORC plant will be used to ease the auxiliary power production on board, with reductions in Diesel oil (DO) consumption as a result. The following data was collected from Stena Scandinavica for the auxiliary plant. Please observe the data used in this study is collected in winter time.

Table 3.1: Power production and DO consumption of Stena Scandinavia November 2014 - February 2015

Month	Mean power produced (kWh)	Bunker Consumption (kg)	Mean bunker consumption (kg/24h)	Mean SFOC (g/kWh)
November	685,583	183,548	6,118	267.72
December	728,514	209,808	6,768	287.99
January	734,989	166,494	5,372	226.53
February	657,391	168,842	6,030	256.84
Total	2,806,477	728,692		
Mean value			6,072	259.77

Stena Scandinavica's main propulsion plant consists of four engines connected with gear to two shafts with two engines each. The measured HT-water used in this study was measured on the pressure side on one of these engines with 89°C and 2.9 - 3.4 bara pressure.

To evaluate the waste heat produced by the main engines; available to the ORC plant, recorded propulsive power data from Stena Scandinavica was analysed. The evaluation is made using fifty-eight days of operational data from Stena Scandinavica. Estimations by this study are made with the data recorded on Stena Scandinavica.

From the operations data provided by Stena Line the power output for a main engine for each sample point. From this the load for each data point was calculated. From the sample loads a mean load for the sample time span was calculated.

With the care free assumption that heat output scale linearly, a scale factor was calculated from the MAN engine rated power divided by the Wärtsilä rated power.

For each of the previously calculated ORC outputs for the model Wärtsilä machine a scaled ORC output was calculated for the MAN machine using the mean load and the scale factor.

3.1.3 Estimated values

This study encountered some difficulties collecting real data needed. This data has been estimated in manners to be so close to circumstances onboard Stena Scandinavica as possible e.g. collected from very similar applications or industrial plants.

The documentations found for MAN-BW 9L40/54 main engines was unsatisfactory why similar engine, Wärtsilä 9LDF50, has been used to evaluate ORC output and then scaled to be realistic according to Stena Scandinavica engine output (Wärtsilä, 2012).

Table 3.2: Data from Wärtsilä DF50 used to estimate heat source for ORC model

Data retrieved from Wärtsilä DF50 manual used in study at 100% load	
HT pump volume flow	160 m3/s
HT pump head	3.2 bara
HT regulating temperature	90°C
HT jacket water heat	1 620 kW
HT charge air cooling water heat	1 860 kW
Lowest ORC output temperature	70.6°C
LT regulating temperature	35°C

For the components efficiencies, values in table 3.3 have been assumed.

Table 3.3: Assumed efficiencies of the ORC plant's components

Assumed efficiencies of the ORC plant's components	
ORC working fluid feed pump	70 %
ORC working fluid expander	80 %
ORC expander driven generator	98 %

The maximum pressure in the ORC plant is set to 16 bara and the minimum pressure is set to 1 bara. This corresponds to class I according to IACS (International Association of Classification Societies, 2011).

For the cost calculations the current, may 2015, Diesel oil bunkerindex.com rating is assumed 632.62 $ per Mton. Considering the expected price development, this is a conservative mean since Diesel oil prices are expected to increase during the lifetime of a potential ORC plant (Krichene, 2008). For the CO_2 emissions reductions the CO_2 content is assumed to be 0.25 $kgCO_2/kWh$ (Eia, 2015).

3.2 Selection of working fluid and dimensioning of plant design

The main objective of this study is to find a solution that is most suitable considering the constraint of this marine propulsive plant, we need to find what the best solution is from an economical point of view. In this instance we want to maximize the capital gain per expenses incurred by the installation: the cost effective performance. Both these parameters can be defined as functions of the selected working fluid. This cost effective performance is determined by the work produced per heat transfer area as defined by previous research done by (Li et al., 2012).

The selection of appropriate working fluids presented in this study have been evaluated using the C++ library CoolProp (Bell et al., 2014). The library implements equations for both pure and pseudo pure fluid's states and component's transport properties. Thermodynamic properties are evaluated using the Helmholtz energy formulations.

