BACHELOR THESIS – ME141502

TECHNICAL AND ECONOMIC ANALYSIS OF ORGANIC RANKINE CYCLE SYSTEM USING LOW-TEMPERATURE SOURCE AND REFRIGERANT LIQUID AS WORKING FLUID TO GENERATE ELECTRICITY IN SHIP

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DOUBLE DEGREE PROGRAM
DEPARTMENT OF MARINE ENGINEERING
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Institut Teknologi Sepuluh Nopember
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2016

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ANALISA TEKNIS & EKONOMIS PEMAKAIAN SISTEM ORGANIC RANKINE CYCLE (ORC) MENGGUNAKAN SUMBER SUHU RENDAH & CAIRAN REFRIGERAN SEBAGAI FLUIDA KERJA UNTUK PEMBANGKIT LISTRIK DI KAPAL

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DOUBLE DEGREE PROGRAM
JURUSAN TEKNIK SISTEM PERKAPALAN
Fakultas Teknologi Kelautan
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Surabaya
2016

APPROVAL FORM

Technical and Economic Analysis of Organic Rankine Cycle System Using Low-Temperature Source and Refrigerant Liquid as Working Fluid to Generate Electricity in Ship

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Submitted to Comply One of The Requirement to Obtain a Bachelor of Engineering Degree
on
Laboratory of Marine Machinery & System (MMS)
S-1 Program Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

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ABSTRACT

Nowadays, the shipping sector has growth rapidly as followed by the increasing of world population and the demands for public transportation via sea. This issue entails the large attention on emission, energy efficiency and fuel consumption on the ship. Waste Heat Recovery (WHR) is one of the solution to overcome the mentioned issue and one of the WHR method is by installing Organic Rankine Cycle (ORC) system in ship. ORC demonstrate to recover and exploit the low temperature waste heat rejected by the ship power generation plant. In this bachelor thesis the main source of heat to be utilized is obtained from container (7900 kW BHP, DWT 10969 mt) ship jacket water cooling system and use R-134a as a refrigerant. The main equipment consists of evaporator, condenser, pump and steam turbine to generate the electricity. The main objective of this bachelor thesis is to quantifying the estimation of electrical power which can be generated at typical loads of the main engine, to design the PFD layout for Organic Rankine Cycle system and to acquires the economic analysis of break-even point with installing ORC system and the percentage of the saving fuel oil consumption from auxiliary engine. As the final result of analysis, the ORC system is able to generate the electricity power ranged from 77,5% - 100% of main engine load producing power averagely 57,69 kW. From the economic analysis, the break-even point occurs in years 10 when comparing the fuel oil saving (profit) and cost of ORC equipment (expenditure).

Keywords - Break-even Point, Economic Ananlysis, Organic Rankine Cycle, Waste Heat Recovery, Jacket Water Cooling System

ANALISA TEKNIS & EKONOMIS PEMAKAIAN SISTEM ORGANIC RANKINE CYCLE (ORC) MENGGUNAKAN SUMBER SUHU RENDAH & CAIRAN REFRIGERAN SEBAGAI FLUIDA KERJA UNTUK PEMBANGKIT LISTRIK DI KAPAL

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ABSTRAK

Saat ini, sektor pelayaran memiliki pertumbuhan yang pesat diikuti dengan meningkatnya populasi penduduk dunia dan banyaknya permintaan transportasi umum melalui laut. Masalah ini memerlukan perhatian yang besar pada segi polusi, efisiensi energi dan konsumsi bahan bakar di kapal. Waste heat Recovery (WHR) adalah salah satu solusi untuk mengatasi masalah yang telah disebutkan dan salah satu metode WHR adalah dengan memasangkan sistem Organic Rankine Cycle (ORC) pada sebuah sistem di kapal. ORC dapat mengkonversi dan mengeksploitasi limbah panas pada suhu yang rendah. Dalam Skripsi kali ini, sumber utama panas yang akan dimanfaatkan diperoleh dari kapal MV. Tanto Tenang pada sistem pendingin mesin dan menggunakan R-134a sebagai refrigeran. Peralatan utama terdiri dari evaporator, kondensor, pompa dan turbin uap untuk menghasilkan listrik. Tujuan utama dari skripsi ini adalah untuk mengukur estimasi daya listrik yang dapat dihasilkan pada skenario beban dari mesin utama kapal, merancang desain sistem ORC dan memperoleh analisa ekonomi waktu terjadinya titik impas dengan menginstal sistem ORC antara penghematan bahan bakar (keuntungan yang diperoleh) dengan biaya peralatan ORC (pengeluaran). Sebagai hasil akhir dari analisis, sistem ORC dapat diaktifkan antara 77,5% - 100% beban mesin utama dan mampu menghasilkan daya listrik rata-rata 57,69 kW. Dari analisis ekonomi, titik impas terjadi pada tahun ke-10 ketika membandingkan antara penghematan bahan bakar minyak (profit) dan biaya peralatan ORC (pengeluaran).

Keywords – Analisa Keekonomian, Jacket Water Cooling System, Organic Rankine Cycle, Titik Impas, Waste Heat Recovery

PREFACE

First and foremost, the author would like to give a huge thanks to Allah SWT the God Almighty for giving intelligent, strength, health and favors so writers can have finished this bachelor thesis.

This bachelor thesis aims to analyze the technical and economic for installing the Organic Rankine Cycle (ORC) system in the ship as an alternative energy to produce electricity energy in ship.

The author also would to express his immeasurable appreciation and deepest gratitude for those who helped in completing this Bachelor Thesis:

1. Bapak Dr. Eng. M. Badrus Zaman, S.T, MT. as Head Department of Marine Engineering FTK-ITS, Surabaya.
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6. All of my friends particularly BISMARCK’12 who had help, cooperate, overcome seatbacks and support the writer to stay focused writes the bachelor thesis until finish**.**
7. Great number of unmentioned friends and parties for all the care, encouragement and moral support.

The author realizes that this thesis remains far away from perfect. Therefore, every constructive suggestion and idea from all parties is highly expected by author for this bachelor thesis correction and improvement in the future.

Finally, may Allah SWT bestow His grace, contentment and blessings to all of us. Hopefully, this bachelor can be advantageous for all of us particularly for the readers.

Surabaya, July 2016
Author

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CHAPTER I
INTRODUCTION

1.1. Background

Emission is a fundamental factor for any transportation vehicle, especially on a global scale as a primary vehicle of transportation at sea. In a world where trade is continuously increasing, it is estimated that 80 % of the goods are transported by sea (UNCTAD, 2012). Worldwide shipping activity produces NOx and SOx emission in approximately 12.57 Mt /y for NOx and 10,54 Mt /y for NOx (BMT Murray Fenton Edon Liddiard Vince Limited, 2000) and this number is increasing begin to if there is no action to prevent this problem.

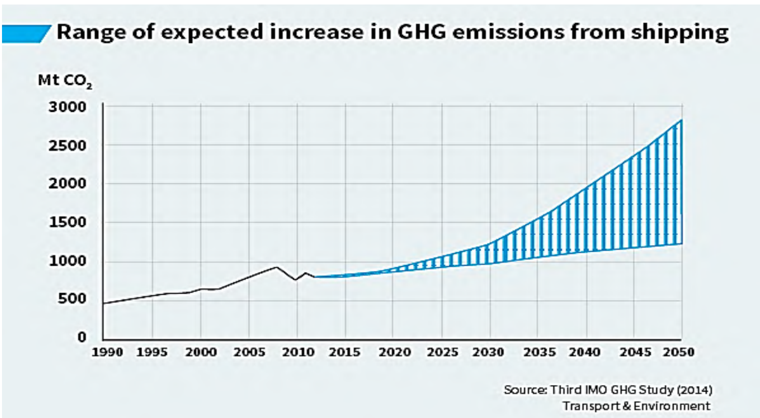

Figure I.1 Range of Expected Increase in GHG Emissions from Shipping (IMO 2014).
Source: Third IMO GHG Study (2014)

As time went from year to year, the fleet operating at sea increased significantly. Ships are the main transportation in the oceans and recorded in 1985 the total numbers of ship operated is 76.395 ships and in 2014, the total number of vessels reached 103.392 units. It was certainly triggered other related issues concerning to the environment that is caused by the emission of exhaust gas from the ships. IMO (International Maritime Organization) noted the estimated exhaust emission from ship formed in of greenhouse gases (GHG) CO2 by 2050 will reach 2800 Mt CO2 compared with 2015 which ranged from 700 Mt CO2, in other words, these emissions will increase by almost 50% - 270% if there is no treatment to reduce these emissions.

IMO (International Maritime Organization) as the main regulator for international shipping take an action firmly to are firmly on to enact regulations with the aim to reduce the level of greenhouse gases, especially CO_2 and improve the energy efficiency of shipping with introducing to Ship Energy Efficiency Management Plan which came into effect on January 1, 2013. One way to improve efficiency and reduce energy in ship exhaust gas that is using Waste Heat Recovery system.

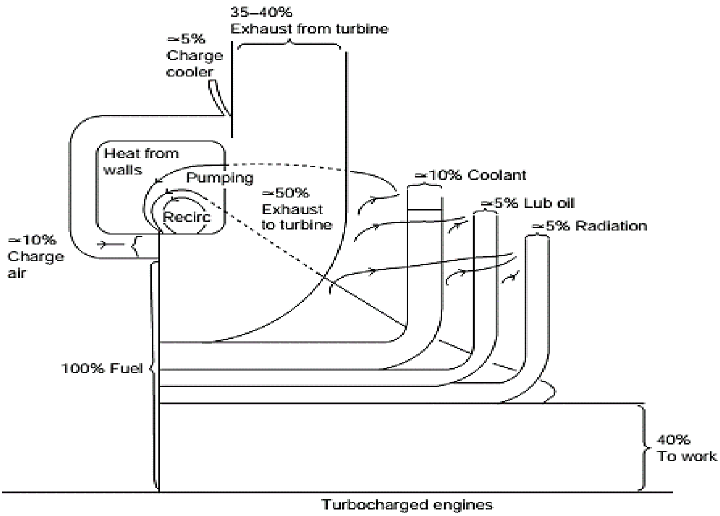

Figure I.2 Sankey Diagram for Turbocharged Engines.
Source: Pounders Marine Diesel Engines & Gas Turbines

Waste Heat Recovery system is a re-use of waste heat from the wasted fuel combustion process into the environment to be reused for more beneficial purposes. As in this case, waste heat recovery system is used to achieve fuel savings, as an alternative energy to generate electricity on the ship and reducing the exhaust gas emission from the ship.

Based on the Sankey diagram, from 100% of energy contained in the fuel oil, about 40% can be used as useful energy that can be used and the 60% remaining becomes heat loss (wasted heat), one of the heat loss percentage is 35-40% wasted in the exhaust gas, 10% of heat is wasted because the cooling system and 5% of waste heat are for the main engine lubricating system. Seeing with the wasted heat then comes the potential for utilizing the wasted heat.

The potential use of this wasted heat energy is by utilizing wasted heat energy in the jacket water cooling system, charge air cooler and lubricating oil system on the ship, but the problems are formed when the wasted heat only producing low temperature by both these systems ranging around 80^{o} C. It takes a method that can be implemented at low temperature is to result in electrical energy.

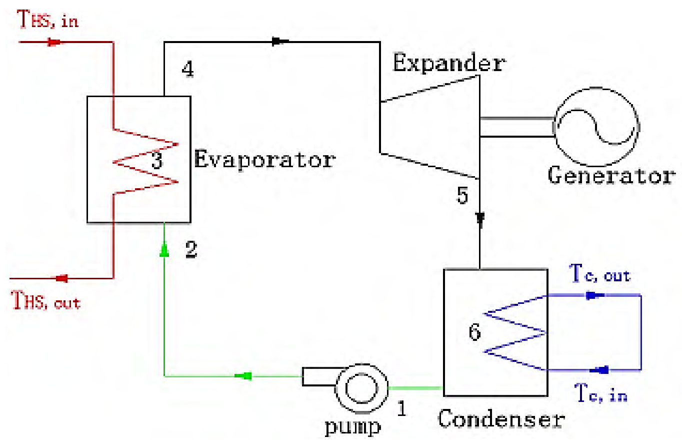

Figure I.3 ORC Schematics Diagram & Main Equipment.
Source: Thermodynamic analysis and performance optimization of an ORC Waste Heat Recovery System for Marine Diesel Engines

Organic Rankine Cycle is one of the methods to recovering the wasted heat. Organic Rankine Cycles is unpopular until the future of fossil fuels degradation has been main concerning issues at the present times, then the wasted recovery from low-grade / low temperature becomes the main interest. This method allows Rankine cycle heat recovery implemented at a lower temperature sources, in this case, is the temperature heat from the cooling system. The reason why used refrigerant in the cooling system as a heat source is because the refrigeration could boil in below 100^{o} C. Water has a very small molecular weight and high boiling point of 100^{o} C. The equipment which is used for this systems contains a pump, evaporator, condenser, and turbine.

The use of ORC system for waste heat recovery on the ship have a process. Firstly, evaporators will vaporize a working fluid such as refrigerant from the liquid phase to vapor phase which will be used to rotate the turbines and drives a generator to produces electricity. Steam that has passed through the turbines will be converted back again to liquid phase through the condenser and return again to the evaporator to converted back into steam. The use of the evaporator at ORC system will be replaced by the heat sourced from main engine jacket water cooling system, charge air cooler and lubricating oil system on the ship

1.2. Problem Formulation & Scope

Potential use of wasted heat energy that is by utilizing heat energy from wasted heat in the jacket water cooling system and lubricating oil system on the ship, but the barrier is formed which is the low temperature is produced by both of these systems ranging around 80^{o} C. Then it requires a method that can be implemented at low temperature to be able generates electrical energy.

One way to take advantage of the low-temperature waste heat that is by using organic Rankine cycle methods. Organic Rankine Cycle gives benefit to improve the efficiency of the energy, generate electricity, reduce the fuel consumption from auxiliary diesel engines and also reduce emission on ships is by using the waste heat recovery system.

ORC system was developed to manage wasted heat from water in the jacket water cooling system after cooling down the main engine and lubricating oil system after the lubricating oil lubricate the main engine as it will take some heat from the ship to be used in other energy (electric energy). There is some heat source available in the ship to obtain the heat and the heat itself will be used as an evaporator to heat the refrigerant The source of heat that will be used for the heat source is from the low-temperature heat from the jacket water cooling system and lubricating oil system from the main engine.

There are some criteria to select the most suitable organic working fluid candidates, the criteria are having a stability of chemical compounds when operated in operating conditions, has a slight impact to the environment (low Global Warming Potential and low Ozone Depletion Potential), provide safety aspect during operated (non-flammable, non-toxic) and also having the low cost.

With both sources that have been described, then at the end of this time the task of comparing the system which one will be better to be implemented on the ship taking into account technical and economic terms. The economic terms will have to compare the fuel savings resulted from using the Organic Rankine Cycle with the capital costs incurred for the installation of equipment's organic Rankine cycle on the ship, so the results would be seen how long it will reach the breakeven point and how many benefits to be gained during the operation after the breakeven point is reached.

Based on the description that has been concluded above, the several number of problems can be concluded are:

1. How much the possibility of power generated in ORC by utilizing the heat from jacket water cooling system?
2. How is the design for energy conversion system using wasted heat from the jacket water cooling system?
3. What is the suitable organic compounds (refrigerant) as a working fluid?
4. How much fuel oil savings able to be done by the system?
5. When the break event point times is happening?

1.3. Problem Limitation

1. Planning system of Organic Rankine Cycle only designed with four main consists of pump, evaporator, turbine and condenser.
2. The economic terms only find the breakeven point times from comparing the fuel savings resulted from ORC with the capital costs incurred for the installation of ORC equipment on the ship and the profit after the breakeven point is reached
3. Head loss and head static of the pump is assuming as zero
4. The analysis of technical and economic is not including piping system and its material
5. System condition is in equilibrium state
6. The control and automation of control valve is not considered.

1.4. Objective

The objectives of writing this bachelor thesis final assignment is:

1. Designing energy conversion system with using Organic Rankine Cycle for one selected ship machinery plant.
2. Utilizing the main engine jacket water cooling system on the ship to generate electricity in the ship.
3. Quantifying estimation of electrical power which can be generated at typical loads of the main engine.
4. Obtaining the economic analysis with installing ORC system and the percentage of the saving fuel oil consumption from auxiliary engine.
5. Designing the PFD layout for Organic Rankine Cycle equipment’s in Engine Room.

1.5. Benefit

The benefit obtained from the writing of this bachelor thesis is:

1. Demonstrate the method to utilizing the wasted heat from low temperature wasted heat use jacket water cooling system of the main engine cooling system in ship using the Organic Rankine Cycle system model.
2. Demonstrate the economical aspect of installing this Organic Rankine cycle systems in the ship.
3. Become as a references green technology topic in the ship.

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CHAPTER II
LITERATURE REVIEW

2.1. Organic Rankine Cycle

Organic Rankine Cycle is one of the methods to recovering the wasted heat. The problems from the water can be prevented by selecting the most appropriate working fluid. The working fluid is use from organic compounds properties and has a high molecular mass and lower boiling point than water. Compared with water, using an organic compound as a working fluid only need low temperature and less heat to evaporate.

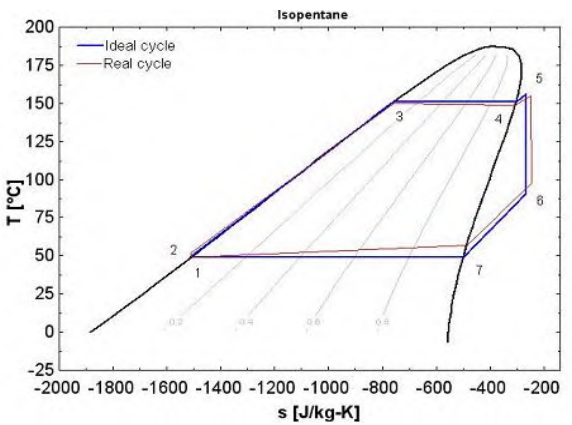


Figure II.1 T-S Diagram of Isopentane Organic Fluid
Source : http://www.wikiwand.com/en/Organic_Rankine_cycle

This method allows Rankine cycle heat recovery implemented at a lower temperature sources, in this case, the temperature heat from the main engine cooling water. The Organic Rankine Cycle uses refrigerant liquid besides water because the refrigerant has a low boiling point. The low temperature itself can be converted into useful work and generates electricity from main engine water cooling temperature outlet after the main engine. The equipment which is used for this systems contains pump, evaporator, condenser and turbine.

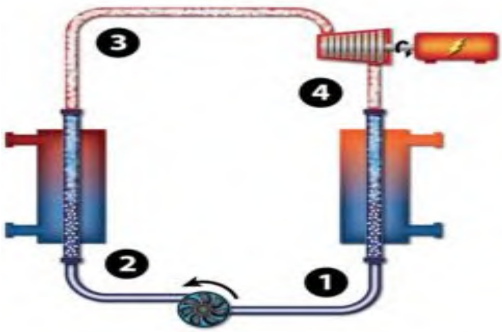


Figure II.2 Organic Rankine Cycle Schematics Diagram & Main Equipment.
Source : http://g-tet.com

2.2. Organic Rankine Cycle Processes

Organic Rankine cycles is a Rankine Cycles where the organic fluid is used replacing the steam in Rankine cycles. The Organic Rankine process is similar with the Rankine cycles. In this cycle the use of organic fluid is selected to optimize the output from heat source which have low temperature and low pressure and this problem which cannot be solved with Rankine cycle and water is used as a working fluid in the system.

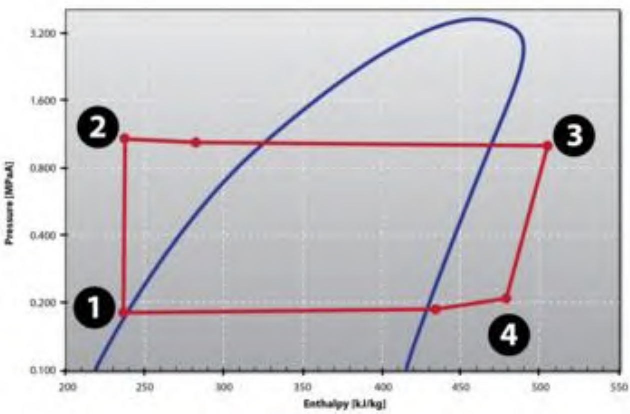


Figure II.3 P-h Diagram of Organic Rankine Cycle.
Source: http://g-tet.com

The picture above shows the Pressure – enthalpy diagram from organic Rankine cycle where this cycle has four process of thermodynamic which is same with normal Rankine cycle.

