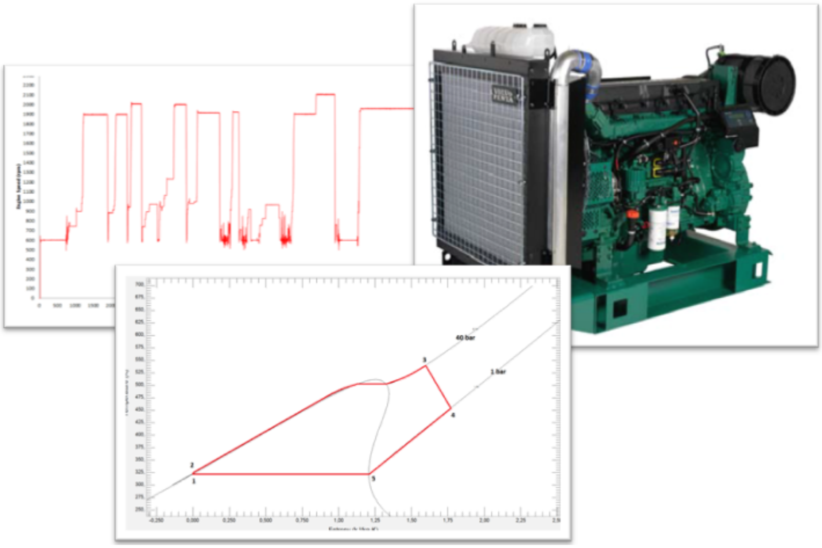
CHALMERS
UNIVERSITY OF TECHNOLOGY



Waste Heat Recovery in Diesel Engines using the Organic Rankine Cycle

Potential of Heat Recovery in the Volvo D13 Engine for Marine and Standby Power Generation Applications

ANIL GOPINATH BHARADWAJ
WAEL AL HAJI

Department of Mechanics and Maritime Sciences,
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2020

Master's thesis in Automotive Engineering

Waste Heat Recovery in Diesel Engines using the Organic Rankine Cycle

Anil Gopinath Bharadwaj
Wael Al Haji

Department of Mechanics and Maritime Sciences
Division of Combustion and Propulsion Systems
Chalmers University of Technology
Gothenburg, Sweden 2020

Waste Heat Recovery in Diesel Engines using the Organic Rankine Cycle
ANIL GOPINATH BHARADWAJ
WAEL AL HAJI

Examiner: Sven B Andersson, Prof. Division of Combustion and Propulsion Systems, Chalmers University of Technology
Supervisors: Jelmer Rijpkema, Chalmers University of Technology; Patrik Landberg, Volvo Penta AB; Björn Wessman, Volvo Penta AB;

Master's Thesis in Automotive Engineering
Department of Mechanics and Maritime Sciences
Division of Combustion and Propulsion Systems
Chalmers University of Technology
SE-412 96 Gothenburg
Telephone +46 31 772 1000

Cover: Pictures showcasing the engine which WHR system is used for, operating points of interest and a T-s diagram of cyclopentane.

Printed by Department of Mechanics and Maritime Sciences
Gothenburg, Sweden 2020

Waste Heat Recovery in Diesel Engines using the Organic Rankine Cycle
Potential of Heat Recovery in the Volvo D13 Engine for Marine and Standby Power Generation Applications
Thesis for the degree of Masters in Automotive Engineering/Marine Engineering
ANIL GOPINATH BHARADWAJ, WAEL AL HAJI
Department of Mechanics and Maritime Sciences,
Division of Combustion and Propulsion Systems
Chalmers University of Technology

Abstract

The current global warming situation and various other environmental challenges call for innovative and easy-to-implement technologies to help counter the effect. One of the ways is to increase the efficiency of already existing systems to help push out more work for the same amount of energy input. In some internal combustion engines, roughly 30% of the input energy could be lost in the exhaust gases with an additional 30% wasted as heat. This wasted energy can be partly recovered using waste heat recovery(WHR) technologies. The recovered energy is then converted to electrical or mechanical energy for further use. The aim of the project is to study the feasibility of waste heat recovery in a Volvo D13 diesel engine with respect to fuel consumption using an organic Rankine cycle for marine and power generation applications. The project also includes brief evaluation of various available alternative WHR technologies. A 0D analysis was conducted using Matlab to select an appropriate working fluid based on the maximum power output and 1D simulations were done with the help of GT-Power and Matlab/Simulink. The operating points were selected based on field test data for both the applications. The 0D and 1D simulations showed that a considerable amount of energy can be recovered from both the applications. The marine and the power generation application yielded roughly 7% of the engine output power from the 1D simulations done using GT-Suite. The two applications had about 9% increase in power output from the 1D simulations done using Simulink. This increase in turn contributes to higher fuel savings hence lowering emissions in general. The 1D simulations highlighted various parameters such as pump speed and expander speed which affected the process. The results from 0D and 1D simulations were also compared for both the applications. The 1D simulations using Simulink had the highest power output and the cause of difference in the power output from other methods were discussed. From the results, it was then concluded that the inclusion of a WHR unit for the Volvo D13 engine has potential and further investigation and fine tuning of this setup will prove advantageous in helping combat the environmental challenges being faced today.

Keywords: Waste Heat Recovery; Volvo D13; Organic Rankine Cycle; Diesel Engine; Recuperator; 0-D Simulations; 1-D Simulations; Marine Engine; Standby Power Generation Engine

Acknowledgements

The complete duration of the thesis work was carried out at the Computer Aided Engineering department at Volvo Penta, Gothenburg between the period of January 2019 to August 2019. We would like to take this opportunity to thank everyone who has helped us to complete the thesis successfully.

We would like to thank our thesis examiner Professor Sven B Andersson at the Division of Combustion and Propulsion Systems for his support and encouragement. The thesis would not be possible without our university supervisor, Jelmer Rijpkema and we thank him for guiding us through his expertise and being patient with us through our mistakes.

The highly experienced team at Volvo Penta were of immense help all throughout the project and we would like to thank Patrik Landberg for being a very understanding and helpful supervisor, Prajwal Shankaregowda for being there and sacrificing much of his time to guide us through our work and Björn Wessman for helping us through the concepts and making sure we never felt out of place. The project would only be half complete if not for the generous help from Thomas Reiche and Francesco Galuppo from Volvo GTT in providing us with the necessary models we required.

Finally, we would like to thank everyone from the department of Computer Aided Engineering for the help and support and guiding us through their vast knowledge and also for the very tasty fikas.

Anil Gopinath Bharadwaj & Wael Al Haji, Gothenburg, August 2019

Contents

Nomenclature

h	Specific Enthalpy (J/kg)
m	Mass (kg)
$\dot{m}$	Mass Flow Rate (kg/s)
P	Pressure (Pa)
$\dot{Q}$	Heat Transfer Rate (W)
s	Specific Entropy (J/kg/K)
t	Time (s)
T	Temperature (K)
V	Volume (m^3)
$\dot{V}$	Volume Flow Rate (m$^3/s$)
$\dot{W}$	Power (W)
η	Efficiency
ρ	Density (kg/m^3)

Subscripts

$coolin$	Coolant In
$coolout$	Coolant Out
D	Displacement
$evap$	Evaporator
$exhout$	Exhaust Out
exp	Expander
th	Thermal
v	Volumetric
wf	Working Fluid

Abbreviations

CAC	Charge Air Cooler
CAE	Computer Aided Engineering
CFC	Chlorofluorocarbons
CP	Critical Point
EMEP	European Monitoring and Evaluation Programme
GWP	Greenhouse Warming Potential
NOISH	National Institute for Occupational Safety and Health
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
WHR	Waste Heat Recovery

1
Introduction

This chapter goes through the main motivation behind the thesis. It gives a brief look at the aim, scope and the limitations of this thesis work.

1.1 Background

The combustion engine has been a leader in powering the industrial, automotive and the marine sector for well over a hundred years. With increasing development, the energy hungry processes world-wide, and the immediate threat of global warming, steps to increase the efficiency of the combustion engine have been going on for quite some time. Increasing the efficiency has many benefits for the engine itself and the system using it. Benefits such as increased unit power and the reduction in fuel consumption have pushed for finding more innovative ways to get that extra percentage of efficiency out of the engine.

Despite the engineering developments in the internal combustion engine during the recent years, a significant part of the fuel energy is still wasted in the exhaust and in cooling the engine. Many ways of recovering this lost energy has been developed over the past century or so. Technologies like turbochargers look at diverting this energy back into the engine in hopes of pumping out extra power. One of the ways to increase the efficiency of the engine is to tap into this wasted heat and try to recover it. This is where the waste heat recovery systems come into play. A waste heat recovery unit is one that transfers heat energy from a process output and utilizes it for another process, thereby increasing the efficiency. The energy which is recovered from the heat sources in a combustion engine can be further utilized by two main ways: either as electrical energy and stored in batteries or as extra mechanical energy to the engine shaft.

1.2 Aim

The aim of the project is to study the potential of a waste heat recovery unit in a Volvo D13 diesel engine with the main focus on increasing the power output of the unit, using an organic Rankine cycle for marine and power generation applications. The project will also include a brief evaluation of various alternative waste heat recovery technologies available on the current market.

1.3 Scope

In this thesis work, an in-depth study of the theory and working principle of an organic Rankine cycle and the components involved was carried out. Due to this being a limited resource project, some conditions were defined in the beginning of the work:

- The thesis work only focused on the theoretical simulation results and not on any results from an experimental setup.
- The thesis work focused only on the waste heat recovery and hence, the simulations were done with the WHR model running as a stand alone unit.
- The simulations were run for a fixed mass flow rate and temperature of exhaust gas for both the applications. In other words, the simulations were run only for steady state engine operation points.

1.4 Limitations

- Only the marine and the power generation application of the Volvo D13 are considered for waste heat recovery.
- Only the high temperature exhaust gas is used as the energy source.
- After the fluid selection, only one fluid which has a higher power output is utilized in all the simulations and no comparison of the fluids in 1D simulations are done.
- The heat exchanger maps were limited to the ones used in the existing models.
- The Matlab 0D model was only run without a recuperator.
- The Simulink 1D model was only run with a recuperator.

2
Theory

2.1 Waste Heat Recovery

With increasing pressure due to environmental concerns, energy industries around the world are now forced to look at greener options if they want to stay ahead of the market. From Figure 2.1, it can be seen that more than half of the fuel energy is wasted in some form of heat. Due to the high amount of energy which would otherwise be expelled, methods to extract this energy are continuously being developed and improved. In this thesis work, the complete focus was towards wasted heat energy recovery from the exhaust gases as it provides a high temperature source.

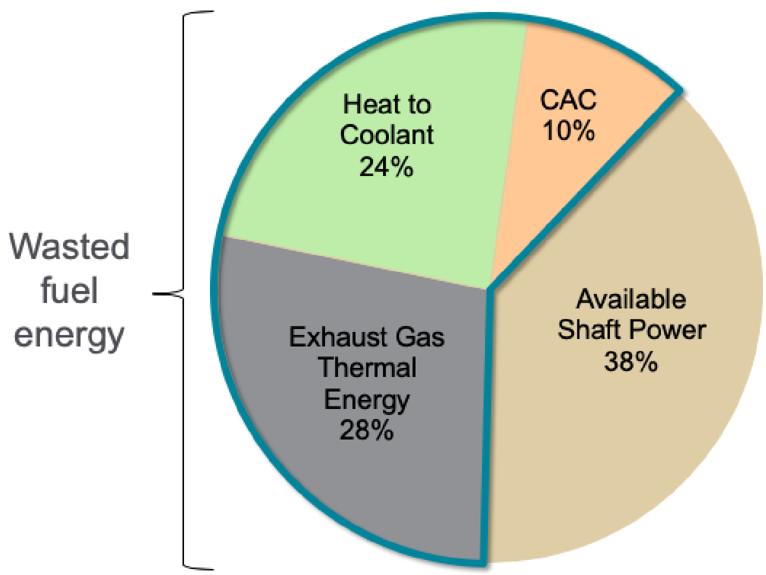


Figure 2.1: A pie chart showing the fuel energy split in a typical heavy duty diesel engine with charge air cooling. Courtesy: Volvo internal document

The concept behind many of the technologies are not new. Concepts which can extract energy from the blow down process of a combustion engine such as two-phase systems (used in steam engines from the 17th century [1]) and turbo compounds (which have been used since the 1940s [2]) are still used in many applications varying from energy generation to small scale refrigeration.

2.2 Rankine Cycle

Named after the renowned Scottish mechanical engineer William John Macquorn Rankine, this cycle is a closed two-phase cycle primarily used with water as the working fluid. For this cycle, the higher the temperature, the better is the power generation. The cycle usually comprises of a boiler (known here as an evaporator), an expander (shown in (a) of Figure 2.2) which generates work, a condenser which facilitates phase change from vapour to liquid, and then a pump to pressurize the liquid before the evaporator.

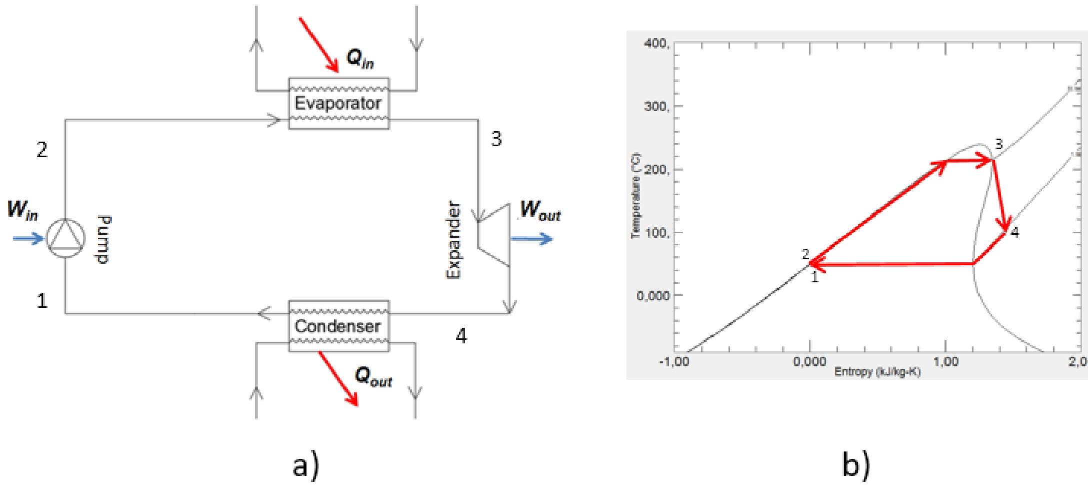


Figure 2.2: (a) Schematic representation of an ideal Rankine cycle (b) T-s diagram of an ideal Rankine cycle [3]

An ideal Rankine cycle is one wherein the friction losses, some pumping losses and irreversible properties of the four components are neglected. Usually, when referring to a Rankine cycle, a steam Rankine cycle is what is considered. This is mainly used for power generation in coal and nuclear reactors which use water as the working fluid. Water has certain properties such as cheap, easy availability, stability at high temperatures etc., which make it the ideal fluid of choice in large scale power generation stations. The processes involved in an ideal Rankine cycle can be better understood with the help of a Temperature Vs Entropy diagram for water as shown in (b) of Figure 2.2. The four processes involved are:

- **Compression**: In step 1-2, there is isentropic compression where the pump increases the pressure in the closed system. Here there is work done on the system (W_{in}).
- **Heat addition**: Step 2-3, a constant pressure heat addition from the evaporator occurs (Q_{in}). This is where the phase change occurs from liquid water to vapour steam.
- **Expansion**: In step 3-4, there is isentropic expansion where the expander generates work (W_{out}).
- **Condensation**: Step 4-1, the isobaric process of condensation from gaseous steam to liquid water (Q_{out}).

In order to calculate the power recovered from heat source in an ideal Rankine cycle, the power output and energy should be calculated for each component.

The heat added to working fluid in the evaporator can be calculated with equation:

$$\dot{Q}_{in} = \dot{m}_{wf} \cdot (h_3 - h_2) \tag{2.1}$$

And the heat rejected from the heat source to the evaporator can be calculated by:

$$\dot{Q}_{source} = \dot{m}_{source} \cdot (h_{in} - h_{out}) \tag{2.2}$$

Since $\dot{Q}_{in} = \eta_{evap} \cdot \dot{Q}_{source}$, working fluid mass flow can be calculated by:

$$\dot{m}_{wf} = \dot{m}_{source} \cdot \frac{(h_{in} - h_{out})}{(h_3 - h_2)} \cdot \eta_{evap} \tag{2.3}$$

where, η_{evap} is the evaporator efficiency.

Similarly for condenser, the heat rejected from working fluid in condenser can be calculated by:

$$\dot{Q}_{out} = \dot{m}_{wf} \cdot (h_4 - h_1) \tag{2.4}$$

And the heat rejected to the coolant in condenser can be calculated from:

$$\dot{Q}_{coolant} = \dot{m}_{coolant} \cdot (h_{coolout} - h_{coolin}) \tag{2.5}$$

where, h_{coolin} and h_{coolin} are the enthalpy of coolant input and output.

The power (useful work transfer) required by the pump can be found from:

$$\dot{W}_{in} = \dot{m}_{wf} \cdot (h_2 - h_1) \tag{2.6}$$

The power output (useful work transfer) by expander can be calculated by:

$$\dot{W}_{out} = \dot{m}_{wf} \cdot (h_3 - h_4) \tag{2.7}$$

The net power output from an ideal Rankine cycle can be calculated with the equation:

$$\dot{W}_{net} = \dot{W}_{out} - \dot{W}_{in} \tag{2.8}$$

2.2.1 Real Rankine cycle

In a non-ideal Rankine cycle, parameters such as friction losses and the physical construction of all the components are considered. In such a case, referring to Figure 2.3, it can be seen from curves 1-2a and 3-4a that the pump and the expander do not operate isentropically, and hence these need to be considered when the net work done is being calculated. It is also noted that there are pressure losses in the condenser and the evaporator causing a pressure drop in the system.

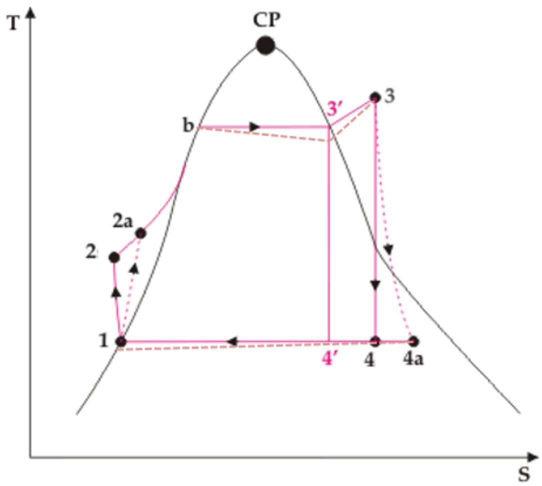


Figure 2.3: T-s diagram of an actual steam Rankine cycle

Referring to the same diagram, power required by the pump is calculated by:

$$\dot{W}_{in} = \dot{m}_{wf} \cdot (h_{2a} - h_1) = \dot{m}_{wf} \cdot \frac{(h_2 - h_1)}{\eta_{pump}} \tag{2.9}$$

And the power output by the expander is calculated by:

$$\dot{W}_{out} = \dot{m}_{wf} \cdot (h_3 - h_{4a}) = \dot{m}_{wf} \cdot (h_3 - h_4) \cdot \eta_{exp} \tag{2.10}$$

Where, η_{exp} and η_{pump} are the combined (mechanical and isentropic) efficiencies of expander and pump respectively.

2.2.2 Organic Rankine Cycle

A Rankine cycle with water as the working fluid is most effective when the heat source has high temperatures. This type of system is usually used in large scale power generation plants due of the large scale availability of water. For a lower temperature input, to utilize the steam Rankine cycle is challenging due to the physical limitations of water. Water has a relatively high boiling point when compared to other fluids such as cyclopentane and ethanol and requires higher energy input to change phase from liquid to gas. And a system using water as the working fluid for smaller, lower power output applications have difficulties with an efficient, reliable, and a cost effective steam expander [4]. For such an application with both low and high temperature heat source, organic fluids are used as the working fluid and such a cycle is called an organic Rankine cycle. For an organic Rankine cycle (ORC), the working fluid selection is of one of the most important parameters as this directly dictates how the cycle performs. Instances where the heat source is not in the ideal water operating temperatures, for example, 500°C or less, other working fluids are selected with low vaporization temperatures which can operate with the same underlying construction of a steam Rankine cycle.

The history of using a working fluid other than water dates back to as early as 1826 where Ofeldt used naphtha as an organic fluid [5]. Since then, technological advancements have helped to develop various working fluids, each suitable for a specific application. ORC is well suited for extracting energy from low heat sources such as geothermal energy and many plants across the world such as the Ngatamariki geothermal plant in New Zealand shown in Figure 2.4 producing 25MW$_e$ utilize this.

Figure 2.4: Aerial view of the Ngatamariki geothermal plant installed in New Zealand. Courtesy: Mighty River Power

Some of the real world uses of the ORC systems are from the following heat sources ranked by their current power outputs [4]:

- Geothermal reservoirs
- Solid biomass combustion
- Exhaust gas from gas turbines or combustion engines

Reports claim that as much as 20,000 GWh of thermal energy can be recovered from industrial waste heat per annum [6] which in turn helps in reducing the carbon emissions of industries. Majority of this heat is between 60°C and 350°C which makes ORC the best suited for extracting this energy [6].