Estimated and collected values in chapter 4.1.3 have been used in relation to the theoretical circumstances presented in chapter 3 to simulate the case of this study.

The fluid's performance has been ranked by net power time the expander outlet density. This ratio indicates net power related to expander cost. This is because if the fluid's outlet density increases the expander size decreases as suggested by (Bao and Zhao, 2013).

3.2.1 Power output

The power output by the plant with the Wsilgine at full load is first calculated for each candidate working fluid. These values are then scaled down to the MAN engine on Stena Scandinavica simply by finding the quotient of rated power for the two engines. These scaled values are then further scaled by the Stena Scandinavica's mean load which is calculated from the operational data collected over a period of fifty-eight days. This value is used to represent the mean expected power output by each of the working fluid candidates.

These scaled down values are to be considered conservative since the MAN engine is both older and running at a lower load. These conditions generally decrease the engines efficiency and a relatively greater portion of the input energy end up as waste heat in the cooling water system.

3.2.2 Performance

The performance metric which this study ranks the candidate working fluids is a proportional representation of each working fluid compared to one and other, relative power output per cost.

The relative cost indicator for this study is each working fluids density after expansion which by (Bao and Zhao, 2013) indicates the size of the expander i.e. the cost of the expander. Working fluids that exhibit a higher expander outlet density are expected to require a relatively lower investment cost.

3.2.3 Environmental restrictions in the marine sector

CoolProp evaluates Global Warming Potential, GWP100, for all working fluids considered so GWP100 will be this studies first environmental criteria (Bell et al., 2014).

Other factors to be considered are the working fluids Ozone depleting Potential, ODP, and Atmospheric Life Time, ALT. Environmental friendly fluids has increased in popularity in this kind of plant because of the stricter international regulations regarding refrigerants, e.g. Kyoto- and Montreal protocols. For example the refrigerants are characteristics of bad ODP and GWP, why both R-11 and R-12 are banned from use by the Copenhagen Amendment to the Montreal Protocol (Bertrand et al., 2008) (IMO, 2015).

In the selection of working fluids for this study the ODP and GWP is considered. Selected working fluids will comply with current and upcoming regulations (Niles, 2010). Low GWP is however hard to define as noted by (Palm, 2013) it is still uncertain which working fluids will be compliant in the long run, but a GWP of under 150, as set by the European F-gas directive (Pavkovic, 2013), can safely be considered at this moment and are expected to be allowed for the foreseeable future. In Pavkovic's article it it also mentioned that refrigerants that are currently produced in the European Union may not contain any chloride. Because of this, the study will only consider candidates with GWP under 150 and ODP of 0.

3.2.4 Safety and handling restrictions in the marine sector

Personnel safety when handling is important on ships as well as it is ashore, this means that working fluids that are less toxic and less reactive are preferred during the evaluation because when handling these fluids special care is required. Furthermore a working fluid that is less flammable is also preferred because of the inherent dangers of working with flammable liquids (Bertrand et al., 2008).

For example all hydrocarbon candidates for this study are flammable and that is a disadvantage when selecting appropriate working fluid for this installation. Even if a hydrocarbon may produce good work in the cycle and is highly available on the market it may not be suitable for a marine ORC plant (DiPippo, 2012).

Working fluids selected by this study are evaluated according to their Hazardous Material Identification System values, HMIS. HMIS indicates the flammability (F), physical hazard (P) and health hazard (H) requirements for each working fluid (Association American Coatings, 2015).

Maximum risk potential for this studies working fluids are set as (Association American Coatings, 2015):

- Flammability (F): 2, this category includes fluids with a flashpoint in the range of 38℃ to 93℃.
- Physical hazard (P): 1, this category states fluids that are normally stable but may self-react at high temperature. This category does not react violently with water nor undergo hazardous polymerization in the absence of inhibitors.
- Health hazard (H): 2, the area where the ORC plant might be stationed is an marine propulsive engine room area for a passenger ferry with constant present personnel why a low health risk potential i required.