Process 1 - 2 (showed in point 1 and point 2) is an isentropic compression from pump. The ideal pump work is given by:

$$W = m\,(h_2 - h_1) \tag{II.1}$$

Process 2 - 3 (showed in point 2 and point 3) is a heating process of organic working fluid with constant pressure in evaporator. The heat absorbed by organic working fluid is given by:

$$Q = m\,(h_3 - h_2) \tag{II.2}$$

Process 3 - 4 (showed in point 3 and point 4) is an expansion process with isentropic expander. Isentropic expansion is given by:

$$Q = m\,(h_4 - h_3) \tag{II.3}$$

Process 4 - is a subcooled condensation of organic working fluid in constant pressure inside a condenser. The heat released by the organic working fluid is given by:

$$Q = m\,(h_4 - h_1) \tag{II.4}$$

2.3. Possibility Usable Wasted Heat Source

According to the U.S Department of Technology, waste heat sources are characterized in the table below:

Table II.1 Classification of Temperature.

Temperatures	Classification
> 650°C	High Temperature
230°C – 650°C	Medium Temperature
< 230°C	Small Temperature

Source : (U.S. Department of Tehnology, 2008)

A study (Serrano, et al., 2012) analyze the possibility usable residual heat sources are analyzed distinctly reflecting the potential uses in different cycle configurations. The following heat sources were considered as a potential use of heat source are the exhaust gas heat, the intercooler of the low pressure air outlet, the exhaust gas recirculation (EGR) cooler, the aftercooler where the high-pressure air is cooled, and the engine block cooling water (jacket cooling water) and this analysis use 12-two-stage turbocharged heavy-duty diesel engine as prime mover.

A study analysis of the correlation between ORC and main engine load (Song, 2015), the net power output will be depending on typical load factor conditions off the marine diesel engine for example if the engines work in 50%, 75%, 85% & 100% engine load. A higher engine load will give the higher temperature for evaporating the working fluid, the higher mass flow rate of a heat source and because the factor of this parameter is positively related to the engine load capacity.

2.4. Working Fluid Selection for Organic Rankine Cycle

The organic working fluids selection is a key vital to design the Organic Rankine Cycle system. Different organic working fluids will result different electricity power generated and the temperature of this organic working fluids is relatively low.

Comparing to the Rankine cycle, water is frequently applied as the working fluid in the system. Water have some beneficial advantages that are the water chemical composition is stable, no special requirements needed for the material construction, cheap, non-toxic and has no negative impact to the environment. However, the water also has a problem if it is used as a working fluid compared with a refrigerant, to prevent condensation during expansion happen in expander (turbines) the water need of superheating (up to 600^{o}C), has a high boiling point of 100^{o} C and have high risk of corrosion in the turbines blades. Regarding this explanation before, the water is more suitable implemented for high-temperature applications & large systems.

Since the environment has becoming a big issue in a recent time, the Montreal Protocol creates the classification to classify the refrigerants according to its potential harming or depleting the earth fragile Ozone layer. The classification is divide into two groups that is called ODP and GDP. ODP (Ozone Depletion Potential) is the potential for a single molecule of the refrigerant to destroy the Ozone Layer. GDP (Global Warming Potential) is a measurement of how much effect the given refrigerant will have on Global Warming in relation to Carbon Dioxide.

For selecting the working fluids for organic Rankine cycle, the analysis was limited to non-flammable, non-toxic and having no potential to ozone layer depletion potential. Some of the now outlawed fluids are compared against some of the newer HFC refrigerants that are currently considered for ORC duty (Brasz & Bilbow, 2004).

The requirements for selecting the refrigerants for the organic Rankine cycle are (Setiawan, 2011) :

1. The refrigerant has a high thermal conductivity. This parameter is important to determine the characteristics of heat transfer.
2. The condensation pressure of the refrigerant is not too high. If the condensation pressure is lower, then the it tends to be safer because the tendency to have a leakage, explosion and damage will relatively low.
3. The refrigerant has a low viscosity in liquid phase or vapor phase. This parameter will give an impact to pressure drop as the working fluid is flowing inside the pipe.
4. The refrigerant is environmental friendly (low ODP and GWP).
5. The refrigerant is non-toxic.
6. The refrigerant is non-flammable and non-explosive.
7. The refrigerant is low cost and has a high availability.

Table II.2 Physical Properties R-152a

Physical Properties of Refrigerants	HCFC R-152a
Product name	Difluoroethane
Chemical formula	CH_3CHF_2
Molecular Weight	-24 ^oC
Boiling Point	-87.8 ^oC
Freezing Point	113. ^oC
Critical Temperature	4.52 Mpa
Ozone Depletion Potential (ODP)	0
Global Warming Potential (GWP) (CO_2=1)	140
Latent Heat of Vaporization at 101 kPa	337.7 kj/kg
Vapor Pressure at 25 ^oC	614.3 kPa
Liquid Density at 25 ^oC	889.2 kg/m^3
Vapor Pressure at 101 Kpa	3.315 kg/m^3
Flammability Limit at 101 Kpa	None
ASHRAE Standard 34-1997 Safety Rating	A2

Source: The Utilization of Waste Energy Recovery System for Generating Electricity 1 MW System by Using Organic Rankine Cycle in PT. PJB Paiton Probolinggo, Haryadi, 2011

2.5. Main Equipment for Organic Rankine Cycle

The main equipment is required to design the simple organic Rankine cycle consist of condenser, pump, evaporator and expander.

2.5.1. Pump

Pump is the equipment that have a function to transfer fluid within the pipe from one location to another location. In this organic Rankine cycle, pump is used to flow the refrigerant that has been cooled from the condenser in liquid phase forms back to the evaporator. Another function of this pump is to creates the pressure for the refrigerants as the working fluid.

Pump has a specification that must be achieved in order to fulfill the maximum operation satisfaction. The specification of the pump is important in the process for selecting the most suitable pumps is figured on the table below:

Table II.3 Required Data for Pump Selection.

No	Required Data	Remarks
1.	Capacity	Required the number for maximum and minimum capacity
2.	Suction condition	Suction height from the suction surface to the height of pump level Height of the suction level Working pressure in suction Suction pipe condition
3.	Discharge condition	Height of the fluid discharge to the pump level Discharge pressure Discharge pipe condition
4.	Total pump head	Defined by the previous explained condition
5.	Fluid type and properties	Sea water, fresh water, chemical, oil, temperature, specific gravity, viscosity, solid particle
6.	Working condition	Continuous, interrupt able, running hour
7.	Prime mover	Electric motor, combustion engine, steam turbine
8.	Vertical or horizontal shaft	Defined by pump manufacturers
9.	Pump installation location	Outside or inside the building, dynamic movement (marine), temperature fluctuations

Source: The Utilization of Waste Energy Recovery System for Generating Electricity 1 MW System by Using Organic Rankine Cycle in PT. PJB Paiton Probolinggo, Haryadi, 2011

2.5.1.1. Pump Classification

Pump is widely generally classified as centrifugal pumps and positive displacement pump. Centrifugal pump is rotating the vane impeller to produce a head and flow by increasing the velocity of the fluid pumped by this machine. This type of the pumps can be classified in below:

- End suction pumps
- In-line pumps
- Double suction pumps
- Vertical multistage pumps
- Submersible pumps
- Self-priming pumps
- Axial-flow pumps
- Regenerative pumps

Positive displacement pump delivers a constant volume of liquid for each cycle of discharge pressure or head. The pump is operating by filling and then replacing a given volume of liquid. The positive displacement pump can be classified as:

- Reciprocating pumps
- Power pumps
- Steam pumps
- Rotary pumps (gear, love, screw, vane, regenerative and progressive cavity)

2.5.1.2. Temperature Rise in Pump

No pump is perfect with 100% efficiency. The energy lost in friction and hydraulic losses are transformed to heat - heating up the fluid transported through the pump. The formula for the temperature rise in the pumps is figured below: (engineering tool box)

$$\Delta T = \frac{H\,(1-\eta)}{102(U)(\eta)} \qquad \text{(II.5)}$$

Where: ΔT = *temperature rise in the pump (^{o}C)*
H = total head (m)
η = *pump efficiency*
U = fluid specific heat (kJ/kg ^{o}C)

2.5.1.3. Pump Capacity

Pump Capacity is the amount of water pumped per unit time. The pump capacity formula is figured below.

$$Q_s = \frac{\dot{m}}{\rho} \tag{II.6}$$

Where: Q_s = *pump capacity* (m^3/s)
$\dot{m}$ = *mass flow rate (kg/s)*
ρ = *fluid density* (kg/m^3)

2.5.1.4. Head of the pump

Assuming the pump is located in the same deck with another equipment, and head of the pump is calculated by the pressure at the discharge section:

$$h = \frac{P\,.\,10{,}197}{SG} \tag{II.7}$$

Where: h = *Head (m)*
P = *Pressure (bar)*
SG = *Specific gravity* (kg/m^3)

2.5.2. Heat Exchanger.

A heat exchanger is a heat transfer device that is used for transfer of internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix. Common examples of heat exchangers familiar to us in day-to-day use are automobile radiators, condensers, evaporators, air preheaters, and oil coolers. Heat exchangers can be classified into many different ways (Thulukkanam, 2013).

2.5.2.1. Condenser

A condenser is an equipment (heat exchanger) or unit used to condense a working fluid. In this organic Rankine cycle system, condenser is used to return refrigerant steam phase from turbine back to the liquid phase so the refrigerant can be use again. A condenser also has a function to create the lower back pressure in the turbine. With lower backpressure, the efficiency of the cycles and turbine work will increase. The condenser widely divided into two type that is surface condenser and direct-contact condenser.

Figure II.5 *Surface Condenser*

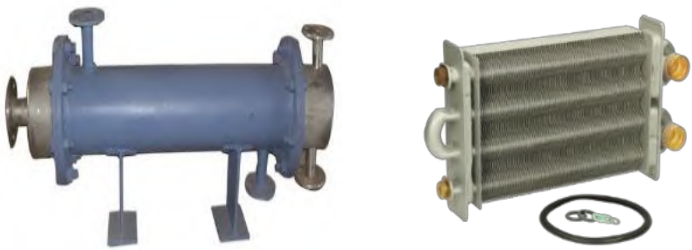
Figure II.5 *Direct Contact Condenser*

Source:http://www.apiheattransfer.com/Product/114/Steam-Surface-Condensers

For the system, water is used as a working fluid to cool down the refrigerant steam so that the refrigerant will return into liquid phase again.

2.5.2.2. Evaporator

The evaporator is a heat exchanger equipment to evaporate the phase from the liquid phase into steam phase. Evaporator has a basic function such as heat exchangers and for separating the vapor formed from the liquid. Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment and the common used type from evaporator is shell and tube heat exchanger type and compact heat exchanger type (Çengel & Ghajar, 2015). Both type of the heat exchanger has advantage and disadvantage. However, the compact heat exchanger will be selected due to the small size and the ship also limited space to install new equipment.



(a) (b)

Figure II.6 Type of Heat Exchanger (a) Shell and Tube & (b) Compact
Source :www.heatexchangerindias.com & http://www.plumbase.co.uk/

In this final assignment, evaporator used to evaporate the liquid refrigerant in ORC system. The refrigerant that become the steam will be channeled to the turbine. The hot side of heat exchanger tube will be replaced by a jacket water cooling system in ship.

2.5.2.3. Brazed Plate Heat Exchanger

Brazed PHE is a compact Plate Heat Exchangers for high-pressure and high-temperature duties.

Figure II.7 Brazed Plate Heat Exchangers
Source: Heat Exchanger Design Handbook Second Edition, 2013

It constructed of a series of corrugated metal plates but without the gaskets, tightening bolts, frame, or carrying and guide bars. It consists of stainless steel plates and two end plates.

The plates are brazed together in a vacuum oven to form a complete pressure-resistant unit. The two fluids flow in separate channels. Brazed PHEs accommodate a wide range of temperatures, from cryogenic to 200°C. Brazed PHE give a number of critical advantages: a sealed, compact system, high temperature and pressure capability, gasket-free construction, high thermal efficiency, and ideal for refrigeration and process applications. (Thulukkanam, 2013)

2.5.2.4. Phase Change in Condenser and Evaporator

The phase change process is also occurred in the evaporator (heat exchanger). During the refrigerant entering heat exchanger (condenser and evaporator) in this organic Rankine cycle system. For this part in the process flow of this organic Rankine cycle, the refrigerant which is in vapor phase enter the condenser as a hot stream.

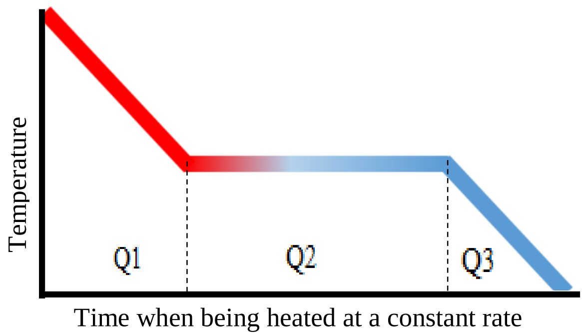


Figure II.8 Illustration of phase change in condenser

By using the cooling water to cooled down the hot refrigerant into liquid phase in this condenser. The refrigerant gradually condensed and become into liquid phase. This phenomenon is also occurred in the evaporator but in the different phase, from liquid phase become to vapor phase.

Table II.4 References and formula for Figure 2.7

Symbol	Condition	Formula	
Q1	Evaporated condition	$\dot{m}.c_p.(\mathrm{T_{in}} - \mathrm{T_{sat}})$	(II.8)
Q2	Saturated condition	$\dot{m}.Lv$	(II.9)
Q3	Subcooled condition	$\dot{m}.c_p.(\mathrm{T_{sat}} - \mathrm{T_{out}})$	(II.10)

Where: $\dot{m}$ = mass flow rate (kg/s)
c_p = specific heat (kJ/kg.°C)
$\mathrm{T_{in}}$ = temperature inlet (°C)
$\mathrm{T_{sat}}$ = temperature at saturated point (°C)
$\mathrm{T_{out}}$ = temperature outlet (°C)
Lv = latent heat of vaporization (kJ/kg)

In example in the process flow of this organic Rankine cycle, the refrigerant which is still in liquid phase enter the evaporator as a cold stream. By using the heat from hot stream which flow in evaporator, the refrigerant gradually evaporates and become a vapor phase. This phenomenon is also occurred in the condenser but in the different phase, from vapor phase become to liquid phase.

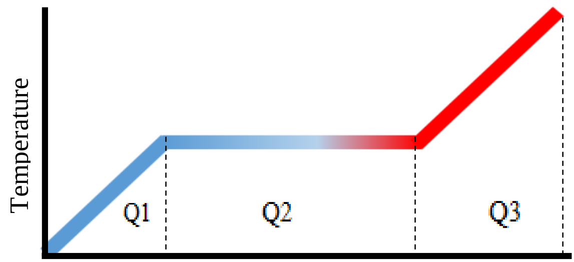


Figure II.9 Illustration of phase change in evaporator

Table II.5 References and formula for Figure 2.10

Symbol	Condition	Formula	
Q1	Subcooled condition	$\dot{m}.c_p.(\mathrm{T_{sat}} - \mathrm{T_{in}})$	(II.11)
Q2	Saturated condition	$\dot{m}.Lv$	
Q3	Evaporated condition	$\dot{m}.c_p.(\mathrm{T_{out}} - \mathrm{T_{sat}})$	(II.12)

Where: $\dot{m}$ = mass flow rate (kg/s)
c_p = specific heat (kJ/kg.°C)
T_{in} = temperature inlet (°C)
T_{sat} = temperature at saturated point (°C)
T_{out} = temperature outlet (°C)
Lv = latent heat of vaporization (kJ/kg)

2.5.2.5. Pressure drop

Pressure drop is an important parameter in heat exchanger design. Limitations may be imposed either by pumping cost or by process limitations or both. The heat exchanger should be designed in such a way that unproductive pressure drop is avoided to the maximum extent in areas like inlet and outlet bends, nozzles, and manifolds. At the same time, any pressure-drop limitation that is imposed must be utilized as nearly as possible for an economic design. The formula which is used to calculate the pressure drop in plate type heat exchanger is (Thulukkanam, 2013):

$$\Delta p = \frac{1{,}5 {G_p}^2 n_p}{2 g_c \rho_i} + \frac{4 f L G^2}{2 g_c D_e}\left(\frac{1}{\rho}\right)_m \pm \frac{\rho_m g L}{g_c} \qquad \text{(II.13)}$$

Where: G_p = fluid mass velocity in port (kg/m^2 s)
n_p = number of plates
g_c = conversion factor (kg·m/N·s^2)
ρ = fluid density (kg/m^3)
f = friction factor
L = plate height (m)
G = mass velocity through the core (kg/m^2. s)
D_e = equivalent diameter (m)
g = gravity acceleration (m/s^2)

2.5.3. Turbine.

The turbine will be used to rotate the motor generator. The principal is that the prime mover in the power field generating of electric power generation. Rotation in the turbine is generated from the vapor which resulted by the expansion to the turbine impeller with significantly pressure drop and the kinetic energy different is happen. The rotation of the turbine is generated from the vapor which resulted by the evaporator. After the vapor passed the turbine, the vapor will flow back again to the condenser where this vapor from the refrigerant will be converted again back to the liquid phase (Yahya, 2002).

Turbine efficiency can be affected by the diameter of the turbine, level of stage, length of impeller, and the surface conducting the vapor. The problem in the turbine will decreasing the efficiency and performance of the turbine. Various types of steam turbines can be classified in following below:

1. In the basis of flow direction:
   - Axial
   - Radial
   - Tangential

2. On the basis of expansion process:
   - Impulse
   - Reaction
   - Compiled impulse and reaction

3. On the basis of number of stages
   - Single Stage
   - Multi-Stage

4. On the basis of steam condition
   - High pressure non-condensing
   - High pressure condensing
   - Back pressure
   - Regenerative
   - Reheating
   - Extraction
   - Mixed pressure
   - Exhaust turbine

2.5.3.1. Temperature from Turbine Outlet

If a gas/steam in turbine is expanded from pressure P1 to pressure P2 and the temperature get colder adiabatically the temperature is given by the formula:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{II.14}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{II.15}$$

Where: T_2 = Temperature outlet turbine (^oC)
T_1 = Temperature inlet turbine (^oC)
P_1 = Pressure inlet turbine (bar)
P_2 = Pressure outlet turbine (bar)
γ = Specific heat ratio (Cp/Cv)

2.6. Sankey Diagrams

Sankey Diagrams are kind of a flowcharts that representing the distribution of an energy process and also the losses of a material or even and energy.

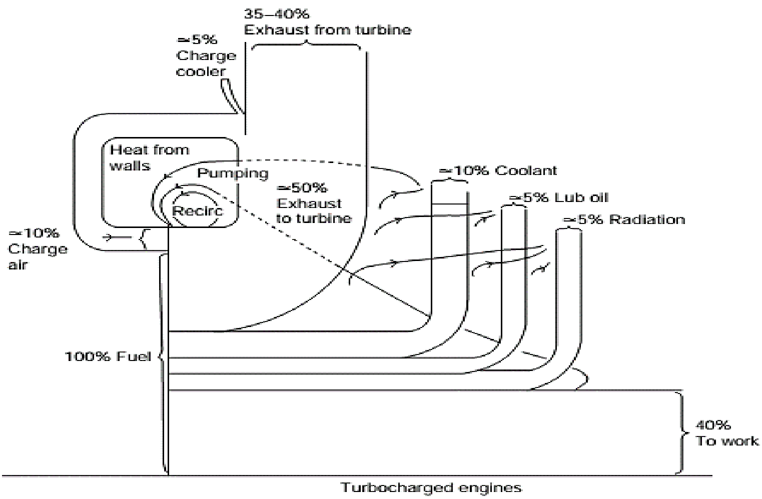


Figure II.10 Typical Sankey Diagram of Turbocharged Diesel Engine
Source: Pounders Marine Diesel Engines & Gas Turbines, 2009

In the terms of the main engine of the ship (i.e MAN B&W), Sankey diagrams is representing that from 100% energy combusted from fuel, cannot be delivered 100 % into the effective energy. The energy cannot be delivered up to 100% from fuel energy into effective energy because the energy is transformed to the heat in several percent regarding to the operation and the engine itself. From the following picture the 100% energy is dissipated up to 6,69% into the heat in cylinder water cooler (jacket water cooler).