2.2.3 Mass Conservation and Energy Balance

In an organic Rankine cycle, the mass of the working fluid across each component remains constant as in the equation:

$$m_{in} - m_{out} = \triangle m_{system} \tag{2.11}$$

where, m_{in} is the mass into the component, m_{out} is the mass out of the component, and m_{system} is the total change in mass in the system.

The mass flow rate in a component of the organic Rankine cycle has to be equal to that in the next component, hence, $\dot{m}_{pump} = \dot{m}_{evap} = \dot{m}_{exp} = \dot{m}_{condenser}$ where $\dot{m}$ is the mass flow rate. The energy balance in the organic Rankine cycle can be evaluated by equations 2.1, 2.4, 2.6, and 2.7. Based on these equations, the energy balance in the closed system can be defined by the equation:

$$\dot{Q}_{in} - \dot{W}_{out} - \dot{Q}_{out} + \dot{W}_{in} = 0 \tag{2.12}$$

2.2.4 Thermal Efficiency

The thermal efficiency of a cycle is defined as the percentage of heat energy that is transformed into work. In other words, the thermal efficiency is a measure of the heat wasted while transforming the heat energy into work. Thermal efficiency is defined by the equation below:

$$\eta_{th} = \dot{W}_{out}/\dot{Q}_{in} \tag{2.13}$$

Considering the energy balance equation

$$\dot{W}_{out} = \dot{Q}_{in} - \dot{Q}_{out} + \dot{W}_{in}$$

The thermal efficiency can also be represented by:

$$\eta_{th} = 1 - (\dot{Q}_{out} + \dot{W}_{in})/\dot{Q}_{in}$$

2.3 Methods to Increase the Efficiency of a Rankine Cycle

A Rankine cycle is limited by the temperature difference between the hot and the cold source. There are a few ways through which efficiency of the cycle can be increased.

2.3.1 Recuperator

An organic Rankine cycle with a recuperator utilizes the high temperature of the working fluid after the expander. An additional heat exchanger is added to the cycle (as shown in Figure 2.5) and heat energy is then transferred after the fluid is pressurized in the pump. The thermal efficiency of the cycle could be increased by such a process. Experiments have shown that recuperators might be beneficial in heavy duty applications with high temperature heat source but they depend on the working fluid chosen [7].

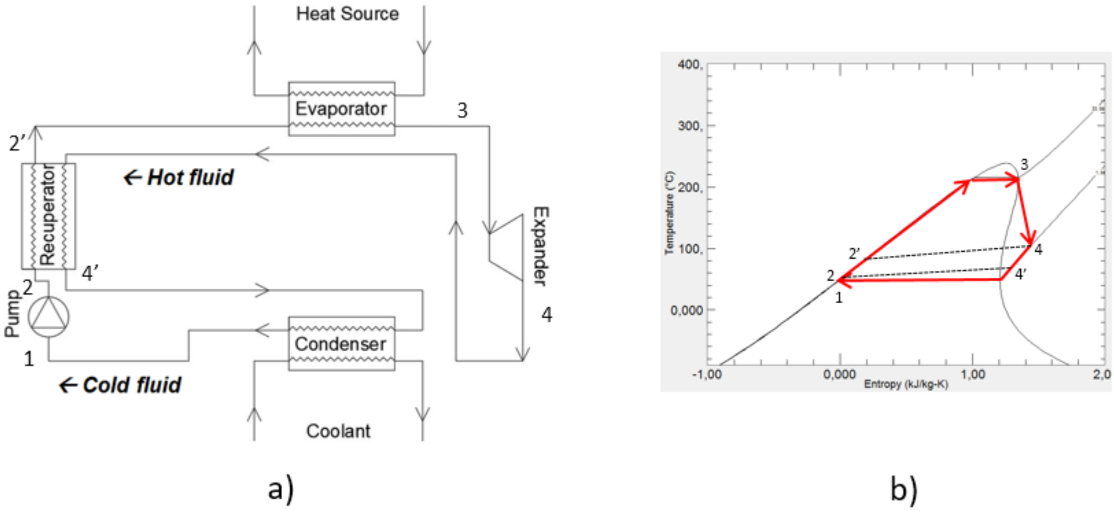


Figure 2.5: (a) Schematic representation of a recuperated Rankine cycle (b) T-s diagram of a Rankine cycle with recuperator

2.3.2 Superheating

Depending on the type of fluid, another way of increasing efficiency of a Rankine cycle is by superheating. This method is one where the working fluid is heated beyond its vaporization temperature. Superheating may ensure that there is vapour state after the expansion process which is necessary for the expander. Ensuring vapour state through the expander is beneficial and results in higher efficiency in the expander. In order for superheating to occur, the flow rate in the system could be reduced from its original value or the heat input into the system could be increased for the same mass flow rate.

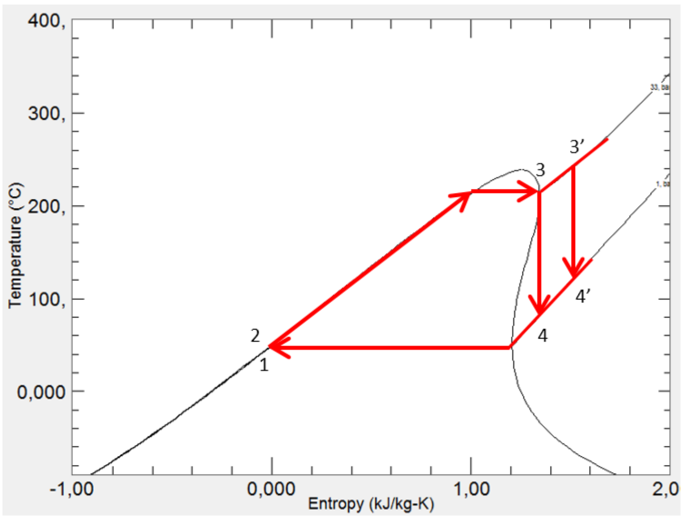


Figure 2.6: A T-s diagram of an organic Rankine cycle with superheating using cyclopentane as the working fluid

As the work done by the system ($\dot{W}_{out}$) increases, from Equation 2.13, this could have a positive effect on the thermal efficiency. As seen in Figure 2.6, the area between the curve 1-2-3-4 which represents the specific work done by the system increases to 1-2-3'-4'.

2.4 Types of Fluids

As stated earlier, ORCs can operate with various organic fluids. Depending on the temperature range of operation, an appropriate working fluid can be selected. The fluids in general can be subdivided into three main types: wet fluids, isentropic fluids and dry fluids.

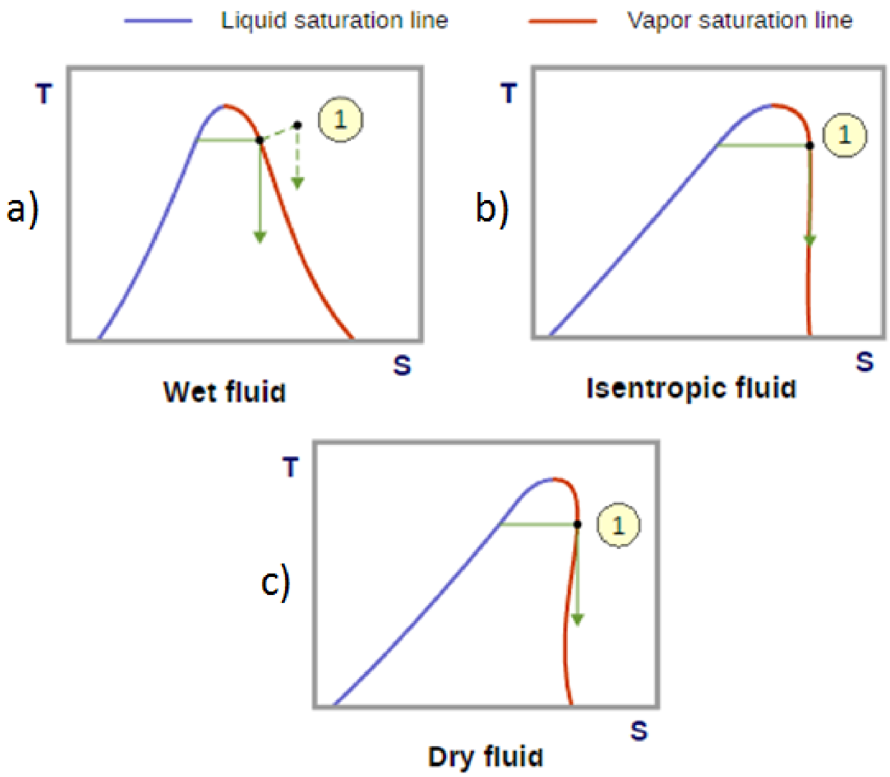


Figure 2.7: T-s diagram of fluids (a) Wet, (b) Isentropic and (c) Dry. Credits: dieselnet.com

The classification of fluids into wet, isentropic and dry is on the basis of the slope of vapour saturation curve as shown in Figure 2.7. Figure 2.7 (a) shows a wet fluid with positive slope, (b) shows a T-s diagram of an isentropic fluid with nearly infinitely large slope, and (c) showing a dry fluid with a negative slope in their vapour saturation curve.

The fluid type chosen will dictate the performance of the system. Dry and isentropic fluids have the advantage of not needing superheating for them to have total vaporization after the expansion device. Wet fluids on the other hand, have a positive slope in the vapor saturation curve which means that after the expansion process, to ensure there is still vapor state after the expansion process, there needs to be superheating. Examples of wet fluids are water and ammonia. Cyclopentane is an example of dry fluid. Though seldom used, some examples of isentropic fluids are R11 refrigerant and fluorinal 85.

2.5 Expanders

An efficient expansion device is usually the difference between a good and an average Rankine system. The chosen technology will depend on the operating conditions and also the size of the Rankine system. There are two main types of expanders: volume type and velocity type expanders. There are other types of expanders like the scroll expander, screw expander etc. They have their own unique area of operation for which they are best suited for. But for this thesis, only the axial turbine expander and the piston expander (specifically, the swash plate piston expander) are used and hence, discussed about.

2.5.1 Volume Type

Volume type expanders such as the piston expanders are widely used in waste heat recovery from an internal combustion engine [8]. Volume type expanders have the advantage of tolerating a percentage of liquid in the expansion chamber. Piston expanders can handle high pressure ratios. Along with low cost of production and their need for a lower operating speed, piston expanders are a good choice as an expansion device [9]. However, reciprocating expanders have some drawbacks:

1. Precise timing necessary for inlet and exhaust valves.
2. The piston itself requires lubrication for its operation.
3. Because this is a reciprocating machine, it required primary and secondary balancing to reduce noise and vibration.
4. Due to its construction, piston expanders have higher friction losses when compared to a velocity type expanders.

2.5.2 Velocity Type

The most popular example of a velocity type expander is the turbine expander. Velocity type expanders like the turbine expander are best suited for steady state operations and are less efficient in off-design cases. The two types of turbines are axial and the radial inflow turbine. According to S. Quoilin et al [10], axial turbines have the advantage of being used in multiple stages but also points out that an ORC does not have the need for a multi stage expander when compared to a steam Rankine cycle.

Turbine expanders used in a steam Rankine cycle have little to no difference with turbine expander used in an ORC. But some research has shown that a few crucial parameters influence the behaviour of a turbine expander [9]:

1. Turbines in general are designed to have high efficiency at a set operating point. The main difference between an ORC and a steam Rankine cycle is the

enthalpy drop for steam is higher during the expansion process. Due to this, a single stage turbine can be employed for an ORC system.
2. Due to organic fluids usually having higher densities and lower specific volume than water, turbine dimensions could be smaller [9].
3. The smaller size of a turbine expander could be affected by drastic overspeed during load shedding where there is a sudden loss in power demand.

Figure 2.8: An axial turbine expander. Credits: KBB Turbo

The advantages of a velocity type expander include light weight design and high efficiency over a short range of operation. Disadvantages include high cost of manufacturing, and low efficiency in off-design conditions with a low tolerance to liquid state in its operation.

2.6 Working Fluid

In an ORC for waste heat recovery, the working fluid plays a major role in the design and performance of the whole system. From the actual sizing to the thermodynamic efficiency of the cycle, the working fluid influences it all. Some fluid properties were identified to play a crucial role in a waste heat recovery application [8]. A few are shown below:

- **Thermodynamic performance**: Due to the complex nature of how a fluid behaves under different thermodynamic states, an appropriate fluid must be chosen to have the required critical point, latent and specific heat, density etc.
- **Type of fluid**: As discussed in Section 2.4, the shape of the vapour saturation curve will influence the circuit and the components to be used. A wet fluid may require superheating which might influence the heat source output conditions.
- **Impact on the environment**: The measure of how much impact a particular working fluid has is measured using ozone depletion potential (ODP) and

greenhouse warming potential (GWP). Many dangerous fluids like the chlorofluoro carbons (CFCs) have been ruled out since the Montreal protocol [11] and due to this, careful selection of the working fluid is to be made to make it usable in commercial usage.

- **Health hazard**: In the event of a leakage, the working fluid might come into contact with human environment and the health hazards must be considered while selecting one.
- **Availability and cost**: A working fluid which satisfies all the above criteria but is sparsely available and extremely expensive is not one which can be used on a wide scale.

2.6.1 Cyclopentane

Cyclopentane is the main working fluids used in this thesis work. It is a alicyclic hydrocarbon with a chemical formula C_5H_{10}. Extensively used as an insulator in refrigerator walls, it is currently replacing the older R245fa and R134a due to environmental concerns [12]. According to National Institute for Occupational Safety and Health (NIOSH), cyclopentane has much more favourable Greenhouse Warming Potential (GWP) numbers of roughly 10 in comparison to the fluids used in the current industry which stand at around 1000 units [13]. This is mainly due to the fact that cyclopentane does not have any chlorine or fluorine. The higher the GWP, the higher is the impact on the environment leading to an increase in the greenhouse gases. For example, GWP of carbon dioxide measured for 20 years has the baseline of 1 and methane has a GWP of 96 units [14]. The ODP of cyclopentane has been concluded to be 0 [15].

Cyclopentane has a vaporization temperature of 49.4°C at atmospheric pressure which makes it ideal as a working fluid in a medium heat WHR system. Critical temperature is about 240°C at 45.7 bar pressure. Cyclopentane has an auto-ignition temperature of about 320°C. The flash point of this fluid is about -20°C. The decomposition temperature of cyclopentane is around 275°C beyond which the compound itself is unstable [16]. Cyclopentane is non lethal but can cause mild irritation if it comes in contact with eyes or the skin [13].

2.7 REFPROP Software

REFPROP or REFerence fluid PROPerties is a program developed by the National Institute of Standards and Technology to calculate thermodynamic properties of various important fluids and fluid mixtures. This type of software is used also as an extensive library for various fluid properties which can be easily plotted.

The convenience in using this particular software is that it is updated regularly to include the newest fluids and their mixtures. An example of how this software can be used is shown in Figure 2.9 which shows a T-s diagram of water generated using REFPROP. The version used for this thesis was REFPROP Version 9.1.

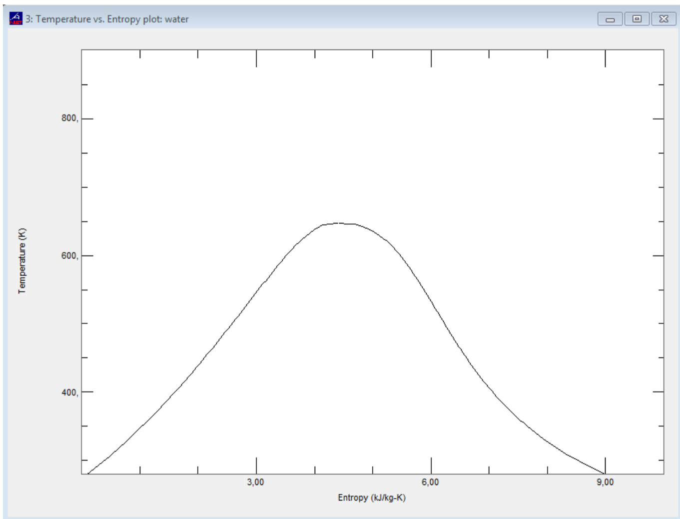


Figure 2.9: T-s diagram of water generated using REFPROP

2.8 GT-Suite Software

GT-Suite software is a physics CAE tool by Gamma Technologies used extensively for 0D/1D/3D simulations of engine and cooling systems. Each engine component can be modelled separately and connected together to mirror a physical system. But unlike a physical setup, this sort of a tool helps in cutting down on development time of a concept or in the investigation of an already existing concept.

The main advantage of this software is that there are many libraries pre-built which helps in integration of various departments like powertrain, electrical, chemical and their individual control systems. For this thesis work, an extended version of GT-Suite was used to build and simulate a waste heat recovery unit. GT-Suite Version 2019 Build 1 was used to create these models.

2.9 Matlab/Simulink Software

Matlab is a software used for solving different types of the technical computing problems. A 0D model can be simulated in Matlab where:

1. The inputs can be written as variables to easily change them later or can be imported from other software.
2. The mathematical equations and functions are inputs to simulate the cycle.
3. The outputs can be shown in different ways such as numbers and graphs and can be exported to other software.

The 0D analysis results give a good estimation for characterizing the components of the 1D model. A set of libraries in Matlab called Simulink is used to create a 1D model. Simulink is a 1D multi-physics simulation tool that has many libraries that can be used to model and connect all the components of the a system. In Simulink, each component is modelled as a box containing all the equations needed for defining this component. For example, in this thesis model, pump, evaporator, expander, condenser, tank, and recuperator were individually modelled. After that, these subsections connect to form a Rankine cycle model that represent a physical setup where the organic working fluid state and the mass flows are computed for each time step in each component, and the results are sent to the adjacent component. The simulation duration in Simulink can be controlled for each process until the model converges.

2.10 Waste Heat Recovery Systems

A brief survey shows that there are many waste heat recovery technologies available for a varied power recovery range. Below is a list of some of these technologies studied solely for comparison purposes:

- Thermoacoustic Convertors
- Turbo-compound
- Rankine Cycle
- Organic Rankine Cycle
- Stirling Engine

As the Rankine cycle and organic Rankine cycle has already been discussed about in Section 2.2, they will not be repeated here and only used for the comparison.

2.10.1 Thermoacoustic Convertors

A thermoacoustic convertor is a WHR device which operates on the principle of resonance to convert heat energy into acoustic waves and subsequently into electrical energy. The convertor uses heat energy (as can be seen in Figure 2.10) arriving from the exhaust to add energy to a 500 Hz acoustic wave in helium, which then amplifies the pressure oscillations. This high powered sonic wave is then sent to an electricity generator module [17].

This type of a WHR unit has been claimed to be have high reliability. Etalim, which is a company that develops thermoacoustic convertors claims that a single unit has a maximum of 2 kW of fixed power output per unit, which is suitable for a smaller automobile like a passenger vehicle or a medium sized truck. Possibility of stacking units next to each other to increase the combined output is an added advantage with such a technology.

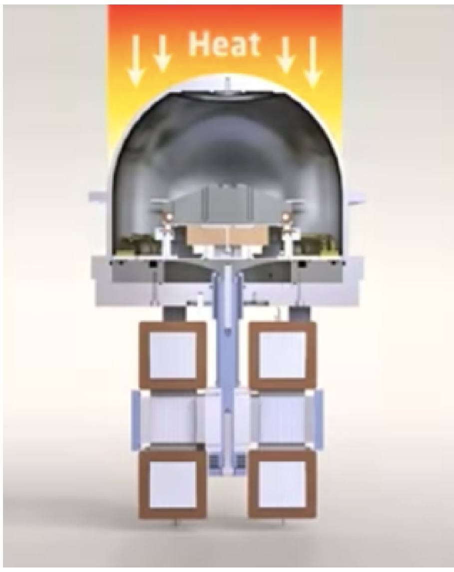


Figure 2.10: A thermoacoustic convertor from Etalim which uses Helium as the working fluid. Credits: Etalim

2.10.2 Turbo-compound

Turbo-compounds are not a new technology. A turbocharger uses exhaust gases to drive the turbine to compress the fresh air into the engine (Shown in Figure A.1). This increases the efficiency of the engine by reducing the pumping loss. Similarly, the turbo-compounds use the same working principle but it uses the exhaust flow to drive a turbine which in turn converts the waste heat energy to either electrical energy or send back mechanical energy to the engine. The technology was first used in the rotary engine for airplanes [18]. Later, automotive companies used this technology to recover energy lost in the exhaust gas. The advantages and disadvantages will be discussed later on in this chapter.