On ships, a fluid with a flash point lower than Diesel fuel, are not currently considered suitable by SOLAS. This means that working fluids in a flammability category higher that Diesel fuel (category 2), i.e. flash point lower than 60℃, are not suitable for use on board ships.

Considering the rise of LNG and other alternative fules with lower flash points, which are currently being evaluated by rule makers. The possibility of using a more flammable working fluid in the future may become a reality although currently it is however not clear how these regulations will specifically apply to working fluids (Ignazio, 2015).

3.2.5 Estimated environmental and bunker savings for Stena Scandinavica

The net mean power calculated by the model directly reduces the needed power production by the auxiliary engines.

The ORC plant needs the main engine running in order to produce power thereby an estimate for ORC operational time can be made from the main engine operational data set. Since Stena Scandinavica uses a shore connection and the ORC is only operational when the main engines are running the ORC will not generate any savings at port.

From the main engine operational data set the operational time of the main engines are calculated to be 60.3% of total hours. If the ORC plant is assumed to be producing mean power for the duration the main engines are running and a year is defined as 8,760 hours, the yearly saving can be calculated.

The Running hours of ORC per year is defined as Rh_{ORC}, the Auxiliary bunker cost is defined as Bc_{aux} and the pay back time as PBT.

Yearly Savings = $\overline{Output}_{ORC} \cdot \overline{SFOC}_{aux} \cdot Rh_{ORC} \cdot Bc_{aux}$

For a desired pay back time for the highest ORC plant purchase and installation cost can now be determined.

Highest purchasing cost(Yearly Savings)= Yearling Saving $\cdot PBT$

An estimate for the potential carbon di oxide reduction per year is calculated. The saved CO_2 emissions per year, $-CO_2/y$, is defined as the amount CO_2 emissions saved by producing power with the ORC instead of an auxiliary generator set. The CO_2 content per Diesel oil combusted (Eia, 2015) is defined as CO_2/DO. This i calculated.

-CO_2/y =$\mathrm{CO}_2/DO \cdot Rh_{ORC} \cdot \overline{Output}_{ORC}$

4 Results

In this chapter the top candidates determined by the model is presented. These results are further discussed in chapter 6. From the top performing candidates, four candidates that fulfill the HMIS, GWP100 and ODP for requirements are further examined for relative economical evaluation.

4.1 Top performing candidates from evaluation

Table 4.1: Top performing refrigerant candidates

Candidate refrigerants and their calculated values sorted by performance							
Working fluid	Power Output (kW)	Performance	GWP100	HMIS H,F,P			ODP
R227EA	59.22	16.54	3220	1	0	0	-
RC318	72.59	16.42	10300	1	0	2	-
R124	73.19	14.25	609	1	1	0	0.022
R236FA	77.43	12.12	9810	1	0	1	-
R1234ze(E)	56.55	11.59	6	1	2	0	-
R12	43.40	11.58	10900	1	1	0	1
R114	79.08	10.32	10000	1	0	0	1
HFE143m	50.93	9.85	0		2		
R142b	75.44	9.61	2310	1	1	0	0.07
R236EA	83.05	9.18	1200	-	-	-	-
R1234yf	33.26	8.80	4	1	2	0	-
R134a	34.75	8.19	1430	1	1	0	-
R218	14.57	7.91	8830	2	1	1	-
R152A	49.24	6.59	124	1	1	1	-
R245fa	80.82	6.18	1030	2	1	1	-
R1234ze(Z)	82.15	6.16	0			1	
R1233zd(E)	82.05	5.25	0				-
DimethylEther	48.20	4.54	1	1	4	2	-
R21	61.17	4.31	151	-	-	-	-
n-Butane	80.91	4.10	3	1	4	0	3
SES36	77.56	3.46	0	Low	-	Low	-

4.2 Environmental and safety evaluation

The four top performing working fluids from table 5.1 that meet the studies environmental and safety criteria are presented in this table. Two extra working fluids are also presented in the table for comparison reasons (R152A and R1234ze(Z))