2.7. Estimating the Engine Load Based on RPM

To estimate the engine load based on RPM of main engine is using a formula:

$$\frac{n_1}{n_2} = \frac{(P_1)^3}{(P_2)^3} \qquad \text{(II.16)}$$

Where: n_1 = *Given RPM*
n_2 = *Max RPM (100% load)*
P_1 = *Load (%) at* n_1
P_1 = *Load (%) at* n_2 *(100% RPM)*

2.8. Economic Terms

2.8.1. Forecast

Forecasting is that the method of constructing predictions of the longer term supported by the past data, present data and analysis of trends. In this research forecasting is used to predict the future price of crude oil.

2.8.2. Installation Cost & Maintenance Cost

a. Maintenance Cost

As long as the equipment get older, the maintenance cost is increasing. The average maintenance cost values are normally used in estimates. Maintenance cost can vary from 1 or 2% to over 15% of the project capital cost per year. For simple plants with relatively mild, noncorrosive conditions, an allowance of 3 to 5% should be adequate. For complex plants and severe corrosive conditions, this factor can be 10 to 12% or even higher. (Humphreys, 2006)

b. Installation Cost

Installation cost (setup cost) is the cost incurred to get equipment ready to process a different batch of goods. According to "Project and Cost Engineers Handbook" installation cost can be obtained by multiplying the purchase cost by cost factor.

$$Installed\ Cost = \ Purchase\ cost\ x\ F \qquad (II.17)$$

Cost factor is varying depend on the equipment which will be installed and the value of the factor is listed in the figure below.

Table II.6 Distributive Labor Factors for Setting Equipment

Equipment type	Factor[a]	Equipment type	Factor[a]
Absorber	20	Hammermill	25
Ammonia still	20	Heater	20
Ball mill	30	Heat exchanger	20
Blower	35	Knockout drum	15
Briquetting machine (with mixers)	25	Lime leg	15
Centrifuge	20	Methanator (catalytic)	30
Clarifier	15	Mixer	20
Coke cutter	15	Precipitator	25
Coke drum	15	Regenerator (packed)	20
Condenser	20	Retort	30
Conditioner	20	Rotoclone	25
Cooler	20	Screen	20
Crusher	30	Scrubber (water)	15
Cyclone	20	Settler	15
Decanter	15	Shift Converter	25
Distillation column	30	Splitter	15
Evaporator	20	Storage tank	20
Filter	15	Stripper	20
Fractionator	25	Tank	20
Furnace	30	Vaporizer	20
Gasifier	30	Water scrubber	20

Source: Project & Cost Engineers Handbook,2006

2.8.3. Break-even Point

The break-even point (BEP) happen when the total cost (expenditure) and the total revenue (profit) are equal, there are no gained profit or loss. In the simple way, the break-even point is happening when the profit is equal to zero.

2.8.4. Inflation

Inflation is an increase in the amount of money necessary to obtain the same amount of goods or services before the inflated price was present (Blank & Tarquin, 2011).

2.8.5. Depreciation

Depreciation is a book method (noncash) to represent the reduction in value of a tangible asset. The method used to depreciate an asset is a way to account for the decreasing value of the asset to the owner and to represent the diminishing value (amount) of the capital funds invested in it (Blank & Tarquin, 2011).

2.8.6. Present Value

Present value is the value today of money that is equivalent to some future or past profit or expense (Humphreys, 2006).

$$Present\ value\ = \frac{P}{(1+i)^n} \qquad \text{(II.18)}$$

Where: P = value of an item after n period
$\boldsymbol{i}$ = interest rate per period

2.8.7. Interest

It is the rate a bank or other lender charges to borrow its money, or the rate a bank pays its savers for keeping money in an account.

2.9. Previous Research

Two separated ORC have presented in ORC systems designed using two waste heat sources to be recover from jacket water cooling system and engine exhaust gas (Yang & Yeh, 2015), with R245fa and benzene as the working fluids, respectively. The temperature of the jacket cooling water is low, which results in low net power output and low thermal efficiency of the ORC system. With R236fa or R245fa as the working fluid and the mass flow rate of the working fluid is ranged between 0,5 – 1 kg/s, the maximum net power output reaches 10.3 kW and the thermal efficiency is 5.2%. In consideration of the impact to the environment, R245fa is selected due to its low GWP value. The temperature of the exhaust gas out of the marine diesel engine is 300^{o}C, and the mass flow rate is 118.9 kg/s. The maximum net power output of the total system reaches 101.1 kW, which increases the marine diesel engine power by 10.2% by using the exhaust gas and the mass flow rate of the working fluid is ranged between 0,62 – 0,67 kg/s (Yang & Yeh, 2015).

As in (Soffiato, et al., 2015) have made a detailed analysis of the engines operation is first performed to evaluate all thermal streams released by the engines in the design optimization of ORC systems for waste heat recovery on board an existing Liquefied Natural Gas (LNG) carrier is performed. Heat associated with the jacket water, lubricating oil and charge air cooling of the engines is found to be available for the ORC, while the heat from the exhaust gasses is already used to generate low-pressure steam for ship internal use. Three layouts of the cooling systems collect the low-grade waste heat of the engine-generator sets and make it available for the ORC system and produce the net powers ranged between 430–580 kW, using two or three heat exchangers in each different system and the mass flow rate of the working fluid is ranged between 30 – 70 kg/s. Installing more heat exchangers also increasing the capital cost for the system. However, in this bachelor thesis, the capital cost, complexity of the system and the electricity generated from the system is giving a main consideration for installing the ORC system onboard.

CHAPTER III
RESEARCH METHOD

3.1 General

The scope of this chapter explains about systematical procedure of the research to obtain the final result. Completion of this research is obtained by following the fundamental framework written in this chapter. The method in this chapter is covers all activities and process analyzing that required to solve the problem in this research.

3.2 Research Method Flowchart

A complete systematical procedure to obtain the final result in this research is presented in a flowchart diagram in figure 3.1.

3.1.1. Problem Identification

The objective at this stage is to find out the problem that can be formulized and identified. The source to formulized and identify the problems is by curiosity from the writer and from various reference such as journal, paper and books.

3.1.2. Literature Study

In this research, literature study is an early stage for planning a process about the basic theories to be learned and a data for supporting the calculation and analysis to obtain the final result of this research. Some basic theories which may require as a fundamental to write and solve the problems in this research such as thermodynamic (heat transfer, properties of refrigerants, rankine cycle, energy balance), fluid machinery (pump, expander, heat exchanger), diesel engine and generator (sankey diagram, generated power to fuel consumption estimation), applied organic rankine cycle in ship (operating conditions and equipment which is used) cooling system in ship and the economic calculation (estimating the price of equipments, fuel and fluid, break-even point) so the data. The source for this stage is taken from books, papers, websites, journals, patent and company data

3.1.3. Collecting Data

The aim of this stage is to collect some data directly from the ship and taken from MV. Tanto Tenang ship which is owned by PT. Tanto Intim Line Shipping Company. The collected data are ship particular, engine log book, cooling system drawing and equipment of the ship.

3.1.4. Data Analyze

After the process of collecting the data is completed, the next step is to analyze the collected data. There are some main data that are important to be analyze to obtain temperature inlet and outlet for jacket water cooling system at typical engine load condition, available heat which can utilized in organic Rankine cycle system, cooling system drawing and equipment of the ship and the refrigerant properties. The process of analyzing the data are concerning about jacket water cooling temperature & pressure at inlet and outlet condition to obtain the available heat and mass flow rate at typical load of the main engine from typical MAN B&W Sankey diagram, properties of the refrigerants which is suitable for desired operating condition, the freshwater and seawater cooling system work from the drawing to determine in which part for installing the suitable ORC technology in the cooling system of the ship.

3.1.5. ORC System Design in Ship Cooling System

This stage aims to design in which suitable location at the ship cooling system that the process organic Rankine cycle system can be installed on the result from previous stage in modelling and simulation. Software that support for the design process at this stage is AutoCAD 2016.

3.1.6. Modelling

From the obtained data, the model from previous step are to be modelled by using ASPEN HYSYS 8.6 to model the equipment of organic Rankine cycle system before the simulation step is begin.

3.1.7. Simulation

From the obtained data, the simulation is simulated by ASPEN HYSYS 8.6 to obtained the optimized operating condition for generating electric power by the equipment and provide that the process is operable and also the most fitted refrigerant. The software also used to simulate many variables from the data and make easier while calculating many variables is required much time when compared to manual calculation to obtained the generated electrical power by the organic Rankine cycle system. Also the software can identify if there are unsuitable or non-optimized for the equipment and the process. The process is defined to be optimized if there are no temperature cross in the heat exchanger, no vapor phase when refrigerant entering the pump and no liquid phase when refrigerant entering the turbine or expander.

3.1.8. Manual Calculation

The process of manual calculation has two purposes. First purpose is act as a comparison to show that the result from simulation stage can be use if the result from the manual calculation also show the same result with some tolerable error. The second is act as a tools to make sizing and selection for each organic Rankine cycle equipment (heat exchanger, pump, expander) that will be used. The aims at this stages is to show that the result from simulation stage can be use if the result from the manual calculation also show the same result with some tolerable error.

3.1.9. Economic Aspect Calculation

The next step after the designing the ORC system in ship cooling system design is the economic aspect calculation. The stage aims to obtain the economic aspect investment with install installing the organic Rankine cycle at MV. Tanto Tenang ship. In this stage, the price of the equipment is required to calculate the break event point of the investment compared between the price of the organic Rankine cycle equipment installed in the ship and the

CHAPTER IV
DATA ANALYSIS

4.1. Ship Particular

The ship which is use for this bachelor thesis is MV. Tanto Tenang and owned by the shipping company named PT. Tanto Intim Line. The ship particular of MV. Tanto Tenang is described in the table below:

Table 4.1 Ship Particular of MV. Tanto Tenang

Parameters	Remarks
Name of Vessel	MV Tanto Tenang
Port of Registry	Jakarta
Nationality	Indonesia
Owners	PT. Tanto Intim Line
Managing Operator	PT. Tanto Intim Line
Official Number	4002
IMO Number	9192040
Built	Xiamen Shipyard-Xiamen China
Keel Laid	27th. Oct. 1997
Gross Tonnage	9030
Net Tonnage	4222
Length Overall	135,6 m
Length Between Perpendiculars	124,97 m
Breadth Moulded	22,49 m
Depth Moulded	11,21 m
Design Draft	7,6 m
Deadweight	10969,0 mt
Displacement	15703,3 mt
Cargo Hold Capacity	15554,3 cbm
Container Capacity	820 TEU's
Reefer Sockets	80 pcs

Parameters	Remarks
Block Coefficient	0,636
Ship Built	Xiamen Shipyard-Xiamen China
Height of highest antenna	31,80 m
Fresh Water Capacity	194,9 t
Ballast Water Capacity	3356,0 mt
Bunker Capacity	962,5 mt
Main Engine Type	MHHM MAN B7W-Type 5S-50MC-C
Auxiliary Engine Type	3xZMDW type 5L23/30H
Propeller Type	LIPS fixed 4 blades, dia 5500mm, pitch 4700mm
Bow Thruster	LIPS=type FT 06H/ Fixed pitched 4 blades 500kW (non-operational
Rudder Type	Semi-Spade Type Rudder Blade
Service Speed	17,0 knots.
Classification Society	B.V.

4.2. Data Analysis

The data analysis is critically required to obtain the & result of how much the electrical power will be generated at the turbine. The required data are listed below:

1. Engine load per RPM
2. Temperature inlet and outlet for jacket water cooling system at typical engine load condition
3. Available heat which can utilized in organic Rankine cycle system
4. Refrigerant selection
5. Cooling system drawing and equipment of the ship and the refrigerant properties

Every details step of the calculation is written in the next sub-chapter.

4.2.1. Estimating the Engine Load Based on RPM

To estimate the engine load based on RPM of main engine is using formula of:

$$\frac{P_1}{P_2} = \frac{(n_1)^3}{(n_2)^3} \tag{IV.1}$$

$$P_1 = \frac{(n_1)^3 P_2}{(n_2)^3} \tag{IV.2}$$

Where : n_1 = *Given RPM*
n_2 = *Max RPM (100% load)*
P_1 = *Load (%) at* n_1
P_1 = *Load (%) at* n_2 *(100% RPM)*

As per equation 4.1, the RPM of main engine can be calculated based on how much the desired load.

$$n_1 = \left(\frac{P_1 (n_2)^3}{P_2}\right)^{\frac{1}{3}} \tag{IV.3}$$

The following example of calculation process to find out the RPM at the 60% load by using equation 4.3:

$$\frac{60}{100} = \frac{(n_1)^3}{(127)^3}$$

$$n_1 = \left(\frac{60\ (127)^3}{100}\right)^{1/3}$$

$$n_1 = (1229029.8)^{1/3}$$

$$n_1 = 107.1 \text{ RPM}$$

For the next step is to determine the RPM at 65, 70, 80, 85, 90, 95 and 100.

Table IV.2 RPM vs Load of the Main Engine

No	Load (%)	RPM
1	**22,29**	77,0
2	**60**	107,1
3	**65**	110,0
4	**70**	112,8
5	**75**	115,4
6	**80**	117,9
7	**85**	120,3
8	**90**	122,6
9	**95**	124,8
10	**100**	127,0

4.2.2. Determine the Jacket Water Inlet & Outlet Temperature of Engine Load

Based on the data obtained from Engine Log Book the temperature at certain RPM are define at following table 4.3:

Table IV.3 Typical Engine Load & Temperature Correlation

No	RPM	Load (%)	Inlet	Outlet
1	77	22,3	69,1	73,8
2	90	36,0	68,0	76,0
3	95	42,0	69,0	78,0
4	97	45,0	73,3	81,6
5	100	48,8	76,0	83,1

To have a complete data for jacket water cooling inlet and outlet temperature until RPM 127 (100% load), the approximation is done by do a calculation which is describe below:

1. **Find out the kW at certain RPM or certain load:**
   In this example is to find kW at RPM 90 (36%), while for the different engine load the step of the calculation is similar.

$$\begin{aligned} kW &= Load\ x\ Maximum\ kW\ of\ Main\ engine \\ &= 36\%\ x\ 7900\ kW \\ &= 8244\ kW \end{aligned}$$

According to the data, the ship is use the MAN B&W S50MC-C series and have the power output maximal at 100% load is 7900 kW. The rest of calculation process to obtain kW of Main Engine at certain RPM is similar and the result is show in the following table.

Table IV.4 Main Engine Load {kW) for certain RPM

No.	RPM	Load	kW
1	77	22,3%	1761,7
2	90	36,0%	2844,0
3	95	42,0%	3318,0
4	97	45,0%	3555,0
5	100	48,8%	3856,8
6	100,5	49,6%	3914,5
7	107	60,0%	4740,0
8	115,5	75,0%	5925,0
9	120,5	85,4%	6748,2
10	122,5	90,0%	7110,0
11	127	100,0%	7900,0

2. **Find Jacket Water Inlet Temperature**
   Assuming for 100 RPM – 127 RPM of the main engine, the outlet temperature of the jacket water cooling is set to remaining constant at 83,1^{o}C. To find jacket water inlet temperature, the steps are described below:

3. **Find kW at typical main engine load**
   This calculation is already explained at the previous section in step 1.

4. Find how much available heat at typical engine load

To calculate the available heat at typical engine load especially in jacket water cooling, a Sankey diagram is used.

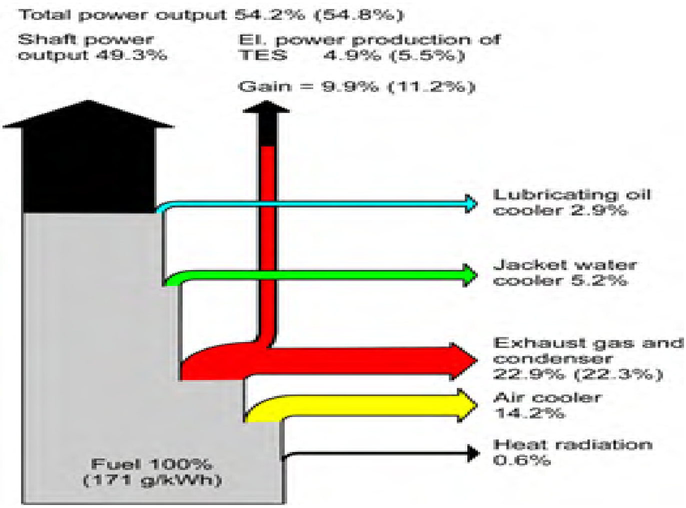


Figure IV.1 Sankey Diagram.
Source: www.sankey-diagrams.com

By looking through the diagram, an estimation could be provided. In this case, the available heat for jacket water cooling system is 10% from typical main engine load. The calculations are described below:

$$Available\ heat\ (kW) = \ 5{,}2\%\ x\ b \qquad \text{(IV.4)}$$

Where:
b = 100% Fuel Energy at typical engine load (kW)

In example, at 90 RPM the main engine load is 2844 kW (36% from 100% main engine load). The first step is to find the 100% fuel energy as the calculation described below:

$$\frac{49{,}3}{100} = \frac{2844}{b}$$

$$b = \frac{2844\,.\,100}{49{,}3} = 5768{,}76\;kW$$

Then the available heat is:

$$\begin{aligned} Available\ heat\ (kW) &= 5{,}2\%\ x\ b \\ &= 5{,}2\%\ x\ 5768{,}76 \\ &= 299{,}98\ kW \end{aligned}$$

For the rest of the typical RPM and main engine load, the formula to calculate the available heat is similar. The complete available heat is written in the following table:

Table IV.5 Available Heat per Main Engine RPM

RPM	Load	kW	100% of Fuel Energy	Qht [kW]
77	22,30%	17612	3573,43	185,82
90	36,00%	2844	5768,76	299,98
95	42,00%	3318	6730,22	349,97
97	45,00%	3555	7210,95	374,97
100	48,80%	3855	7819,88	406,63
100,5	49,60%	3918	7948,07	413,30
107	60,00%	4740	9614,60	499,96
116,7	77,50%	6123	12418,86	645,78
117,9	80,00%	6320	12819,47	666,61
120,5	85,40%	6747	13684,79	711,61
122,5	90,00%	7110	14421,91	749,94
124,8	95,00%	7505	15223,12	791,60
127	100,00%	7900	16024,34	833,27

5. **Find temperature different at each typical main load engine**

The actual data from engine log book obtained only from RPM 77 – RPM 100, then the rest of temperature inlet should be calculated. Heat flow formula is used to calculate temperature different at typical load of main engine, the calculation step is figured below:

$$Qht = \dot{m}\ x\ cp\ x\ \Delta t \qquad \text{(IV.5)}$$

Where:

Qht = Available heat in jacket water cooler at typical main engine load [kW]

ṁ = Mass flow rate of hot fluid (jacket water cooling water) [k/s]

Δt = Temperature inlet and outlet different

Cp = Specific heat [kj/kg ^{o}C]

To calculate the mass flow rate of the jacket water cooling through the main engine is by look at the pump capacity and convert it to calculate the mass flow rate of the jacket water cooling.

