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System-level comparison of energy consumption for maritime SOx reduction technologies

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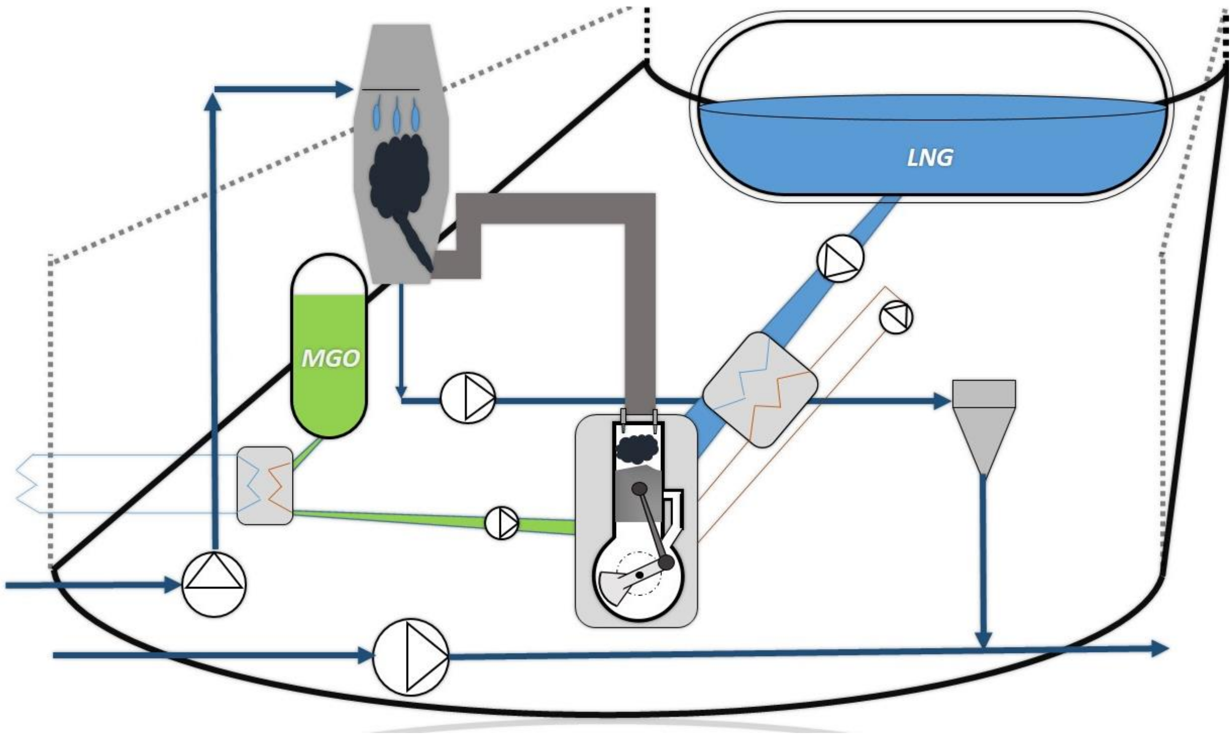

Preface

This Master's Thesis is written during the spring semester 2014, at the Norwegian University of Life Sciences (NMBU), Department of Mathematical Sciences and Technology. The Thesis is written in cooperation with DNV GL, Environmental Advisory.

The subject of Thesis is highly relevant in as it will affect all ship owners trading in large parts of the EU coasts, in less than a year. For a machine process engineer, the subject have a lot of interesting and challenging obstacles related to engines, energy balances and process flows.

I would like to express my great gratitude to: My advisor at DNV GL, environmental consultant Dr. Océane Balland, for her great effort in advising and supporting me through the whole process. My advisor at NMBU, Associate professor Dr. Jan Kåre Bøe for his help and advice. Senior environmental consultant at DNV GL Martin Christian Wold, for initiating the cooperation with DNV GL, and for helpful input. Marianne Charlotte Svendsen for her patience and support.

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Pål Evang Nundal

Abstract

Increasingly stringent regulations of emissions from the marine sector is coming into effect. Amongst other, a limit for SOx content in marine fuels will soon come into effect in large parts of Europe and North America, referred to as Emission Control Areas (ECA).[1]

This thesis aim to analyze and compare the energy consumption from three of the most common options to reduce SOx emissions to a compliant level. These are; Switching to a gas engine and use Liquefied Natural Gas (LNG) as fuel, installing a SOx scrubber and continue using heavy fuel oil (HFO) as the main fuel, or switching from HFO to a low sulfur diesel while being inside an ECA. HFO which are the most common used fuel today, are also included to have a known reference to compare with.

These technologies are quite new (the LNG engines revived are not yet installed on an operating vessel) so the access to literature is somewhat limited. A large part of the material used is found in articles, some is found in projects done by DNV GL, and some from manufacturer websites. However, the components consuming most of the energy is pumps and heat exchangers, subjects that are curriculum for process engineers.

The data material needed to perform the analysis is somewhat limited. This is because manufactures do not supply very detailed information in their brochures, a large part is confidential, and as earlier mentioned the technologies are quite new and have not been researched a lot. A large proportion of the data is taken from internal projects done at DNV GL. Some from open literature. Where reliable data required is not found, there were done assumptions.

The results of this thesis summarized:

Two-stroke low pressure LNG has an energy consumption of 0,03 % of installed engine power. The required heat transfer to vaporize and heat the fuel is 264,7 kW.

Two-stroke high pressure LNG has an energy consumption of 0,32 % of installed engine power. The required heat transfer to the fuel is 180,9 kW.

The open and closed loop scrubbers had an energy consumption of respectively 1,9-2,2 % and 0,9-1,1 % of the engine power scrubber. This is in addition to the consumption from the HFO system.

The energy consumption for MGO and HFO varies from 0,7-1,5 %. It is assumed that the real number is higher for HFO since several components were excluded, and because it high viscosity will give a higher pressure loss in pipes and components.

Sammendrag

Strengere og strengere krav til utslipp fra skip trer løpende i kraft. Blant annet innføres det om kort tid nye grenser for SOx-innholdet i marine drivstoff, i store deler av Europa og Nord Amerika.

Målet med oppgaven er å analysere og sammenligne energiforbruket fra de tre vanligste tiltakene for å redusere SOx-utslipp til et tillatt nivå. Disse er: Et midlertidig bytte fra tungolje til diesel med lavt nok svovelinnhold når skipet entrer et regulert område, et permanent bytte til flytende naturgass (LNG) som drivstoff, eller benytte tungolje som drivstoff og rense eksosen for SOx i en såkalt skrubber. HFO som er det klart mest utbredte drivstoffet inkluderes også for å ha ett bedre sammenligningsgrunnlag.