Table 4.2: Suitable candidates for Stena Scandinavica

Suitable candidate refrigerants and their calculated values sorted by performence								
Refrigerant	Power output (kW)	Performance	GWP 100	HMIS H,F,P			ODP	Ton CO2 saved per year
R1234ze(E)	56,55	11,59	6	1	2	0	-	75
HFE143m	50,93	9,85	0		2		-	67
R1234yf	33,26	8,80	4	1	2	0	-	44
R152A	49,24	6,59	124	1	1	1	-	65
R1234ze(Z)	82,15	6,16	0			1		108

Observe these studies results are in accordance with this studies safety- environmental and handling criteria described in section 3.2.3 and 3.2.4. Also noted in the results are the loss in performance for using environmentally approved working fluid over the top rated performing working fluid. E.g. R227EA (GWP100 $<$ 150) have a superior performance of approx. 30 % higher than optional top rated R1234ze(E) (GWP100 $>$ 150).

4.3 Economical evaluation

Estimated savings calculations from section 3.2.5.

Table 4.3: Economical evaluation of the top performing candidates ordered by performance

Saving estimates for top performing candidates					
Refrigerant	Power output (kW)	Performance	Yearly savings ($)	Pay back time (years)	Highest purchasing cost for a specific pay back time (k$)
R1234ze(E)	56.55	11.59	49,089	5	245
HFE143m	50.93	9.85	44,211	5	221
R1234yf	33.26	8.80	28,872	5	144
R152A	49.24	6.59	42,744	5	214
R1234ze(Z)	82.15	6.16	71,312	5	357

It is observed in the results that the working fluid with the less performance (e.g. more expensive), R1234ze(Z), is the most power generating one and so it is also the most CO_2 reducing one. Even so the performance indicates the investment cost of each kW produced is higher, why R1234ze(E) may be a more attractive alternative due to its relatively low purchasing cost for a specific pay back time.

5 Discussion

Although ORC technology is not a new technology in worldwide power generating, marine applications are not yet very common. This may be caused by the marine industry being known as conservative but with future environmental demands and needs for optimizing energy efficiency there might be a development for this technology even in the marine sector. It can be believed that the increasing demands of green shipping and emission restrictions will push research for this WHR technology forward. Pros for this technology development is the fact ORC plants already exists in one for this study known merchant vessel supporting the results evaluated by this study and, as said before, the heat source being as unutilized as it is now.

5.1 Discussion of method

The method used in this study is not optimal for determine the potential savings for the ORC plant. Because the amount of assumptions required to produce workable data, the validity of this model can certainly be questioned. For a real application further investigation into these candidate working fluids are necessary, in particular regarding regulations specific to shipping in Europe. For example we have set the bunker price per MTon to be a fixed value which will not correspond with reality for a very long time because of the general volatility of bunker prices. This will lead to large deviations in the amount of actual savings.

When estimating the mean load for use the in the calculation of the ORC installation expected mean power output. Data collected over a fifty-eight day period for a fairly regular trade route was used for this mean load calculation. The study considers this to be a fairly good representation of reality. This is because of the data's high resolution of a sampling rate of one shaft power readout per every ten minutes over what this study consider to be normal operation of the ship. Thereby this study consider this mean is to be a reliable representation of future expected power output.

This study has considered that the ORC plan would be operating at design point during every hour of main engine operations. Naturally this is a simplification of reality, but the authors believe that this has a negligible affect on the results. This is because on review of the main engine data this off design point running time is relatively a small part of the total running hours. For a more precise determination of the power output more advanced calculations are required. These computation would not be possible on this research groups available hardware in a reasonable time frame.

Total plant installation cost is not adequately considered in this study for simplicity reasons and for the difficulty in estimation. Examples of costs not evaluated are the components individual costs, installation work and the actual cost of the working fluids themselves. The total costs may also vary from a lot of considerations such as manufacturer of components, what corporation hired for installation, the regulatory development considering use of various refrigerants etc. For finding a more exact total plant cost, further evaluation of a specific working fluid and corresponding plant design should be individually evaluated, e.g. getting cost estimates by manufacturer.