Capacity of the pump = 70 m^3/h

Mass flow rate (ṁ) = Q x ρ (IV.6)

= 70 x 1000

= 70000 kg/h

= 19.44 kg/s

In this example is to find out the inlet temperature at RPM 100,5 (49,60% of main engine load), the calculation process is figured below:

$$\Delta \mathrm{t} = \frac{Qht}{\dot{m}\ x\ Cp} \qquad \text{(IV.7)}$$

$$= \frac{413{,}30}{19.44\ x\ 4.19}$$

$$= 5{,}07\ ℃$$

Since the temperature outlet is constant, then the inlet temperature is:

$$\Delta \mathrm{t} = 5{,}07\ ℃$$

$$T_{outlet} - T_{inlet} = 5{,}07\ ℃$$

$$T_{inlet} = T_{outlet} - 5{,}07\ ℃$$

$$T_{inlet} = 83{,}1 - 5{,}07\ ℃$$

$$T_{inlet} = 78.03\ ℃$$

The formula is valid to calculate the jacket water cooling temperature inlet for typical engine load condition. Following table are show the complete jacket water inlet temperature at different main engine load:

Table IV.6 Inlet and Outlet of Jacket Water Cooling

RPM	Load	kW	Qht (10%) [kW]	ΔT (°C)	Tin (°C)	Tout (°C)
77	22,30%	1762	176,17	4,7	69,1	73,8
90	36,00%	2844	284,4	8	68	76
95	42,00%	3318	331,8	9	69	78
97	45,00%	3555	355,5	8,3	73,3	81,6
100	48,80%	3857	385,68	7,1	76	83,1
100,5	49,60%	3915	391,45	4,81	78,3	83,1
107	60,00%	4740	474	5,82	77,3	83,1
115,5	75,00%	5925	592,5	7,27	75,8	83,1
120,5	85,40%	6748	674,82	8,28	74,8	83,1
122,5	90,00%	7110	711	8,73	74,4	83,1
127	100,00%	7900	790	9,7	73,4	83,1

4.2.3. Refrigerant Selection

The refrigerant is used as a working fluid to generate electricity in ship through the expander. In this research the refrigerant which being used is R-152a & R-134a. Based on the simulation, different type of refrigerant gives an effect to the different electric power that will be produced while the best result is achieved using R-134a as a working fluid in the organic Rankine cycle.

4.2.4. Cooling System Drawing

The organic Rankine cycle system proposed to be installed at cooling system in the jacket water cooling system. For the full drawing system is already attached in the attachment page.

4.5. Simulation

After the complete data and the manual calculation has been collected, then the simulation process can be start with inputting the data to the software. By varying the mass flow rate of jacket water cooling water at typical engine load, the generated electrical output power will have differences. The maximum mass flow rate of jacket water cooling is 19,44 kg/s.

The mass flow rate of each different main engine load will also be different in order to keep the organic Rankine cycle component well-performed.

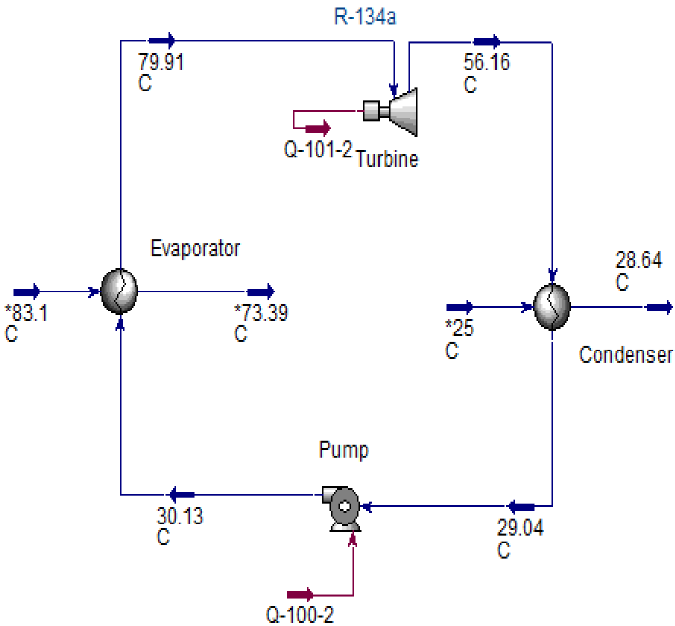


Figure IV.2 Simulation Result from Hysys Software with R-134a as a Working Fluid

Table IV.7 Simulation Result

Load	ṁht from simulation [kg/s]		ṁht average [kg/s]	ṁref [kg/h]	Power Produced from Simulation [kW]
	Min	Max			
75,0%	19,44	19,44	19,44	10200	57,54
80,0%	18,10	18,9	18,5	10200	61,11
85,4%	17,00	17,8	17,4	10200	61,32
90,0%	16,15	17,1	16,625	10200	61,49
95,0%	15,45	16,1	15,775	10200	61,68
100,0%	14,60	15,4	15	10200	61,72

4.3. Organic Rankine Cycle Design in Ship

As mentioned before, the organic Rankine cycle system proposed to be installed at cooling system in jacket water cooling system. The equipment of organic Rankine cycle is to be installed after the jacket water cooling system leaving from the main engine. Detail stage of the cold stream and hot stream process are listed below:

a. Hot Stream (Jacket water):

1. The jacket water will be entering the evaporator (heat exchanger) and give heat for refrigerant to be utilized as a heat source.
2. As the refrigerant receive heat from jacket water heat per time, the phase of the refrigerant will change from liquid phase to vapor phase.
3. After jacket water flow out from the heat exchanger, the jacket water is flow in normal conditions again.

b. Cold Stream (Refrigerant R-134a):

1. Refrigerant is pumped in liquid phase through to the evaporator.
2. Evaporator change the phase of the refrigerant from liquid phase into vapor phase by using heat from jacket water.

3. In vapor phase, the refrigerant steam is entering the turbine. With typical mass flow rate and pressure, the refrigerant in vapor phase is able to give force to rotate the turbine. From rotating turbine, the turbine will generate electric power in various power according to the temperature, mass flow rate and pressure.

   To regulate the mass flow rate, a three-way valve is act as a control valve. The various mass flow rate will be determined by the various engine load. If the engine load is become higher or increase, the mass flow rate should be decreased in order to maintain the process is operable and make sure if the organic Rankine cycle equipment perform desired performance (does not become trouble) while operation is running.

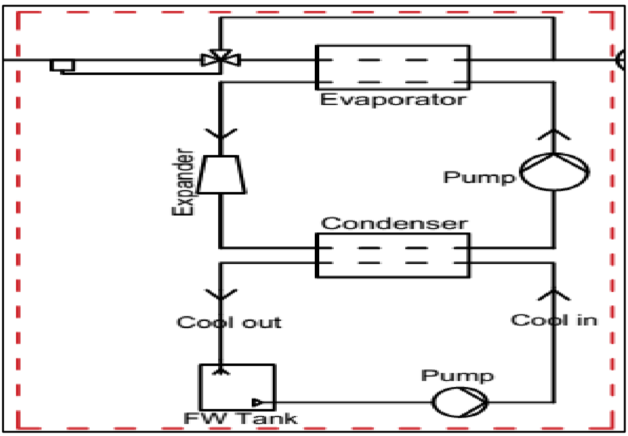


Figure IV.3 Organic Rankine Cycle Proposed System

In example, if the mass flow rate flows are increase, the heat load that could be utilized also will increased. In order to keep the equipment works properly, if the engine load is increased then the mass flow rate of the jacket water cooling should be reduced.

For the jacket water cooling system, the ship fresh water generator equipment is not used any longer because the ship sail in the local Indonesia region only and the jacket water could be utilized more effectively. The three-way valve has a function for regulating the flow of the jacket water cooling. If the organic Rankine cycle equipment is not readily used, then the valve will have passed jacket water cooling flow and the jacket water will flow normally through the circuit.

4.4. Manual Calculation

The first step before start the manual calculation is to find out the operating conditions for the organic Rankine cycle. Operating conditions for this research is acquired from patented technology that already built. The operating conditions are described below on the table:

Table IV.8 Proposed Operating Condition of ORC System

Parameters	Value
Pressure inlet turbine	21,6 bar
Pressure inlet evaporator	22 bar
Pressure inlet condenser	7,5 bar
Mass flow rate of working fluid	2.811 kg/s

The adapted operating conditions obtained from Site of Nant-de Châtillon (Genève, Switzerland) organic Rankine Cycle experiment as the operating conditions is listed below on the table:

Table IV.9 Operating Condition References for ORC System

Parameters	Value
Pressure inlet turbine	18-22 bar
Hot inlet evaporator	95 ^oC
Pressure inlet condenser	7-8 bar
Electric Power Measured	5-6 kW

For the first step to begin the manual calculation, the refrigerant should be selected first. The refrigerant which is used in this calculation is R-134a. After the refrigerant has been selected, the list of steps has written below:

4.4.1. Determine Mass Flow Rate of Working Fluid

The mass flow rate is determined by trial and error in HYSYS Software. For the optimum mass flow rate of the refrigerant, the value that has been selected is 2,7861 kg/s. Actually the value of the mass flow rate itself could be lower or higher than 2,7861 kg/s. For the lower mass flow rate, the power that will be generate also will be lower, but the ORC system could be used in lower engine load. In the other hand, the higher mass flow rate of the refrigerant in ORC system will generate more electrical power, but the ORC system could be used in higher engine load.

As an example, the value of mass flow rate which is used in this system are valued 2,833 kg/s and with this mass flow rate the ORC system still could be activated at 77.5% - 100% of the main engine load. Conversely, if the mass flow rate increase to 2,833 kg/s could be activated at 77,5%-100% of the main engine load with more electricity that could be produced by the system.

4.4.2. Determine Refrigerant Temperature After Leaving Evaporator.

The temperature outlet of the evaporator can be calculated using the heat balance formula. In this case, the heat source is taken from main engine jacket water cooling system at 85,4% engine load (637,238 kW).

From Hysys simulation, the available heat (637,238 kW) cannot be utilized up to 100% because it will give an effect to the refrigerant temperature after leaving the evaporator higher than 83,1 ^{o}C (inlet temperature of hot fluid). Once this effect is ignored, hence the heat exchanger will have temperature cross condition.

Temperature cross means where the outlet temperature of the cold stream is higher than the inlet temperature of hot stream. Table below show the operating conditions as a reference for the calculation.

Table IV.10 Operating Conditions in 85,4% Engine Load

Hot Stream (water)		Cold Stream (R-134a)	
Inlet Temperature	83.1 °C	Inlet Temperature	29.28 °C
Outlet Temperature	74,36 °C	Outlet Temperature	x
Mass Flow Rate	17,4 kg/s	Mass Flow Rate	2,833 kg/s

Using heat balance formula:

$$\dot{Q}_{hot} \quad = \dot{Q}_{cold} \tag{IV.8}$$
$$\dot{Q}_{hot} \quad = \dot{m}.c_p.(T_{in} - T_{out}) \tag{IV.9}$$

Since in the evaporation process is includes the phase change of the refrigerant from liquid to vapor phase, then the equation of the $\dot{Q}_{cold}$ becomes:

$$\begin{aligned}
\dot{Q}_{hot} &= \dot{Q}_{cold} \\
\dot{Q}_{hot} &= \dot{m}.c_p.(\mathrm{T_{in}} - \mathrm{T_{out}}) \\
\dot{Q}_{hot} &= 17{,}4 \,.\, 4{,}2.\,(83{,}1 - 74{,}36) \\
\dot{Q}_{hot} &= 17{,}4 \,.\, 4{,}2.\,(83{,}1 - 74{,}36) \\
\dot{Q}_{hot} &= 637{,}238\ kW
\end{aligned}$$

$$\begin{aligned}
\dot{Q}_{hot} &= \dot{Q}_{cold} \\
637{,}238 &= Q1 + Q2 + Q3 \\
&= 2{,}83\ x\ 1{,}532(71{,}66 - 29.28) + 2{,}83\ x152{,}5 \\
&\qquad + 2{,}83\ x\ 1{,}532(T_{out} - 71{,}66) \\
624{,}775 &= 180{,}87 + 424{,}8819 + 4{,}268\ (T_{out} - 71{,}66) \\
T_{out} &= 76{,}5498\ ^{o}C
\end{aligned}$$

Since the heat load of the evaporator has already known, then the evaporator can be selected according to the operating conditions which has been determined based on the previous calculation. The illustration of the condenser is figured in the figure 4.4.

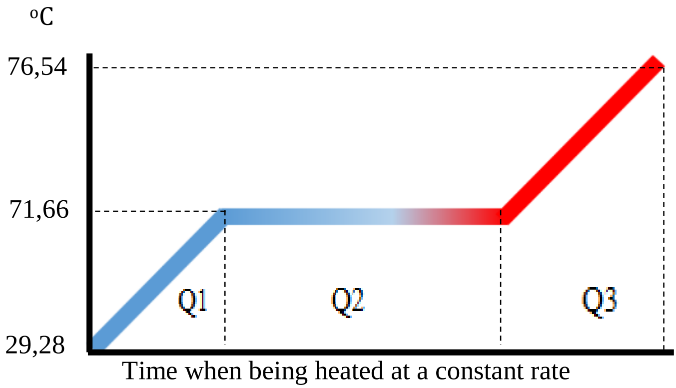


Figure IV.4 Illustration of Phase Change in Evaporator at 85 % Main Engine Load

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The evaporator should be selected to withstand at the maximum operating conditions. Operating conditions of the Jacket Water Cooling is written below:

Max. Heat Load	= 790 kW
Max. Flow Rate	= 19,44 kg/s
	= 69,984 m^3/h
Max. Temperature	= 83,1 °C
Max Pressure	= 22 bar

Based the operating conditions of the jacket water cooling conditions, the specification of the selected evaporator is:

Manufacturers	= Funke ViFlow
Series	= TPL-02K
Application	= Evaporating
Type	= Brazed Plate Heat Exchanger
Max Pressure	= 30 bar
Temperature Range	= -160 – 200 °C
Heat Capacity	= 2000 kW
Max. Flow Rate	= 70 m^3/h

4.4.3. Determine the Temperature Outlet & Electrical Power Generated of the Turbine

The temperature outlet of the turbine is required to calculate how much electricity that will be generated by the turbine by assuming that the outlet pressure from the turbine is 7,5 bar. Pressure drop is occurred when the fluid entering the heat exchanger (evaporator). The pressure drop in the evaporator is occurred into two fluid flows. The first is Jacket water (Hot stream) and the second is refrigerant (Cold stream). In this chapter, the calculations are taken to obtained the maximum pressure drop.

1. **Pressure Drop at Jacket Water (hot stream)**

   Maximum pressure drop is set at the mass flow rate of the jacket water reach 19,44 kg/s. The provided data for pressure drop calculations refers to the selected evaporators which has been selected. Pressure drop at the jacket water side can be calculated by the formula:

$$\Delta p = \frac{1{,}5{G_p}^2 n_p}{2 g_c \rho_i} + \frac{4fLG^2}{2g_c D_e}\left(\frac{1}{\rho}\right)_m \pm \frac{\rho_m g L}{g_c}$$

np = 1
Np = 120 (number of plates)
w = 0,225 m
L = 0.486 (plate height)
Dp = 0.075 m (port diameter)
b = 0.004 m (plate thickness)
De = 0.008 m (equivalent diameter)
gc = 1 (conversion factor)
g = 9.87 m/s^2 (gravity acceleration)
ṁ = 20 kg/s
μ = 0.0003411 kg/ms (dynamic viscosity)
ρ = 970 kg/m^3 (fluid density)

The mass velocity through the port (Gp) can be calculated by:

$$Gp = \frac{\dot{m}}{(\pi/4){D_p}^2} \tag{IV.10}$$

$$Gp = \frac{20}{(0.7857)\ x\ 0.075^2} = 0{,}1433\ kg/m^2s$$

The mass velocity (G) through the core can be calculated by:

$$G\ = \frac{\dot{m}}{A_o} \tag{IV.11}$$

$$G\ = \frac{20}{0.108} = 185{,}185\ kg/m^2s$$

Where:

$$A_o\ =\ Np\ x\ w\ x\ b \tag{IV.12}$$

$$=\ 1\ x\ 0.225\ x\ 0.004\ = 0.108$$

Then calculate the Reynold Number (Re) and friction factor:

$$R_e = \frac{G\ D_e}{\mu} \tag{IV.13}$$

$$R_e = \frac{185{,}185\ x\ 0{,}008}{0.0003411} = 4343{,}2647$$

$$f\ \ = 0{,}8\ Re^{-0.25}$$

$$f\ \ = 0{,}8\ (4343{,}2647)^{-0.25} = 0{,}09854$$

$$\Delta p\ = \frac{1{,}5(0{,}1433\ )^2 120}{2x1x970} + \frac{4x0{,}09854x0.486x185{,}185^2}{2x1x0{,}008} x \left(\frac{1}{970}\right) + \frac{970x9{,}87x0.486}{1}$$

Δp = 2344,7857 kg/m^{2}s

= 23,44 kPa

= 0,23 bar ≈ 0,2 bar

2. Pressure Drop at Refrigerant (cold stream)

Maximum pressure drop is set at the mass flow rate of the refrigerant at 2,83 kg/s. The provided data for pressure drop calculations refers to the selected evaporators which has been selected in the previous explanations. The calculations process is similar but the only thing that makes a differences is mass flow rate. Pressure drop at the refrigerant side can be calculated by the formula:

$$\Delta p = \frac{1{,}5{G_p}^2 n_p}{2g_c\rho_i} + \frac{4fLG^2}{2g_cD_e}\left(\frac{1}{\rho}\right)_m \pm \frac{\rho_m gL}{g_c}$$

np = 1
Np = 120 (number of plates)
w = 0,225 m
L = 0.486 (plate height)
Dp = 0.075 m (port diameter)
b = 0.004 m (plate thickness)
De = 0.008 m (equivalent diameter)
gc = 1 (conversion factor)
g = 9.87 m/s^2 (gravity acceleration)
ṁ = 2,83 kg/s
μ = 0.0002301 kg/ms
ρ = 1227 kg/m^3 (fluid density)

The mass velocity through the port (Gp) can be calculated by:

$$Gp = \frac{\dot{m}}{(\pi/4){D_p}^2}$$

$$Gp = \frac{2{,}83}{(0.7857)\ x\ 0.075^2} = 0.0202\ kg/m^2s$$

The mass velocity (G) through the core can be calculated by:

$$G = \frac{\dot{m}}{A_o}$$

$$G = \frac{2.83}{0.108} = 26{,}203\ kg/m^2s$$

Where A_o (Area),

$$A_o = Np\ x\ w\ x\ b$$
$$= 1\ x\ 0.225\ x\ 0.004 = 0.108\ \text{m}^2$$

Then calculate the Reynold Number (Re) and friction factor:

$$R_e = \frac{G\ D_e}{\mu}$$

$$R_e = \frac{26{,}018\ x\ 0.008}{0.0002301} = 911{,}037$$

$$f = 0{,}8\ Re^{-0.25}$$
$$f = 0{,}8\ (904{,}5986)^{-0.25} = 0.1456$$

$$\Delta p = \frac{1{,}5(0{,}0201)^2 120}{2x1x1227} + \frac{4x0.1458x0.486x26{,}018^2}{2x1x0{,}008}x \left(\frac{1}{1227}\right) + \frac{1227x9{,}87x0.486}{1}$$

Δp = 2943,3003 kg/m^{2}s
= 29,43 kPa
= 0,29 bar ≈ 0,3 bar

Once the calculations of the pressure drop at evaporator are already finished, then the summary is:

Pressure drop at jacket water = 0,2 bar
Pressure drop at refrigerant flow = 0,3 bar

3. **Electrical Power Generated by the Turbine**

Electrical power will be generated by the rotating. Before the calculations begin, the operating conditions in the turbine should be determined first. As previously calculated, the pressure drop in evaporator from the cold stream (refrigerant flow) is 0,23 bar which is affect to the pressure of the refrigerant after pumped at 22 bar becomes 21,6 bar. Below are figured the operating conditions at 85,4 % of engine load.

Table IV.11 Operating Conditions in 85,4% Engine Load

Parameters	Value
Pressure inlet turbine	21,6 bar
Pressure outlet turbine	7,5 bar
Temperature Inlet turbine	76,5498 °C
Mass flow rate	2,78 kg/s
Specific Heat	0,933
Cp/Cv (γ)	1,464

After the operating conditions already completed, the outlet turbine outlet temperature is:

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 77{,}12 \left(\frac{7{,}5}{21{,}6}\right)^{\frac{1{,}464-1}{1{,}464}}$$

$$T_2 = 55{,}157\ ^{\mathrm{o}}\mathrm{C}$$

Then the electrical power produced by the generator are:

$$kW_{generated} = \dot{m}.c_p.(T_{in} - T_{out}) \qquad \text{(IV.14)}$$

$= 2{,}78 \times 0{,}933\ (77{,}126\text{-}55{,}157)$

$= 55{,}2195$ kW

In order to obtain the electrical power generated by the turbine at the different main engine load, the same calculations and process is similar. After the electrical power already obtained, the next step is select the turbine.