Disse systemene er relativt nye (LNG motorene brukt i oppgaven er ennå ikke installert på noe skip), så tilgangen på litteratur er noe begrenset. En del er tatt fra artikler, en del er tatt fra prosjekter gjort hos DNV GL, og noe er tatt fra produsentenes nettsider. Det viser seg at det mye av energien går med til pumping og oppvarming, temaer som er pensum for en prosessingeniør.

Datagrunnlaget for analysene er noe begrenset. Dette skyldes at produsenter ikke oppgir veldig spesifikk informasjon i sine brosjyrer, en god del er konfidensielt, og som tidligere nevnt at teknologiene er så nye at det ikke er skrevet veldig mye åpent tilgjengelig litteratur om det. En god del av datamaterialet er hentet fra interne prosjekter gjort hos DNV GL. Noe er hentet fra åpne rapporter. Der nødvendige verdier ikke er funnet, er det gjort antagelser.

Resultatene av denne masteroppgaven er kort oppsummert:

To-takts lavtrykk-LNG har et energiforbruk på 0,03 % av installert motoreffekt. Varmeoverføringen som må til for fordampe og gi rett temperatur på gassen er 264,7 kW.

To-takts høytrykk-LNG har et energiforbruk på 0,32 % av installert motoreffekt. Nødvendig varmeoverføring til drivstoffet er 180,9 kW.

I SOx-skrubberene er energiforbruket for åpen- og lukket modus henholdsvis 1,9-2,2 % og 0,9-1,1 % av ytelsen til motoren de renser eksosen på. Dette kommer i tillegg til energiforbruket fra HFO systemet.

Energiforbruket ved bruk av MGO og HFO som drivstoff varierer fra 0,7-1,5 %. Det er antatt at verdiene vil være en del høyere for HFO da en del komponenter ikke er tatt med i beregningene, samt at dets høye viskositet vil gi et større trykktap gjennom rør og komponenter.

Contents

1. Introduction

1.1. List of abbreviations

A lot of abbreviations are used in technological literature. Table 1 shows the full form of all abbreviations used in this thesis.

Table 1: List of abbreviations used in text.

Abbreviation	Full form
SOx	Sulfur Oxide
ECA	Emission Control Area
IMO	International Maritime Organization
MARPOL	The International Convention for the Prevention of Pollution from Ships/ Maritime Pollution
DNV GL	Det Norske Veritas Germanischer Lloyd
IMO	International Maritime Organization
HFO	Heavy Fuel Oil
IFO	Intermediate Fuel Oil
RM	Residual Marine Fuel
DM	Distillate Marine Fuel
LNG	Liquefied Natural Gas
MGO	Marine Gas Oil
MDO	Marine Diesel Oil
LSF	Low Sulphur Fuel
BSFC	Brake Specific Fuel Consumption
LCV	Lower Calorific Value
TDC	Top Dead Center
BDC	Bottom Dead Center
kWh	Kilo Watt Hours
ATM	Atmospheric Pressure
HP	High Pressure
LP	Low Pressure
CAPEX	Capital Expenses
OPEX	Operational Expenses
NaOH	Sodium Hydroxide
PBU	Pressure Buildup Unit
RPM	Revolutions Per Minute
P&ID	Process & Instrumentation Diagram

1.2. SOx regulations forcing changes

All fossil fuels contains sulphur. When fossil fuels are combusted, sulphur reacts with oxygen and forms sulphur oxides (SOx) that are released with the exhaust. Studies have shown that exposure to SOx, or particles created when SOx reacts with other compounds in the atmosphere, can cause or worsen diseases such as emphysema and bronchitis, and aggravate existing heart disease.[2]

The International Maritime organization (IMO)[a] has agreed upon a set of regulations to limit SOx emissions from ships, through the Marpol Annex VI.[1] Annex VI has one fuel oil sulphur limit applying worldwide and one far more stringent applying inside Emission Control Areas (ECAs). Today these ECAs are parts of the Baltic Sea, the North Sea, the coast of North America and the United States Caribbean Sea. Figure 1 show the precise location of current ECAs and possible future ECAs.

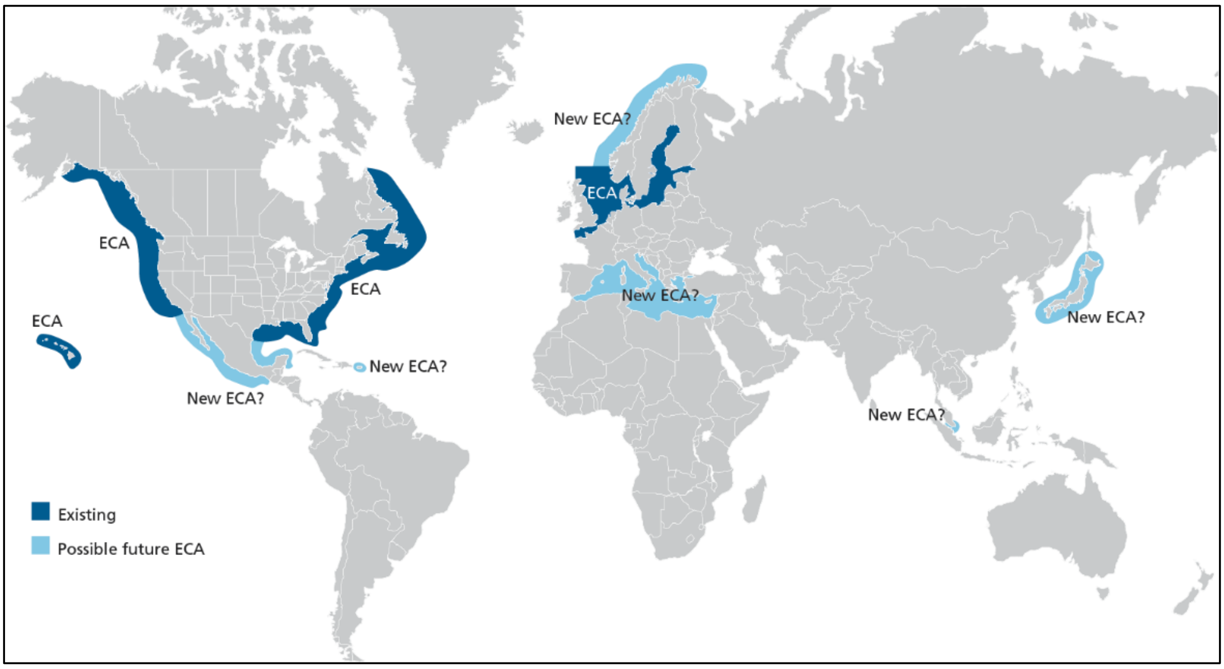


Figure 1: Location of current and possible future ECAs[3]

The current limit of sulphur content in fuel oil, inside any ECA is per this date 1,0 % m/m[b]. From 1st January 2015 this limit will be lowered to 0,1 % m/m.[1] The worldwide fuel oil sulphur limit is currently 3,5 % m/m. However, a worldwide fuel oil sulphur limit of 0,5 % m/m is expected to come in to effect by 1st January 2020[c]. These limits apply to fuel being used in all

[a] IMO is the United Nations agency responsible for safety, security and pollution in the shipping industry
[b] % m/m – Mass percentage of sulphur in fuel.
[c] Could be postponed to 1. January 2025, depending on the outcome of a review to be concluded in 2018.

combustion equipment and devices onboard. Regulation 4 of the Marpol VI allows flag administrations to approve alternatives to a fuel switch if it reduce the SOx emissions to the equivalent of a fuel complying to the sulphur limits.