5.2 Discussion of results

This study found that the ORC technology do look very promising from a economical point of view, although further investigation is required. Environmental restrictions, such as GWP100, has shown by this study leads to lower possible fuel savings. Research and development of modern working fluid alternatives are recommended.

Results presented in this work is dependent on the data for one merchant vessel with specific running data over a limited period and is therefore not to be necessary applicable for any vessel. The output of the ORC is heavily dependent of vessel's specific running data and merchant route which need to be considered for evaluation of possible plant investments on board other vessels.

The application of this technology is still quite young in the marine sector, which indicates not to be hasty if considering instalment of this WHR system. But, on the other hand, the technology itself is historically utilized in other energy generating plants and is not very complicated. Combined with this studies results an application of this technology in the shipping industry should not prove to be difficult and do look like a promising alternative for greener and more profitable shipping in the future.

In this study the performance was measured as a function of the net power output by the potential ORC installation and the relative size of the expander unit. The authors of this study were unable to find a reliably and extensive source for the thermal conductivity of each of the fluids investigated. Thereby no approximation of the relative size of the heat exchangers i.e. evaporator and condenser could be made. If the thermal conductivity, or even a relative thermal conductivity, of more fluids were available a more reliable performance metric could have been calculated. This would certainly have increase the weight of this studies result.

Because of the absolute nature of environmental and safety regulations in Europe considerable compromises are needed which results in sub optimal performance for any potential ORC installation. The authors of this study do see the point of these regulations but if some kind of dispensation for these kinds of installation were to be an option, it would greatly improve performance. Considering that one of the benefits of an installations of this kind is a reduction in green house gas emissions.

We believe that under the right circumstances and precautions a net decrease in environmental impact could be achieved even with a high GWP working fluid.

Previous research in this field is primarily limited to on-shore applications, such as solar- and geothermal power generation, which can be considered as an obstacle for marine development. Subsequently more research aimed for the marine sector would be of great interest if ORC technology is to be an applied WHR technology on board the merchant fleet in the future.

6 Conclusion

An ORC solution recovering waste heat from the main engine high temperature cooling water system utilizing a modern working fluid such as R1234ze(E) is both profitable and environmentally sound. If such a plant it is technically possible to construct, waste heat from Stena Scandinavica's propulsion plant could save tens of thousands of dollars in auxiliary power generation costs every year. This reduction in auxiliary power generation will also mean a proportional reduction in environmentally harmful emissions from the auxiliary power generation plant adding to the benefit of an organic Rankine cycle installation.

To improve upon this research greater knowledge about the candidate working fluids thermal capacity would be of great interest. If the thermal capacity for a greater part of working fluids were known better total plant cost estimates could be made. Thereby increasing this research's applicability for the cost estimation of a real project. It would also be of great benefit to further investigate how the technical and mechanical requirements set by these newer working fluids, like R1234zd(E) affect plant construction and its total cost.

Bibliography

Association American Coatings (2015). Explanation of the HMIS ® Ratings.

Bao, J. and Zhao, L. (2013). A review of working fluid and expander selections for organic Rankine cycle. *Renewable and Sustainable Energy Reviews*, 24:325–342.

Bell, I. H., Wronski, J., Quoilin, S., and Lemort, V. (2014). Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. *Industrial & Engineering Chemistry Research*, 53(6):2498–2508.

Bertrand, T. F., Papadakis, G., Lambrinos, G., and Author, C. (2008). Criteria for working fluids selection in low-temperature solar organic Rankine cycles. *1st International Congress on Heating, Cooling and buildings*, pages 1–8.

DiPippo, R. (2012). *Binary Cycle Power Plants*. Butterworth-Heinemann.

DNV (2003). BOILERS, PRESSURE VESSELS, THERMAL-OIL INSTALLATIONS AND INCINERATORS.

Eia (2015). Carbon Dioxide Uncontrolled Emission Factors.

Hung, T. C., Wang, S. K., Kuo, C. H., Pei, B. S., and Tsai, K. F. (2010). A study of organic working fluids on system efficiency of an ORC using low-grade energy sources. *Energy*, 35(3):1403–1411.