Table IV.12 Operating Conditions of Turbine

Parameters	Value
Inlet Pressure	22 bar
Outlet Pressure	7,5 bar
Max Power Produced	56 kW
Max Inlet Temperature	83 °C
Min Outlet Temperature	52 °C

The selection is described in the table below for the expander:

Table IV.13 Expander Selection

Parameters	Value
Type	Micro Steam Turbine
Manufacturer	G-Steam
Series	TR-Hi 150
Max Pressure	3 – 40 bar (inlet) 0,5 – 14 bar (outlet)
Max Temperature	420 ^{o}C
Max Power Output	150 kW
RPM	30000 RPM
Base Plate Dimension	400 x 780 (mm)

The summary of power produced by the manual calculation are showed in the table below:

Table IV.14 Power Produced per Main Engine Load

Load [%]	ṁht from simulation [kg/s]		ṁht average [kg/s]	ṁref [kg/h]	Power Produced [kW]	Power Produced from Simulation [kW]	Error
	Min	Max					
75	19,44	19,44	19,44	10200	55,54524	57,54	3,4667
80	18,10	18.9	18,50	10200	57,19702	61,11	6,4032
85,4	17,00	17,80	17,40	10200	57,6404	61,32	6,0007
90	16,15	17,10	16,63	10200	58,43601	61,49	4,9666
95	15,45	16,10	15,78	10200	58,61269	61,68	4,9729
100	14,60	15,40	15	10200	58,71498	61,72	4,8688

4.4.4. Determine Condenser Heat Load After Turbine

Before the calculations to determine the heat load of the condenser begins, the temperature outlet of a condenser should be determined first. The temperature is to be set at the point which refrigerant is change to be a liquid phase, because in the next process this refrigerant will be pumped using pump. The temperature is to be set at 28 ^{o}C in 7,48 bar. Below is figured the heat load of a condenser at 85,4 % of main engine load.

Table IV.15 Operating Conditions of system at 85,4% Engine Load

Parameters	**Value**
Pressure inlet condenser	7,5 bar
Temperature inlet condenser	55,157 °C
Temperature outlet condenser	28 °C
Mass flow rate	2,694 kg/s
Specific heat of refrigerant	0,8995 kJ/kg °C
Latent heat of refrigerant	184,5 kJ

Since in the condensation process is includes the phase change of the refrigerant from vapor to liquid phase, then the equation of the $\dot{Q}_{hot}$ becomes:

$$\dot{Q}_{hot} = Q1 + Q2 + Q3 \qquad \text{(IV.15)}$$

$$\begin{aligned} Q1 &= \dot{m}.c_p.(\mathrm{T_{in}} - \mathrm{T_{sat}}) \\ &= 2{,}694 \,.\, 0{,}8995\,(55{,}157 - 29{,}11) \\ &= 63{,}13\ kW \end{aligned}$$

$$\begin{aligned} Q2 &= \dot{m}\,.\,L \\ &= 2{,}694\,.\,184{,}5 \\ &= 497{,}13\ \mathrm{kW} \end{aligned}$$

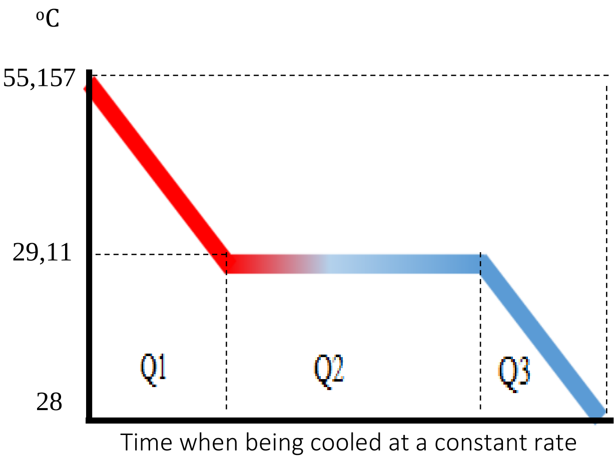

Figure IV.5 Illustration of Phase Change in Condenser at 85 % Main Engine Load

$$\begin{aligned} Q3 &= \dot{m}.c_p.(\mathrm{T_{sat}} - \mathrm{T_{out}}) \\ &= 2{,}694\,.0{,}8995.(29{,}11 - 28) \\ &= 2{,}69\ kW \end{aligned}$$

$$\dot{Q}_{hot} = 63{,}13 + 497{,}13 + 2{,}69$$
$$= 562{,}95\ \ kW$$

So the heat load of the condenser at 85,4 % of the main engine load is 562,95 kW. Maximum heat load at the condenser is reach at 100% of the main engine load (563,71 kW). The value of the heat load at the condenser is required to select a suitable condenser to be installed. The value of the maximum operating conditions is written below:

Table IV.16 Condenser Maximum Operating Conditions

Parameters	Value
Max. Heat Load	523 kW
Max. Flow Rate (refrigerant)	2,78 kg/s
Max. Flow Rate (water)	36,11 kg/s (130 m^3/s)
Max. Temperature	60 ^oC
Max Pressure	7,5 bar

As the maximum operations has already known, then selected which is suitable to fulfill the operating conditions is written below:

Table IV.17 Condenser Selection Specification

Parameters	Value
Manufacturers	Alva Laval
Series	CB400
Application	Condensing
Type	Brazed Plate Heat Exchanger
Max Pressure	35 bar
Temperature Range	-196 – 220 ^oC
Heat Capacity	1,5 – 600 kW
Max. Flow Rate	200 m^3/h

1. **Pressure Drop at Refrigerant Side in Condenser (cold stream)**

   Maximum pressure drop is set at the mass flow rate of the refrigerant reach 2,78 kg/s. The provided data for pressure drop calculations refers to the selected condensers which has been selected. Pressure drop at the refrigerant side can be calculated by the formula:

$$\Delta p = \frac{1{,}5{G_p}^2 n_p}{2 g_c \rho_i} + \frac{4 f L G^2}{2 g_c D_e}\left(\frac{1}{\rho}\right)_m \pm \frac{\rho_m g L}{g_c}$$

np = 1
Np = 240 (number of plates)
w = 0,300 m (width)
L = 0.900 (plate height)
Dp = 0.07 m (port diameter)
b = 0.004 m (plate thickness)
De = 0.008 m (equivalent diameter)
gc = 1 (conversion factor)
g = 9.87 m/s^2 (gravity acceleration)
ṁ = 2,78 kg/s
μ = 0.0001356 kg/ms (dynamic viscosity)
ρ = 34,4 kg/m^3 (fluid density)

The mass velocity through the port (Gp) can be calculated by:

$$Gp = \frac{\dot{m}}{(\pi/4){D_p}^2}$$

$$Gp = \frac{2{,}78}{(0.7857)\ x\ 0.07^2} = 0{,}017\ kg/m^2s$$

The mass velocity (G) through the core can be calculated by:

$$G = \frac{\dot{m}}{A_o}$$

$$G = \frac{2,78}{0,288} = 9,6527\ kg/m^2 s$$

Where

$$A_o = Np\ x\ w\ x\ b$$
$$= 240\ x\ 0.300\ x\ 0.004 = 0.288\ \text{m}^2$$

Then calculate the Reynold Number (Re) and friction factor:

$$R_e = \frac{G\ D_e}{\mu}$$

$$R_e = \frac{9,6527\ x\ 0.008}{0.0001356} = 569,4854$$

$$f = 0,8\ Re^{-0.25}$$
$$f = 0,8\ (569,4854)^{-0.25} = 0,1637$$

$$\Delta p = \frac{1,5(0.017)^2 240}{2x1x34.4} + \frac{4x0,09854x0.900x9,6527^2}{2x1x0,008} x \left(\frac{1}{34,4}\right) + \frac{34,4x9,87x0.900}{1}$$

Δp = 153,593 kg/m²s

= 1, 54 kPa

= 0,02 bar

2. **Pressure Drop at Water Cooling Side in Condenser (Hot Stream)**

   Maximum pressure drop is set at the water cooling to cooled down the refrigerant reach at 36,11 kg/s. The provided data for pressure drop calculations refers to the selected condensers which has been selected. Pressure drop at the water cooling side can be calculated by the formula:

$$\Delta p = \frac{1,5{G_p}^2 n_p}{2g_c\rho_i} + \frac{4fLG^2}{2g_cD_e}\left(\frac{1}{\rho}\right)_m \pm \frac{\rho_m gL}{g_c}$$

np = 1

Np = 240 (number of plates)

w = 0,300 m (width)

L = 0.900 (plate height)
Dp = 0.07 m (port diameter)
b = 0.004 m (plate thickness)
De = 0.008 m (equivalent diameter)
gc = 1 (conversion factor)
g = 9.87 m/s^2 (gravity acceleration)
ṁ = 36,11 kg/s
μ = 0.0003411 kg/ms (dynamic viscosity)
ρ = 1000 kg/m^3 (fluid density)

The mass velocity through the port (Gp) can be calculated by:

$$Gp = \frac{\dot{m}}{(\pi/4){D_p}^2}$$

$$Gp = \frac{36{,}11}{(0.7857)\ x\ 0.07^2} = 0{,}2600\ kg/m^2s$$

The mass velocity (G) through the core can be calculated by:

$$G = \frac{\dot{m}}{A_o}$$

$$G = \frac{36{,}11}{0{,}288} = 144{,}6527\ kg/m^2s$$

Where:

$$A_o = Np\ x\ w\ x\ b$$
$$= 240\ x\ 0.300\ x\ 0.004 = 0.288\ \mathrm{m}^2$$

Then calculate the Reynold Number (Re) and friction factor:

$$R_e = \frac{G\ D_e}{\mu}$$

$$R_e = \frac{144{,}6527\ x\ 0.008}{0.0003411} = 3392{,}6186$$

$$f \;= 0{,}8\, Re^{-0.25}$$

$$f \;= 0{,}8\,(3392{,}6186)^{-0.25} = 0{,}1048$$

$$\Delta p = \frac{1{,}5(0{,}2600)^2 240}{2x1x1000} + \frac{4x0{,}1048x0.900x144{,}6527^2}{2x1x0{,}008}x \left(\frac{1}{1000}\right) + \frac{1000x9{,}87x0.900}{1}$$

Δp = 4496,1694 kg/m^{2}s
= 44,96 kPa
= 0,45 bar

Once the calculations of the pressure drop at evaporator are already finished, then the summary is:

Pressure drop at refrigerant flow = 0,02 bar
Pressure drop at water cooling = 0,45 bar

4.4.5. Determine the Pump Selection for Fluid

The pump is selected to fit the operating conditions of the system. Actually, two pumps should be selected in order to pump the refrigerant fluid and to pump the water as a refrigerant cooler to become liquid phase again. The operating conditions of the pump that will be installed in organic Rankine cycle system is written below:

Table IV.18 Operating Conditions of Refrigerant in Pump

Parameters	Remarks
Refrigerant Type	R-134a
Fluid Density	1197 kg/m^3
Mass Flow Rate	2,833 kg/s (8,521 m^3/h) (37,5184 Gpm)
Temperature inlet	28 ^{o}C
Pressure Inlet	7,48 Bar
Pressure Outlet	22 Bar
State of Aggregation	Liquid

1. Pump Selection for Refrigerant Pump

In this part, the hear required by the pump should be calculated first. The head loss, head static and head velocity is assumed to be zero. So the head pressure is still remaining to be calculated.

$$Hreq = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 {V_2}^2 - \alpha_1 {V_1}^2}{2g} + (Z_2 - Z_1) + Hloss \qquad \text{(IV.16)}$$

$$Hreq = \frac{P_2 - P_1}{\rho g} + 0 + 0 + 0$$

$$Hreq = \frac{P_2 - P_1}{\rho g}$$

$$Hreq = \frac{2200000 - 748000}{1197\ x\ 9{,}78}$$

$$= 124{,}032m$$

$$= 406{,}929\ Ft$$

After the head pressure calculation has already completed, the selection of the pump can be done. The selected pump should be fulfilling the required operating conditions. The selected pump is written below.

Table IV.19 Selected Pump Specification

Parameters	Remarks
Type	Seal-Less Mag Drive Pump
Series	WMTA 2020
Manufacturers	Warrender, Ltd
Max Pressure	500 bar (725 Psi)
Max Capacities	24-45 Gpm
Max Temperature	316 ºC (600 F)
Max Head	213m (700 Ft)
RPM	3500 RPM
Power	12,1 kW

2. Pump Selection for Water Cooler Pump

Water cooler pump is used to pump the freshwater as cooler to cooling down the refrigerant in vapor phase to becomes liquid phase. The operating conditions of the pump is written below. Mass flow rate of the water is set up at 100 m^3/h to make the refrigerant vapor is turned back again to the liquid phase.

Operating Conditions of Water Cooler

Inlet Temperature	: 25 °C
Mass Flow Rate	: 100000 kg/h
	: 27,778 kg/s
ρ	: 1000 kg/m^3
Flow Rate	: 100 m^3/h

Temperature Outlet of the Water Cooler

After the water cooling down the refrigerant, the temperature of the water cooler is increase. The temperature of the increased water cooler can be calculated by using the heat balance formula. Assuming the temperature inlet to the condenser of the freshwater is always set to 25°C after the tank entering the condenser. Main engine load will increase the temperature of the water cooler gradually. The calculations to determine the outlet temperature of water cooling at 85% of main engine load is figured below.

$$Qtot = \dot{m}.c_p.(\mathrm{T_{in}} - \mathrm{T_{sat}})$$
$$592{,}90 = 27{,}778\,.\,4{,}182\ (\mathrm{dT})$$
$$dT = \frac{562{,}7}{116{,}167} = 4{,}843\ ℃$$
$$dT = Tin - Tout$$
$$4{,}843 = 25 - Tout$$
$$Tout = 25 + 4{,}843 = 30.07\ ℃$$

To obtain the water cooler temperature outlet of another main engine load, the calculations process is similar. After the operating conditions data for pump selection is completed, then the selected pump is written below.

Pump Selection

Type	: Centrifugal Pump
Series	: B100 Electric Drive
Manufacturers	: BBA Pumps
Max Head	: 19 mwc
Max Flow	: 140 m^3/h
RPM	: 1450 RPM
Power	: 7,5 kW

4.6. Determining the Cost Saving in Auxiliary Engine Fuel Based on Generated Power

The specification of the auxiliary engine which is used in MV. Tanto Tenang ship is:

Auxiliary Engine	: MAN B&W
Series	: 5-8L23/30H
RPM	: 720 RPM
kW (1 Cyl)	: 130 kW
kW (5 Cyl)	: 650 kW

Specific fuel oil consumption data from specific auxiliary engine can be obtained from the project guide of its engine. The SFOC data for typical main engine load is figured at table 4.19:

Table IV.20 SFOC of MAN B&W 5-L23/30H Engine

% Load	kW	SFOC
25 %	162,5	217
50 %	325	196
75 %	487,5	192
85 %	552,5	193
100 %	650	194

From the table above, the graphic of the SFOC at typical main engine load for MAN B&W 5-L23/30H Engine is figured below.

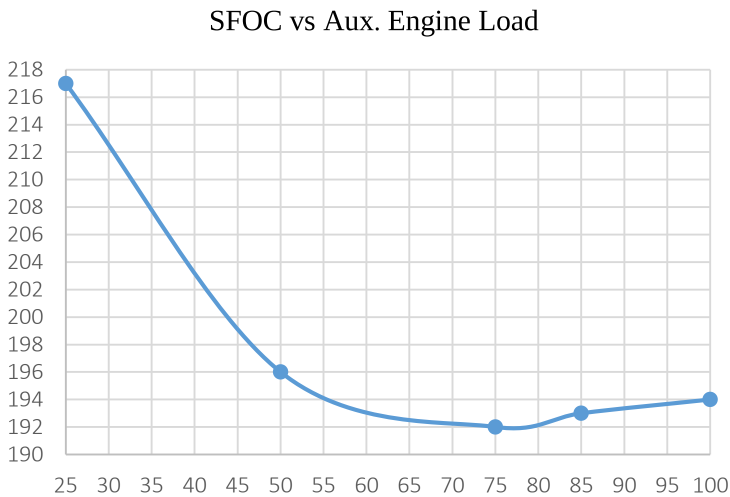

Figure IV.6 SFOC vs Aux. Engine Load Graphic

Assume if the generator load is operated at 85% load (552,5 kW) and the SFOC is 193 g/kW.h. If the generator is operated in hours, so the fuel consumption will be:

$$Gr \;=\; SFOC\ x\ kW\ x\ Hr\ (duration) \qquad \text{(IV.17)}$$

= 193 x 552,5 x 1

= 106633 gr

= 0.1066 Ton

Electrical power generated by the ORC system is various depending on Main Engine Load. In this example, the 85% of main engine load will produced 57,26 kW by the ORC system. So it will make the generator run lower than 85% of auxiliary engine load. The example of calculations is figured below:

kW after saving with ORC = Aux. Engine load – Electrical generated by ORC at 85% of engine load (IV.18)

= 552,5 – 57,64

= 494,4 kW

The 494,4 kW is equal to:

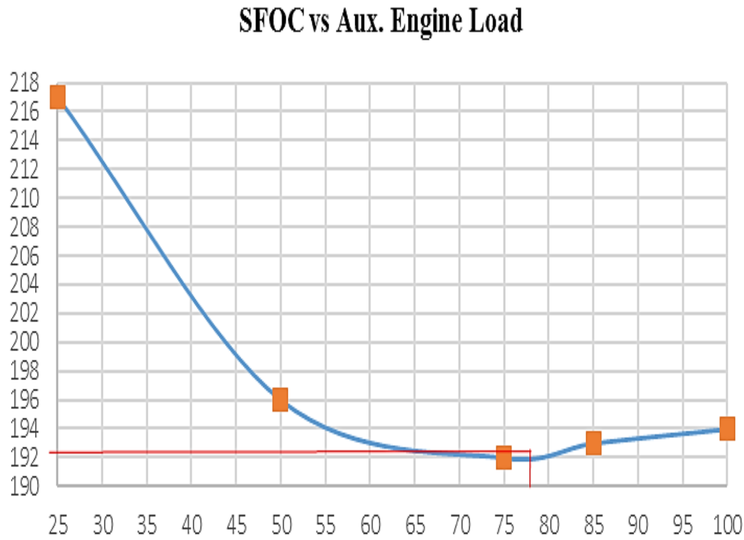

Figure IV.7 SFOC at 76,5% Auxiliary Engine Load

$$\frac{494{,}4\ kW}{650\ kW} x100\% = 76{,}13\%\ \text{Aux Engine Load}$$

From the SFOC vs Aux. Engine Load graphic, the SFOC could be determined by pulling of the lines at 76,13 % engine load and the result of the SFOC is at 191,94 gr/kW. h.

After SFOC have already obtained, the FOC at the 76,13% for 1 hour can be determined as the calculation figured below.

Gr = SFOC x kW x Hr (duration)
= 191,94 x 494,4 x 1
= 94938 gr/hr
= 0.0949Ton/hr

So the fuel oil savings which can be done by the ORC system at 85% of the main engine load per hour are:

Fuel savings = FOC (85% AE Load) – FOC (76,5% AE Load)
= 0,1066 – 0,0950
= 0.0116 Ton/hr
= 11,649 kg/hr

By using the similar process and calculation, the complete result for fuel saving that can be obtained by the ORC system are figured at table below:

Table IV.21 Fuel Oil Saving by ORC System per Main Engine Load

Main Engine Load %	Fuel Oil Saving (kg) / hr
77,5	11,2510
80	11,5650
85	11,6491
90	11,7994
95	11,8328
100	11,8519
Average	**11,6582**

4.6.1. Added Scenario:

Assuming that the ships is operated for 330 days (7920 hours) per year and the rest of the 30 days will be at docking times, then the fuel that can be saved (85% Main Engine Load) are:

Gr = Fuel saving per hour x Hr (duration) (IV.19)
= 11,6491 x 7920
= 92261,25 kg per year
= 92,26125 tons per year

The ship is using Pertamina HSD as a fuel for auxiliary engine, therefore the density of the fuel is 843 kg/m^3. To calculate the liters saving per year is figured below:

$$V = \frac{m}{\rho} = \frac{92261{,}25}{843}$$

$$V = 109{,}444 \text{ m}^3/\text{yr}$$

= 109444 liters/yr

Table IV.22 Fuel Saving per year per Main Engine Load

Main Engine Load %	Fuel Oil Saving (kg) / hr.	Fuel Oil Saving (Ton) / yr.	Fuel Oil Saving liters/yr.
77,5	11,2510	89,1077	105703,1456
80	11,5650	91,5950	108653,73
85	11,6491	92,2612	109443,9542
90	11,7994	93,4511	110855,4571
95	11,8328	93,7158	111169,4304
100	11,8519	93,8674	111349,2526
Average	11,6582	92,3331	109529,1617

As the main engine load increased, the saving of the fuel is also increased. Because this correlate with the total of heat that can be utilized, the higher of the main engine load the higher of the total heat that can be utilized and also the more electricity power can be produced by the organic Rankine cycle system

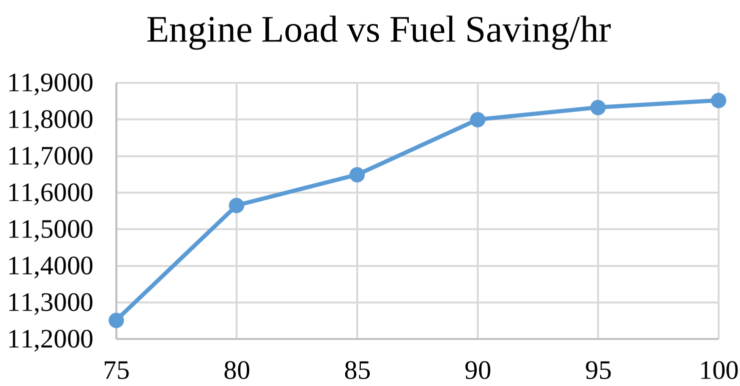

Figure IV.8 Main Engine Load vs Fuel Saving/hr

4.6.2. Price Forecasting:

Forecasting is required to predict the future value of the HSD fuel prices. To predict the future HSD prices is required the forecasting data and the data which is used in this part obtained from IMF Spot Crude Oil/ barrel.