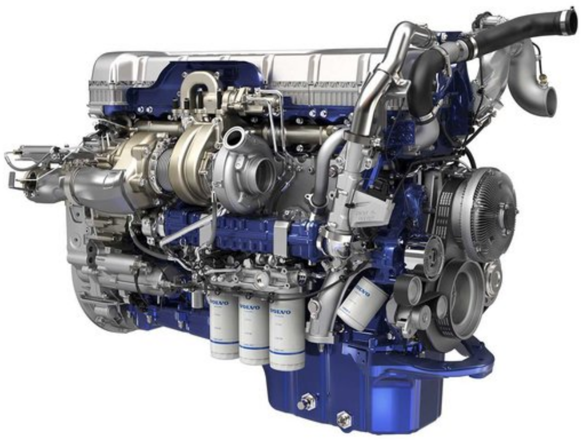


Figure 2.11: Volvo Trucks' VNL series of engines which has a turbo-compound device. Volvo Trucks claim an increase of 7.5% increase in fuel efficiency. Credits: Volvo Trucks

2.10.3 Comparison of Different WHR Technologies

Having had a quick look at the different WHR technologies, each have their own pros and cons. As many of them are seldom used extensively, start up companies which are investigating on the technologies were referred and they were compared. Below is a brief comparison between them:

Table 2.1: Different WHR technologies and their individual advantages and disadvantages [1, 7]

WHR Technology	Pros	Cons
Thermoacoustic convertors (claimed)	Compact design, Helium is easily available	Low power output, Only high temperature circuits, High noise
Turbocompound	Both high and low temperature circuits, Higher power output per unit	High noise, Cost for integration
Rankine cycle	Working fluid availability	Only high temperature circuits, No lubricating property of fluid
Organic Rankine cycle	Varied range of temperature circuits, High power output	Cost of integration

For the two applications at hand, the Volvo D13 engine has varied temperature ranges at which energy is dissipated. Instead of having two different technologies for different temperatures of wasted heat, a single organic Rankine cycle is chosen as the WHR technology of choice and is used in this thesis work. Research which compared different thermodynamic cycles such as Rankine cycle, the organic Rankine cycle, trilateral flash cycle and others deemed that for a similar heat source, they produced similar power output [19]. As this particular setup was built to include a lower temperature circuit for future work, the organic Rankine cycle was selected.

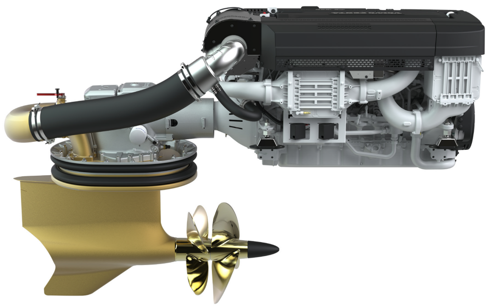
2.11 Volvo D13 Engine

The famed Volvo D13 engine is a versatile combustion engine used in many types of applications depending on how the engine is configured. They are used in marine, industrial power generation, and other industrial applications. The Volvo D13 engine is a 12.8L in-line six engine diesel engine. A version of this engine is also used in Volvo Trucks but that will not be considered for this thesis work.

Due to their engine size and the high temperatures that the exhaust gases have, waste heat recovery seems like a very beneficial addition to the setup to make the engine even more efficient. For this thesis work, the marine leisure application and the industrial power generation will be of interest.

2.11.1 Marine Leisure Application

The marine application engines variant of the D13 engine which is studied here is the D13-1000hp. It has a rated power of about 735 kW for a medium sized yacht.



+

Figure 2.12: The Volvo D13 engine variant used in marine application. Credits: Volvo Penta

Some features and properties of the engine variant used for this application are:

- Popular engine variant used in many yachts because of the high power figures.
- Marine variants have a varied speed range. During its operation, this engine operates at roughly 75% of its load for most of its operation.
- Considering this point, high amounts of energy is wasted in the exhaust which a WHR unit can hope to recover.
- As this is a marine based engine, there is availability of sea water for cooling.

2.11.2 Industrial Power Generation Application

Power generators are commonly used in emergency situations in hospitals and companies or to have extra power when needed. Such an application requires a stable and reliable source of electricity. Volvo D13s are used in such applications with varied range of output possibilities.

Figure 2.13: The Volvo D13 engine used in industrial power generation systems. The variant here is a 1500 rpm 50 Hz unit. Credits: Volvo Penta

Some features of this variant are:

- This engine variant is produced both in 1500 rpm 50 Hz unit and the 1800rpm 60Hz unit for different requirements. The 1500 rpm 50 Hz will be considered for this thesis work.
- During prime operation, the engine has an output of 409 kVa with a standby power increase of 10%.
- High temperature exhaust and the high power output are some of the reasons why a WHR can prove to be most beneficial in such an application.

Table 2.2: Performance figures for Volvo D13 engine used in marine and the standby power generation applications

Application	Marine - Leisure	Power Generation
Max Power (kW)	735.0	500.0
Max Torque (Nm)	2950.0	3000.0
Engine Speed (rpm)	600-2400	1500
Max Exhaust Temperature (°C)	449.0	488.0
Max Exhaust MFR (kg/s)	1.150	0.630

3

Methodology

This chapter will look at the steps taken in designing the organic Rankine cycle starting with the 0D analysis and then moving onto the 1D simulations using Matlab/Simulink and GT-Suite.

3.1 Operating Point

For the marine engine, the operating point of interest was determined by looking at test cycle data for the D13. The engine was run in a typical boat driving cycle in a predetermined route. From the test data it was observed that the operating point at which the engine runs for a majority of the cycle was around the 2200 rpm range, better seen in Appendix A.2. This point is about 75% of the engine load and the exhaust temperatures and mass flows for this point were used for designing the 1D models. The model was also run at various other operating points to simulate different conditions during the test cycle and they are shown in Table 3.1.

Table 3.1: Input conditions for the Volvo D13 engine used in the marine application

Load Point %	Exhaust Temperature (°C)	Exhaust Mass Flow (kg/s)	Engine Speed (rpm)
100	449.2	1.150	2400
75	432.2	0.877	2185
50	347.2	0.680	1920
25	255.2	0.441	1512

For the power generation application, European Monitoring and Evaluation Programme (EMEP) guidelines were used. According to this, the stationary power generation engines [20] belong to Type D2. The D2 test cycle shows that for stationary engines, 75% load has a weightage factor of 0.25. Hence, this was used as the operating point of interest for future simulations. The WHR unit was designed to operate for the exhaust parameters (temperature and mass flow rate) at 75% engine load but the models were also tested for 10%, 25%, 50% and 100% load and this can be seen in Table 3.2 wherein all the load points are run at 1500 rpm.

Table 3.2: Input conditions from the power generation variant Volvo D13 engine at 1500 rpm

Load Point %	Exhaust Temperature (°C)	Exhaust Mass Flow(kg/s)	Engine Power (kW)
100	488.5	0.641	500.0
75	459.0	0.495	375.0
50	431.0	0.380	250.0
25	369.0	0.246	125.0
10	266.0	0.178	50.0

3.2 0D Modelling

The most important step in an organic Rankine cycle design is the fluid selection. Fluid selection is better understood by 0D analysis. The Matlab program had water, ethanol and cyclopentane predefined in the code and these were used for comparison purely from a thermodynamic perspective and this program was originally created by Volvo Trucks. Some of the fluid properties are shown in Table 3.3. REFPROP was used as a function called by the 0D program to obtain the fluid properties.

Table 3.3: Critical point and boiling point of water, ethanol and cyclopentane

Fluid	Critical Point	Boiling Point
Water	374°C at 220 bar	100.0°C at 1.013 bar
Ethanol	240°C at 62 bar	78.4°C at 1.013 bar
Cyclopentane	240°C at 45 bar	49.4°C at 1.013 bar

The steps involved in the 0D analysis were:

- An organic Rankine cycle was created without considering the physical construction of the subsystems.
- The subsystems use efficiency as a parameter to help simulate a realistic cycle where pump and expander efficiencies are assumed 0.65.
- Subcooling of 5°C was input in the condenser before running the analysis.
- REFPROP software is called used as a function to obtain each of the fluid properties such as boiling point, critical temperature and pressure.
- The program then determines each fluid's performance in terms of net output power.
- Fluid selection is done after assessing the output values from the cycle.

Referring to Section 2.6, it discusses about the properties that a working fluid must have in order for it to be used in a real world scenario. Table 3.4 shows the operating points chosen for the two applications where 75% load for the marine system and 75% load for the industrial power generation system were chosen.

Table 3.4: Input conditions for fluid selection for both the industrial and power generation application at the chosen operating points

Application	Exhaust Temperature (°C)	Exhaust Mass Flow (kg/s)	Coolant Temperature (°C)
Marine	432.0	0.877	25.0
Power Gen	459.0	0.494	35.0

The Matlab code was tweaked to mirror the system that would be present in the engine applications. Some changes that were made are:

- Input conditions such as temperature and mass flow rate were changed from the original values.
- The pump, evaporator and expander's efficiency were set to 65%, 90% and 65% respectively.
- The minimum expansion ratio was set to 2.
- Pinch condensation point was fixed to have a 5°C difference.

0D analysis also helps in having a benchmark for 1D analysis. The values of net output power, the mass flow rate from the pump and volume flow rate from the 0D analysis were used later on for comparison with the 1D analysis.

3.3 1D Modelling

1D modelling for WHR was carried out in both Matlab/Simulink and GT-Suite software. The difference in the way the two software handle simulations is what prompted the use of both. Simulink is a model based design approach which depends heavily on how the user programs the logic of the cycle. Hence, the user has more control of the components. The parts will have a user defined discretization and the accuracy of the model will then depend on the input conditions. GT-Suite on the other hand is a standard engine performance simulator which has a wider range of options with components already defined and lesser opportunity to tweak them when compared to Matlab.

3.3.1 1D Modelling in GT-Suite

GT-Power which is a part of the GT-Suite was used for building an organic Rankine cycle model for two applications of the Volvo D13 engine. The model was based off of an example model in GT-Power. The example model was chosen instead of building the model from scratch as this would save time along with providing a baseline working model complete with a recuperator.

Example Model

The actual simulations using cyclopentane as the working fluid was based off of an example model in GT-Power software as shown in figure 3.1. Some properties of this model are:

- The system uses R245fa as the working fluid which is a refrigerant used for low temperature organic Rankine cycles. Due to this, the system component sizing is suited for R245fa along with the pipe dimensions.
- Each individual component is configured to work best with R245fa
  - The pump is a normal PumpRefrig model with a performance map of a physical test run which gives more realistic results.
  - The tank is a ReceiverDryerRefrig model with a capacity to store the working fluid to be later sent into the pump.
  - The condenser is a plate type heat exchanger using ethylene glycol as the coolant.
  - The recuperator is a shell and tube type heat exchanger.
  - The expander used for this model is an axial turbine type with an performance map tuned to R245fa refrigerant specifications.
  - The evaporator is a shell and tube type heat exchanger between the exhaust gases and R245fa in the cycle.

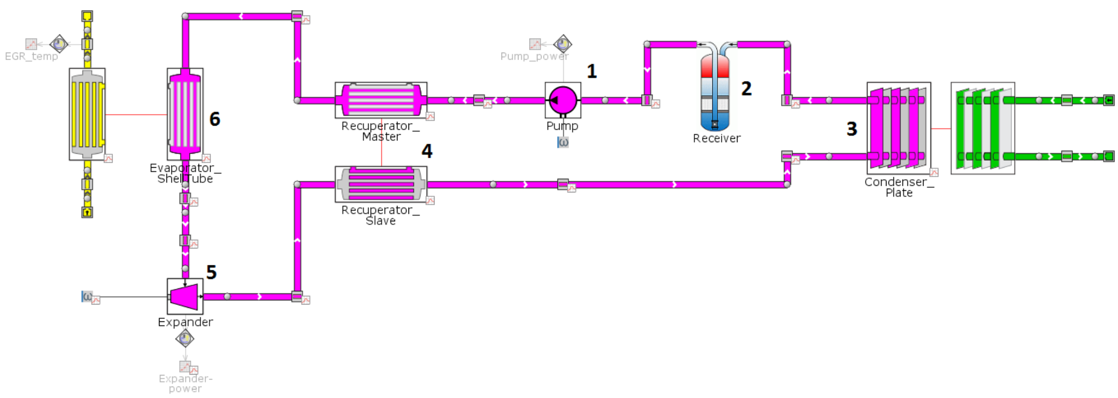


Figure 3.1: Skeleton model of an organic Rankine cycle used as a baseline for further simulations using R245fa as the working fluid. 1-Pump, 2-Tank, 3-Condenser, 4-Recuperator, 5-Turbine Expander, 6-Evaporator

Parts of Organic Rankine Cycle

The organic Rankine cycle model in GT-Power consisted of various components. GT-Suite has the advantage of having an extensive library with pre-defined components and so, the model is not limited in using the specifications of the example model it is based on. The parts are discussed in detail below along with the other alternative models for that component.

Pump

The pump controlled the mass flow in the organic Rankine cycle and there were a number of parameters which could be altered to have a better fitting component. The example model had a performance map from a physical model (Figure 3.2). Some properties of this unit are:

- Figure 3.2 shows the performance map of the pump. The physical pump was tested to run until 2500 rpm. For this speed, other fields were calculated such as volumetric flow, the pressure rise possible, and how they influenced the isentropic efficiency.
- The map is then used for further calculating how the unit behaves at different speeds and how it could be altered to get the desired output pressure.
- For applications which required higher pressure outputs and/or higher volumetric flows, parameters such as 'Pressure Rise Multiplier' and 'Flow Rate Multiplier' were utilized to extrapolate the map and simulate a bigger pump.
- When the pump speed requirement is outside the performance data which was specified, a linear extrapolation method is done by GT-Suite.
- Finally, the pump speed of the unit was altered to suit the cycle.

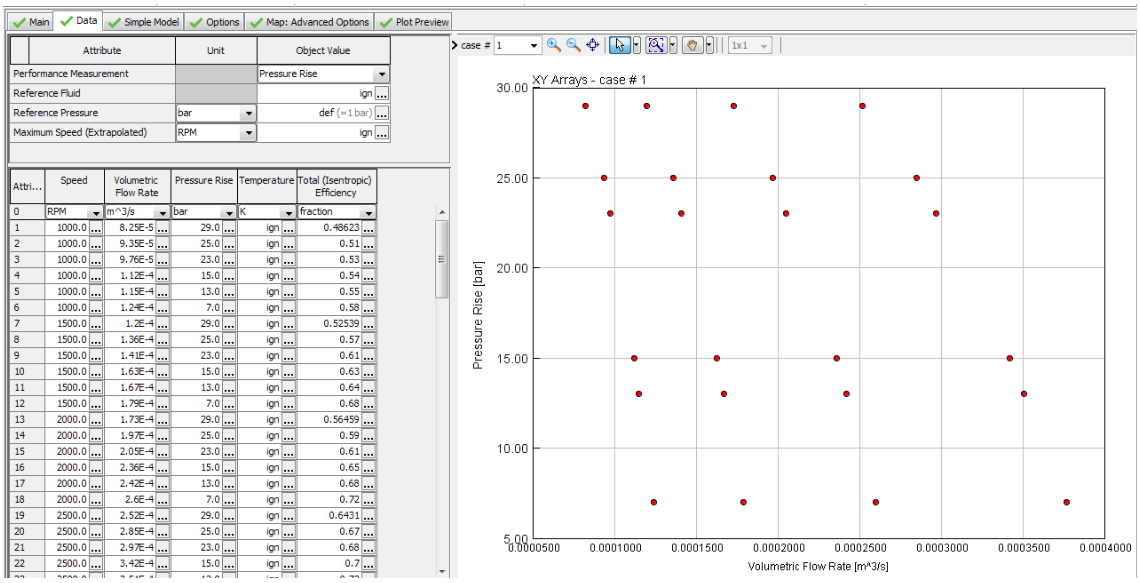


Figure 3.2: The example model pump map depicting performance figures from an actual pump

The compression power ($\dot{W}_{pump}$) required by this pump template is calculated by the equation 3.1. The pressure value is calculated by taking the measurement at the end of each cycle and relaying back the value.

$$\dot{W}_{pump} = \frac{\triangle P \cdot \dot{V}}{\eta_{total}} \tag{3.1}$$

where, $\triangle P$ is the static pressure rise between the inlet and outlet of the pump, $\dot{V}$ is the inlet volume flow rate of the pump, and η_{total} is the total efficiency of the pump.

The shaft mechanical friction $\eta_{mechanical}$ is calculated by the equation 3.2.

$$\eta_{mechanical} = \frac{\dot{m}(h_{out} - h_{in})}{\dot{W}_{shaft}} \tag{3.2}$$

where, $\dot{W}_{shaft}$ is the power transmitted by the pump shaft.

Other alternative pump models available in GT-Suite library which were tested was the 'PumpPosDispRefrig', a simple positive displacement pump. In the absence of a performance map, this type of a pump can be used.

Expander

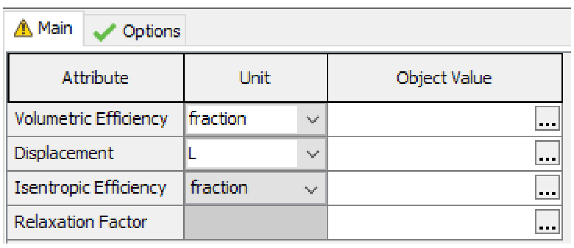
The expander is an integral part of organic Rankine cycle and GT-Suite offers many types. From section 2.4, it is known that there are mainly two types of expanders. The velocity type and the volumetric type. Due to the complexity of defining a volumetric type expander from scratch, a generic type expander was used for simulations. The turbine expander model used in the example model features a performance map. These measurements were specific to R245fa refrigerant and the turbine model will behave specific to that vapour properties. Due to the unavailability of a physical model suited for cyclopentane, 'TurbPosDispRefrig' was used as the expander model. This is a positive displacement generic type expander template which uses input conditions (Figure 3.3) and does not require test data. The outputs of this template are the mass flow rate and the enthalpy change and is governed by the equation:

$$\dot{m}_{wf} = \rho \cdot \eta_v \cdot n \cdot V_D \tag{3.3}$$

Where, ρ is the inlet density, η_v is the volumetric efficiency, n is the rotational speed of the expander, and V_D is the volume of the expander. The change in the enthalpy is calculated by the equation:

$$\Delta h = \Delta h_{isentropic} \cdot \eta_{isentropic} \tag{3.4}$$

The power from the expander is calculated by multiplying the mass flow rate of the working fluid calculated from Equation 3.3 and the enthalpy change which is calculated from the Equation 3.4.


Main Options

Attribute	Unit	Object Value
Volumetric Efficiency	fraction	...
Displacement	L	...
Isentropic Efficiency	fraction	...
Relaxation Factor		...

Figure 3.3: The parameters which can be altered in a positive displacement type turbine expander

Boiler/Evaporator

The type of evaporator type used is the shell and tube type. Figure 3.4 shows the master fluid (working fluid) inlet and outlet and the slave fluid (exhaust gases) inlet and outlet. The master fluid runs in the tubes which are encapsulated by the shell where the slave fluid flows.

The heat exchangers in this model were governed by the equation 3.5 through which the outlet temperature of the fluid passing through it was calculated.

$$T_{fluid} = c \cdot T_{out} + (1 - c) \cdot T_{in} \tag{3.5}$$

where, T_{fluid} is the temperature of the fluid exiting the heat exchanger, c is a weighting factor and is taking as 0.5, and T_{out} and T_{in} are the temperatures of the fluid measured initially in the unit.

Some properties of this unit are:

- The exhaust gas flows through the shell of the heat exchanger.
- The heat exchanger is based on test data wherein R245fa was used as the fluid.
- The geometry of the heat exchanger can be altered to minimize pressure losses.
- The heat exchanger map displays various thermodynamic data.
- Heat transfer rate in the map is used as a reference to define how much energy enters into the system.

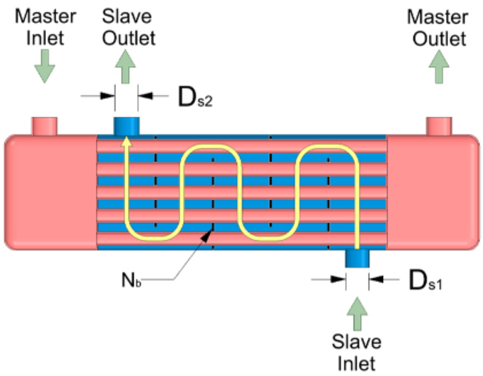


Figure 3.4: Construction of a shell & tube type heat exchanger used for the evaporator

Recuperator

The working principle of a recuperator is described in Section 2.3.1 and from this, it is seen that a heat exchanger which transfers heat from the hot side to the cold side of the cycle could be beneficial. The example model uses a shell and tube type heat exchanger (Figure 3.5) and the thesis work also uses the same setup.

Some properties of the recuperator are:

- The setup has a single tube encased by a shell.
- Some of the heat energy from the hot side of the cycle after the expander is transferred to preheat the pressurized working fluid out of the pump.
- A heat rate performance map based on R245fa is used as a template for the thesis work.

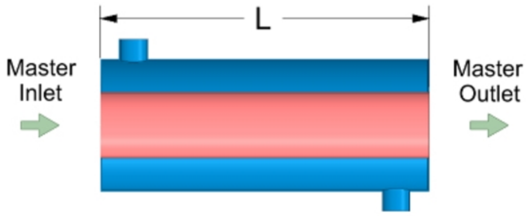


Figure 3.5: A simple Shell and Tube type heat exchanger setup used in the recuperator

Condenser

The condenser in an organic Rankine cycle is where the phase change from gaseous vapour back to liquid occurs. In marine applications, a plate type heat exchanger was used. The master fluid and the slave fluid (the coolant) flows in alternative plates and heat exchange occurs.