Table 2: Sulphur limits Worldwide and inside ECAs. [1]

Outside an ECA established to limit sox and particulate matter emissions	Inside an ECA established to limit sox and particulate matter emissions
4.50% m/m prior to 1 January 2012	1.50% m/m prior to 1 July 2010
3.50% m/m on and after 1 January 2012	1.00% m/m on and after 1 July 2010
0.50% m/m on and after 1 January 2020	0.10% m/m on and after 1 January 2015

The majority of commercial vessels are fueled by heavy fuel oil (HFO) which has a sulphur content higher than the upcoming limits. A vessel running on HFO will therefore not be able to sail inside an ECA after 2015 without taking actions to reduce sulphur emissions.

1.3. How to comply with SOx regulations

There are several methods to comply with the forthcoming ECA SOx regulations, but there are currently three alternatives regarded as the most reliable:

- **Low sulphur fuels:** Switching to a low sulphur fuel as Marine Gas Oil (MGO) or Marine Diesel Oil (MDO). In this thesis the term “low sulphur fuel” is referring to MGO with less than 0,1 % sulphur. Most engines can switch between HFO and MGO with very few modifications. This allow vessels to keep their engine as it is and use HFO as fuel outside ECAs, while switching to a low sulphur fuel when entering one. MGO’s low viscosity and lubricity can increase wear and tear on equipment, especially if used over long periods of time.[4]

  The low sulphur fuels are quite expensive compared to HFO and LNG. For example, in Singapore the current price of MGO is more than 50 % higher than the IFO380 price.[5] This price gap causes high operational expenses (OPEX), which makes it a less attractive option for vessels with a lot of sailing time inside ECAs.

- **Liquefied Natural Gas:** Using a gas engine with Liquefied Natural Gas (LNG) as the main fuel. Engines running on LNG emits up to 95 % less SOx emissions than an engine running on HFO.[6] Liquefying natural gas reduces its volume significantly, giving it an energy density high enough to be a realistic option to HFO.

LNG is stored in a specially designed tank at approximately - 163°C. Before being injected to the cylinders of the engine, LNG is pumped from the tank, vaporized, and regulated to the required pressure.

Some of the obstacles related to LNG are the capital expenses of purchasing a LNG engine, or alternatively rebuilding an existing engine, purchasing an LNG tank, and finding room for the LNG tank which is quite large. Another challenge is the missing infrastructure for bunkering LNG. As LNG is a relative new marine fuel there are limited numbers of LNG bunkering stations and most of the operational ones are located in Norway and the Northern part of Europe.

Operational expenses of running on LNG are quite difficult to predict, as there does not exist a global LNG price.[6]

- **Wet scrubber:** Installing a scrubber and continue to use HFO as the main fuel. An approved scrubber reduces the sulphur content in the exhaust to the equivalent of a fuel complying to the sulphur limits, making it possible to sail in an ECA without switching fuel.

  Inside the scrubber unit, alkali water is sprayed over the exhaust. The alkali water reacts with and removes SOx from the exhaust.

  In an open loop scrubber, seawater which has a natural alkalinity, is used as scrubbing water. Scrubbing water exits at the bottom of the scrubber tower before its cleansed and discharged to the ocean. Sludge created in the process is stored in a tank until proper disposal is possible.

  In a closed loop scrubber, fresh water added which a chemical, typically NaOH, is used as scrubbing water. Used scrubbing water is cooled by seawater, boosted with more NaOH, and circulated back into the scrubber unit.

  A third variant of the scrubber is the hybrid scrubber. This one has the necessary piping and instrumentation to switch between both open and closed loop mode.

  Since all energy consumptions from a scrubber are in addition to the energy consumption from running on HFO, the OPEX are quite high. They are still lower than the OPEX of MGO due to the fuel price difference. A scrubber is expensive, but less than installing a LNG system. Due to the scrubbers weight and high vertical placement, the vessels center of gravity hence stability might be affected.[7]

  Table 3 sums up the costs of these different SOx reduction options.

Table 3: CAPEX and OPEX: MGO, Scrubber and LNG.

	MGO	Scrubber	LNG
CAPEX	$	$$	$$$
OPEX	$$$	$$	$ / $$

1.4. DNV GL

This Master thesis is written for DNV GL, Environment Advisory. This section is dealing with reducing emissions from ships. Amongst many other areas, this department specializes in helping ship owners chose the right option to comply with the forth-coming ECA regulations. Based on a vessels operational profile and layout, DNV GL estimates future fuel consumption, required space for new equipment, CAPEX, OPEX and compare the different options for each specific ship. The ability to give precise estimations is a key factor for this type of business.

This thesis aims at analyzing the energy consumption of the different ECA compliance options. While the fuel consumption of the engines are well documented, there is a need to better understand and capture the energy consumption of all the other components required for the different options. Increased knowledge about energy consumption would enables DNV GL to give more accurate estimates of the OPEX related to the different options.

1.5. Technical difficulties

- Make broad and general analysis which is applicable to all scenarios.
- Presenting the results and findings in a way that they can be compared for the different technologies.
- Obtain data on different components that are not in DNV GLs archives or described literature open to public.

2. Project plan

The project plan describes the steps towards achieving the aim of this thesis and report it, in time for the deadline. This was done in an early stage of the process, but have been updated some during the process.

2.1. Aim

As the title states, the main aim of this thesis is to do a:

"***System-level comparison of energy consumption for maritime SOx reduction technologies"***

To be more specific, the aim is to analyze and compare the total energy consumption from all components required to use the SOx reduction technologies described in chapter 1. "All components required" are in this case components that are specific to each system. Components that are required for all the systems, as cylinder lubrication and engine cooling, are not included. This is specified in the limitations chapter.

Energy consumption is a broad term. In this thesis energy consumption is mainly presented as required power for a component as percentage of the engines shaft power.