From the data, the trendlines is representing the future prices for crude oil within next year.

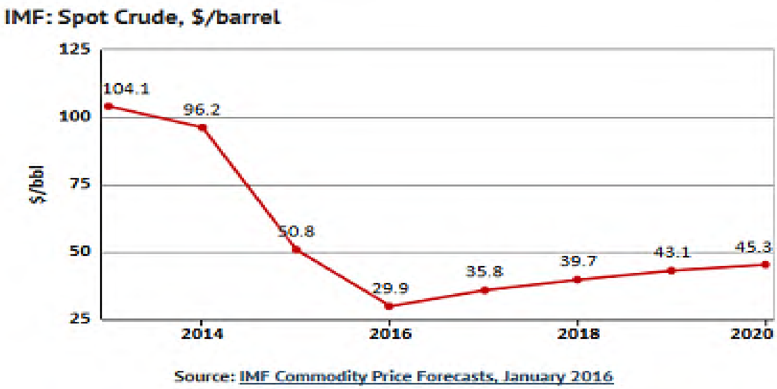

Figure IV.9 IMF Spot Crude Oil /Barrel Price Forecasting

According to the data in the figure above (from International Monetary Fund), the crude oil prices from 2015 is going down to 2016 and from 2016 is forecasted to be increase gradually per year. The data is figured below on the table:

Table IV.23 IMF Spot Crude Oil Price Forecasting

Year	2015	2016	2017	2018	2019	2020
$	50,8	29,9	35,8	39,7	43,1	45,3

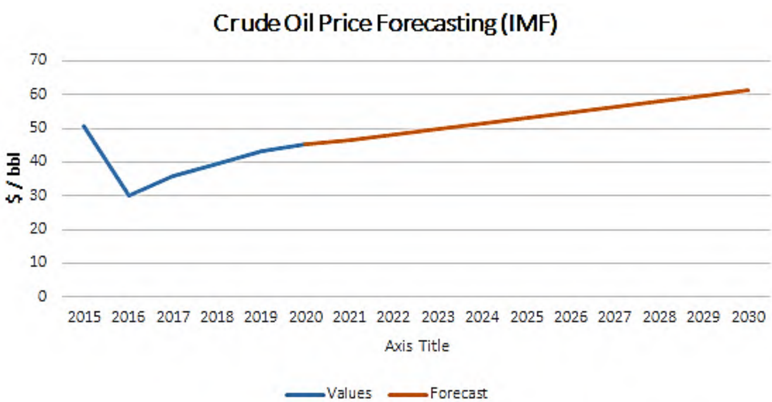

Figure IV.10 Crude Oil Price Forecasting (Next 20 Years)

For the next 20 years of the crude oil price, the price can be predicted by using the trendline which the data is obtained by crude oil prices from the previous year (according to the table above). The crude oil prices increase rate (%) from 2016 until 2035 from the result of the trendline above are tabulated below on the table:

Table IV.24 Crude Oil Prices Increase Rate per Year Forecasting

Year	2016-2017	2017-2018	2018-2019	2019-2020	2020-2021
% Increase	5,9	3,9	3,4	2,2	1,35

Year	2021-2022	2022-2023	2023-2024	2024-2025	2025-2026
% Increase	1,64	1,64	1,64	1,64	1,64

Year	2026-2027	2027-2028	2028-2029	2029-2030
% Increase	1,636	1,636	1,636	1,636

After the data have already obtained from the previous crude oil forecasting result, so the increasing price of HSD is assumed to follow the increasing price of crude oil that have already forecasted by IMF (International Monetary Fund) and the dollar price is to be locked at Rp. 13.000,00 per US Dollar. The HSD price data in 2015 obtained from the average of Pertamina HSD price in every month. For the annual HSD price increase can be seen in the table below:

Table IV.25 HSD Price per Litre (Forecasting Result)

	HSD Price Liter	
Year	**Rupiah**	**$**
2015	10504	0,808
2016	8309	0,639
2017	8799	0,677
2018	9142	0,703
2019	9453	0,727
2020	9661	0,743
2021	9792	0,753
2022	9952	0,766
2023	10115	0,778
2024	10280	0,791
2025	10448	0,804
2026	10619	0,817
2027	10793	0,830
2028	10970	0,844
2029	11149	0,858
2030	11331	0,872

4.7. Defining the Organic Rankine Cycle Equipment Price

List of organic Rankine cycle equipment are obtained from supplier manufacturers information and some equipment price data are obtained from estimation using website for the estimation.

Table IV.26 Price of Organic Rankine Cycle Equipment

Equipment	Manufactures	Type	Price ($)
Refrigerant Pump	Warrender, Ltd Pump	Seal-Less Mag Drive Pump	16000
Steam Turbine	G-Team Micro Steam Turbine	TR-Hi 150	114000
Evaporator	Funke ViFlow TPL	TPL 02-K	6320
Condenser	ALVA LAVAL	CB4000	10920
Refrigerant Cooling Pump	BBA Pumps	B100 Centrifugal	7900
Valve	Samson	Three Way Control	2800

According to the "Cost and Project Engineering Handbook" the maintenance cost is approximated 10% from the price of each equipment's and the installation cost of each equipment estimated to be 20% of static equipment (valve, condenser, evaporator) and 25% of rotary equipment steam turbine, pump) (Humphreys, 2006). The list of maintenance and installation cost are listed below:

Table IV.27 Organic Rankine Cycle Equipment Maintenance & Installation Cost

Equipment	Price ($)	Maintenance Cost	Installation Cost
Refrigerant Pump	16000	1600	4000
Steam Turbine	114000	11400	28500
Evaporator	6320	632	1264
Condenser	10920	1092	2184
Refrigerant Cooling Pump	7900	790	1975
Valve	2800	280	280
Total Price	157940	15794	38203

4.5.1. Profit per year

The profit is calculated by multiplying the fuel that can be saved by using ORC system (liters/year) and the price of the fuel per liters. As the price of the fuel is already forecasted to be increased per year, hence the money that can be saved also increased per year. The tables below are show how much fuel oil and money that can be saved per main engine load and per year.

Table IV.28 Profit per Year at Each Main Engine Load (2016 - 2018)

Main Engine Load %	Fuel Oil Saving (Ton) / yr.	Fuel Oil Saving (liter)/yr.	Profit 2016 ($/yr.)	Profit 2017 ($/yr.)	Profit 2018 ($/yr.)
77,5	89,1078	105703,14	67558	71544	74334
80	91,5951	108653,73	69444	73541	76409
85	92,2613	109443,95	69949	74076	76965
90	93,4512	110855,45	70851	75031	77957
95	93,7158	111169,43	71051	75244	78178
100	93,8674	111349,25	71166	75365	78304
Average	**92,3331**	**109529,16**	**70003,15**	**74133,34**	**77024,54**

Table IV.29 Profit per Year at Each Main Engine Load (2018-2024)

Main Engine Load %	Profit 2019 ($/yr.)	Profit 2020 ($/yr.)	Profit 2021 ($/yr.)	Profit 2022 ($/yr.)	Profit 2023 ($/yr.)	Profit 2024 ($/yr.)
77,5	76861	78552	79616	80918	82242	83588
80	79007	80745	81838	83177	84538	85921
85	79581	81332	82434	83782	85153	86546
90	80608	82381	83497	84863	86251	87662
95	80836	82614	83733	85103	86495	87910
100	80967	82748	83869	85241	86635	88053
Average	79643,37	81395,53	82497,72	83847,35	85219,07	86613,23

Table IV.30 Profit per Year at Each Main Engine Load (2025-2030)

Main Engine Load %	Profit 2025 ($/yr.)	Profit 2026 ($/yr.)	Profit 2027 ($/yr.)	Profit 2028 ($/yr.)	Profit 2029 ($/yr.)	Profit 2030 ($/yr.)
77,5	84955	86345	87758	89193	90652	92135
80	87327	88755	90207	91683	93183	94707
85	87962	89401	90863	92350	93861	95396
90	89096	90554	92035	93541	95071	96626
95	89349	90810	92296	93806	95340	96900
100	89493	90957	92445	93958	95495	97057
Average	88030,20	89470,34	90934,05	92421,70	93933,70	95470,42

4.8. Financing Scenario

The first capital is obtained from 40% from own investment & 60 % capital obtained from loan at Bank BNI from 2016 until 2020 with interest rate 11% per year as the details are shown in the table below:

Table IV.31 Credit Scenario for 5 year

Credit Scenario for 5 year	
Total equipment price	$ 157940
Self-financing (40%)	$ 63176
Bank credit financing (60%)	$ 94764

As the cost from interest and the duty to pay the bank loan (exclude taxes) are shown in the table below:

Table IV.32 Cost for Added Interest and Load Repayment

Year	Interest	Interest ($)	Loan Repayment	Total Loan Repayment
2016		Rp 10.424	Rp. 18.953	Rp. 29.377
2017		Rp. 10.424	Rp. 18.953	Rp. 29.377
2018	11%	Rp. 10.424	Rp. 18.953	Rp. 29.377
2019		Rp. 10.424	Rp. 18.953	Rp. 29.377
2020		Rp. 10.424	Rp. 18.953	Rp. 29.377

By the time the equipment run through year by year, therefore the component asset value going to depreciate. The depreciation value will be estimated 10% per year. Inflation factor is to be added to the expenditure calculation. The value for this inflation is set to 6% per year as per Bank Indonesia average inflation occurred. As the list of expenditure cost are listed below:

Table IV.33 Expenditure Cost from 2016 - 2020

Year	2016	2017	2018	2019	2020
Equipment Price	157940	0	0	0	0
Installation Cost	38203	0	0	0	0
Depreciation (10%)	-	15794	15794	15794	15794
Interests (11%)	10424	10424	10424	10424	10424
Loan Repayment	18953	18953	18953	18953	18953
Maintenance cost (10%)	-	15794	16584	17413	18284
Total Expenditure before Inflation	225519	60964	61754	62584	63454
Inflation (6%)	13531	3657	3705	3755	3807
Total Expenditure after Inflation	239051	64622	65459	66339	67261

Table IV.34 Expenditure Cost from 2021 - 2025

Year	2021	2022	2023	2024	2025
Equipment Price	-	-	-	-	-
Installation Cost	-	-	-	-	-
Depreciation (10%)	15794	15794	15794	15794	15794
Interests (11%)	-	-	-	-	-
Loan Repayment	-	-	-	-	-
Maintenance cost (10%)	19198	20158	21165	22224	23335
Total Expenditure before Inflation	34992	35952	36959	38018	39129
Inflation (6%)	2100	2157	2218	2281	2348
Total Expenditure after Inflation	37091	38109	39177	40299	41477

Table IV.35 Expenditure Cost from 2026 - 2030

Year	2026	2027	2028	2029	2030
Equipment Price	0	0	0	0	0
Installation Cost	0	0	0	0	0
Depreciation (10%)	15794	15794	15794	15794	15794
Interests (11%)					
Loan Repayment					
Maintenance cost (10%)	24502	25727	27013	28364	29782
Total Expenditure before Inflation	40296	41521	42807	44158	45576
Inflation (6%)	2418	2491	2568	2649	2735
Total Expenditure after Inflation	42713	44012	45376	46807	48310

4.6.1. Break-even Point

The break-even point (BEP) happen when the total cost (expenditure) and the total revenue (profit) are equal, there are no gained profit or loss. In this case, the total profit value (fuel oil saving) is equal with the spending of an expenditure.

Table IV.36 Spending vs Profit Result of Proposed ORC System in MV. Tanto Tenang Ship in dollar

No.	Year	Income	Spending	Profit	Present Value
1	2016		239051	-239051	-215361
2	2017	74133,34	64622,73	-229540	-186300
3	2018	77024,54	61754,54	-214270	-156673
4	2019	79643,37	66338,75	-200966	-132382
5	2020	81395,53	67261,63	-186832	-110876
6	2021	82497,72	37091,21	-141425	-75612
7	2022	83847,35	38108,69	-95687	-46088
8	2023	85219,07	39177,04	-49645	-21542

No.	Year	Income	Spending	Profit	Present Value
9	2024	86613,23	40298,81	-3330,3	-1302
10	2025	88030,20	41476,67	43223,3	15223
11	2026	89470,34	42713,42	89980,2	28549
12	2027	90934,05	44012,01	136902	39132
13	2028	92421,70	45375,53	183948	47369
14	2029	93933,70	46807,22	231075	53608
15	2030	95470,42	48310,5	278235	58152

According to the table above, the break-event point for the cost to install the proposed organic Rankine cycle system in typical ship is happened in years 10.

fuel saving which can be saved from electric power generated by the organic Rankine cycle system.

3.1.10. Conclusion and Suggestion

After all the stages have completed, the result from the research that obtained at the previous stage will be concluded. The conclusions are the answer for the problem that formularized from the earlier stage at Chapter 1 and it is a summary from the research and data analysis at the previous stage.

3.2 Flowchart of Research Method

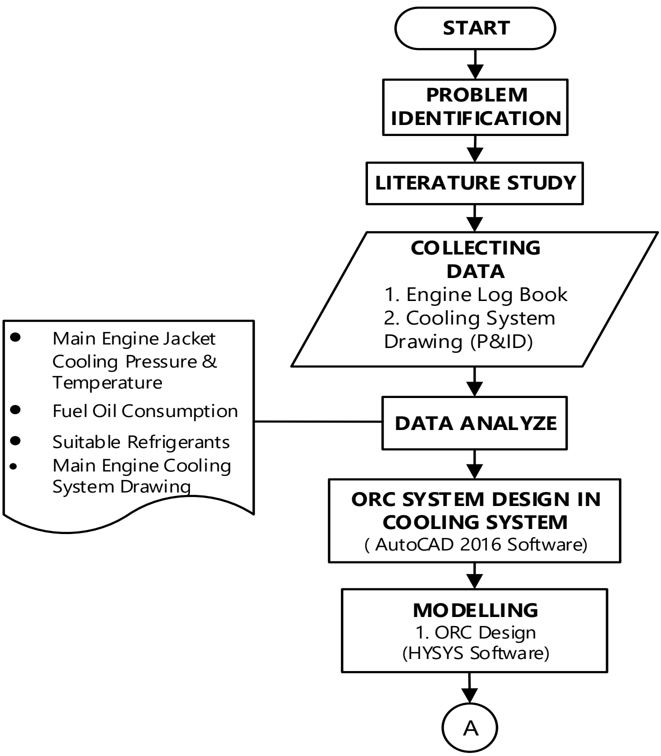

Figure 3.1 Flowchart of Research Method

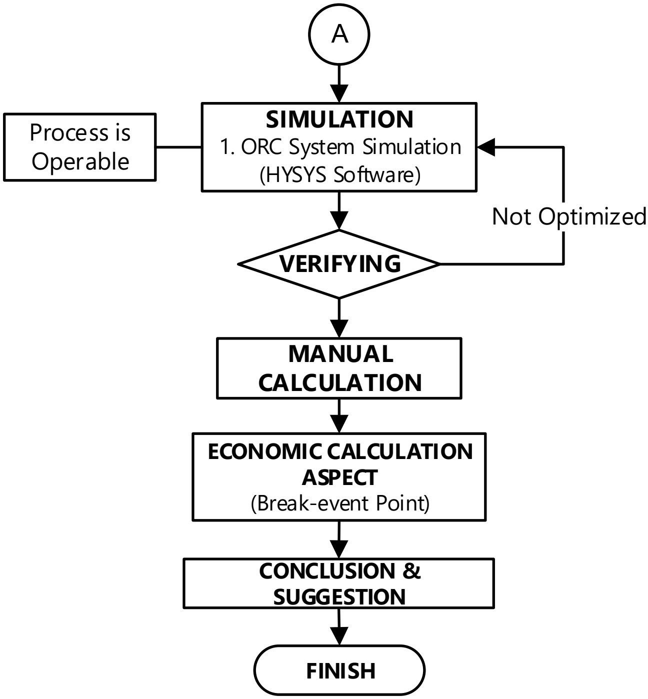

Figure 3.1 Flowchart of Research Method (continued)

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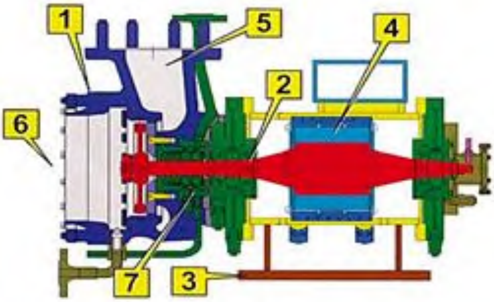
Enclosure A
Equipment

Expander (Turbine)

TR Hi 150

The TR Hi 150 turbine is a special machine for reducing the water steam pressure developed for extremely low steam flows with high requirements for mechanical operation safety and high efficiency. The turbine casing together with the electric generator stator forms an assembly without a coupling and secondary rotating parts. The turbine wheel overhangs the rotor of the high-frequency electric generator. The rotor tightness against steam leakage is ensured by means of a special mechanical seal. A complete oil system for the supply of the control and lubrication oils is a part of the turbine.



Description:

1. Turbine body
2. Turbine rotor
3. Welded steel frame
4. High-frequency electric generator
5. Steam admission
6. Steam exhaust
7. Steam seal

Parameters	TR Hi 150
Output	maximum 150 kW
Admission steam pressure	0,3 – 4,0 MPa a
Exhaust steam pressure	0,05 – 1,4 MPa a
Admission steam temperature	maximum 420 °C
Concrete foundation size	400 x 780 mm

Heat Exchanger (Condenser)

CB400

Brazed Plate Heat Exchanger

General information
Alfa Laval introduced its first brazed plate heat exchanger (BHE) in 1977 and has since continuously developed and optimized its performance and reliability.

Brazing the stainless steel plates together eliminates the need for gaskets and thick frame plates. The brazing material seals and holds the plates together at the contact points ensuring optimal heat transfer efficiency and pressure resistance. The plate design guarantees the longest possible life.

The design options of the brazed heat exchanger are extensive. Different plate patterns are available for various duties and performance specifications. You can choose a standard configuration BHE, or a unit designed according to your own specific needs. The choice is entirely yours.

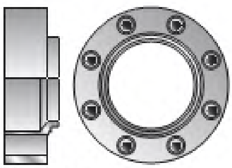
Typical applications
- HVAC heating/cooling
- Process heating/cooling
- Hydraulic oil cooling
- Oil cooling

Working principles
The heating surface consists of thin corrugated metal plates stacked on top of each other. Channels are formed between the plates and corner ports are arranged so that the two media flow through alternate channels, usually in countercurrent flow for the most efficient heat transfer process.

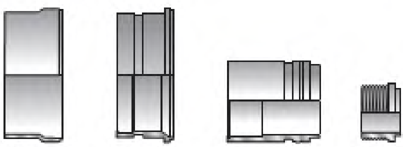
Standard design
The plate pack is covered by cover plates. Connections are located in the front or rear cover plate. To improve the heat transfer design, the channel plates are corrugated.