Tank

The tank is primarily used in an organic Rankine circuit to ensure constant flow of the working fluid into the pump and so, is situated just before the pump. As the pump operates most efficiently with a constant supply of working fluid in the liquid phase, the tank provides this saturated flow at a steady rate. Figure 3.6 shows the construction of a typical receiver unit.

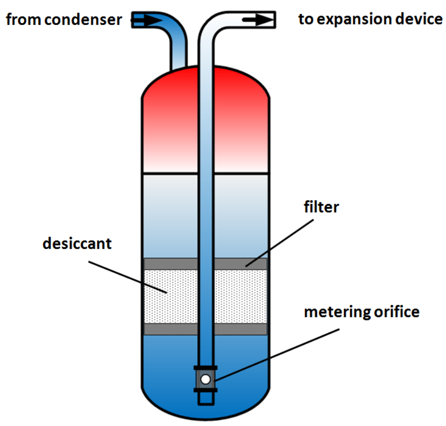


Figure 3.6: Diagram of a tank

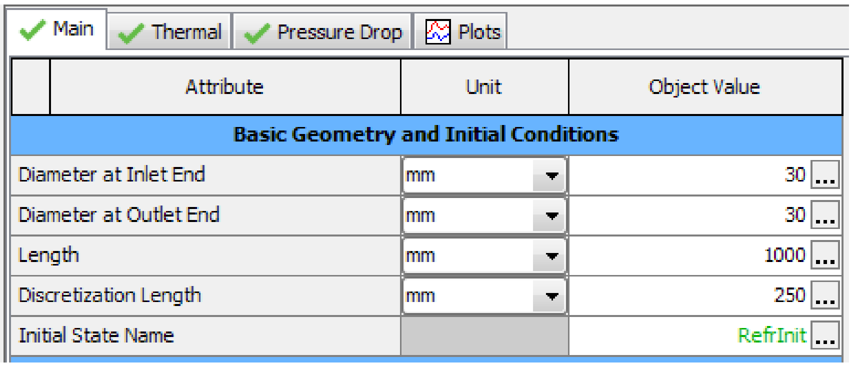
The liquid volume fraction in the tank, which can be thought of as the non-dimensional height of the liquid was calculated by 3.6. The value of the fraction may vary from 0 (pure vapour) to 1 (pure liquid).

$$Liquid\,Volume\,Fraction = \frac{V_{liquid}}{V_{liquid} + V_{vapour}} \tag{3.6}$$

where, V_{liquid} is the volume of liquid and V_{vapour} is the volume of vapour in the tank.

Piping

The piping construction is crucial in a 1D model. The discretization length of a pipe will determine the quality of result. Parameters shown in Figure 3.7 were used to control the construction of the pipes. Regions where phase change of the working fluid occurs, pipe diameter plays a huge role and a pipe geometry not suitable for that process will affect the cycle negatively.



Attribute	Unit	Object Value
Basic Geometry and Initial Conditions		
Diameter at Inlet End	mm	30
Diameter at Outlet End	mm	30
Length	mm	1000
Discretization Length	mm	250
Initial State Name		RefrInit

Figure 3.7: Parameters which were altered in the piping construction to suit the mass flow and energy input

1D modelling in GT-Suite for Marine application

Section 3.3.1.2 gives a detailed look into how each part of an organic Rankine cycle work in GT-Suite. For designing a WHR unit for the marine application, the setup is better understood in section 3.2.1. Due to the high availability of fresh water supply for the cooling unit, a condenser which uses sea water as a coolant was used. The steps taken in designing the model are:

- The model is built on the existing model for the R245fa system.
- A different working fluid, one that was selected from the 0D analysis is used for this layout.
- The pump used was the same from the example model but the 'Flow Multiplier', which simulates a pump handling higher or lower mass flow rate through it was used to mimic a different size pump and it had a value of 3. The existing performance map of the pump was used for the current model. The pump speed also changed appropriately to meet the temperature and flow demands.
- Due to the complexities involved in tweaking an already existing expander to fit the new working fluid vapour properties, the 'TurbPosDispRefrig' was used

to design an expander for this model. The volumetric efficiency was set to 0.7 [21] and the isentropic efficiency was set to 0.65. The displacement parameter was used as a case to be further changed for various values in Case Setup along with the expander speed.

- In the evaporator, the 'Heat Transfer Rate' parameter was changed to simulate a bigger sized unit without changing the dimensions of the part.
- The condenser had sea water as the coolant and the volume flow rate was input in the Case Setup to be changed for different settings.
- The input temperature for the coolant was set to 25°C.
- The recuperator, tank and the pipe dimensions were changed accordingly to the get the best power output from the system.

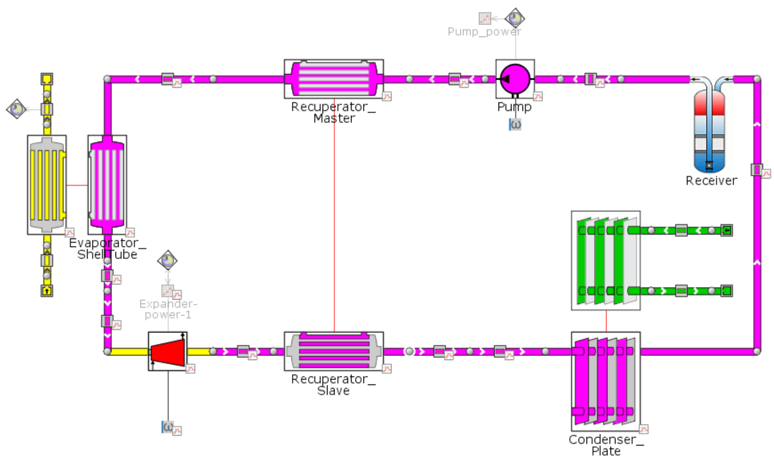


Figure 3.8: GT-Power model of a single turbo waste heat recovery cycle for marine application

As can be seen from the Figure 3.8, the setup of the organic Rankine cycle for the marine application is very similar to the example model. For further understanding of the influence of a recuperator in the model, a setup without the recuperator was also tested and the setup can be seen in Figure 3.9. All the models were created along with the different cases and various types of optimizers were run to help achieve a higher power output while maintaining the working fluid temperature. An optimization was run where parameters such as maximum temperature of working fluid out of the evaporator and maximum pressure in the system were obtained which resulted in higher power output from the expander. This helped narrow down the pump speed and expander speed values which would lead to higher power output while meeting the fluid requirements. The results were then created and processed using GT-Post.

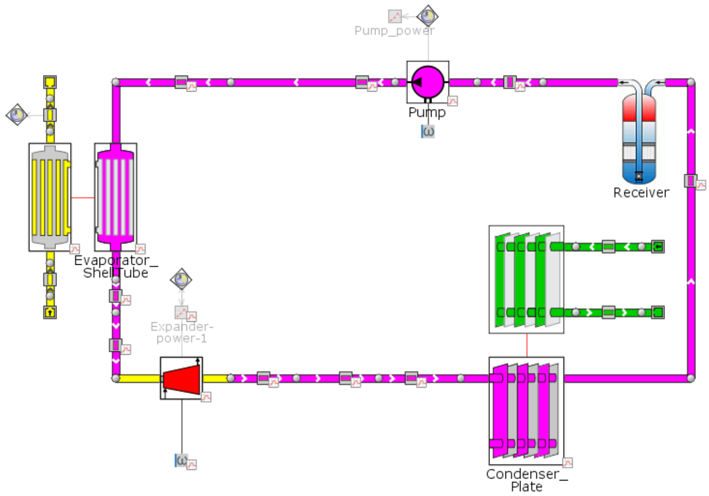


Figure 3.9: GT-Power model of a single turbo waste heat recovery cycle without recuperator

1D Modelling in GT-Suite for Power Generation Application

Similar to the marine setup, an organic Rankine cycle model for the power generation application was based on the example WHR model with R245fa. This type of application is fixed to a constant engine speed for optimal electricity generation and frequency matching. Some changes to the example model made were:

- As the energy input into the system is similar to the marine application, appropriate changes to the evaporator, expander, and the pump were copied from the marine model.
- For this particular setup, a similar layout as used in the marine application was used. The condenser used water as the coolant instead of air as in the physical setup. This was due to the complexity in modelling an air cooled radiator setup.
- The coolant water temperature was set to 35°C.

The influence of recuperator for this particular application was tested as was done previously for the marine application. Various loads for this particular engine were tested and simulations were run to find the best output for that case. The results were then processed using GT-Post and presented in the next chapter.

3.3.2 1D Modelling in Matlab/Simulink

Simulink was used for building an organic Rankine cycle for marine and power generation application of the Volvo D13 engine. The model was created by Volvo Truck, but it was built to run with the exhaust mass flow of the Volvo Truck engines which is lesser in comparison to the Volvo Penta D13 engine. To make this model handle the higher temperature and mass flow of exhaust gas for the Volvo Penta D13 engine, the condenser was adjusted.

Volvo Truck Simulink Model

The actual simulations using cyclopentane as the working fluid were based on the Simulink model of the Volvo Truck that is shown in Figure 3.10. Some properties of this model are:

- The system can use various types of working fluids where the fluid properties should be added to the workspace before running the simulation.
- The pump is a volumetric pump model.
- The tank is a receiver model with a capacity to store working fluid to be later sent into the pump.
- The condenser is a plate type heat exchanger using air as the coolant.
- The recuperator is a plate type heat exchanger.
- The expander is a volumetric expander model.
- The evaporator is a plate and fin type heat exchanger.

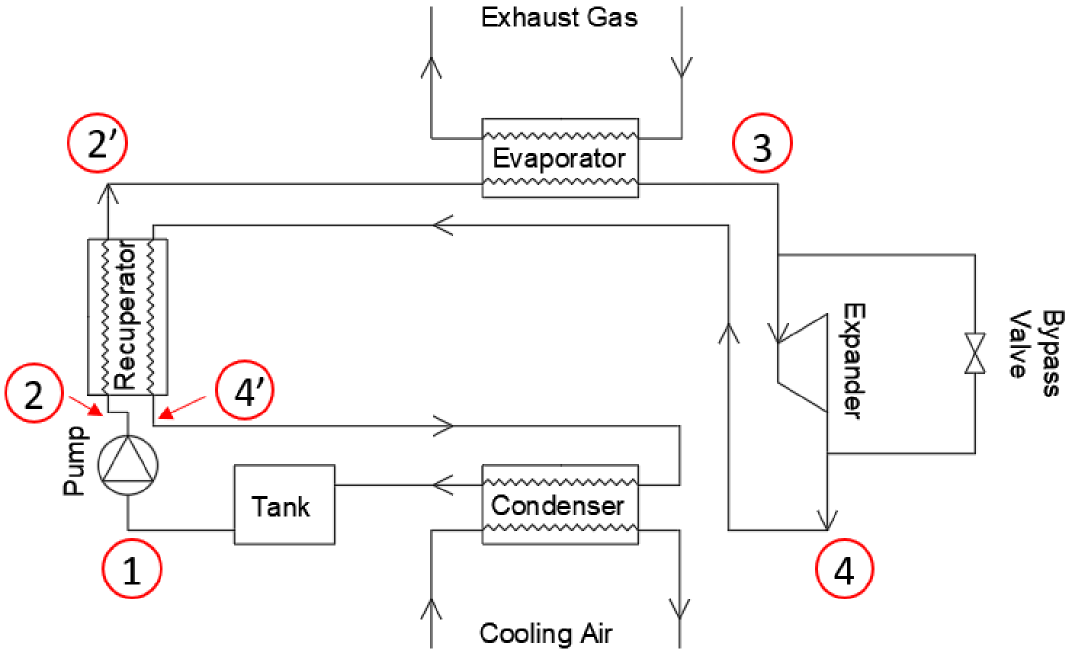


Figure 3.10: Schematic representation of the ORC model developed by Volvo

Parts of Organic Rankine Cycle in Simulink Model

The organic Rankine cycle model in Simulink consisted of various components. The model uses cyclopentane as the working fluid, and can use different types of working fluids such as water, and ethanol. The parts are discussed in detail below.

<u>Pump</u>

The pump speed controls the mass flow in the organic Rankine cycle and the pressure is also changed as a consequence of this. In this thesis work, the pump speed is altered manually until the highest net output power in a stable system is reached.

The pump is a volumetric pump type with a constant isentropic efficiency of 0.6 and a mechanical efficiency of 0.9 for all pump speeds. The pressure before the pump is entered manually, and the pressure out of the pump is calculated in the expander. The pump power is calculated later as in Equation 2.9. After calculating the fluid state out of the pump, the values are used as inputs into the recuperator. The fluid state includes the main thermodynamic variables of the working fluid such as pressure, density, enthalpy, vapor quality, and mass flow rate.

Recuperator

The recuperator is a plate type heat exchanger with an efficiency of 0.7. The heat transferred from the working fluid in the vapor phase to the working fluid in the liquid phase is calculated based on the energy balance in a similar way with Equations 2.1, 2.2, and 2.3. After calculating the fluid state out of the recuperator, the values are used as inputs into the evaporator.

Evaporator

The evaporator is a plate and fin type heat exchanger. In this heat exchanger, the thermal losses to the ambient are not considered. The heat transferred from the exhaust gas to the working fluid is calculated based on heat transfer coefficients calculated for both the exhaust and the working fluid. The fluid state out of the evaporator is calculated and entered as inputs into the expander.

Expander

The expander speed controls the pressure out of the pump in the organic Rankine cycle as in equations 2.7, where there is a specific volumetric flow for each expander speed. The expander speed is altered manually along with the pump speed until the highest net output power in a stable system is reached. The expander is a volumetric expander model with a isentropic efficiency of 0.82 and a mechanical efficiency of 0.92. After calculating the pressure out of the pump (in the expander), the value is sent to the pump. The fluid state is also calculated in the expander and used as inputs into the condenser, and the expander power is calculated as in Equation 2.10.

Condenser

The condenser is a plate type heat exchanger that uses air as a coolant. The thermal losses to the ambient are also not considered as in the evaporator. The heat transferred from the working fluid to the cooling air is calculated based on heat transfer coefficients calculated for both the cooling air and the working fluid. The fluid state out of the condenser is calculated later on and entered as inputs into the tank.

Tank

The working fluid enters the tank after the condensation, in the condenser, in order to supply the pump with a constant mass flow rate of the working fluid which is in a liquid phase. The fluid state out of the tank entered as inputs into the pump. There is no pressure drop in the cycle because the pipes are not considered in this model, and the mass flow rate is constant in these components. Since there are two applications for the Volvo Penta D13 engine marine and power generation, the existing 1D model is adjusted to create two 1D models in Simulink for each:

- 1D model for Marine application
- 1D model for Power Generation application

1D Modelling in Matlab/Simulink for Marine Application

The components of the 1D model of Volvo Truck were kept except the condenser which was air cooled and replaced by a water cooled condenser that uses seawater as a coolant. The average temperature of seawater that adopted in this model simulation is 25°C based on Volvo Penta documentation. The 1D model for marine application consists of the components shown in Figure 3.10. The condenser in this model however, is water cooled, and is a plate type heat exchanger. The thermal losses to the ambient are also not considered as in the Volvo Truck model. The bypass valve was completely closed during the simulation because the engine was run in steady state operation.

The mass flow of the organic working fluid increases by increasing the pump speed, and the pressure in the cycle decreases by increasing the expander speed. The 1D model is used for computing the net output power, pump power, the mass flow of organic working fluid and fluid state in each component in the cycle. In terms of the inputs, the 1D model has more inputs than the 0D model which has only the fluid selected, exhaust mass flow and exhaust temperature as inputs, whereas the 1D model has the pump speed, and expander speed as extra. Increasing the pump speed leads to increasing the mass flow of cyclopentane which then decreases the temperature of working fluid out of the evaporator. If the pressure in the cycle becomes close to the critical pressure (when it exceeds 40 bar), it could be decreased by increasing the expander speed. To choose the appropriate pump speed and expander speed of Rankine cycle, two parameters were taken into account:

- Cyclopentane should be subcooled liquid in the pump because the pump has a higher efficiency when there is a subcooled liquid, and superheated vapor in the expander.
- Cyclopentane temperature in the evaporator which should not exceed 275°C as it leads to decomposition of cyclopentane.

The water cooled condenser was not able to change the phase of cyclopentane from vapor to saturated or subcooled liquid and so, the dimensions were adjusted to increase heat transfer rate of the condenser thus ensuring that cyclopentane is in saturated liquid phase into the pump.

1D Modelling in Matlab/Simulink for Power Generation Application

The components of the 1D model of Volvo Truck were completely kept without any change to be used in the 1D model for power generation. The average temperature of the air used in this model is 35°C based on Volvo Penta documentation.

The 1D model for the power generation application is similar to the 1D model for the marine application except for the water cooled condenser, which is replaced by an air cooled condenser. The air cooled condenser was not able to change the phase of cyclopentane from vapor to saturated or subcooled liquid in the beginning, and there were many attempts to resize it and make the heat exchange area between the working fluid and the coolant (air) bigger than before. But the air cooled condenser in this model was not responding to any change in its dimensions and heat exchange area, even though after increasing the air mass flow. Therefore the water cooled condenser which was used in the marine application was used for power generation application as well.

4

Results and Discussion

This chapter will look at the results from the 0D analysis and the 1D simulations using Matlab/Simulink and GT-Suite. The chapter will also discuss the obtained results to understand the process as a whole.

4.1 0D Modelling Results

The 0D analysis is an excellent way for selecting the fluid to be used in an organic Rankine cycle. For the purpose of WHR, the fluid selection is the one of the most important part of the process and this will determine how efficient the cycle is in extracting energy from the heat source. As described previously in the chapter Methodology, 0D analysis was done using Matlab software in conjunction with REFPROP. Table 3.4 points out the input conditions which were used for testing the fluids where 75% load is used for the marine and 75% load is used for the power generation application.

4.1.1 Marine Application

Marine application had the condenser unit using water as the coolant at 25°C and using this, the 0D analysis was run. The most important results obtained from the analysis can be seen in Table 4.1, where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure out of the condenser, T_{evap} is the temperature out of the evaporator, T_{cond} is the temperature out of the condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator, and η_{th} is the thermal efficiency of the system.

Table 4.1: 0D analysis results for water, ethanol, and cyclopentane

Fluid	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
Water	33.8	0.071	50.5	1.0	362.2	94.7	175.0	0.149
Ethanol	34.3	0.160	35.0	1.0	223.8	73.2	155.0	0.141
Cyclopentane	39.0	0.355	40.6	1.0	260.3	44.0	150.0	0.157

Cyclopentane has the maximum power output amongst the three fluids and solely on this parameter, it was chosen as the fluid of choice. The temperature to which cyclopentane has superheating in the evaporator is chosen by the 0D model to achieve the maximum power output while still meeting the fluid limitations such as its decomposition point of 275°C. Water on the other hand being a wet fluid, requires superheating to ensure that there is complete vaporization after the evaporator. A way of achieving this high temperature is by reducing the mass flow rate. This is observed in Table 4.1. Though water is more favourable due to its availability and handling, cyclopentane has an advantage due to the fact that the fluid is a dry fluid. Other parameters which help understand the temperature and pressure limits in the system were also obtained from the 0D analysis as shown in the tables 4.2 and 4.3. The maximum pressure in the cycle for cyclopentane was limited at 40 bar with 5 bar safety margin from the critical pressure of 45 bar. Using REFPROP software, a temperature vs entropy diagram was plotted for the cycle using cyclopentane as the working fluid and can be seen in Figure 4.1.

Table 4.2: 0D marine results of the WHR model using cyclopentane

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	44.0	1.0	0.355	-10.38
2	46.8	40.6	0.355	0.00
3	260.3	40.6	0.355	694.10
4	166.0	1.0	0.355	577.45

Table 4.3: 0D results of the source parameters for cyclopentane

Source	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)
Inlet	432.2	1.0	0.877
Outlet	150.0	1.0	0.877

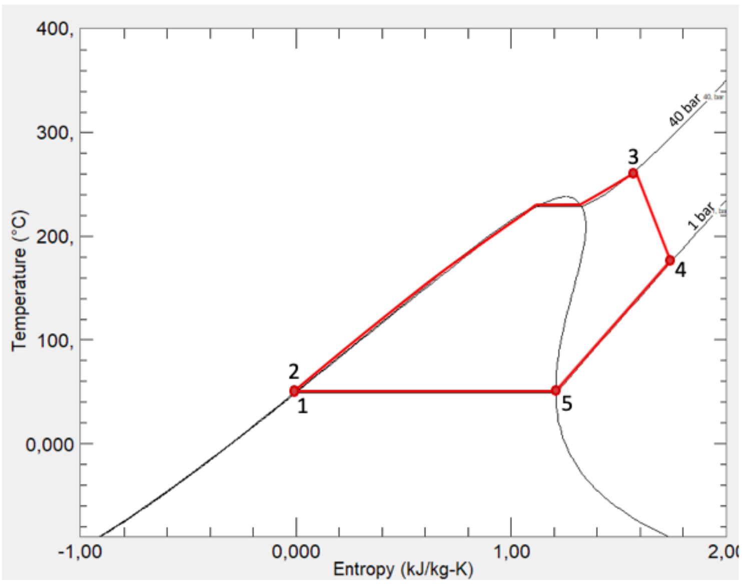


Figure 4.1: T-s diagram of organic Rankine cycle for the marine application using cyclopentane as the working fluid. 1-2: Compression; 2-3: Evaporation and superheating; 3-4: Expansion; 4-5: Condensation and subcooling

From Figure 4.1, it can be observed that the temperature of the fluid coming out of the expander is higher than the input temperature of the pump unit. This shows that the system may benefit from having a recuperator unit to boost the cycle efficiency and help output higher power. Similar to a Temperature vs Entropy diagram, a Temperature vs Heat rate or a Q-T diagram also helps in understanding the energy transfer from the source to the system as seen in figures 4.2, 4.3, and 4.4 for water, ethanol and cyclopentane respectively.