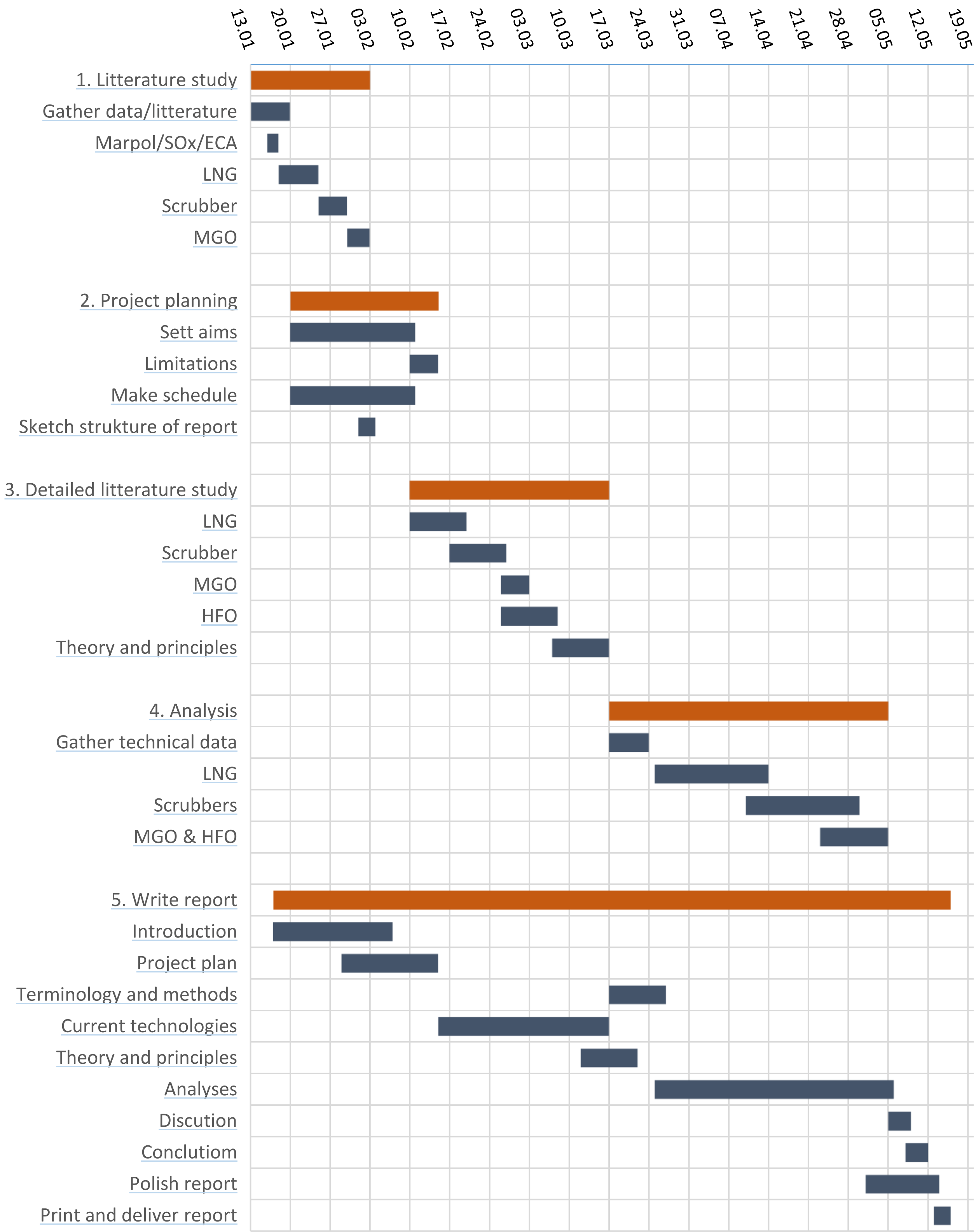
2.2. Partial aims

- Understand the technologies and their processes.
- Identify all components in a typical setup.
- Determine which components consumes significant amount of energy and should be included in the analysis.
- Acquire necessary input for analysis.
- Evaluate energy consumption.
- Use brochures and information from suppliers to verify calculations.
- Compare different technologies.
- Conclude

2.3. Project schedule

The Gantt chart in Table 4 shows when the main part of an activity is planned to begin and be finished. Most activities will be reviewed and updated several times later in the process.

Table 4: Schedule.



2.4. Limitations and terms for analysis

This Section is listing the limitations that have been considered in this work.

Applying to all:

- The analysis is based on a scenario where the engines have an output of approximately 7 MW.
- Pressure losses in pipes and over equipment (unless specified) are not included.
- All equipment except from in the scrubber, are assumed to be placed at the same height.
- Assumed efficiency are 0,75 for all pumps.
- Economic factors will not be included, beyond chapter one.
- Consumption from computers, monitoring systems and other electronics, are not included.

Specific to LNG:

- The LNG engines included are dual fuel, two-stroke with high- and low pressure.
- LNG will be regarded as pure methane, when required heating is calculated.

Specific to Scrubbers:

- The scrubber systems included are wet scrubbers in open loop and closed loop.
- There will be done there calculations for placement of the scrubber. These are 10, 20 and 30 meters above the seabed.
- In a closed loop system, there will be a fourth scenario for an alternative placement of the mixing tank. More details are in chapter 6.

Specific to Low sulphur diesel:

- MGO is the only low sulphur fuel liquid fuel included.
- There will be done two scenarios for fuel injection pressure. One with 1000 bar and one with 2000 bar.

Specific to HFO:

- Heating of storage tank is not calculated.
- Centrifuges, and filters are not included.

3. Terminology and methods

3.1. Definitions

Table 5: Definitions.

Term	Definition
Bar	1 Bar = 100 000 Pascal
Barg	Gauge pressure. Pressure in bar above atmospheric pressure
ATM	The pressure at sea level. Defined as 1 bar in this thesis.
Efficiency	Theoretic power / Real power Useful work done/ energy spent
Ideal	Means that the equipment it refers to has 100 % efficiency
Top Dead Center	When a piston is at the highest point of a cycle inside the cylinder.
Bottom Dead Center	When a piston is at the lowest point of a cycle inside the cylinder.
Auxiliary fluid	The secondary fluid running through a heat exchanger to increase or decrease the temperature of the primary fluid.
Energy	Energy is the ability to do work. E.g. the water above a water fall has potential energy and a moving train has kinetic energy.
Work	Work is the displacement of something, done by a force in the direction of the force.
Power	Power is the rate work is done at. If one Joule of work is done in one second, 1 watt is the power.
Low pressure	$\leq$ 16 Bar
High pressure	$\geq$ 16 Bar
Adiabatic	A process where no heat is exchanged with the surroundings.
Isovolumetric	A process with constant volume.
Isobaric	A process with constant pressure.
Isothermal	A process with constant temperature.

3.2. Symbols and units

Table 6: Symbols and units.