Ignazio, M. (2015). LNG as Marine Fuel Introduction Rules Technology Training.

IMO (2015). Air Pollution, Energy Efficiency and Greenhouse Gas Emissions.

International Association of Classification Societies (2011). Requirements concerning pipes and pressure vessels.

Invernizzi, C. M. (2013). *Closed power cycles: thermodynamic fundamentals and applications*, volume 11. Springer Science & Business Media.

Krichene, N. (2008). *IMF Working Papers : Crude Oil Prices : Trends and Forecast*. International Monetary Fund, Washington, DC, USA.

Kuiken, K. (2012). *Diesel Engines: for ship propulsion and power plants. From 0 to 100 000kW Part 1*. Target Global Energy Training, 2nd rev. e edition.

Leibowitz, H., Smith, I. K., and Stosic, N. (2006). Cost Effective Small Scale ORC Systems for Power Recovery From Low Grade Heat Sources. *2006 ASME International Mechanical Engineering Congress and Exposition*, 2006:521–527.

Li, Y. R., Wang, J. N., and Du, M. T. (2012). Influence of coupled pinch point temperature difference and evaporation temperature on performance of organic Rankine cycle. *Energy*, 42(1):503–509.

Niles, S. (2010). Refrigerant Update 2010 and Beyond.

Palm, B. (2013). Defining "Low GWP".

Pavkovic, B. (2013). Refrigerants – Part 2 : Past , present and future perspectives of refrigerants in air-conditioning applications.

Quoilin, S., Broek, M. V. D., Declaye, S., Dewallef, P., and Lemort, V. (2013). Techno-economic survey of organic rankine cycle (ORC) systems. *Renewable and Sustainable Energy Reviews*, 22:168–186.

Siuru Bill (2013). Turning waste heat into electricity saves fuel, reduces CO2 emissions.

Stena Lines (2015). Energy.

Vélez, F., Segovia, J. J., Martín, M. C., Antolín, G., Chejne, F., and Quijano, A. (2012). A technical, economical and market review of organic Rankine cycles for the conversion of low-grade heat for power generation. *Renewable and Sustainable Energy Reviews*, 16:4175–4189.

Wärtsilä (2012). Cooling Water System.

Wong, C. S., Meyer, D., and Krumdieck, S. (2013). Selection and Conversion of Turbocharger As Turbo-Expander for Organic Rankine Cycle (Orc). *35th New Zeakand Geothermal Workshop*, 2013(November):1–8.