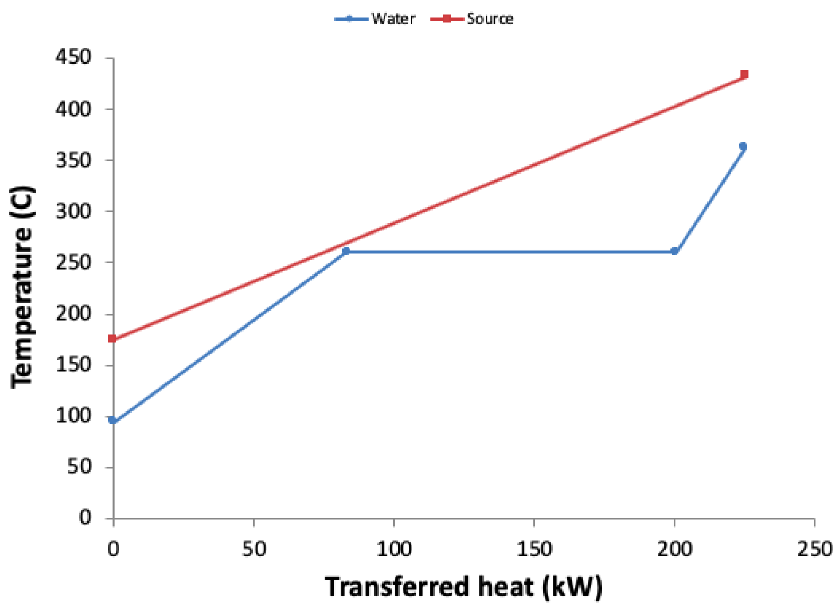


Figure 4.2: A Q-T diagram for the marine ORC using water as the working fluid

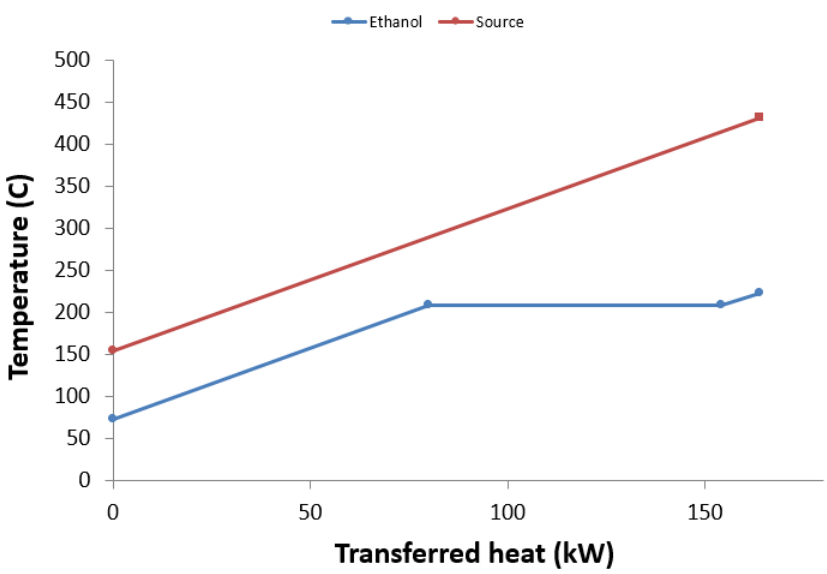


Figure 4.3: A Q-T diagram for the marine ORC using ethanol as the working fluid

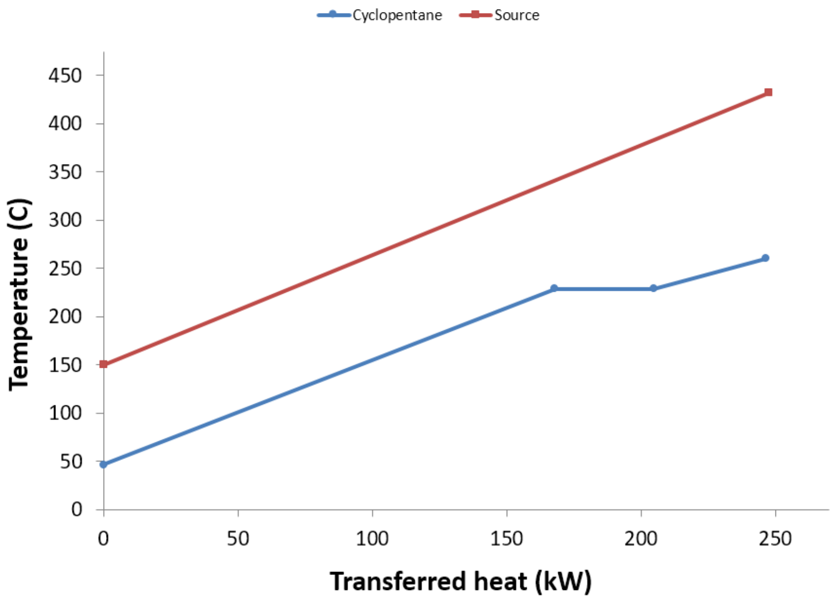


Figure 4.4: A Q-T diagram for the marine ORC using cyclopentane as the working fluid

Figures 4.2, 4.4 and 4.3 show how the heat added to the system differs by changing the working fluid. The heat added into the system is highly dependant on the mass flow rate and the pressure in the system. Observing from Equation 2.1, we note that the mass flow rate and the change in enthalpy dictate the heat input. Hence, the heat added to cyclopentane is the highest. Observing 4.2, it is inferred that the more energy could be extracted from the heat source but the parameters might be limited to this point due to the higher power output available. Looking at the three Q-T diagrams, it is also observed that the exhaust gas does not go below 150°C. This is a limitation specified in the 0D analysis and influences the heat extraction for the working fluids. If no limit was specified, the mass flow rate of the fluids could be increased to extract more heat energy from the exhaust gases until $\dot{Q}_{in}$ is equal to $\dot{Q}_{source}$ beyond which no energy can be exchanged.

4.1.2 Power Generation Application

For the power generation application, the 0D model was run with the 75% load as chosen previously. The performance figures of the three fluids were obtained and can be seen in Table 4.4, where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure out of the condenser, T_{evap} is the temperature out of the evaporator, T_{cond} is the temperature out of the condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator, and η_{th} is the thermal efficiency of the system.

Table 4.4: 0D analysis results for water, ethanol, and cyclopentane

Fluid	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
Water	23.0	0.077	50.5	1.0	430.7	94.5	203.0	0.179
Ethanol	21.1	0.152	35.0	1.0	223.4	73.2	150.0	0.138
Cyclopentane	23.8	0.226	40.6	1.0	258.9	44.0	154.0	0.157

Observing the power figures for the three fluids, cyclopentane had the highest power output and this was chosen as the fluid of choice. Similar to that of the marine application, the temperature of the working fluid out of the evaporator was calculated by the model such that it achieved the maximum power. Though water has a very similar power output figure, the temperature out of the evaporator is higher which meant that in this case, the energy into the system is higher with water in turn leading to lower temperature of exhaust exiting the evaporator. In the marine application, the exhaust temperature after the evaporator is lower than for 200°C for both cyclopentane and ethanol but this is acceptable as there are no after treatment systems in place (which require temperatures higher than a specific point for efficient operation) and the exhaust is directly let out into the sea water.

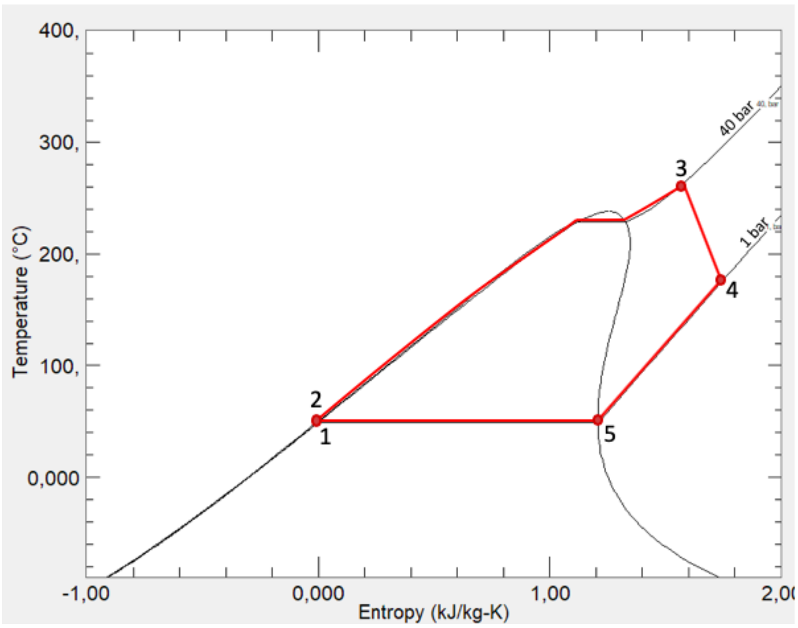


Figure 4.5: T-s diagram of organic Rankine cycle for the power generation application using cyclopentane as the working fluid. 1-2: Compression; 2-3: Evaporation; 3-4: Expansion; 4-5: Condensation and subcooling

As was done previously, REFPROP was used to plot a temperature vs entropy diagram to better understand the cycle and is shown in Figure 4.5 and the values from the cycle are seen in tables 4.5 and 4.6. From this, it can be noted that the temperature of the fluid after the expansion process is higher than when it enters the pump. Due to this, the system might benefit from including a recuperator into the system.

Table 4.5: 0D power generation results for the WHR model using cyclopentane

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	44.0	1.0	0.226	-10.38
2	46.8	40.6	0.226	0.00
3	258.9	40.6	0.226	689.57
4	164.0	1.0	0.226	573.73

Table 4.6: 0D power generation results of the source parameters for cyclopentane

Source	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)
Inlet	459.0	1.0	0.490
Outlet	150.0	1.0	0.490

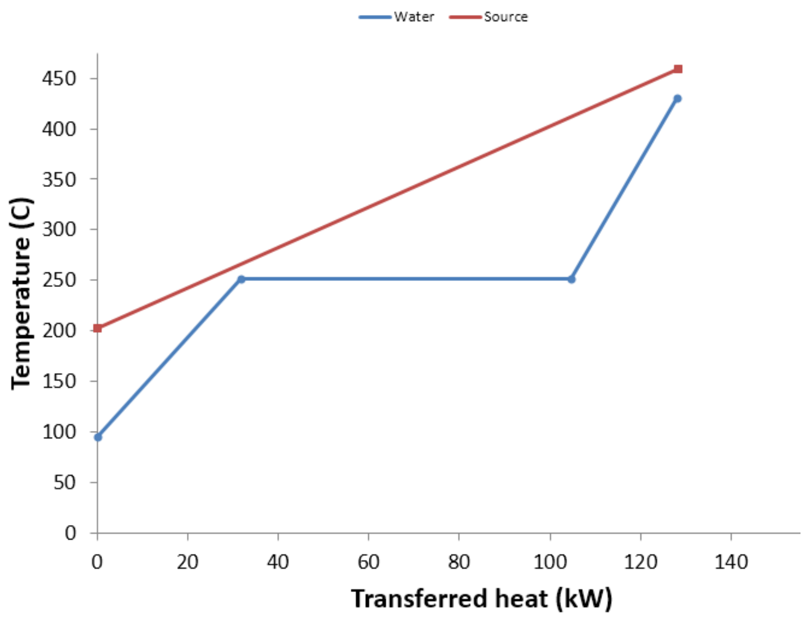


Figure 4.6: A Q-T diagram for the power generation ORC using water as the working fluid

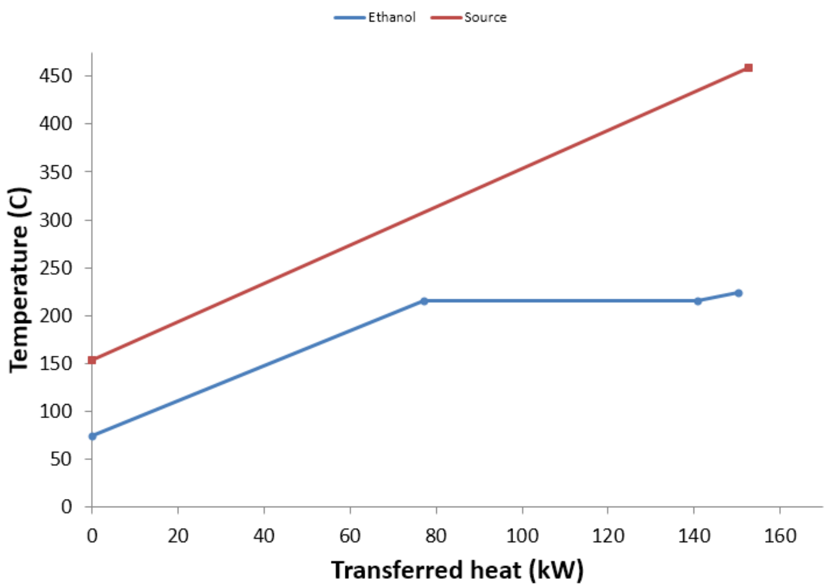


Figure 4.7: A Q-T diagram for the power generation ORC using ethanol as the working fluid

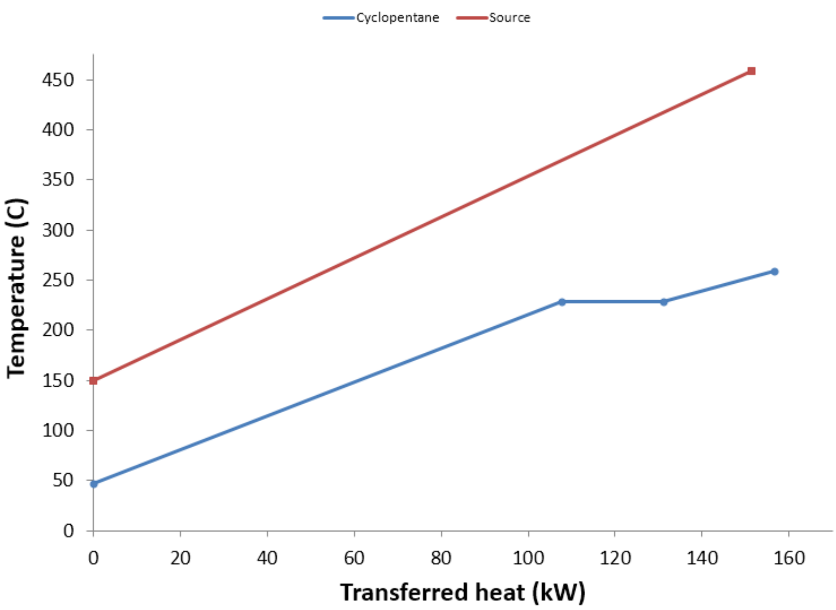


Figure 4.8: A Q-T diagram for the power generation ORC using cyclopentane as the working fluid

A Q-T diagram was drawn for each fluid to analyze the heat exchange from the source to the system and this was helpful in understanding each working fluid's characteristics. Similar to the marine application, Figure 4.8 shows that more heat energy from the exhaust could be extracted to the working fluid but due to the limiting critical pressure of 40 bar, it is limited to this point. Due to the varying limits at which the maximum power output was achieved for the three fluids, they were not compared at the same mass flow rates which also affected the Q-T curves and are seen in figures 4.6, 4.7, and 4.8. The limit of 150°C for the exhaust is observed in these diagrams and this also limits the energy into the system.

4.1.3 Limitations

Some of the limitations of the 0D modelling are listed below:

- Fluid list limited to the three available ones in the program.
- Simplified models do not consider the physical construction of the components.
- The flow properties in the pipes are highly simplified and their construction is not considered.
- The lower limit of the exhaust temperature for the 0D analysis was limited to 150°C.
- The cooling system is not simulated. The fluid is assumed to reach a certain temperature without a condenser.
- No recuperator was used for the analysis to analyse its effect on the system.

The 0D modelling results give an insight into the capability of a WHR unit and serves as baseline for the 1D modelling. With the working fluid selected as cyclopentane for both the applications, 1D analysis was then carried out and the results are presented in the next section.

4.2 1D Modelling Results

From the 0D analysis, cyclopentane was chosen as the working fluid. The T-s diagram indicated that a model built for the two applications could benefit from having including a recuperator unit. The effect of recuperator is further studied and the 1D simulation results are presented and discussed in this chapter.

4.2.1 Results from GT-Suite Modelling

As previously discussed, the GT-Suite model is based on the example WHR model. With the heat source temperature of 450°C and mass flow of 0.170 kg/s, the example model dealt with an energy input of 60 kW into the system. As the Volvo D13 applications have higher temperature and mass flows meaning higher energy input, the model was similarly altered and run.

4.2.1.1 Marine Application

The 75% load values of the Volvo D13 marine engine was used as input and the working fluid in the system was changed to cyclopentane as per the 0D analysis. During the course of the thesis work, some parameters were crucial for the proper working of the cycle such as mass flow rate of the working fluid, energy input into the system, influence of the recuperator unit, and of course, the power ouput from the system. These will be talked about in detail later in this section.

On optimizing the pump speed and the expander speed based on the output values of power and the maximum fluid temperature and pressure, the cycle for the marine application was run and the various parameters of interest are seen in Table 4.7 where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure out of the condenser, T_{evap} is the temperature out of the evaporator, T_{cond} is the temperature out of the condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator heat exchanger, and η_{th} is the thermal efficiency of the system. The cycle was largely influenced by the pump speed and the expander speed where, the increase in pump speed increased the mass flow in the system thereby decreasing the temperature out of the evaporator unit. The expander speed influenced the pressure in the system.

Some specific parameters helped fine tune the system to produce a stable output. Due to its fluid properties, cyclopentane has a temperature limit of operation at 275°C and if the system has an input energy higher than that temperature limit, it will lead to decomposition of the fluid leading to loss in efficiency of the cycle. As the organic Rankine model works best with the maximum possible temperature, the cycle had the maximum temperature limit of 260°C with a safety margin of 15°C.

Table 4.7: Results from 1D analysis for the marine application with a recuperator

Load Point	$\dot{W}_{net}$	$\dot{m}$	P_{evap}	P_{cond}	T_{evap}	T_{cond}	T_{exhout}	η_{th}
75 %	38.2 kW	0.358 kg/s	33.8 bar	0.8 bar	253.3°C	41.2°C	214.3°C	0.199

Table 4.8 shows other parameters such as Mach number, pressure instability index and the maximum pressure drop in the system. Initially, the system had very high factor of real time and convergence was not possible. The Mach number parameter helped to understand which point in the model there were constrictions to the flow. The higher the Mach number, the higher was the obstruction leading to instability in the mass flow across the cycle. This would lead to higher simulation time and convergence cannot be reached. The pressure instability index is calculated from the pressure from three consecutive time steps. The higher the instability index in a particular region, the higher will be the simulation time at that region. Pipe dimensions in and out of different components were changed with respect to the instability index.

Table 4.8: Others parameters which were taken into consideration for designing the model

Parameter	Value
Max Mach Number	0.32
Max Pressure Instability Index	16.2
Max Pressure Drop	0.035 bar

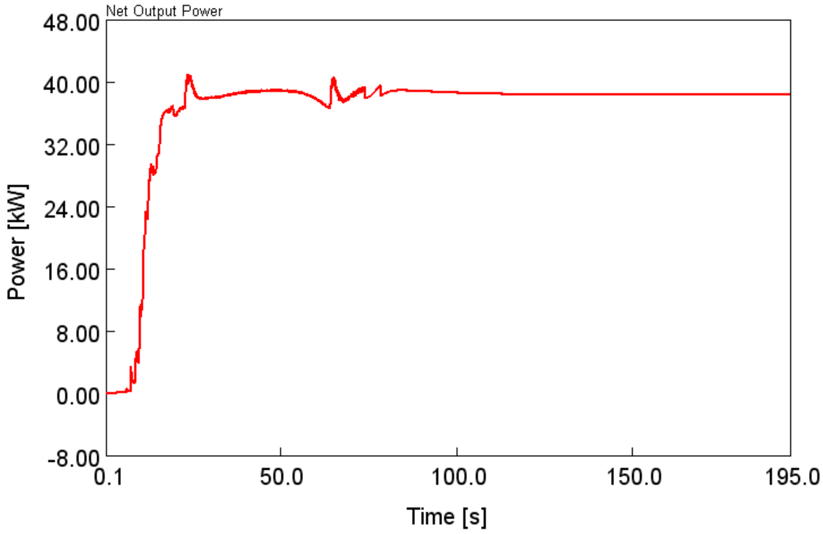


Figure 4.9: Power out from the turbine expander for the main operating point showing the convergence in the system

It can be seen from Figure 4.9 that the model reaches convergence at around 200 simulation seconds. The pump speed for this result was 1675 rpm and the expander speed was fixed at 1000rpm which were obtained from the optimization process. Figure 4.10 shows the isentropic efficiency map of pump unit. The point in the red circle the efficiency which the pump was operating at and is around 60%. The pump unit consumes 2.65 kW and 60% efficiency means that only 1.6 kW is used for compression and the rest 1.05 kW is lost due to friction and dissipated as heat.