Symbol	Represents	Unit
E	Energy	J
W	Work	J
P	Power	W
Q	Heat transferred	J
$\dot{Q}$	Heat transfer rate	W
U	Internal energy	J
H	Enthalpy	J/kg
$\dot{m}$	Mass flow rate	kg/s
$\dot{V}$	Volumetric flow	m^3/s
p	Pressure	N/m^2
A	Square meters/Area	m^2
V	Cubic meters/Volume	m^3
m	Mass	kg
h	Distance	m
v	Speed	m/s
G	gravity	m/s^2
η	Efficiency	
ρ	Density	kg/m^3
Δ	Difference	

3.3. Formulas and equations

Table 7: Formulas and equations.

Category	Equation	Quantity calculated	Equation number
Heat Exchangers	$Q = \dot{m}_c C_{p,c}(T_{C,out} - T_{C,in})$	Energy balance for cold side of heat exchanger.	(1)
	$Q = \dot{m}_c \Delta H_C$	Energy balance for cold side of heat exchanger, expressed by change in enthalpy.	(2)
	$Q = \dot{m}_h C_{p,h}(T_{H,in} - T_{H,out})$	Energy balance for hot side of heat exchanger.	(3)
	$Q = \dot{m}_h \Delta H_H$	Energy balance for hot side of heat exchanger, expressed by change in enthalpy.	(4)
	$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\Delta T_1/\Delta T_2\right)}$	Logarithmic mean temperature difference.	(5)
	$A = Q/(U\ \Delta T_{lm})$	Required surface area of heat exchanger.	(6)
Pumps	$P_h = \dot{m} * g * h$	Hydraulic power	(7)
	$P_h = \Delta p * \dot{V}$	Hydraulic power	(8)
	$P_S = P_h/\eta$	Shaft power	(9)
Flow	$\dot{V} = \dot{m} * \rho$	Volumetric flow rate	(10)
	$\dot{m} = BSFC * P_{engine}$	Mass flow rate of fuel to engine.	(11)
General physics	$\Delta U = Q + W$	The first law of thermodynamics.	(A)
	$\eta = W/Q_{in}$	Efficiency for a combustion engine.	(B)
	$\Delta E_{System} = E_{in} - E_{out} + Q + W$	The law of conservation of energy	(C)

3.4. Tools and analysis

Microsoft excel:

All equations in chapter 5 are inserted to a spreadsheet in Excel. All variables of the equations are assigned to a cell where a value is inserted. That way, the results can easily be updated when more precise data is obtained. This also makes it faster to evaluate scenarios with e.g. different efficiencies and lifting heights.

Excel is used for supporting functions as e.g. the schedule, a table of tasks by category, a contacts list sorted after field of competence, and organizing of sources

Microsoft power point:

Microsoft power point is used to make illustrations, flow sheets and process charts.

3.5. Analysis – Process chart

Figure 2 on the next page, shows have the analysis will be approached. For each of the system there will be chosen an engine with approximately 7 kW power. From data sheets for the engines, fuel consumption and sometimes injection pressure and temperature can be found. Using the literature, experts and assumptions, the initial pressure and temperature is found. From this information, flow rate pressure increase and temperature increase can be estimated. The basis of the analysis, which are pumping and heating requirements, can be found. From this point it is searched for more specific information. The more is found and the more credible it is, the more precise the results get.

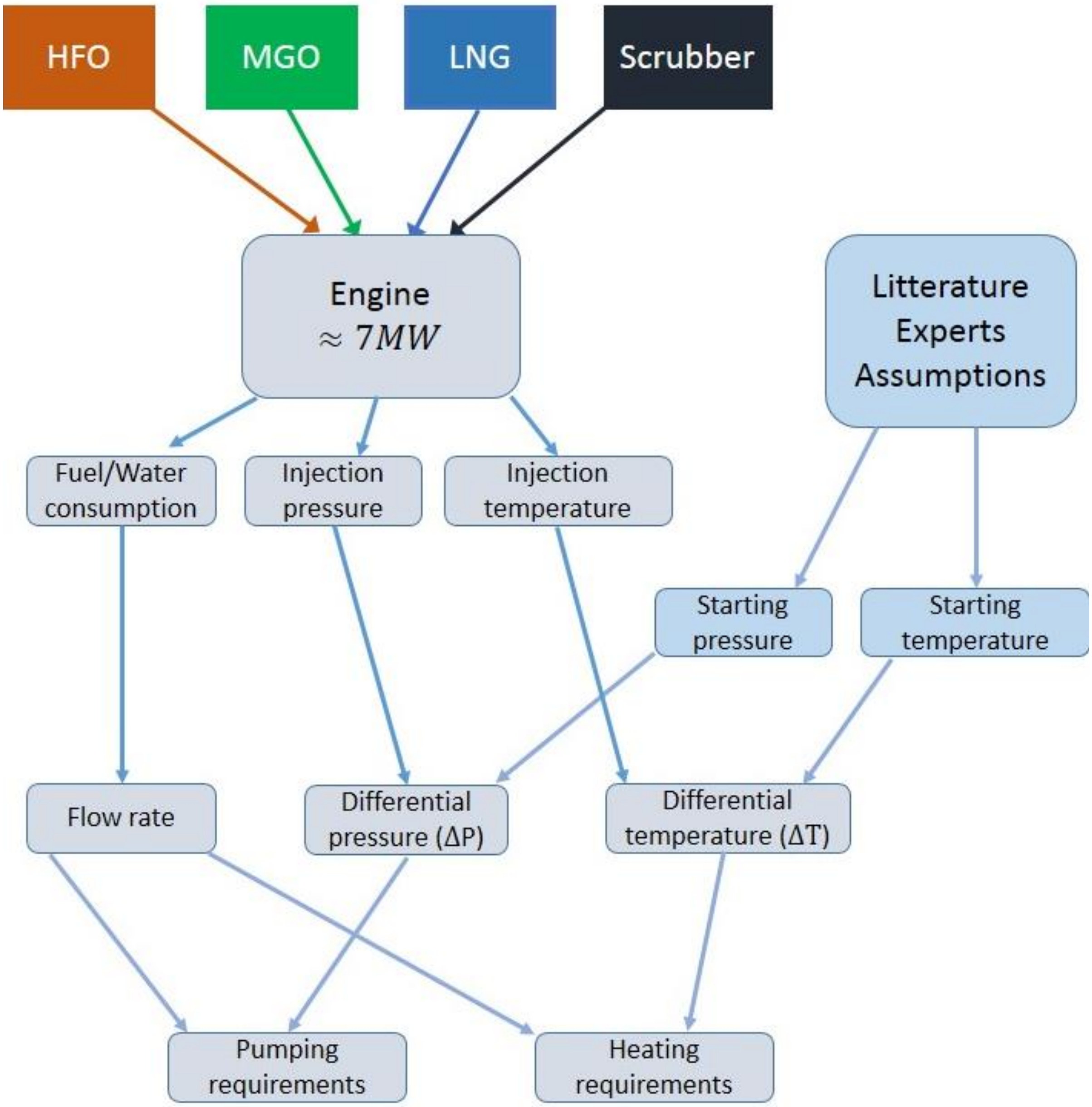


Figure 2: Process chart

4. Theory and principles

In this chapter the theory and principles that the analysis are base on, will be explained. Practical equations that are used directly in the analysis are indexed with number. Fundamental equations which the practical equations are derived from are indexed with a letter. Expressions that are used for explanatory reasons and are transformed in several steps may be temporary indexed with a roman letter for the purpose of keeping track of them.