A Appendix 1

A.1 Cycle calculation python code

```
import matplotlib
import numpy as np
from CoolProp.CoolProp import FluidsList
from CoolProp.CoolProp import PropsSI
#from scipy.optimize import newton

k = 0 + 273.15
fluids = np.array(FluidsList())
thtin = k + 90
thtout = k + 70.6
tltin = 35
tmax =  thtin - 5
tmin = k + tltin + 5
presission = 2
tests = np.array([(fluid,super_heated,sub_cooled,tmax,tmin,.8,thtin,thtout)
        for fluid in np.nditer(fluids) for super_heated in np.arange(0,45,presission)
         for sub_cooled in np.arange(0,45-super_heated,presission)]
        , dtype=('a20,f1,f1,f1,f1,f1,f1,f1'))
cand=[]
def discard_fails(results):
    for result in results:
        if result[0] != 'Fail test':
            cand.append(result)

#savetxt('newcandidates0705.csv', cand, delimiter=',',fmt='%s')


def SC(Ref,DTsh,DTsc,Tmax,Tmin,eta_a,THTin,THTut,Ts_Ph='Ph',skipPlot=False,axis=None):
    """

    Modified Simple Cycle function from the CoolProp library

    This function plots a simple four-component cycle,
     on the current axis, or that given by the optional parameter *axis*

    Required parameters:

    * Ref : A string for the refrigerant
    * Te : Evap Temperature in K
    * Tc : Condensing Temperature in K
    * DTsh : Evaporator outlet superheat in K
    * DTsc : Condenser outlet subcooling in K
    * eta_a : Adiabatic efficiency of compressor (no units) in range [0,1]

    Optional parameters:

    * Ts_Ph : 'Ts' for a Temperature-Entropy plot, 'Ph' for a Pressure-Enthalpy
    * axis : An axis to use instead of the active axis
    * skipPlot : If True, won't actually plot anything, just print COP

    """
    T=np.zeros((8))
    h=np.zeros_like(T)
    p=np.zeros_like(T)
    s=np.zeros_like(T)
    T[1]=Tmax
    T[3]=Tmin
    Te = T[1] - DTsh
    Tc = T[3] + DTsc
    pe=PropsSI('P','T',Te,'Q',1,Ref)
    pc=PropsSI('P','T',Tc,'Q',1,Ref)
    h[1]=PropsSI('H','T',T[1],'P',pe,Ref)
    s[1]=PropsSI('S','T',T[1],'P',pe,Ref)
#    T2s=newton(lambda T: PropsSI('S','T',T,'P',pc,Ref)-s[1],T[1],maxiter=100)
    T2s=PropsSI('T','S',s[1],'P',pc,Ref)
    h2s=PropsSI('H','T',T2s,'P',pc,Ref)
    h[2]=h[1]-((h[1]-h2s)/eta_a)
    T[2]=PropsSI('T','H',h[2],'P',pc,Ref)
```

```
    s[2]=PropsSI('S','T',T[2],'P',pc,Ref)

    sbubble_c=PropsSI('S','P',pc,'Q',0,Ref)
    sdew_c=PropsSI('S','P',pc,'Q',1,Ref)
#    sbubble_e=PropsSI('S','P',pe,'Q',0,Ref)
    sdew_e=PropsSI('S','P',pe,'Q',1,Ref)
    h[3]=PropsSI('H','T',T[3],'P',pc,Ref)
    s[3]=PropsSI('S','T',T[3],'P',pc,Ref)
    h4s=PropsSI('H','S',s[3],'P',pe,Ref)
    h[4]=h[3]+((h4s-h[3])/0.7) #after pump

    p=[np.nan,pe,pc,pc,pe,pe]
    COP=(h[1]-h[4])/(h[1]-h[2]) #evap over exp
    COPH=(h[2]-h[3])/(h[1]-h[2]) #cond over exp

    hsatL=PropsSI('H','T',Te,'Q',0,Ref)
    hsatV=PropsSI('H','T',Te,'Q',1,Ref)
    ssatL=PropsSI('S','T',Te,'Q',0,Ref)
    ssatV=PropsSI('S','T',Te,'Q',1,Ref)
#    vsatL=1/PropsSI('D','T',Te,'Q',0,Ref)
#    vsatV=1/PropsSI('D','T',Te,'Q',1,Ref)
    x=(h[4]-hsatL)/(hsatV-hsatL)
    s[4]=x*ssatV+(1-x)*ssatL
    T[4]=x*Te+(1-x)*Te

    h[5]=PropsSI('H','P',pe,'Q',0,Ref) #befor evap
    #h5=h(P5,x=0),
    h[6]=PropsSI('H','P',pe,'Q',1,Ref) #after evap
    #h6=h(P6,x=1),
    #h10?h13=cp(T10?T13),
    Tht2 =THTin
    Tht3 = THTut