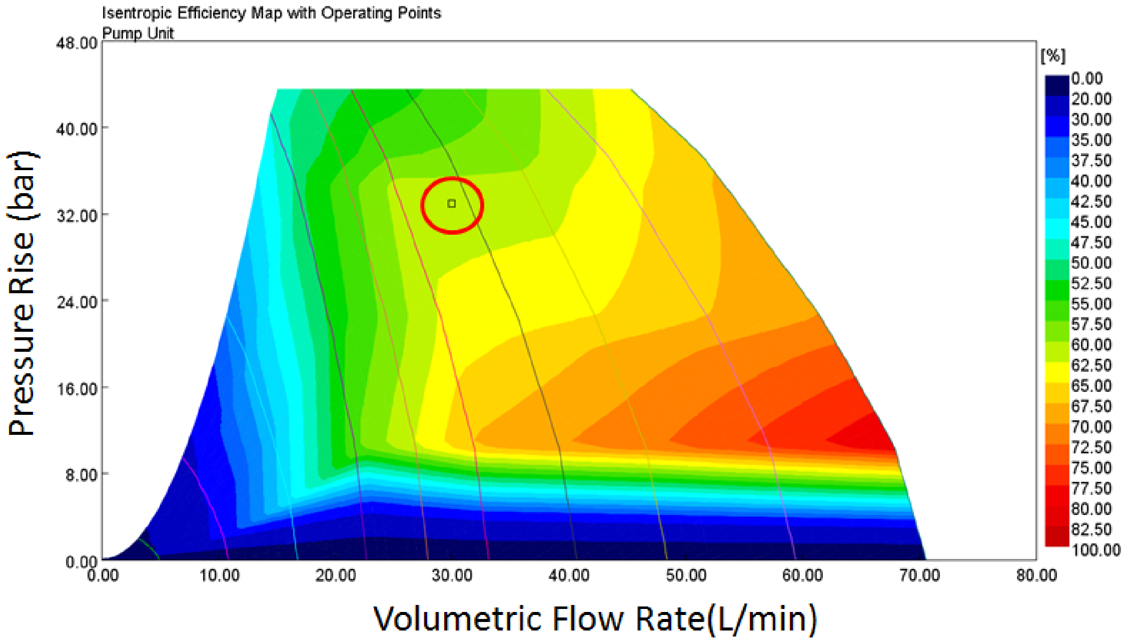


Figure 4.10: Isentropic efficiency map for the positive displacement pump for the marine application

A Temperature vs Entropy diagram was plotted for the results and is shown in Figure 4.11. This plot is better understood by also looking at Figure 4.12. Point 1-2 shows the compression in the pump unit. Here, there is pressure increase from 1 bar to 33 bar from the chosen pump speed. The pressurized fluid then enters the recuperator. This leads to a preheating and the temperature increase from 2-2'.

The fluid then enters the evaporator and is heated up further by the exhaust energy and there is some superheating to the point 3. The fluid now has high temperature and pressure and then enters the expander unit. here, the expansion process occurs and work is produced. This reduces the pressure in the system from 3-4. There is an increase in the entropy due to irreversibility in the expander unit due to the friction and other losses. It is observed that the fluid is still in the vapour phase here. After point 4, the fluid enters the recuperator and this leads to precooling to 4' where the energy is transferred to point 2-2'. After 4', the fluid then enters the condenser and is cooled to the initial temperature and pressure. It was also observed here that the system did not perform well after the pressure was pushed beyond 33.0 bar output from the pump and this is seen as a limitation from the modelling software.

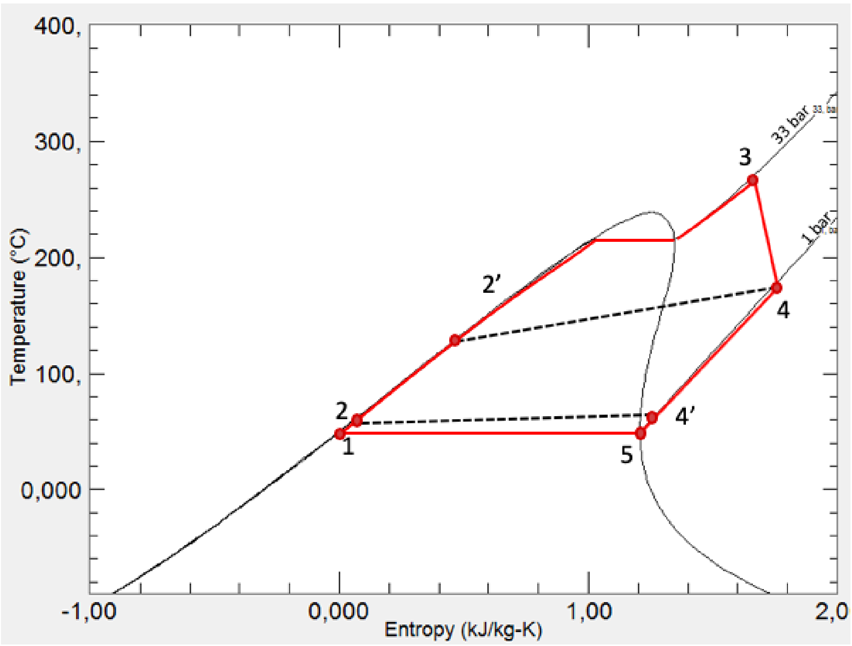


Figure 4.11: A T-s diagram to understand the recuperated organic Rankine cycle for the marine application

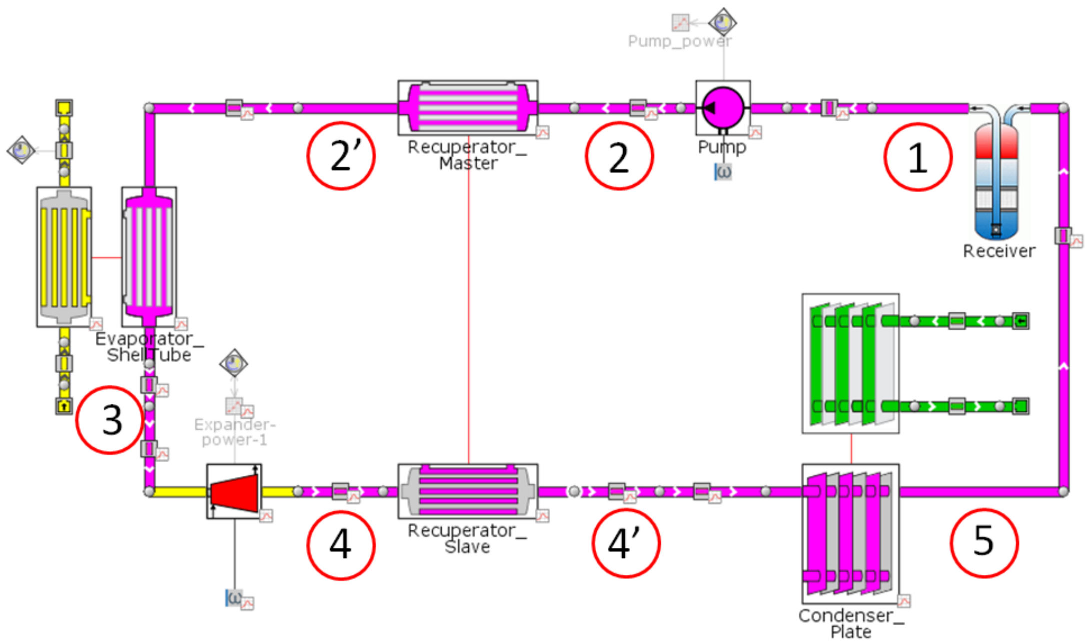


Figure 4.12: Schematic representation of the ORC model in GT Suite for the marine application

Table 4.9: 1D results for the marine application with recuperator - GT Suite model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	41.2	0.8	0.358	-15.41
2	43.8	33.8	0.358	-7.99
2'	128.2	33.8	0.358	171.31
3	253.3	33.8	0.358	695.38
4	170.4	1.2	0.358	588.59
4'	63.4	1.2	0.358	409.27

A temperature vs heat rate diagram was also plotted to understand the trend in the heat exchange process and is seen in Figure 4.13. The model was also run for different operating points for the marine engine chosen in Table 3.1. The results are seen in Table 4.10 and visualized in Figure 4.14. The graph shows that as the engine load increases due to which input energy increases, the net output power from the ORC also increases. Lower energy inputs at 25% and 50% have lower power outputs of 8 kW and 23 kW respectively. The 100% load point has a maximum power output of 49 kW. As the Volvo D13 marine engine works at 75% load for longer time, the power output of 38.7 kW is of interest and is used for future comparison between the models.

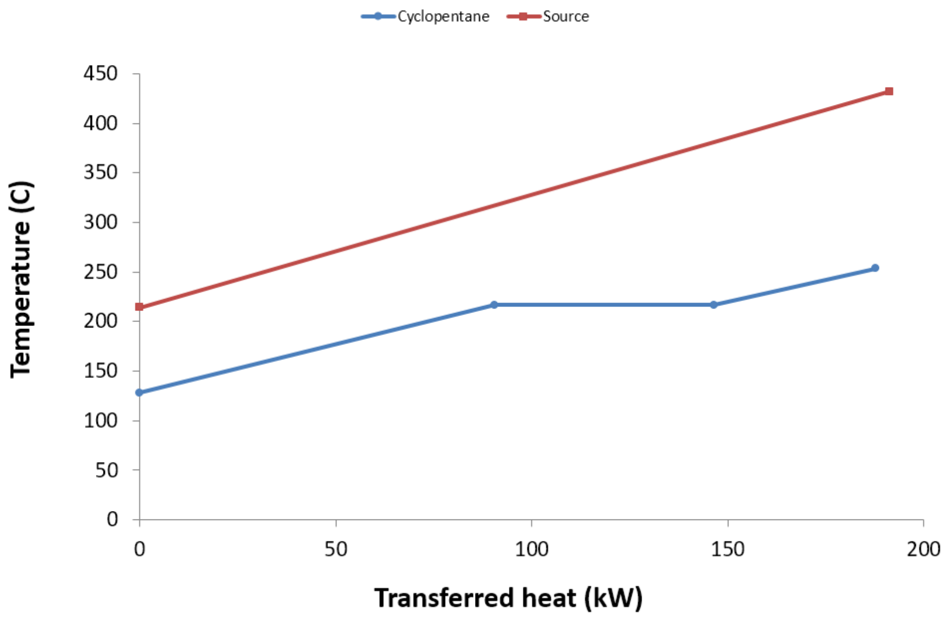


Figure 4.13: A Q-T diagram to analyze the heat exchange process for the recuperated system

Table 4.10: Comparison of the engine load percentage to the organic Rankine percentage power output for the marine application

Engine Load %	ORC Power Output %
25	4.4
50	6.3
75	6.9
100	6.7

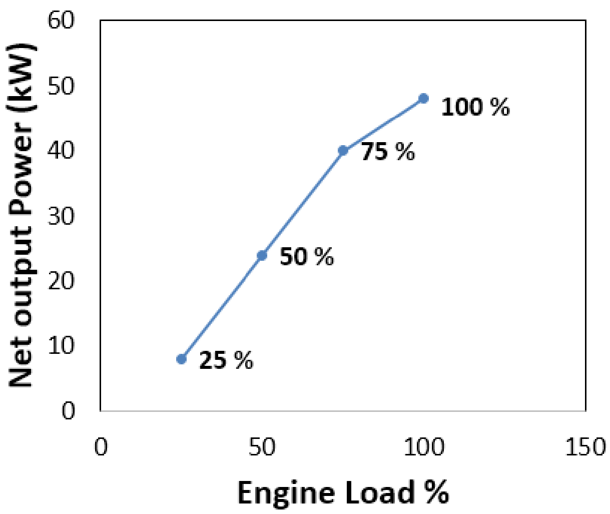


Figure 4.14: Variation of power output between the different load points for the recuperated model for the marine application

A model without the recuperator was also simulated and the power output difference is seen in Figure 4.15. The non-recuperated model has roughly 8 kW lower power output. The difference in the power can be understood by the difference in the mass flow rates. There is roughly 20% lower mass flow rate in the non-recuperated model. The recuperated model had a higher mass flow rate as the temperature out of the evaporator unit had to be maintained and due to the added heat exchanger in the model, the higher mass flow rate was required to keep the working fluid temperature below the decomposition temperature.

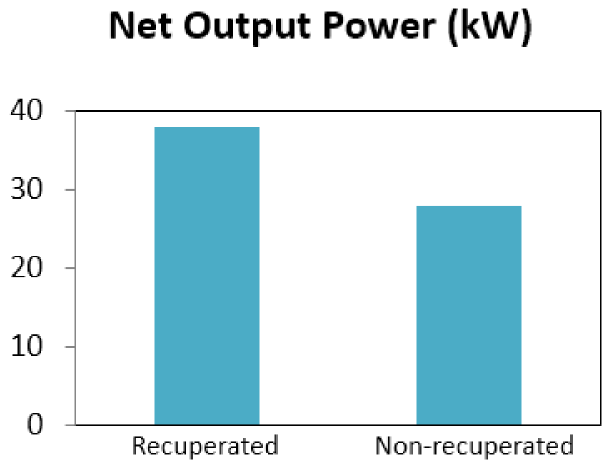


Figure 4.15: Comparison of output power between a model with and without a recuperator unit

As the mass flow rate is directly proportional to the power output, a 20% lower power output is seen in the non-recuperated model. The difference in power is better understood through a temperature vs entropy diagram as shown in Figure 4.16. Hence in this particular case, a recuperator was beneficial.

Comparing Figure 4.17 to Figure 4.13, it is observed that the non-recuperated model seems to have lower exhaust temperature which might mean that there is higher energy extracted from the exhaust. But the mass flow rate of the recuperated system is higher which directly translates to higher power produced by the system. The cycle temperatures and pressures for the different points are seen in 4.11.

Table 4.11: 1D results for the marine application for a non-recuperator - GT Suite model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	46.1	0.9	0.283	-6.07
2	48.8	32.8	0.283	1.45
3	253.3	32.8	0.283	699.63
4	180.2	1.6	0.283	604.50

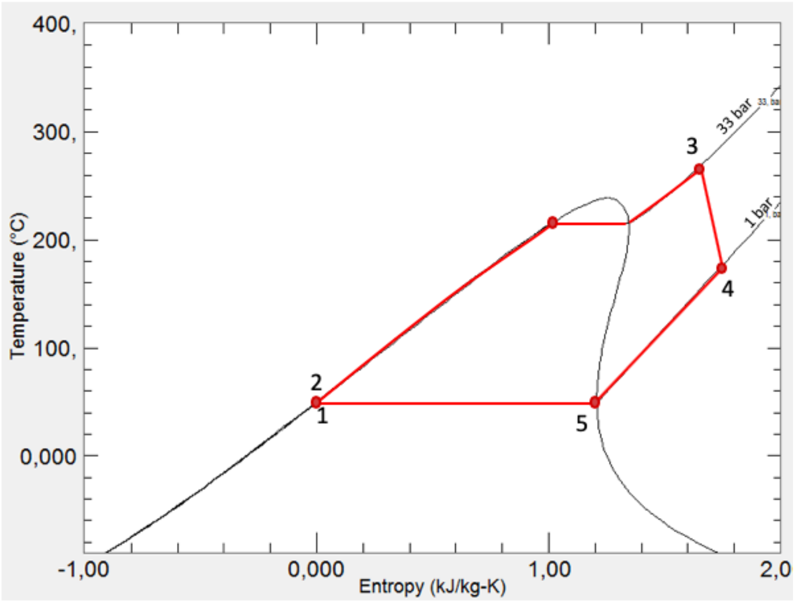


Figure 4.16: A T-s diagram to understand the non-recuperated organic Rankine cycle for the marine application

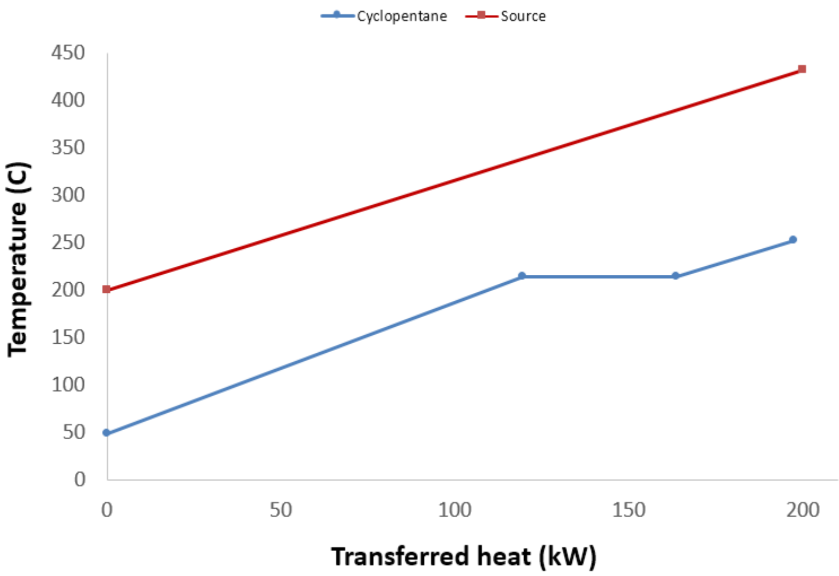


Figure 4.17: A Q-T diagram to analyze the heat exchange process for the non-recuperated system

4.2.1.2 Power Generation Application

Similar to the marine application, the power generation WHR cycle was based off of the R245fa example model. The changes done to the model are mentioned in Section 3.3.1.4 and the working fluid was changed to cyclopentane. The results thus obtained as shown in Table 4.12 where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure out of the condenser, T_{evap} is the temperature out of the evaporator,

T_{cond} is the temperature out of the condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator, and η_{th} is the thermal efficiency of the system.

Table 4.12: Results from 1D analysis for the power generation application with recuperator

Load Point	$\dot{W}_{net}$	$\dot{m}$	P_{evap}	P_{cond}	T_{evap}	T_{cond}	T_{exhout}	η_{th}
75 %	28.4 kW	0.280 kg/s	39.8 bar	0.9 bar	253.0°C	43.9°C	196.0°C	0.189

The net power output from the power generation application is lower at 28.4 kW. The pressure is higher compared to the results from the marine application and this is due to the difference in the energy into the system. This has an effect on the temperature of the working fluid and hence, the higher pressure of 39.8 bar could be reached. Table 4.13 shows the other parameters which were used to fine tune the model. The Mach number study as well as the pressure instability index around the recuperator region helped better understand the phase composition of the fluid and how the model needed to be changed. The pump speed for this power output was 2200 rpm and the expander speed was 3000 rpm. Figure 4.18 shows the simulation run and the convergence in the system after 250 simulation seconds.

Table 4.13: Parameters which helped fine tune the model

Parameter	Value
Max Mach Number	0.157
Max Pressure Instability Index	16.9
Max Pressure Drop	0.032 bar

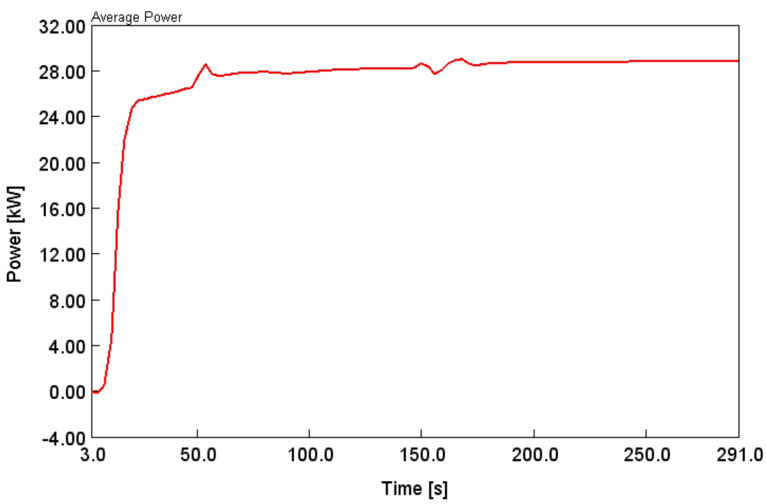


Figure 4.18: Power output from turbine expander for the power generation application showing the convergence in the system

From a temperature vs entropy diagram seen in Figure 4.19, the cycle behaviour can be observed. Due to the higher coolant temperature of 35°C and lesser mass flow rate as well, lesser heat energy is rejected in the condenser as lesser heat energy is extracted from the source. Combined with the lower energy input into the system, this model has a lower power output compared to the marine application. The values for the different points in the system is seen in Table 4.14.