4.1. Combustion Engines

In this thesis, the engines installed with these systems are two-strokes, running in both Otto and Diesel cycle. HFO, MGO, high pressure LNG are Diesel cycle, while low pressure LNG is Otto cycle.

4.1.1. Otto cycle engines

The Otto cycle is a thermodynamic description of a method of combusting petrol in an engine. Now a days there are several fuels that can be used in an Otto Cycle engine, but the first one was the petrol engine.

Figure 3 and Figure 4 illustrates the principle of a four-stroke, Otto Cycle, petrol engine. The Otto Cycle describes the compression and work stroke, and applies to both two- and four-stroke engines.

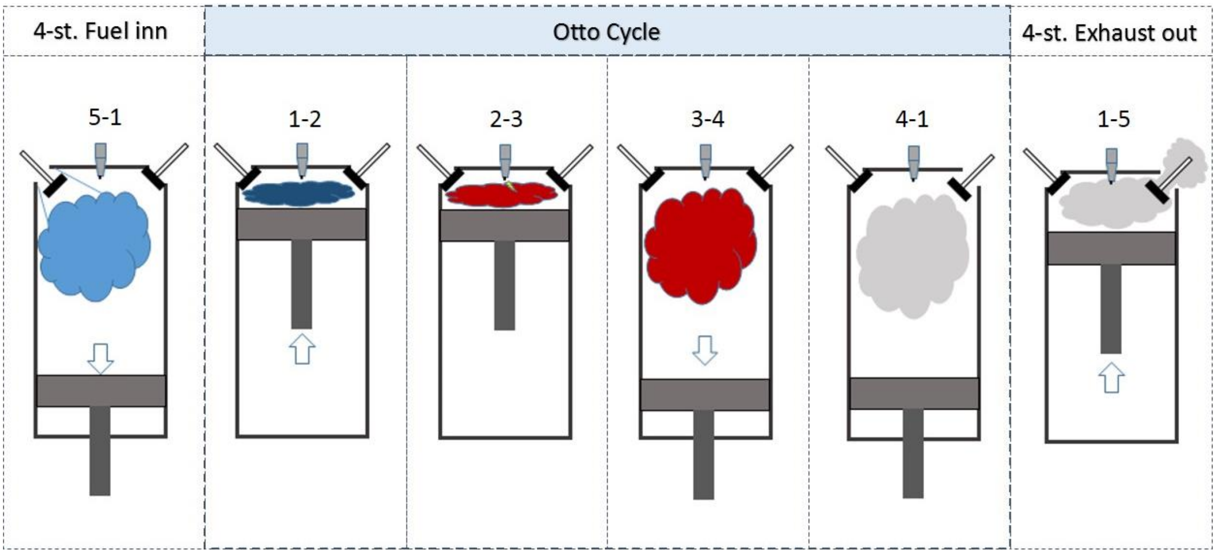


Figure 3: 4-stroke, Otto Cycle petrol engine.

5-1: A mixture of petrol gas and air is sucked into the cylinder when the piston moves down and increases the cylinder volume. An automated valve makes sure the mixture flows into the cylinder at the right time. This step is only present in a four-stroke engine.

1-2: The piston moves up and compresses the air and petrol mixture. This is called the compression stroke and is a adiabatic process, which means that no heat is added.

2-3: Right before the piston reaches its top point, called top dead center, a sparkplug ignites the mixture. Heat is raised by the burning fuel while the volume stays constant, which increases the pressure.

3-4: The increased pressure pushes the piston down with great force. This is called the work stroke, and is an adiabatic process as well.

4-1: The exit valve opens, temperature and pressure decreases rapidly, while the volume remains constant.

1-5: As the piston moves back up, exhaust is pushed out of the cylinder. This step is only present in a four-stroke engine.

Efficiency of an ideal Otto cycle engine:

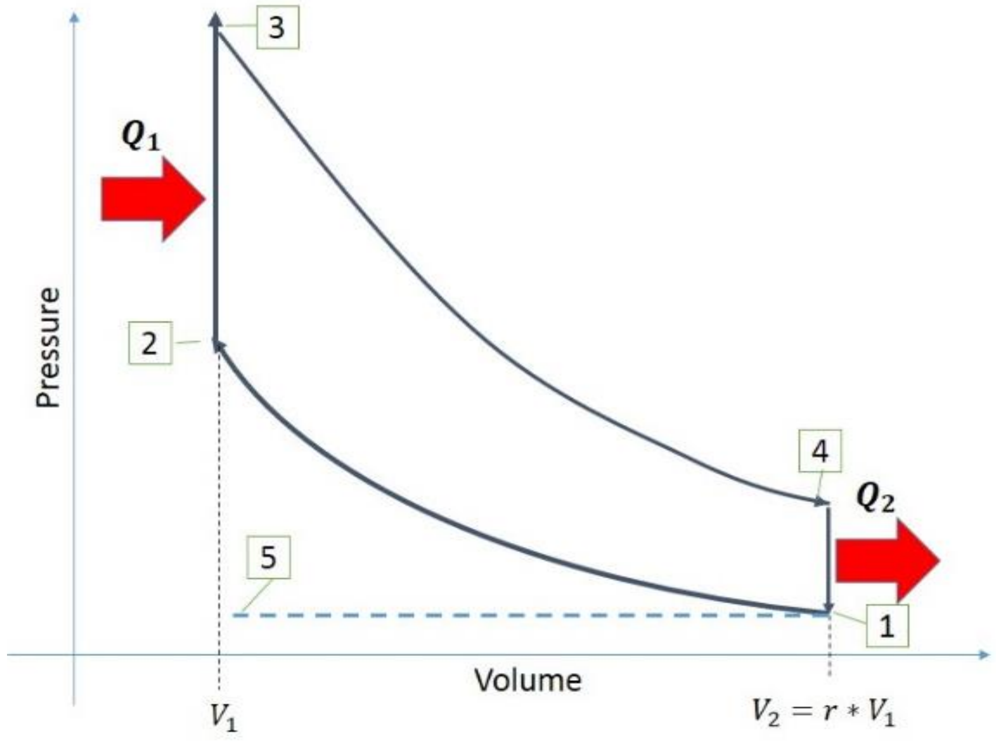


Figure 4: Ideal Otto Cycle

The first law of thermodynamics states that the change internal energy (ΔU) of a closed system equals the heat added (Q) minus the work (W) done by it.[8]

$$\Delta U = Q - W \qquad \text{(A)}$$

A cyclic process can be regarded as a closed system and be expressed by the first law of thermodynamics, where $\Delta U = 0$.[9]

$$\Delta U = Q - W \quad \rightarrow \quad W = Q = Q_1 - Q_2 \qquad \text{(A)}$$

Where Q_1 is the heat added through combustion of the fuel and Q_2 is the heat loss when the piston is at the bottom. W is the useful work done by the gas on the piston.