Particulars required for quotation
To enable Alfa Laval's representative to make a specific quotation, specify the following particulars in your enquiry:
- Required flow rates or heat load
- Temperature program
- Physical properties of liquids in question
- Desired working pressure
- Maximum permitted pressure drop

Examples of connections



Compact flanges



Welding Clamp Soldering External threaded

Important components in refrigeration applications

Installed for a wide range of duties in refrigeration applications worldwide, Alfa Laval's high performance Brazed plate heat exchangers (BHEs) offer highest heat transfer performance with maximum reliability and cost efficiency.

The two typical equipment used in refrigeration are chiller and heat pump.

Chiller
Chillers are cooling water or brine and rejecting the heat to air or water. The water is transported by a hydraulic system through different types of heat exchanger to cool air in an air conditioning system or to cool manufacturing or industrial processes. Two basic systems are normally used to drive chillers: a compressor driven by an electric motor, based on a vapour compression refrigeration cycle; or a heat-driven system (steam, burning natural gas), based on an absorption refrigeration cycle.

Heat pump
Heat pumps are a type of water chillers which can also run in a reverse cycle, also called a water-source heat pump. In this case the primary function is heating water and rejecting the heat to air or water. The heated water warms up air in the air conditioning system. Another variation of this system is ground source heat

Technical specifications

AC230 brazed plate heat exchanger with a single refrigerant circuit unit. It is suitable to work as an evaporator and as condenser.

CB60, copper brazed plate heat exchanger. The brazing material seals and holds the plates together at the contact points ensuring optimal heat transfer efficiency and pressure resistance.

AXP52 is a brazed plate heat exchanger with thin external frames that withstands operating pressures of 130 bar. AXP52 is specially designed to fulfill the need when using CO_2 as refrigerant in subcritical and transcritical applications.

Evaporator

AC	AC16	AC18	AC30EQ
Capacity, kW/(HP)	1-5 (1,3-6,7)	2-10 (2,7-13,4)	3-30 (4-40)
Double circuit	No	No	No
Design pressure, Bar/(Psig)	32 (464)	32 (464)	35 (507)
High Pressure ACH, Bar/(Psig)	45 (653)	45 (653)	50 (650)
Height, a, mm/(inch)	210 (8,27)	316 (12,4)	325 (12.8)
Width, b, mm/(inch)	74 (2,91)	74 (2,91)	93 (3.66)
Vertical connection distance, c, mm/(inch)	172 (6,78)	278 (10,9)	269 (10.59)
Horizontal connection distance, d, mm/(inch)	40 (1,57)	40 (1,57)	39 (1.53)

Condenser

CB	CB16	CB18	CB30
Capacity kW/(HP)	1-5 (1,3-6,7)	2-10 (2,7-13,4)	5-40 (6.7-54)
Design pressure, Bar/(Psig)	10 (145)	10 (145)	40 (450)
High Pressure CBH, Bar/(Psig)	32 (464)	32 (464)	50 (653)
Height, a, mm/(inch)	210 (8,27)	316 (12,4)	313 (12.32)
Width, b, mm/(inch)	74 (2,91)	74 (2,91)	113 (4.45)
Vertical connection distance, c, mm/(inch)	172 (6,78)	278 (10,9)	250 (9.84)
Horizontal connection distance, d, mm/(inch)	40 (1,57)	40 (1,57)	50 (1.97)

Evaporator, gas cooler, economizer and desuperheater for transcritical C

AXP, CBXP	AXP10	AXP14	CBXP27
Capacity, kW/(HP)	2-15 (2.7-20)	10-35 (13.4-47)	40-70 (53.6-94)
Pressure, Bar/(Psig)	154 (2233)	140 (2030)	90 (1305)
Height, a, mm/(inch)	190 (7.48)	190 (7.48)	310 (12.20)
Width, b, mm/(inch)	76 (2.99)	76 (2.99)	111 (4.37)
Vertical connection distance, c, mm/(inch)	154 (6.06)	154 (6.06)	250 (9.84)
Horizontal connection distance, d, mm/(inch)	40 (1.57)	40 (1.57)	50 (1.97)

pumps, using the earth or water surface to take the heat.

Refrigeration systems

The BHEs provide an efficient solution for a range of functions in the equipment in these refrigeration systems. The most common of these involve transferring heat from two basic media: the refrigerant as the primary fluid (HFC or natural gas) and water or brines as the secondary fluid.

Alfa Laval offers a complete portfolio of BHEs for the following applications:

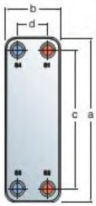
- Evaporator (dry expansion) to cool water,
- Condenser to reject or recover heat to water,
- Desuperheater for partial heat recovery to water,
- Economizer to cool liquid refrigerant and superheat vapour refrigerant,
- Subcooler to cool down the liquid refrigerant,
- Intermediate heat exchanger in the absorption cycle to preheat the diluted solution and to pre-cool the concentrated solution
- NEW! Gas cooler (transcritical CO_2) to reject or recover heat to water.

AC70X	AC112	AC220EQ	AC230DQ AC230EQ	AC232DQ	CB300X	AC500DQ AC500EQ
10-90 (13,4-120)	30-200 (40-270)	50-200 (67-270)	50-200 (67-270)	30-250 (40-330)	150-450 (200-605)	150-600 (200-805)
No	No	No	Yes	Yes	No	Yes
32 (464)	37 (537)	37 (537)	32 (464)	47 (682)	33 (370)	32 (464)
45 (653)	51 (740)	51 (740)	45 (653)	53 (769)		45 (653)
526 (20.71)	616 (24.25)	616 (24.25)	490 (19.29)	490 (19.29)	990 (38.98)	739 (29.09)
112 (4.41)	191 (7.52)	191 (7.52)	250 (9.84)	250 (9.84)	366 (14.41)	322 (12.68)
466 (18.35)	519 (20.43)	519 (20.43)	400/369 (15.75/14.53)	400/369 (15.75/14.53)	816/861 (32.12/33.90)	632/568 (24.88/22.36)
50 (1.97)	92 (3.62)	92 (3.62)	155 (6.1)	155 (6.1)	213.5 (8.40)	205 (8.07)

CB60	CB62	CB110	CB112	CB200	CB300	CB400
50-100 (67-134)	50-100 (67-134)	50-220 (67-295)	50-220 (67-295)	150-350 (200-469)	150-450 (200-605)	150-600 (200-805)
40 (450)	40 (450)	37 (537)	37 (537)	30 (363)	33 (370)	35 (464)
50 (653)	50 (653)	51 (595)	51 (595)	37 (406)		
527 (20.75)	531 (20.91)	616 (24.25)	616 (24.25)	742 (29.21)	990 (38.98)	990 (38.98)
113 (4.45)	115 (4.53)	191 (7.52)	191 (7.52)	324 (12.76)	366 (14.41)	390 (15.35)
466 (18.35)	476 (18.74)	519 (20.43)	519 (20.43)	622 (24.49)	816/861 (32.12/33.90)	825 (32.48)
50 (1.97)	60 (2.36)	92 (3.62)	92 (3.62)	205 (8.07)	213.5 (8.40)	225 (8.86)

O_2 applications

CBXP52	AXP27	AXP52
40-100 (53.4-134)	10-100 (13.4-134)	10-150 (13,4-200)
90 (1305)	130 (1885)	130 (1885)
526 (20.71)	362 (14.25)	582 (22.91)
111 (4.37)	160 (6.30)	160 (6.30)
466 (18.35)	250 (9.84)	466 (18.35)
50 (1.97)	50 (1.97)	50 (1.97)



Brazed plate heat exchangers – compact and cost-efficient

The first Alfa Laval brazed plate heat exchangers (BHEs) were developed in the seventies. Today they are well-established components in refrigeration systems due to their compactness, durable designs, ease of installation and cost efficient operation.

Material

The brazed plate heat exchanger (BHE) consists of thin corrugated stainless steel plates vacuum brazed together using copper as the brazing material.

Design

Brazing the stainless steel plates together eliminates the need for sealing gaskets and thick frame plates. As well as holding the plates together at their contact points, the brazing material seals the package. Alfa Laval's BHEs are brazed at all contact points, ensuring optimal heat transfer efficiency and pressure resistance. The plates are designed to provide the longest possible lifetime.

Since virtually all surfaces of the brazed plate heat exchanger actively contribute to heat transfer, the BHE is very compact in size, and it has a low weight and a low hold-up volume.

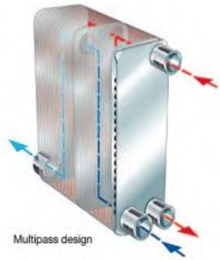
Alfa Laval offers a wide range of standard heat exchanger models and sizes, tailor-made for a wide scope including refrigeration applications. Standard configurations are available from stock and customer-specific designs are available on request.

Flow principle

The basic flow principle in a brazed plate heat exchanger for refrigeration applications is parallel or diagonal flow to achieve the most efficient heat transfer process.

In a single pass design, all connections are located on one side of the heat exchanger, making installation very easy.

Multipass design and different types of connections are available. Optionally, the location of connections can be chosen.



Multipass design

Flow principle in Evaporator design

The channels formed between the corrugated plates and corners are arranged so that the two media flow through alternate channels, always in opposite directions (counter current flow).

The two phase refrigerant (vapour and liquid) enters the bottom left of the exchanger with a vapour quality depending on the operating condition of the plant. Evaporation of the liquid phase takes place inside the channels and some degrees of superheat are always requested, which is the reason why the process is called "dry expansion".

In the illustration of an evaporator the dark and light blue arrows show the location of the refrigerant connections. The water (brine) to be cooled flows counter current in the opposite channel; the dark and light red arrows show the location of the water (brine) connections.

Evaporator, showing flow principle.

Flow principle in Condenser design

The main components are the same as for the evaporator. The refrigerant enters at top left of the exchanger as hot gas and starts to condense on the surface of the channels until fully condensed, and is then slightly subcooled. The process is called "free condensation".

In the illustration of a condenser the light and dark blue arrows show the location of the brine connections. The refrigerant flows counter current in the opposite channel and is cooled. The light and dark red arrows indicate the locations of the refrigerant connections.

Condenser, showing flow principle.

Heat Exchanger (Evaporator)

FUNKE

TPL The powerful unit for top heat transfer rates in case of media with average viscosity and high viscosity

The TPL unit is especially developed for mechanical engineering and plant engineering (for cooling of hydraulic oil and motor oil etc.)
Volume of the flow gap is max. 80 % bigger compared to classical heat transfer plates.
By means of special turbolators placed between the flow gaps and by diagonal media flow together with large diameter connections very high heat transfer rates are reached.

In case of media with higher viscosity the unit can be much smaller in size compared to conventional plate heat exchangers!

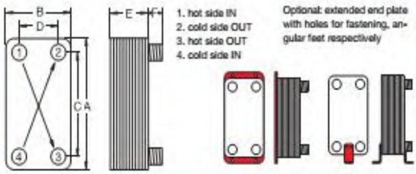
Media

- oil/water
- water/water
- gas/liquid (condensation)
- specific media on request

Applications
(heating, cooling, condensing)
System separation, use of waste heat and heat recovery in mechanical engineering, process technology as well as refrigeration engineering and domestic technology, for instance
- cooling of lubricating-oil
- district heat transfer stations
- system separation on cooling ceilings

Limit conditions
- operating pressure max. 30 bar
- operating temperature min. -160°C, max. +200°C
- heat duty 2,0 to 2000 kW

Location of the connections:
standard: on front plate
optional:
on front plate and end plate



Caution: on principle brazed plate heat exchangers are unsuited for: seawater, ammonia, demineralized water, silicone oil and high-chloride media (exception: NPL series)

Type	Dimensions (mm) A	B	C	D	E	F	Connections	max. number of plates	Empty weight kg	max. volume-flow m³/h	Volume/ channel (liter)
TPL 00-K	274	111	213	50	6+4xN	50	G 1"	60	1,7+0,23xN	13	0,098
TPL 00-L	439	111	378	50	6+4xN	50	G 1"	60	2,4+0,40xN	13	0,134
TPL 01-K	383	168	309	94	6+4xN	50	G 1 1/2"	90	2,9+0,48xN	45	0,206
TPL 01-L	631	168	557	94	6+4xN	50	G 1 1/2"	90	4,8+0,87xN	45	0,321
TPL 02-K	488	225	403	140	6+4xN	50	G 2"	120	5,0+0,83xN	70	0,351
TPL 02-L	818	225	733	140	6+4xN	50	G 2"	120	8,3+1,50xN	70	0,574

N = number of plates

Advantages

With the FUNKE brazed plate heat exchangers TPL a high heat transfer rate at a low pressure loss can be obtained. The thermodynamically and hydraulically optimized alignment of the turbulence sheets generates high turbulent flow rates even at low volume flows.

This allows for efficient use of the heat exchange area available and leads to a perfectly optimised heat transfer. The high turbulent flow also results in an efficient self-cleaning effect, which greatly reduces maintenance and time-out. FUNKE TPL have a compact design and are used for high pressures and temperatures.

As the whole surface of the plates is coated with solder, the resulting brazing allows for a higher integrity joint. Thus, stresses arising from e.g. variations in pressure or temperature can be absorbed more effectively compared to conventional brazed plate heat exchangers where only single dots of solder are applied.

Applications

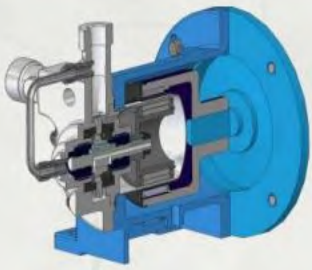
Typical applications for brazed plate heat exchangers are heating, cooling, condensing

- System separation
- Heating engineering (solar thermal systems, central heating, floor heating)
- Cogeneration units
- Heat pumps
- Heat extraction and heat recovery in domestic and process technology
- Hot water / Process water
- Refrigeration engineering
- Evaporation / Condensing in cooling systems
- Mechanical engineering
- Oil cooling
- Air drying
- Hydraulic oil cooling
- Cooling of machines and motors
- Mold machine temperature control
- Economizing

Refrigerant Pump Specification

WMTA - *Machined Billet, Single Stage*

ALLOY



Cross-sectional view

WMTA Machined Billet, Single Stage

Performance Range

Flow	0.25-45 GPM	0.5-10 M^3/H
Head	To 700 Feet	213 M
Temp	-148 to 600°F	-100 to 316°C
System Pressures	To 7250 PSIG	500 BAR

WMTA machined billet alloy regenerative turbine sealless mag-drive pumps can be constructed for high system pressures and special alloy configurations. All WMTA pumps are equipped with zero leakage magnetic couplings to meet the latest toxic emissions regulations. The absence of elaborate mechanical seal systems eliminates costly pump maintenance, lost production time and process contamination.

Variations in head calculations have minimal effect on the flow of a turbine pump. Also, turbine pumps can be throttled to a required duty point without by-passing.

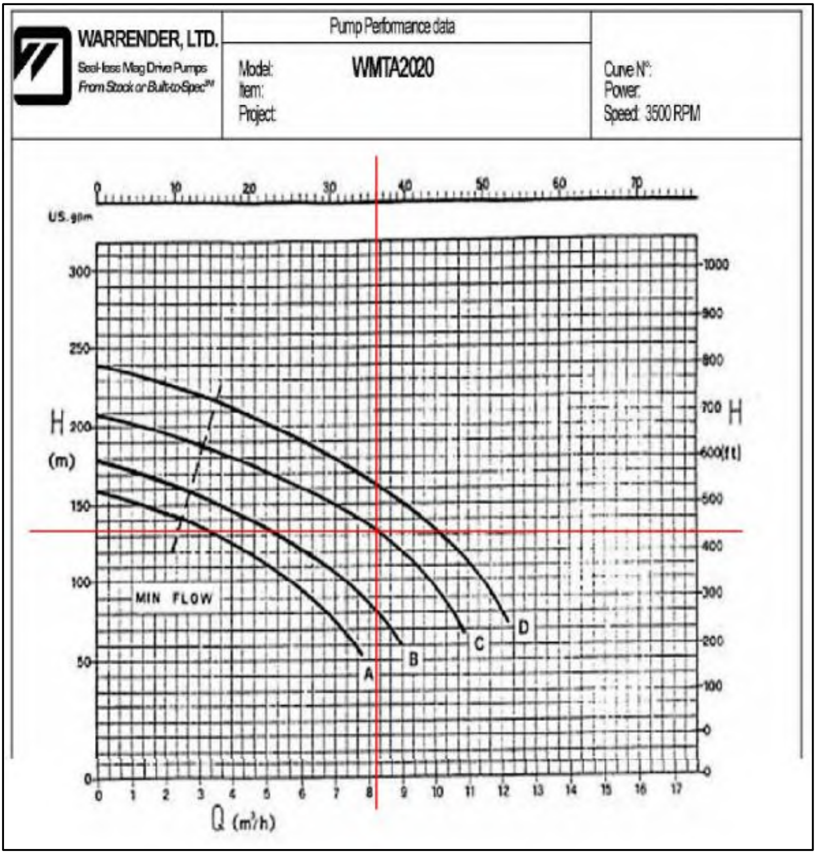
DESIGN FEATURES

- High head / low flow capability minimizes by-pass requirements and prevents overheating of centrifugals and high head cavitation
- Self balancing impeller—zero axial thrust loading
- Impeller design handles up to 20% entrained gas—ideal for pumping liquefied gases
- No galling or metal to metal contact
- Heavy duty alloy containment shell
- High torque magnets, suitable for direct starting motors
- Heavy walled casings withstand extreme system pressures.

MATERIALS

- **AISI SS-316 Stainless Steel**
- **Alloy-20**
- **Incoloy-825**
- **Hastelloy-C27**

Refrigerant Pump Specification (continue)



Refrigerant Water Cooler Pump

BBA Pumps

Pumps for results

B100 Electric Drive
Self-Priming Centrifugal Pump
Max. 140 m^3/hour, Max. 19 mwc

Pump specifications:

Type	B100 BVGMC
Max. flow	140 m^3/hour
Max. head	19 mwc
Connections	4" BSP
Impeller type	Open impeller
Free passage	46 mm
Pump speed	1450 rpm
Electric motor	7.5 kW - 4 pole
Voltage	400/690V 50 Hz
Assembly	Monobloc
Weight (net)	186 kg

FEATURES

B self-priming centrifugal pump

The B range of pumps has been designed with a clear focus on reliability, efficiency and durability, ranging from 1-1/2" to 12". Featuring an open impeller configuration in combination with a changeable wear plate, facilitating servicing and minimising costs of spares. All B series pumps are fitted with a non return valve in the suction port, eliminating the need for a foot valve in suction line and allowing quick priming.

Complete package designed & built by BBA Pumps

- Complete in-house design & production
- Over 60 years of experience in the market
- Extensive testing facility in-house
- Contemporary & functional design
- Durable & eco-friendly materials
- Custom builds available

Design features

- Basic, reliable technology
- Large solids handling capacity
- Suitable for use with hot fluids
- Cooled shaft seal
- Extended life due to pump design
- Available with ATEX approval

After sales service & product support

- Single supplier for parts, spares & accessories
- Dedicated customer help-desk (24h service)
- Dedicated service department in-house
- Global parts distribution network
- Optional global on-site servicing
- Extensive training options available (technical & commercial), on-site or in-house

Fields of application

- Shipping
- Industry
- Construction
- Waste water management
- Emergency pump units
- Irrigation
- Agriculture
- Horticulture

- *Refrigerant Water Cooler Pump Curves*

Three Way Valve

Series 240
Type 3244-1 and Type 3244-7 Pneumatic Control Valves
Type 3244 Three-way Valve

DIN and ANSI versions

Application

Mixing or diverting valve for process engineering and industrial applications

Valve size	**DN 15 to 150 · NPS ½ to 6**
Pressure rating	**PN 10 to 40 · Class 150 and 300**
Temperatures	**−196 to +450 °C · −325 to +842 °F**

Type 3244 Three-way Valve with

- Type 3271 Pneumatic Actuator (Fig. 1)
- Type 3277 Pneumatic Actuator (Fig. 2) for integral positioner attachment

Valve body made of

- Cast iron (DIN version only)
- Cast steel
- Cast stainless steel

Undivided valve bonnet

The control valves, designed according to the modular assembly principle, can be equipped with various accessories: Positioners, limit switches, solenoid valves and other accessories according to IEC 60534-6 and NAMUR recommendation (see Information Sheet ► T 8350 for more details).