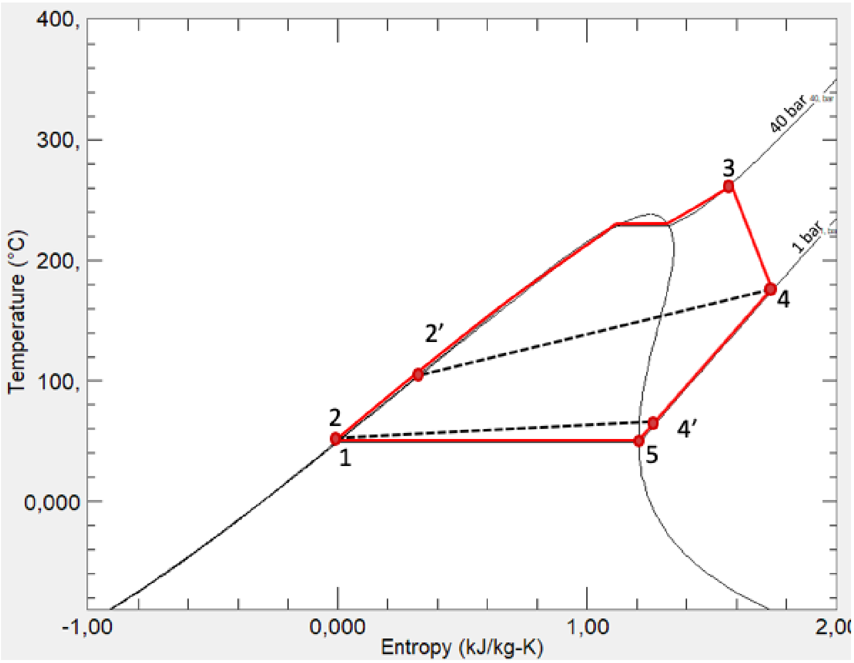


Figure 4.19: A T-s diagram to understand the recuperated organic Rankine cycle for the power generation application

Table 4.14: 1D power generation results - GT Suite model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	43.9	0.9	0.280	-10.33
2	46.6	39.8	0.280	-2.11
2'	121.0	39.8	0.280	155.83
3	253.0	39.8	0.280	675.00
4	159.0	1.2	0.280	565.42
4'	62.7	1.2	0.280	407.44

A temperature vs heat rate diagram was plotted to understand the heat exchange process in the system better and is seen in Figure 4.20. The system could be changed to further increase the heat energy extraction into working fluid as it is visible from the diagram that there is still no limit to the pinch point yet. But beyond this point, cyclopentane was reaching its critical point which meant that the liquid could start to disintegrate. Hence, the maximum power output at this point was chosen.

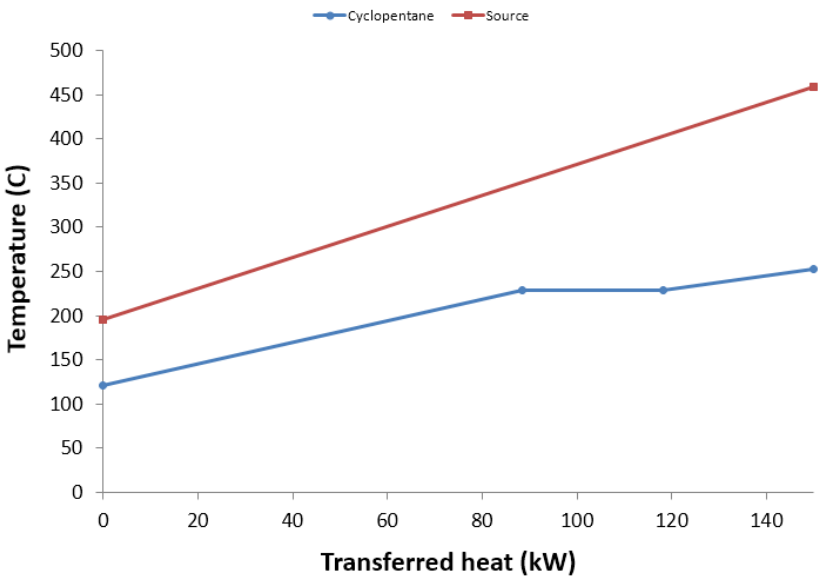


Figure 4.20: A Q-T diagram to analyze the heat exchange process for the recuperated system

Similarly to the marine application, a model without the recuperator was compared with the main recuperated model and the values for different points are seen in Table 4.15. For this application, there was a 5 kW difference in the output power between the two models. This is better understood with figures 4.21 and 4.22. Similar to the case of the marine system, the systems had different mass flow rates due to the added heat exchanger and this affected the power output. Hence, a recuperator was beneficial for this application as well.

Table 4.15: 1D results for the power generation application without recuperator - GT Suite model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	45.1	0.9	0.234	-7.97
2	48.2	40.6	0.234	1.02
3	254.0	40.6	0.234	674.95
4	159.7	1.0	0.234	565.15

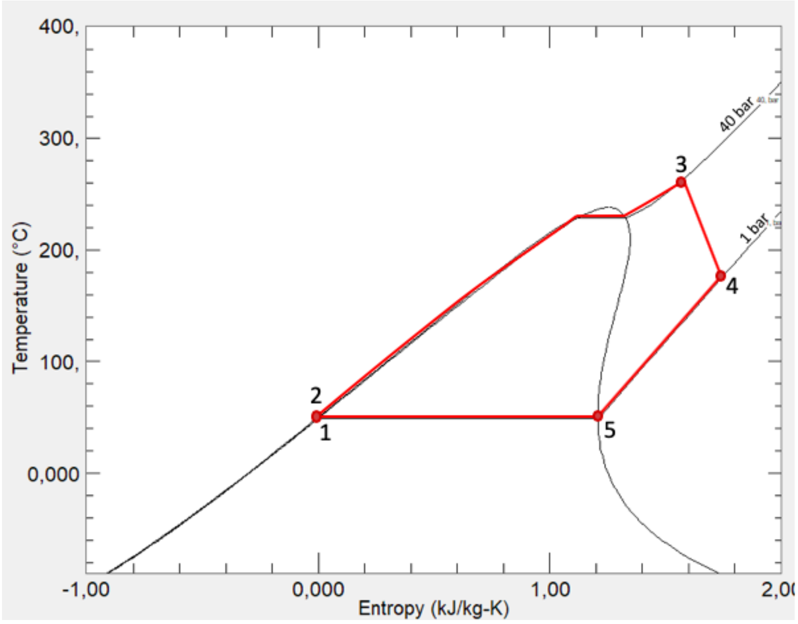


Figure 4.21: A T-s diagram for understanding the non-recuperated organic Rankine cycle for the power generation application

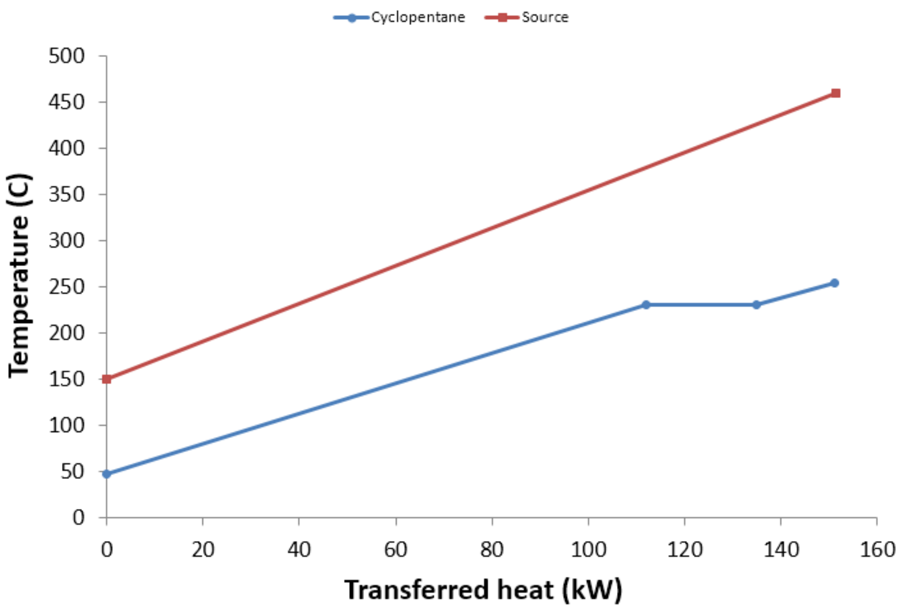


Figure 4.22: A Q-T diagram to understand the heat exchange process for the non-recuperated system

Figure 4.23 shows the various comparison of power figures for the different loads. As expected, the lower loads have lower powers due to lower exhaust energy. The 100% load has higher exhaust energy compared to the main load point of interest (75%) and yielded a higher power output of 39 kW. But this point is only used in the engine as standby extra power and so, the focus was on the 75% load point which produced 28.4 kW from the WHR system.

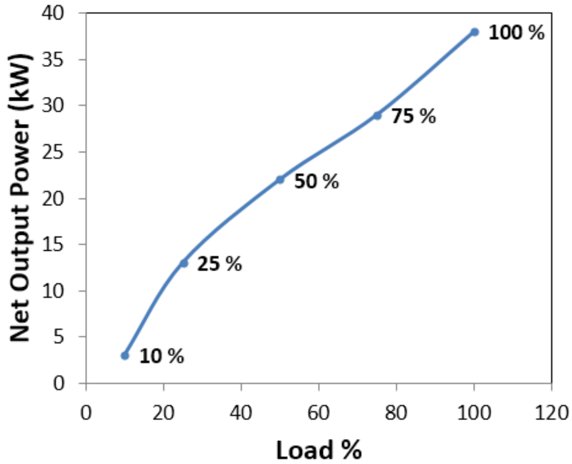


Figure 4.23: Variation of output power between the different load cases for the power generation application

4.2.2 Results from Matlab/Simulink

4.2.2.1 Marine Application

The model was tuned for the chosen engine operating point based on Table 3.4 where the pump speed and expander speed were altered to reach the suitable pressure and mass flow that give the highest net output power where cyclopentane should remain subcooled before the pump and superheated before and after the expander, and the temperature of cyclopentane does not exceed the decomposition temperature (275°C). Figure 4.24 shows the mass flow rate effect on the power output of an ORC at constant pressure (40 bar).

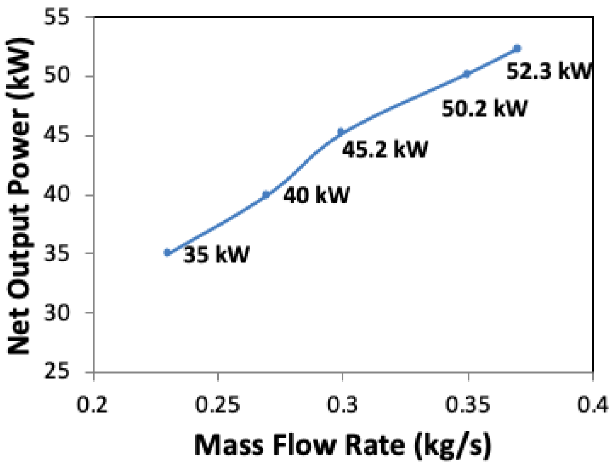


Figure 4.24: Mass flow effect on the net output power at constant pressure

Due to the increased mass flow, the work done by the expander will increase proportionally as observed with the help of Equation 2.10, thereby increasing the net output power as observed through equation 2.8. With respect to the temperature, it is decreased by increasing the mass flow and the temperature is maintained to be less than the decomposition temperature and higher than the saturated vapor temperature for each testing point. Figure 4.25 shows the net output power of ORC for each testing point. This shows the effect of pressure out of the pump on the power output of ORC at constant mass flow (0.37 kg/s). Increasing the pressure ratio of cyclopentane causes an increase in the work done by the expander thereby increasing the net output power.

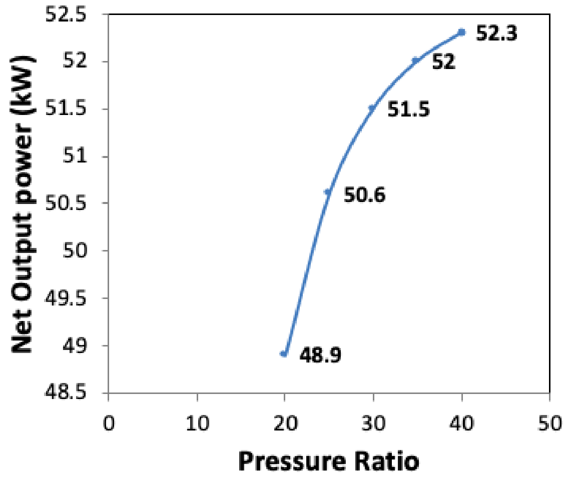


Figure 4.25: Pressure ratio effect on the net output power at constant mass flow

The expander power is controlled by the mass flow rate and the difference in enthalpy between points 3 and 4 as observed through Equation 2.10. It is seen in figures 4.24 and 4.25 that the pressure ratio has a lesser effect on the net output power than the mass flow rate. Figure 4.26 shows the different pressure lines for each case. As observed from the T-s diagram, the pressure lines are very close to each other and the difference in enthalpy between points 3 and 4 for each point has low difference in value. Hence the change in the net output power is also small.

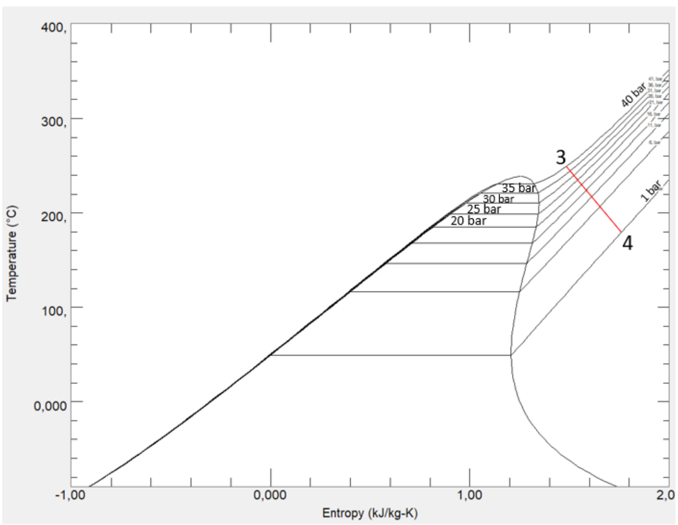


Figure 4.26: The pressure ratio for each testing point at constant mass flow rate

Figure 4.28 shows the schematic representation of the organic Rankine cycle, and Figure 4.27 shows the thermodynamic processes in the temperature vs entropy diagram for cyclopentane at the chosen engine operating point for the marine application: the adiabatic compression process in the pump (1-2), the heat absorption process in the recuperator (2-2'), the heat absorption process in the evaporator (2'-3), the adiabatic expansion process in the expander (3-4), the heat rejection process in the recuperator (4-4'), and the heat rejection process in the condenser (4'-1).

Point a is called the saturation liquid point in which cyclopentane starts to vaporize at a constant temperature. When the heat still added, the evaporation lasts until cyclopentane reaches point b in which it completely vaporized, this point is called saturation vapor point. At point 3, cyclopentane is superheated. Point 5 is the saturation vapor point in which cyclopentane starts to condense at a constant temperature. The condensation lasts while the heat is still rejected until cyclopentane reaches point c in which it completely condensed, this point is called the saturation liquid point. At point 1, cyclopentane is subcooled.

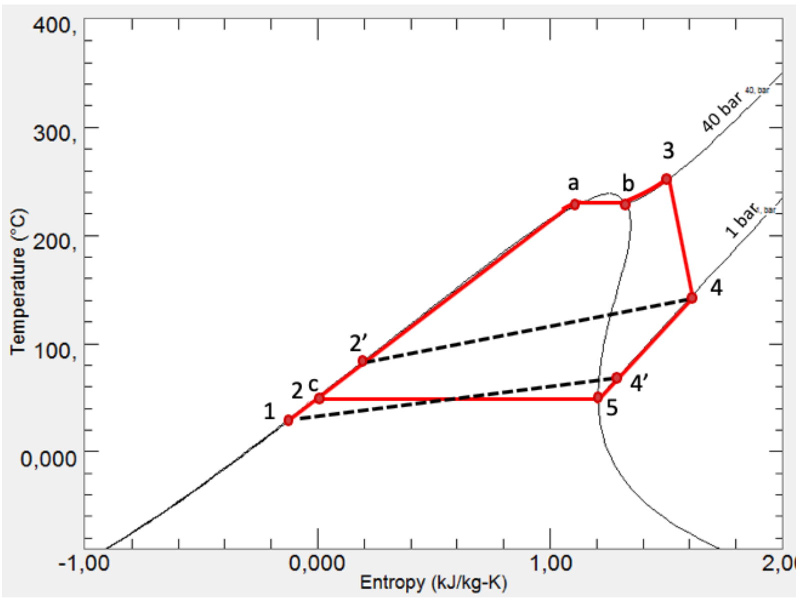


Figure 4.27: T-s Diagram of ORC for the marine application

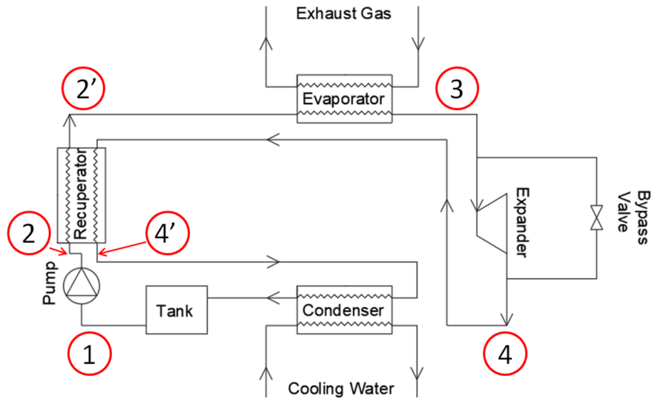


Figure 4.28: Schematic representation of the ORC model in Simulink for the marine application

Figure 4.29 shows the inlet and outlet temperatures for both exhaust gas (heat source) and cyclopentane and the heat transferred between them. The difference in temperature between the exhaust gas and cyclopentane did not reach the pinch point limit in temperature between the exhaust gas and the working fluid. Therefore, increasing the pressure could increase the heat transferred. But in this scenario, the pressure in the cycle was close to the critical pressure. Therefore, it was limited to 40 bar hence, limiting the heat transferred into the system.

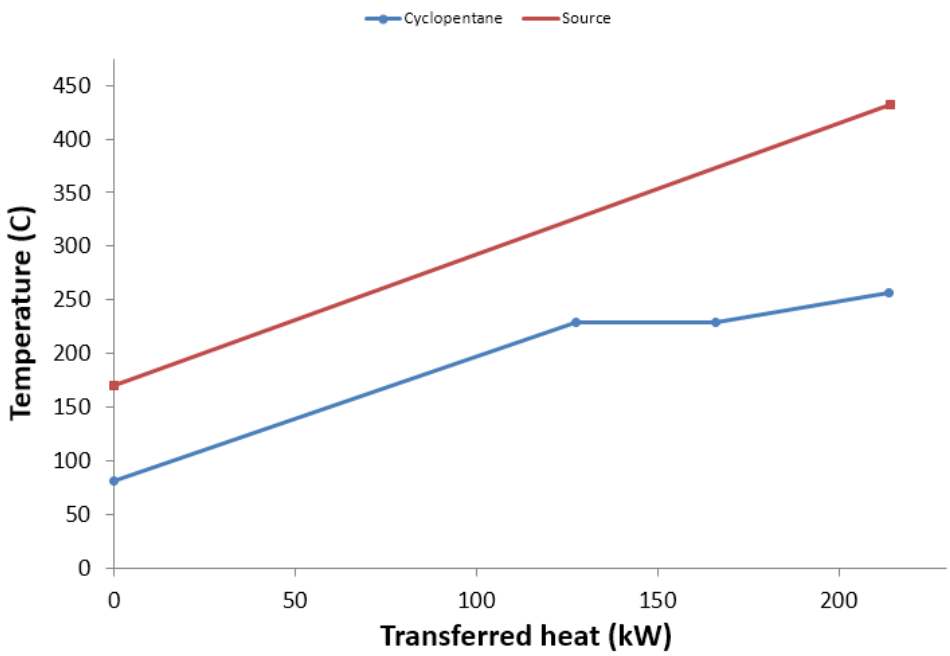


Figure 4.29: A Q-T diagram to analyze the heat exchange process for the marine system

Table 4.16 shows the values of the thermodynamic variables of the working fluid at the points mentioned in Figure 4.28 at 75% engine load, such as temperature, pressure, mass flow rate, and enthalpy.

Table 4.16: 1D marine results - WHR simulink model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	25.4	1.0	0.372	-44.75
2	28.2	40.0	0.372	-36.35
2'	81.7	40.0	0.372	68.59
3	257.5	40.0	0.372	683.30
4	143.0	1.0	0.372	538.70
4'	62.4	1.0	0.372	388.70

Table 4.17 shows the results of chosen engine operating point based on Table 3.4 for the Marine D13 Engine used in the simulation where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure in the condenser, T_{evap} is the temperature out of evaporator, T_{cond} is the temperature out of condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator heat exchanger, and η_{th} is the thermal efficiency of the system.