Efficiency of an engine is the relationship between the useful work done and the heat added to the system.

$$\eta = \frac{W}{Q_{In}} = \frac{Q_1 - Q_2}{Q_1} \qquad \text{(B)}$$

The process from 2-3 and 4-1 is isovolumetric, which means the volume is constant, hence W=0.

$$Q_1 = nC_V(T_3 - T_2), \qquad Q_2 = nC_V(T_4 - T_1)$$

Where *n* is number of moles and C_V is molar heat capacity at constant volume.

Equation B for efficiency can be rearranged to:

$$\eta = \frac{nC_V(T_3 - T_2) - nC_V(T_4 - T_1)}{nC_V(T_3 - T_2)} = \frac{(T_3 - T_2) - (T_4 - T_1)}{(T_3 - T_2)}$$

When a process is adiabatic, the relationship between temperature and volume stays constant.

$$TV^{\gamma-1} = Constant$$

Where $\gamma = {}^{C_P}/_{C_V}$ and C_P is the heat capacity at constant pressure.

As Figure 4 shows the volume at 2 and 3, *V*, are equal. The volume at 4 and 1 is also equal and can be written as $r * V$, where r is the compression ratio. Temperature and pressure at the adiabatic processes between 3-4 and 1-2 can be expressed as:

$$\boldsymbol{I}: \quad T_1(rV)^{\gamma-1} = T_2V^{\gamma-1}$$

$$\boldsymbol{II}: \quad T_3V^{\gamma-1} = T_4(rV)^{\gamma-1}$$

Both equations are divided by $V^{\gamma-1}$

$$\boldsymbol{I}: \quad T_1r^{\gamma-1} = T_2$$

$$\boldsymbol{II}: \quad T_3 = T_4r^{\gamma-1}$$

If the new expression for T_3 and T_2 is inserted to equation B, the equation for efficiency is:

$$\eta = \frac{(T_4r^{\gamma-1} - T_1r^{\gamma-1}) - (T_4 - T_1)}{(T_4r^{\gamma-1} - T_1r^{\gamma-1})}$$

$$\eta = \frac{T_4(r^{\gamma-1} - 1) - T_1(r^{\gamma-1} - 1)}{r^{\gamma-1}(T_4 - T_1)}$$

$$\eta = \frac{(r^{\gamma-1} - 1)(T_4 - T_1)}{r^{\gamma-1}(T_4 - T_1)}$$

$$\eta = \frac{(r^{\gamma-1} - 1)}{r^{\gamma-1}}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

This shows that the compression ratio, the ratio between the volume of the air at BDC and TDC, is the determining factor for an engine’s efficiency.

An increased compression ratio will give a higher pressure which leads to higher temperatures in the cylinder. In an Otto cycle the fuel is inside the cylinder before ignition. If the temperature gets to high the fuel might auto ignite before the sparkplug fires, what is referred to as knocking. This is why Otto cycle engines have a lower compression ratio and lower efficiency than the Diesel cycle engine, where the fuel is injected after the compression. Typical compression ratios for Otto-cycle engines are around 10:1.

4.1.2. Diesel cycle

In a diesel engine there is no spark plug. Instead diesel is injected into highly compressed air where it self-ignites due to the compression heat.

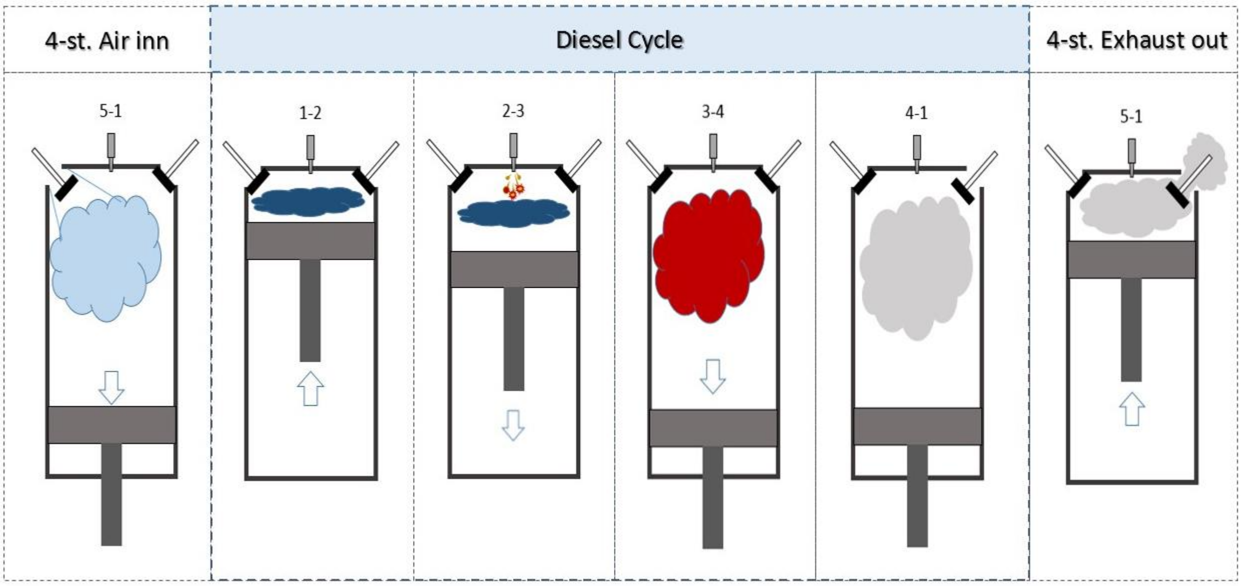


Figure 5: 4-stroke, Diesel Cycle petrol engine.

5-1: Pure air is drawn into the cylinder when the piston moves down.

1-2: The air is compressed adiabatically by the piston moving up.

2-3: Diesel is sprayed into the cylinder when the piston is at top dead center. Heat from the compressed air makes the diesel self-ignite immediately. The piston is pushed down increasing the volume at constant pressure, the process is isobaric. This the first part of the work stroke.

3-4: In the second part of the work stroke, the piston keeps moving down and the volume increases while the pressure decreases. This process is adiabatic.