    hMEin = Tht2*PropsSI('C', 'T',Tht2,'P',3.2e5, 'water')
    hMEut = Tht3*PropsSI('C', 'T',Tht3,'P',3.2e5, 'water')
    #m?1=m?10(h10?h13)/(h1?h4),
    vdot=160.0/3600 #m3/s
    mdotHT=vdot*PropsSI('D','T',Tht2,'P',3.2e5, 'water')
    mdotwf =mdotHT * ((hMEin-hMEut)/(h[1]-h[4]))
    #h11=h10?m?1(h1?h6),
    hME11=hMEin-(mdotwf*(h[1]-h[6]))
    #h12=h11?m?1(h6?h5),
    hME12=hME11-(mdotwf*(h[6]-h[5]))
    #T11=h11/cp,
    TME11=hME11/PropsSI('C', 'T',Tht2,'P',3.2e5, 'water')
    #T12=hME12/cp,
    TME12=hME12/PropsSI('C', 'T',Tht3,'P',3.2e5, 'water')
    h[7]=h[1]
    s[7]=s[1]
    T[7]=T[1]

    #Qme = (1620+1860)*1000
    mCpHT = (PropsSI('C', 'T',Tht2,'P',3.2e5, 'water')
    +PropsSI('C', 'T',Tht3,'P',3.2e5, 'water'))/2
    Qme=mdotHT*mCpHT*(Tht2-Tht3)
    mdotwf = Qme/(h[1]-h[4])
    Wexp = mdotwf*(h[1]-h[2])
    #mRhoPump = (PropsSI('D','T', T[3], 'P',pe, Ref)+PropsSI('D','T', T[4], 'P',pc,Ref))/2
    #Wwfp = ((pe-pc)*mdotwf*(1/mRhoPump))/0.7 #70% eta ->kw
    Wwfp = mdotwf*(h[4]-h[3])
    Pnet = (Wexp - Wwfp) * 0.98 # eta gen 98%
    #k = PropsSI('L','T',Tc,'P',pc,Ref)
    #pref = Pnet *k
    rho_exp_out = PropsSI('D','H',h[2],'S',s[2], Ref)
    cost_pref = (Pnet*rho_exp_out)/1e6
    #J. Bao - higher density at turbine outlet
    #negatively impact size parameter => high rho_exp_out = cheap expande

    if pc > 1e5 and Pnet > 0 and pe < 16e5 and Wexp > 0 and Wwfp > 0:
       return  np.array(["candidate",Ref,Pnet,cost_pref,rho_exp_out,
       COP,COPH,pe, pc, Te, Tc, DTsh,DTsc,Wexp,mdotwf,Wwfp,PropsSI('GWP100',Ref),Qme])
    else :
        return np.array(["Fail test",Pnet,Ref,pe, pc, T[1], T[2],T[4], mdotwf,Wexp,Wwfp,Qme])
    if skipPlot==False:
        if axis==None:
            ax=matplotlib.pyplot.gca()
        if Ts_Ph in ['ph','Ph']:
            ax.plot(h,p)
        elif Ts_Ph in ['Ts','ts']:
            s=list(s)
            T=list(T)
            s.insert(5,sdew_e)
            T.insert(5,Te)
            s.insert(3,sbubble_c)
            T.insert(3,Tc)
            s.insert(3,sdew_c)
            T.insert(3,Tc)
            ax.plot(s[1::],T[1::],'b')
        else:
            raise TypeError('Type of Ts_Ph invalid')
results = []
failiurs = []
for test in tests:
    try:
        results.append(SC(test[0],test[1],test[2],test[3],
        test[4],test[5],test[6],test[7],Ts_Ph='Ph',skipPlot=False,axis=None))
```

```
    except ValueError:
        failiurs.append(test)
```

A.2 Candidate sorting and scaling script

```
from __future__ import division
import numpy as np
import matplotlib
#topcandidates in order of Pnet * rho_out
data = np.loadtxt('data.csv', delimiter=';', skiprows=1)
modelloutput =np.loadtxt('output.csv', delimiter=';', skiprows=1,dtype=(str,float), usecols=(0,1))
me_rpm = np.array(data[:,-2])
me_kW = np.array(data[:,0])/2
load = me_kW/6480
sum_me_kW = np.sum(me_kW)
loadScale=np.mean(load)
duration = len(me_kW)*10/60/24 #sample size in days
kw_h = sum_me_kW/duration/24
engineScale = 6480.0/9000.0
#orcOutput = np.array(modelloutput[:,1],dtype=float)
orcOutput = np.array(newcandidates)
orcOutputScaled = orcOutput/1000 * engineScale * loadScale # mean scaled orc out put.
scalefactor = engineScale * loadScale
topcandScaled = orcOutputScaled[0:9]/1000
def output():
    for i in orcOutputScaled:
        print i
```