The mass flow of coolant for this application is 7 kg/s which is chosen based on the previously selected engine operating point (75% load) as mass flows less than 7 kg/s are not enough to completely condense cyclopentane. The coolant inlet temperature is 25°C.

Table 4.17: Results from 1D analysis for the marine application

Load Point	$\dot{W}_{net}$	$\dot{m}$	P_{evap}	P_{cond}	T_{evap}	T_{cond}	T_{exhout}	η_{th}
75 %	50.7 kW	0.372 kg/s	40.0 bar	1.0 bar	257.5°C	25.4°C	170.0°C	0.236

Tables 4.1 and 4.17 show that the input and the output conditions in the 0D and 1D models are almost identical except for the net output power, which is higher in the 1D model because the 1D model has a higher expander efficiency than the 0D model.

4.2.2.2 Power Generation Application

The model was tuned in the same way in the marine application by altering the pump and expander speed to reach the suitable pressure and mass flow that give the highest power output.

The T-s diagrams for both applications are seen to be similar. But the mass flow of cyclopentane in the power generation application is less than that in the marine application because the heat input is also less. The pressure out of the pump and the superheating temperature were identical in both diagrams, but the temperature in which cyclopentane was subcooled in the power generation application (36.2°C) is higher than it in the marine application (25.4°C) because the coolant temperature is also higher as chosen from Table 3.4.

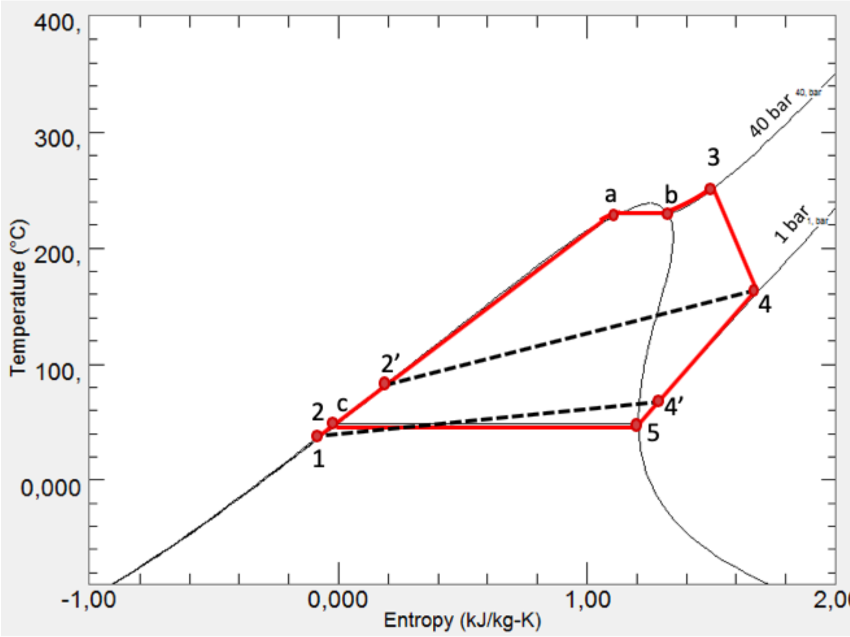


Figure 4.30: T-s Diagram of the ORC for power generation application

Figure 4.31 shows the inlet and outlet temperatures for both exhaust gas (heat source) and cyclopentane and the heat transferred between them. Similarity to the marine application, the heat transferred into the system is limited by the critical pressure.

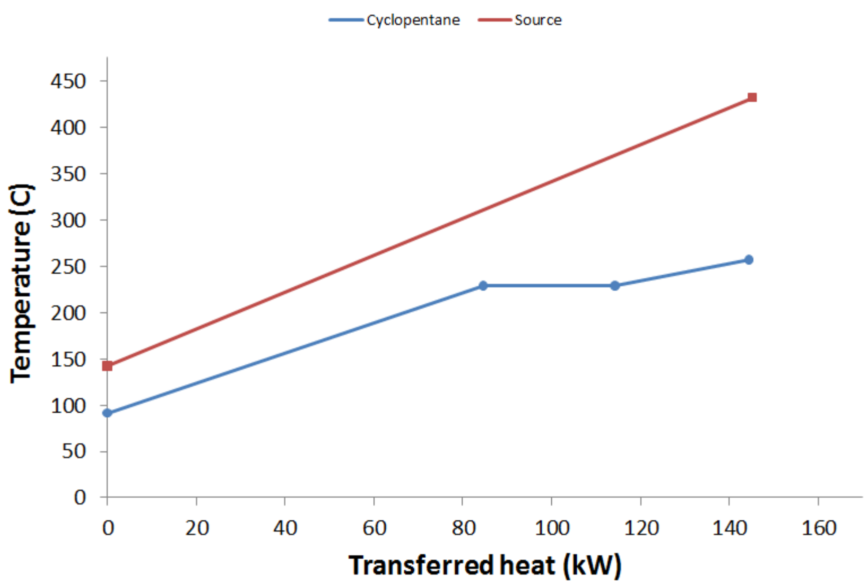


Figure 4.31: A Q-T diagram to analyze the heat exchange process for the power generation system

Table 4.18 shows the values of the thermodynamic parameters of the working fluid at the points mentioned in Figure 4.30 at 75% engine load, such as temperature, pressure, mass flow rate, and enthalpy.

Table 4.18: 1D power generation results - WHR simulink model

Point	Temperature (°C)	Pressure (bar)	Mass Flow Rate (kg/s)	Enthalpy (kJ/kg)
1	36.2	1.0	0.285	-24.89
2	39.0	40.0	0.285	-16.02
2'	91.3	40.0	0.285	88.70
3	257.5	40.0	0.285	682.90
4	142.8	1.0	0.285	538.30
4'	62.4	1.0	0.285	388.70

Table 4.19 shows the results of chosen engine operating point based on Table 3.4 for the Genset D13 Engine used in the simulation where P_{evap} is the pressure in the evaporator, P_{cond} is the pressure in the condenser, T_{evap} is the temperature out of evaporator, T_{cond} is the temperature out of condenser, T_{exhout} is the temperature of the exhaust after exiting the evaporator heat exchanger, and η_{th} is the thermal efficiency of the system.

The mass flow of coolant for this application is also 7 kg/s which is chosen based on the previously selected engine operating point (75% load) as mass flows less than 7 kg/s is not enough to completely condense cyclopentane at the pump unit. The coolant temperature as mentioned in the methodology is 35°C. Figure 4.30 shows the temperature-entropy diagram for cyclopentane at the chosen engine operating point of the Genset D13 Engine.

Table 4.19: Results from 1D analysis for the power generation application

Load Point	$\dot{W}_{net}$	$\dot{m}$	P_{evap}	P_{cond}	T_{evap}	T_{cond}	T_{exhout}	η_{th}
75 %	38.7 kW	0.285 kg/s	40.0 bar	1.0 bar	257.5°C	36.2°C	142.2°C	0.266

Tables 4.4 and 4.19 show that the input and the output conditions in the 0D and 1D models are almost identical except for the net output power, which is higher in the 1D model because the 1D model has a higher expander efficiency than the 0D model.

4.2.3 Comparison of the Simulation Results

The comparison of the models is done separately for models with and without the recuperator unit.

Marine Application

The marine application results are compared by separating the recuperated and the non-recuperated models and can be seen in tables 4.21 and 4.22.

Table 4.20: Comparison of the energy added and rejected by the system, the thermal efficiency, the expander efficiency and the net output power for the various models for the marine application

Model	$\dot{Q}_{in}$ (kW)	$\dot{Q}_{out}$ (kW)	η_{exp}	η_{th}	$\dot{W}_{net}$ (kW)
0D Analysis	254.7	215.7	0.65	0.157	39.0
1D GT-Suite (Recuperated)	254.8	216.2	0.65	0.199	38.2
1D GT-Suite (non-recuperated)	200.6	172.8	0.65	0.137	29.1
1D Simulink	267.7	217.0	0.75	0.236	39.0

Table 4.21: Result comparison of the recuperated models for the marine application with Simulink and GT-Suite software

Model	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
1D Simulink	50.7	0.372	40.0	1.0	257.5	25.4	170.0	0.236
1D GT-Suite	38.2	0.358	33.8	0.8	253.3	41.2	214.3	0.199

Table 4.22: Result comparison of the non-recuperated models for the marine application with Matlab and GT-Suite software

Model	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
0D Analysis	39.0	0.355	40.6	1.0	260.3	44.0	150.0	0.157
1D GT-Suite	29.1	0.283	32.8	0.9	254.3	46.1	203.3	0.137

The models are varied in the power output and also the other parameters in the system. The 0D model had the ideal case wherein the components did not represent the physical construction. The difference in the net power output in some cases where the input conditions are very similar was due to the expander efficiency. The Simulink model had an efficiency of 0.75 whereas the 0D model and GT-Suite model had an efficiency of 0.65 each. In both the models, the main limitation to the net power output was the working fluid limitations and this meant that the temperature and pressure of the working fluid could not be exceeded belong its critical limit. This limited the heat energy extracted from the exhaust and thus, limited the work output from the expander. The recuperated models in Table 4.21 shows the effect of a recuperator in the system. The limitation in the modelling with GT Suite affected this model too and hence, the system was not able to extract more energy from the exhaust gas and hence, the temperature of the exhaust after the evaporator unit is lower in the Simulink model. The Simulink model also had a bigger recuperator to handle the heat energy into the system which was higher than the GT-Suite model. Apart from this, the GT- Suite software could not reach the set 40 bar pressure for this application because the fluid had reached the maximum temperature limit.

Power Generation Application

Similarly, the results are compared for the power generation models separately for with and without the recuperator unit and are shown in tables 4.24 and 4.25.

Table 4.23: omparison of the energy added and rejected by the system, the thermal efficiency, the expander efficiency and the net output power for the various models for 75% load of the D13 power generation engine

Model	$\dot{Q}_{in}$ (kW)	$\dot{Q}_{out}$ (kW)	η_{exp}	η_{th}	$\dot{W}_{net}$ (kW)
0D Analysis	155.8	132.0	0.65	0.152	23.8
1D GT-Suite (Recuperated)	189.6	161.2	0.65	0.149	28.4
1D GT-Suite (non-recuperated)	160.0	134.3	0.65	0.158	25.1
1D Simulink	199.1	160.5	0.75	0.194	38.7

Table 4.24: Result comparison of the recuperated models for the power generation application with Simulink and GT-Suite software

Model	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
1D Simulink	38.7	0.285	40.0	1.0	257.5	36.2	142.5	0.266
1D GT-Suite	28.4	0.280	39.8	0.9	253.0	43.9	196.0	0.189

Table 4.25: Result comparison of the non-recuperated models for the power generation application with Matlab and GT-Suite software

Model	$\dot{W}_{net}$ (kW)	$\dot{m}$ (kg/s)	P_{evap} (bar)	P_{cond} (bar)	T_{evap} (°C)	T_{cond} (°C)	T_{exhout} (°C)	η_{th}
0D Analysis	23.8	0.226	40.6	1.0	258.9	44.0	154.0	0.157
1D GT-Suite	25.1	0.235	40.6	0.9	254.0	45.1	210.2	0.165

It is observed that the 0D model is able to extract higher energy from the exhaust. But the mass flow rate in the GT-Suite model is higher. Due to the difference in the heat exchanger efficiencies in the models, the GT-Suite model is able to produce a higher net output power. In Table 4.24, it is seen that the input conditions are very similar for both the models. The difference in power output can be attributed to the difference in the expander efficiency.

For the non-recuperated models, the 0D analysis and the GT-Suite had the same expander efficiency. But the mass flow rate in the 1D model was higher. This translated to a higher net output power. The pressure in all the models were limited to roughly 40 bar due to the fluid limitations.

5
Conclusions

The organic Rankine cycle first conceptualized by Ofeldt in 1826 using naphtha as a working fluid focused on extracting energy from low temperature sources. This same technology was used in this thesis work for modelling and analyzing the potential of waste heat recovery from a Volvo D13 engine used in marine and power generation application. Initially, various other WHR technologies were investigated from Rankine cycles to thermo-acoustic converters. The organic Rankine cycle had the advantage of being versatile in the temperature required to operate as well as ease of integration to an existing system.

The main conclusions of this thesis work are:

- Water, ethanol and cyclopentane were analyzed and compared for the operating points best suited for two applications. The fluid properties of these fluids were studied using REFPROP software. Cyclopentane was chosen as the working fluid best suited for these applications as it output the increase in the highest net power of 39.0 kW for the marine application and 23.8 kW for the power generation application.
- For the 1D modelling, the Simulink model was based off of an already existing model from Volvo Trucks. Parameters such as pump speed and expander speed were tweaked to handle the input conditions for the two applications. The model was run with a recuperator unit. For the marine application, the model had a 9% increase in power output and for the power generation, the model was able to help increase the power output by 10%.
- The GT-Suite model was based off of an example model and the various components of the system were changed to suit the applications at hand. Due to the help of a water cooled condenser, the marine application was helped achieve a power increase of 6.9%. The power generation system had a power increase of 7.6%.
- The GT-Suite models were simulated with and without a recuperator unit to see the influence it had on the system. The models were then compared and analyzed. It was concluded that for this particular model, a recuperator was beneficial as it helped achieve 7 kW more power in the marine application and about 5 kW more power in the power generation application in comparison to the non-recuperated model using the same software.

The simulations which were done helped understand the cycle as well as the limitations of the modelling process. Inferring from the results thus achieved from the various cases run in this thesis work, it can be concluded that a waste heat recovery system based on an organic Rankine cycle is feasible and will help increase the power output thereby reducing the fuel consumption and the emissions.

Development Directions

Further work can be carried out to address some limitations that were observed during the thesis work. The models could be based off of physical components for better results in the simulations. Currently, the models run at fixed load points across the engine range. The simulations would greatly improve if there were a transient cycle as an input and this this would need the help of a controller to change the other parameters to get the best output. This particular model was based only on the high temperature output from the exhaust. Further work can be done by creating another temperature cycle with low heat sources such as charge air cooler and heat from the cooling system.

Bibliography

[1] Johnson, Steven. "The invention of air: a story of science, faith, revolution, and the birth of America. Penguin", 2008

[2] Engines, W. A. "Facts about the Wright Turbocompound.", 2014

[3] Prof. R.K.Sullerey, "Rankine Cycle", IIT Kharagpur, viewed 20 September 2019, <http://home.iitk.ac.in/ suller/lectures/lec29.htm>

[4] Colonna, Piero, et al. "Organic Rankine cycle power systems: from the concept to current technology, applications, and an outlook to the future." Journal of Engineering for Gas Turbines and Power, 2015: 100801

[5] Galloway, Elijah. "History and progress of the steam engine: with a practical investigation of its structure and application" T. Kelly, 1834

[6] Campana, Francesco, et al. "ORC waste heat recovery in European energy intensive industries: Energy and GHG savings." Energy Conversion and Management 76, 2013: 244-252

[7] Zhao, Mingru, et al. "The investigation of the Recuperative Organic Rankine Cycle models for the waste heat recovery on vehicles." Energy Procedia 129, 2017: 732-739

[8] Wang, Tianyou, et al. "A review of researches on thermal exhaust heat recovery with Rankine cycle." Renewable and sustainable energy reviews 15.6, 2011: 2862-2871

[9] Bao, Junjiang, and Li Zhao. "A review of working fluid and expander selections for organic Rankine cycle." Renewable and sustainable energy reviews 24, 2013: 325-342

[10] Quoilin, Sylvain, et al. "Techno-economic survey of Organic Rankine Cycle (ORC) systems." Renewable and Sustainable Energy Reviews 22, 2013: 168-186

[11] Chen, Huijuan, D. Yogi Goswami, and Elias K. Stefanakos. "A review of thermodynamic cycles and working fluids for the conversion of low-grade heat." Renewable and sustainable energy reviews 14.9, 2010: 3059-3067

[12] Significant New Alternatives Policy 2016, *Transitioning to Low-GWP Alternatives in Domestic Refrigeration*, Environmental Protection Agency, viewed 20 October 2019, <https://www.epa.gov/snap/transitioning-low-gwp-alternatives-domestic-refrigeration>

[13] National Institute for Occupational Safety and Health 2019, *Cyclopentane*, Centers for Disease Control and Prevention, viewed 20 July 2019, <https://www.cdc.gov/niosh/npg/npgd0171.html>

[14] Stocker, Thomas, ed. "Climate change 2013: the physical science basis: Working Group I contribution to the Fifth assessment report of the Intergovernmental Panel on Climate Change. Cambridge University Press", 2014

[15] Burkhart, G., M. Klincke, and K. Willoughby. "Do the Existing Silicone Surfactants Match the Requirements of HFC-245fa-Blown Appliance Systems?." Proceedings of the Polyurethanes Expo, 2001

[16] Ginosar, Daniel M., Lucia M. Petkovic, and Donna Post Guillen. "Thermal stability of cyclopentane as an organic Rankine cycle working fluid." Energy & Fuels 25.9, 2011: 4138-4144

[17] Etalim 2018, *Etalim Solutions*, Etalim Inc., viewed 10 March 2019, <https://www.etalim.com/solutions.html>

[18] Dellacha, J., et al. "Turbocompound technology on hybrid power trains: urban buses take inspiration from F1 power units." Energy Production and Management in the 21st Century II: The Quest for Sustainable Energy 205, 2016: 179-190

[19] Rijpkema, Jelmer, Karin Munch, and Sven B. Andersson. "Thermodynamic potential of Rankine and flash cycles for waste heat recovery in a heavy duty Diesel engine." Energy Procedia 129, 2017: 746-753

[20] DieselNet 2001, *ISO 8178*, DieselNet, viewed 20 April 2019, <https://www.dieselnet.com/standards/cycles/iso8178.php>

[21] Dumont, Olivier, Rémi Dickes, and Vincent Lemort. "Experimental investigation of four volumetric expanders." Energy Procedia 129, 2017: 859-866

[22] Cao, Yue, et al. "Optimum design and thermodynamic analysis of a gas turbine and ORC combined cycle with recuperators." Energy Conversion and Management 116, 2016: 32-41

[23] Moreno, Jesus, and Thomas Lang. "Advantages of Dual Enhanced Heat Transfer Tubes in Organic Rankine Cycle Based Geothermal Plants."

[24] Badr, O., S. D. Probert, and P. W. O'callaghan. "Selecting a working fluid for a Rankine-cycle engine." Applied Energy 21.1, 1985: 1-42

A

Appendix

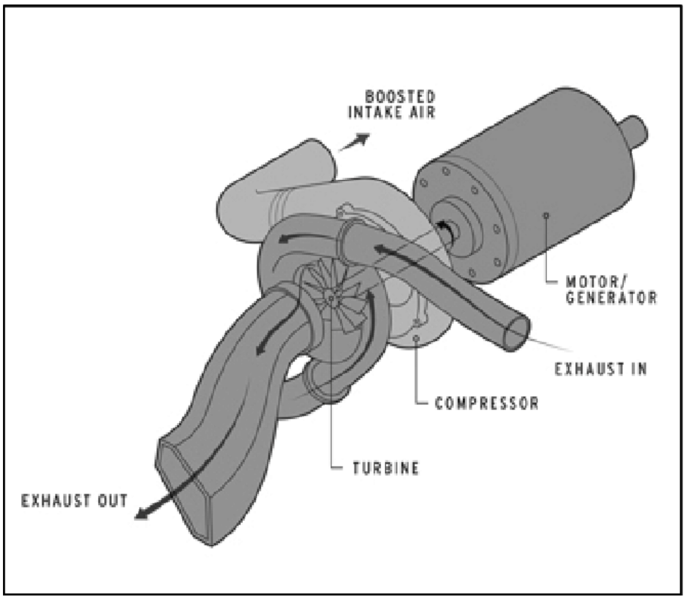


Figure A.1: Another look at how the turbocompound system works

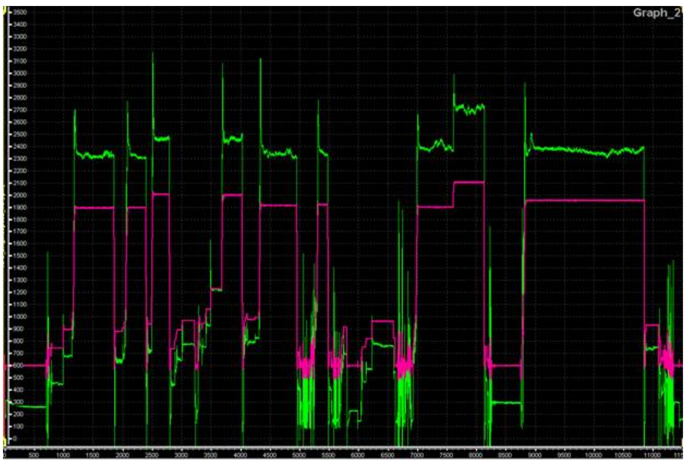


Figure A.2: A quick look at the transient drive cycle which the main operating points were selected for the marine application