Versions

Standard version designed for temperatures from −10 to +220 °C (15 to 430 °F) with pneumatic actuator

- **Type 3244-1** (Fig. 1) · Type 3244 Valve with Type 3271 Actuator (see Data Sheet ► T 8310-1)
- **Type 3244-7** (Fig. 2) · Type 3244 Valve and Type 3277 Pneumatic Actuator for integral positioner attachment (see Data Sheet ► T 8310-1)

Further versions

- **Insulating section or bellows seal** · See Technical data
- **Adjustable packing** · Details on request
- **Heating jacket**
- **Additional handwheel** · See Data Sheet ► T 8310-1
- **Type 3244-2 Electric Control Valve** · Details on request
- **Type 3244-3 Manually Operated Valve** with Type 3273 Hand-operated Actuator · See Data Sheet ► T 8312

Fig. 1: *Type 3244-1 Pneumatic Control Valve with Type 3271 Actuator*

Fig. 2: *Type 3244-7 Pneumatic Control Valve with Type 3277 Actuator*

Principle of operation (Fig. 3 and Fig. 4)

Depending on the version, the three-way valve can be used for either mixing or diverting service.

In mixing valves, the process media to be mixed enter at valve ports **A** and **B**. The combined flow exits the valve at port **AB** (see Fig. 3). The flow rate from ports **A** or **B** to **AB** depends on the cross-sectional area of flow between the seats and plugs.

In diverting valves, the process medium enters at the valve port **AB** and the partial flows exit at ports **A** and **B** (see Fig. 4).

Note: The design of the mixing and diverting valves in sizes DN 15 to 25 (NPS ½ to 1) is identical.

Fail-safe position

Depending on how the springs are arranged in the pneumatic actuator (▶ T 8310-1), the valve has two different fail-safe positions effective upon air supply failure.

- **Actuator stem extends:** when the supply air fails, port **B** is closed in mixing valves and port **A** is closed in diverting valves.
- **Actuator stem retracts:** when the supply air fails, port **A** is closed in mixing valves and port **B** is closed in diverting valves.

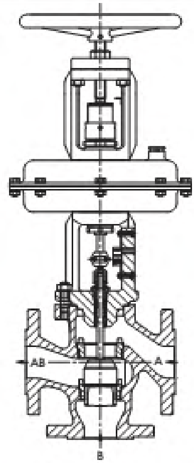


Fig. 3: *Type 3244-1 Pneumatic Control Valve with Type 3244 Three-way Valve (plug arrangement for mixing service, DN 15 to 25 for diverting service as well, and Type 3271 Actuator with additional handwheel)*

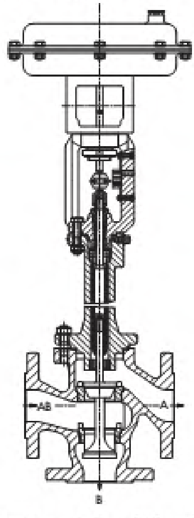


Fig. 4: *Type 3244-7 Pneumatic Control Valve with Type 3244 Three-way Valve, DN 32 to 150 (plug arrangement for diverting service), additional bellows seal and Type 3277 Actuator*

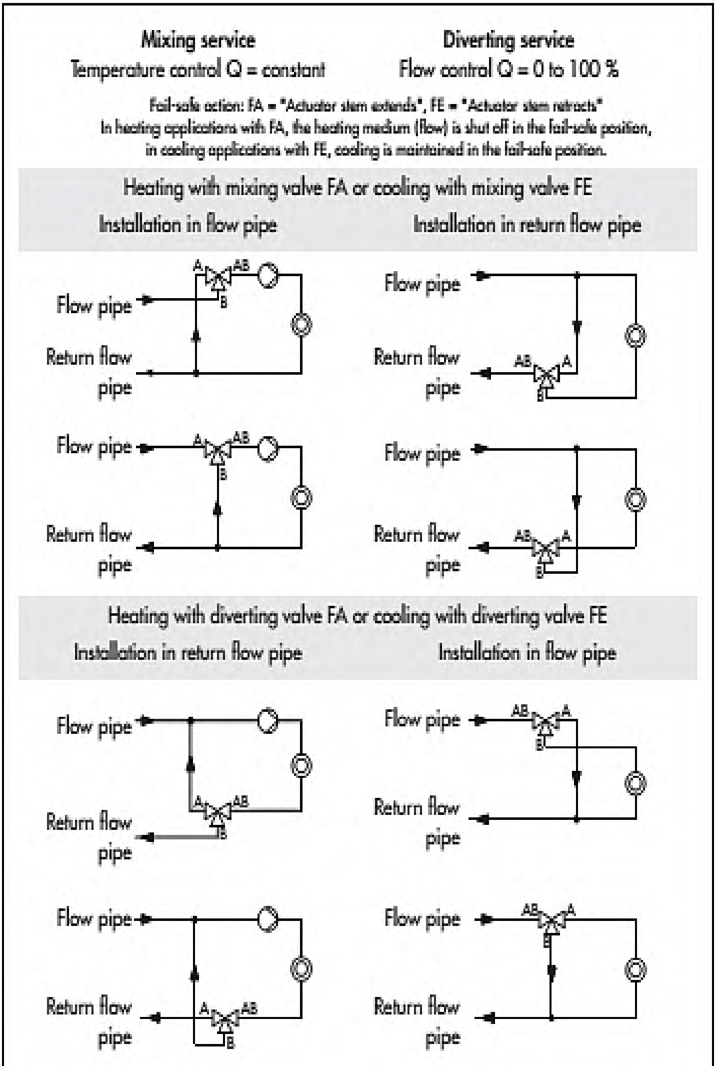
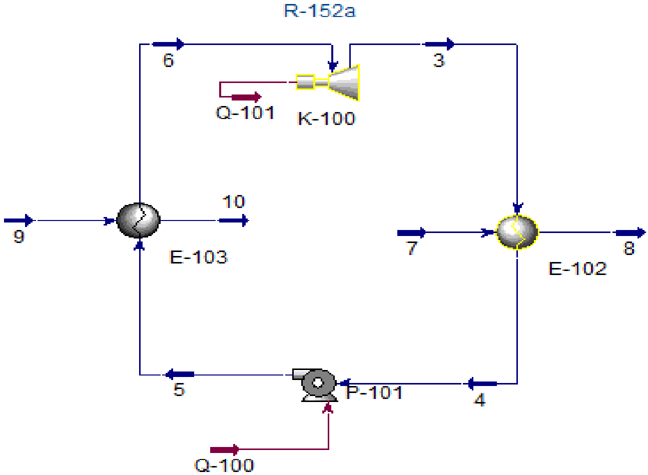


Fig. 2: *Typical installations*

Enclosure B
Simulation

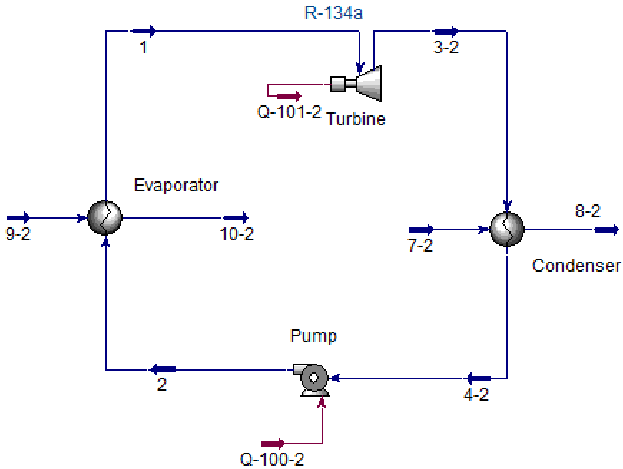
Running Simulation Result

- ***Simulation using R-152 as a Working Fluid***



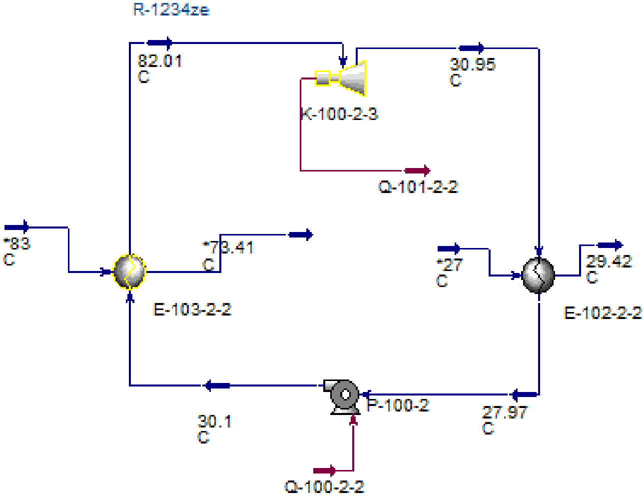
Error occurred: Liquid in Inlet Stream (Turbine), Temperature Cross (Condenser)

- ***Simulation using R-134a as a Working Fluid***



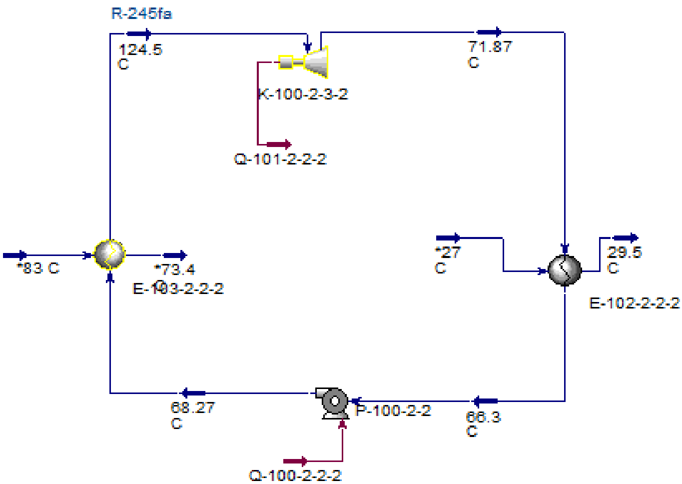
Error occurred: -

- ***Simulation using R-1234ze as a Working Fluid***



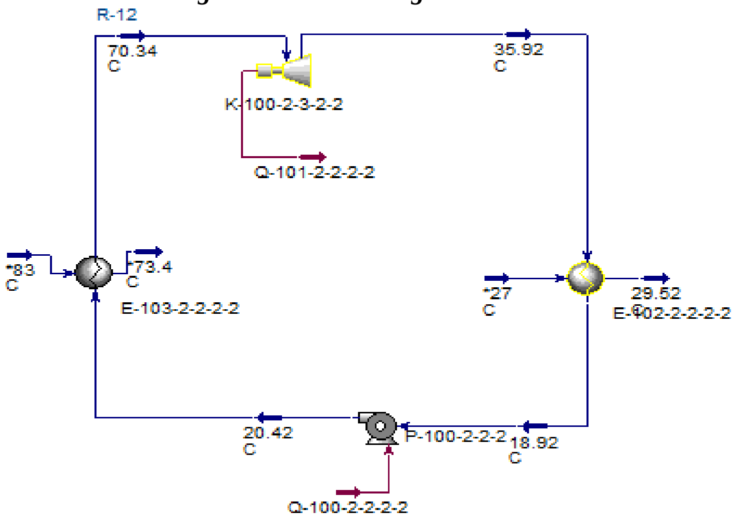
Error occurred: Liquid in Inlet Stream (Turbine), Temperature Cross (Evaporator)

- ***Simulation using R-245fa as a Working Fluid***



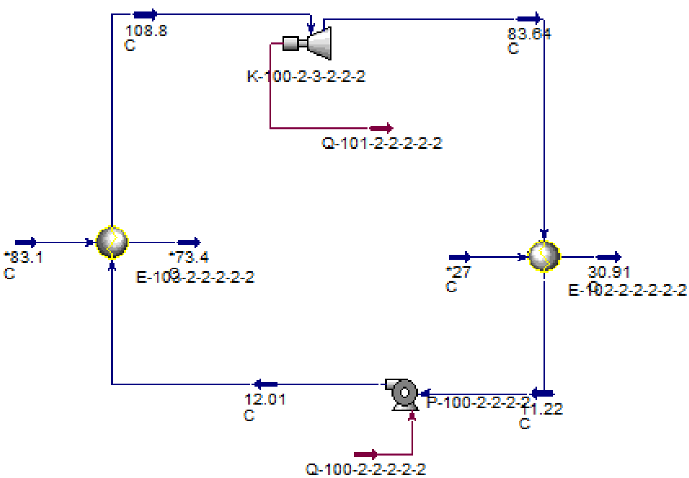
Error occurred: Liquid in Inlet Stream (Turbine), Temperature Cross (Evaporator)

- ***Simulation using R-12 as a Working Fluid***



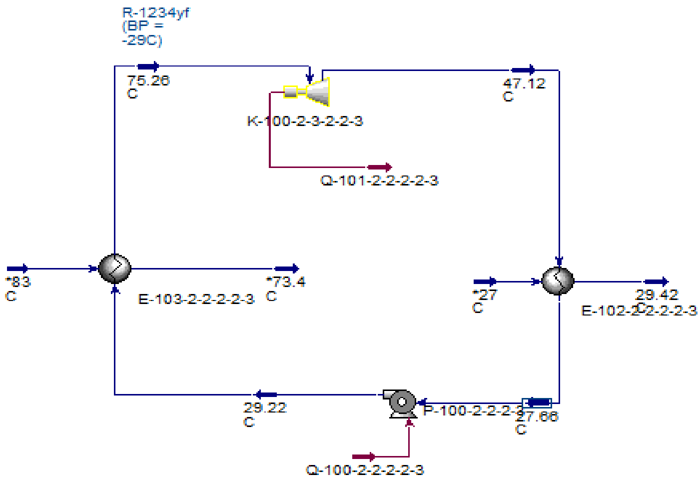
Error occurred: Liquid in Inlet Stream (Turbine), Temperature Cross (Condenser)

- ***Simulation using R-22 as a Working Fluid***



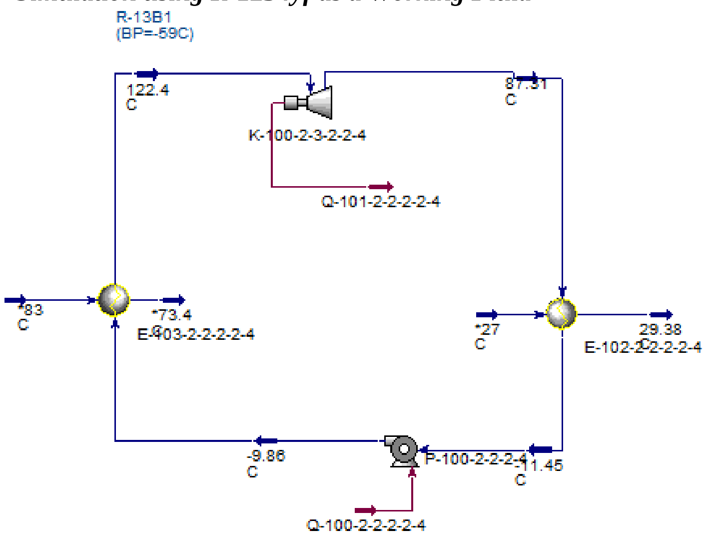
Error occurred: Temperature Cross (Evaporator), Temperature Cross (Condenser)

- ***Simulation using R-1234yf as a Working Fluid***



Error occurred: Liquid in Inlet Stream (Turbine)

- ***Simulation using R-1234yf as a Working Fluid***



Error occurred: Temperature Cross (Evaporator), Temperature Cross (Condenser)

Enclosure C
P-h Diagram

P-h Diagram of R-134a refrigerant implemented with ORC system.

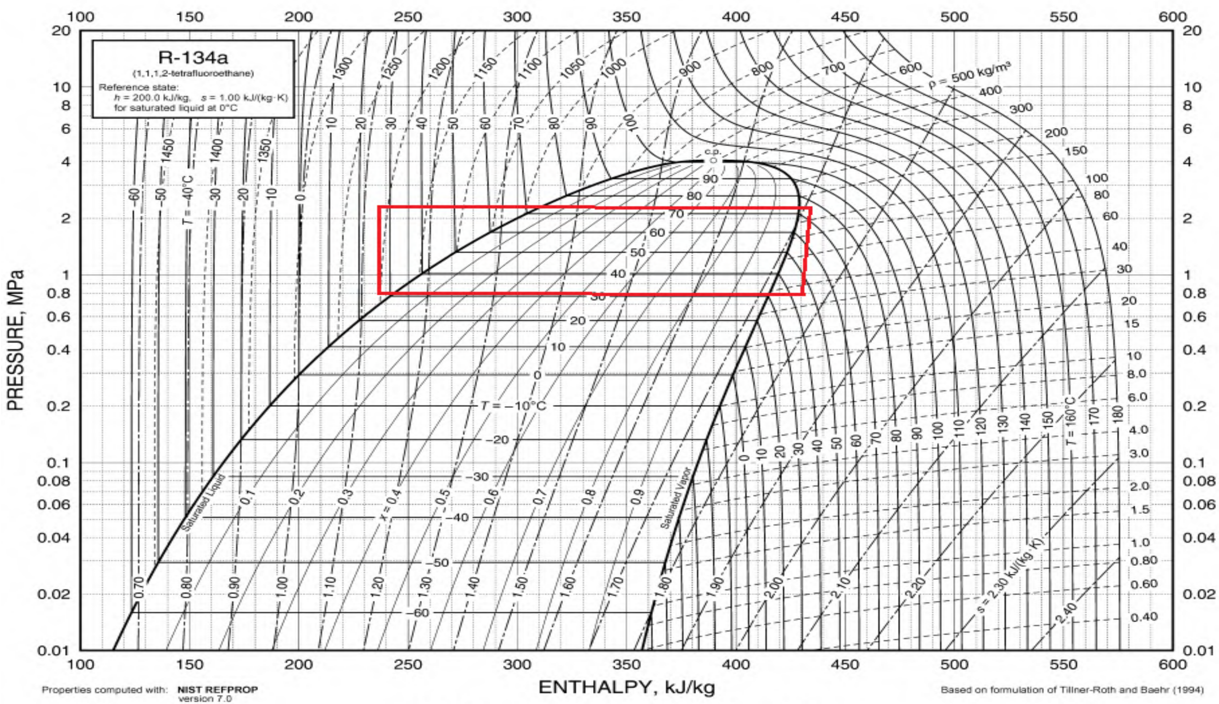


Fig. 8 Pressure-Enthalpy Diagram for Refrigerant 134a

CHAPTER V
CONCLUSION & SUGGESTION

5.1. Conclusion

Based on data analysis and result which has been done in the previous section can be concluded as follows:

1. The electricity power generated by the ORC system is different depends on main engine load. As the main engine load increase, the electricity power generated also increase. At 77,5% of main engine load electricity power generated reach 55,55 kW, at 80% of main engine load electricity power generated reach 57,20 kW, at 85% of main engine load 57,64 kW, at 90% of main engine load electricity power generated reach 58,44 kW, at 95% of main engine load electricity power generated 58,61 kW and at 100% of main engine load electricity power generated reach 58,71 kW. Averagely, the electricity power that can be generated is 57,69 kW.
2. The best working fluid to use is R-134a
3. The fuel oil which can be saved by installing ORC system is different depends on main engine load. As the main engine load increase, the fuel savings is also increase. At the 77,55% the fuel oil saving reach 11,2510 kg/hr, 11,5650 kg/s at 80% of main engine load, 11,6491 kg/hr at 85% of main engine load, 11,7994 kg/hr at 90%, 11,8328 kg/hr at 95% of main engine load and 11,8519 kg/hr at 100% of main engine load.
4. Based on the economic analysis which has been described in the previous part, break-event point of proposed organic Rankine cycle installation in typical ship is happen in years 10.

5.2. Suggestion

As for suggestions which writer can be deliver correlate with this bachelor thesis are and for the further improvement of this research are:

1. To make the system performs well and reliable, the automation and control of the control valve in the jacket water cooling system still should be designed as the main engine load increased or decreased.
2. The sailing duration of a ship (hour) might be adapted in a real sailing time of a typical ship.
3. The placement of organic Rankine cycle equipment onboard should consider the available space in ship engine room.
4. The analysis of the economic should be considering the piping system cost also the organic refrigerant cost and to be more detailed in order to make it more realistic.

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Motto: Fortune favors the bold.