4-1: The exit valve opens and the pressure decreases at constant volume, which is called a isovolumetric process.

Efficiency of an ideal Diesel cycle engine:

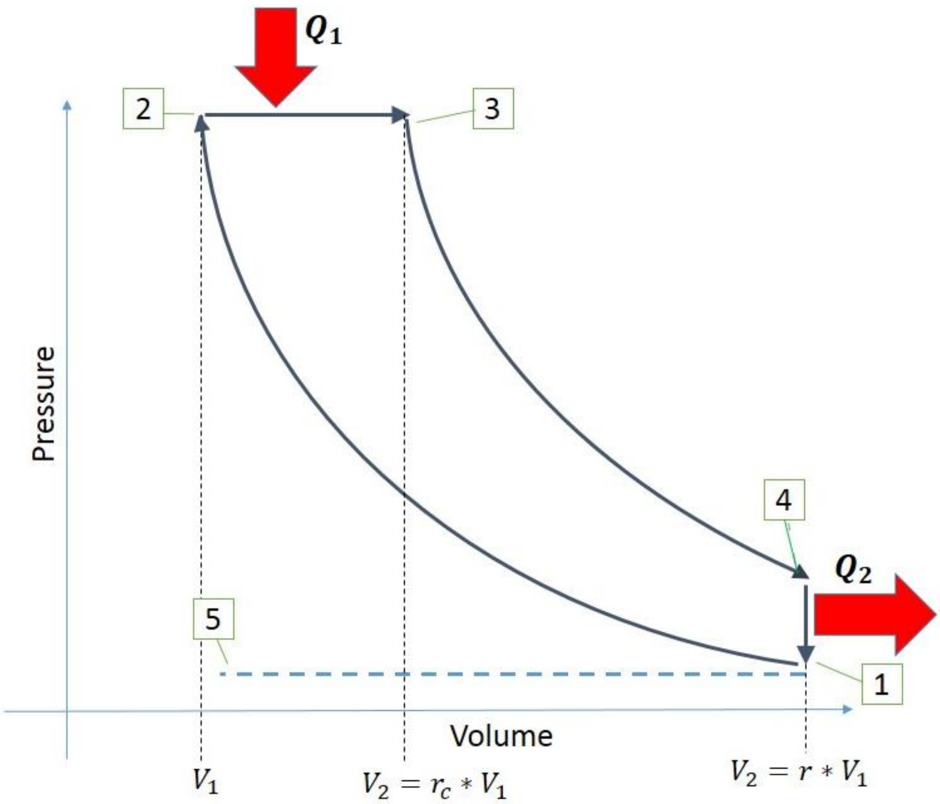


Figure 6: Ideal Diesel cycle

Figure 6 illustrates the ideal Diesel cycle in a pressure-volume diagram. The volume at point two, which is the TDC is called V.[10] Point three where the diesel injection is cut of, has a volume of V* r_c. r_C is called the cut off ratio. Point four and one has the same volume, which is the initial volume times the compression ratio, V*r.[10]

The efficiency of an Diesel cycle engine is also the useful work divided by the heat supplied, and can be expressed by equation B:

$$\eta = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

Heat inn to the system during the isobaric process from 2-3 is expressed as:

$$Q_1 = C_P(T_2 - T_3)$$

The heat loss at the isovolumetric process from 4-1 is expressed as:

$$Q_2 = C_v(T_4 - T_1)$$

The expressions for $Q_1$and Q_2 are inserted to equation B:

$$\eta = 1 - \frac{C_v(T_4 - T_1)}{C_P(T_3 - T_2)} = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$

For the adiabatic processes between 1-2 and 3-4 the relationships between temperature and volume is:

$$\boldsymbol{I}: \quad T_1(rV)^{\gamma-1} = T_2V^{\gamma-1}$$

$$\boldsymbol{II}: \quad T_3(r_cV)^{\gamma-1} = T_4(rV)^{\gamma-1}$$

$$\boldsymbol{I}: \quad \frac{T_2}{T_1} = \frac{(rV)^{\gamma-1}}{V^{\gamma-1}} = r^{\gamma-1}$$

$$\boldsymbol{II}: \quad \frac{T_3}{T_4} = \frac{(rV)^{\gamma-1}}{(r_cV)^{\gamma-1}} = \left(\frac{r}{r_c}\right)^{\gamma-1}$$

$$\boldsymbol{III}: \quad \frac{T_4}{T_1} = \frac{T_3 * {r_c}^{\gamma-1}}{r^{\gamma-1}} * \frac{r^{\gamma-1}}{T_2} = \frac{T_3 * {r_c}^{\gamma-1}}{T_2}$$

The relationship between temperature and pressure in the isobaric process from 2-3 is:

$$\frac{V}{T_2} = \frac{r_cV}{T_3} \quad \rightarrow \frac{T_3}{T_2} = r_c$$

$$\boldsymbol{III}: \quad \frac{T_4}{T_1} = r_c * {r_c}^{\gamma-1} = {r_c}^{\gamma}$$

Equation B can be rearranged and combined with I, II and III:

$$\eta = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)} = 1 - \frac{1}{\gamma}\frac{T_1}{T_2}\frac{\left({T_4}/{T_1} - 1\right)}{\left({T_3}/{T_2} - 1\right)}$$

$$= 1 - \frac{1}{\gamma}\frac{1}{r^{\gamma-1}}\frac{\left({T_4}/{T_1} - 1\right)}{\left({T_3}/{T_2} - 1\right)}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\frac{({r_c}^{\gamma} - 1)}{\gamma(r_c - 1)}$$

This shows that for the Diesel cycle engine a higher compression ratio means higher efficiency. For Diesel engines the cut of ratio is also quite important. A lower cut off ratio means higher efficiencies, which is why the injection timing in a diesel engine is crucial. Lowering the cut of ratio is also the reason why many manufacturers strive to pressurize the diesel before injection as much as possible.

The cut of ratio can never be exactly one which means that the theoretic efficiency for a diesel engine is lower than for a Otto cycle engine. However, as earlier mentioned, the Diesel is injected after the compression and there is now danger of knocking due to high temperatures from the pressure increase. Therefore Diesel cycle engines can have a much higher compression ratios and higher efficiencies than Otto cycle engines.

Efficiency of a real engine will be lower than the ideal efficiency, because there are friction between the cylinder and piston. Volume does not stay constant during the combustion and expulsion of hot air. This means none of the processes are adiabatic, isovolumetric or isobaric as described in the Otto and Diesel cycles.

4.1.3. 2-stroke and 4-stroke

As illustrated in Figure 3 and Figure 5, a four-stroke engine uses one piston stroke to fill the cylinder with fuel and one to push the exhaust out. In a two stroke engine, filling and emptying of the cylinder is done in one single stroke. Figure 7 illustrates the basic principle of a two-stroke engine.

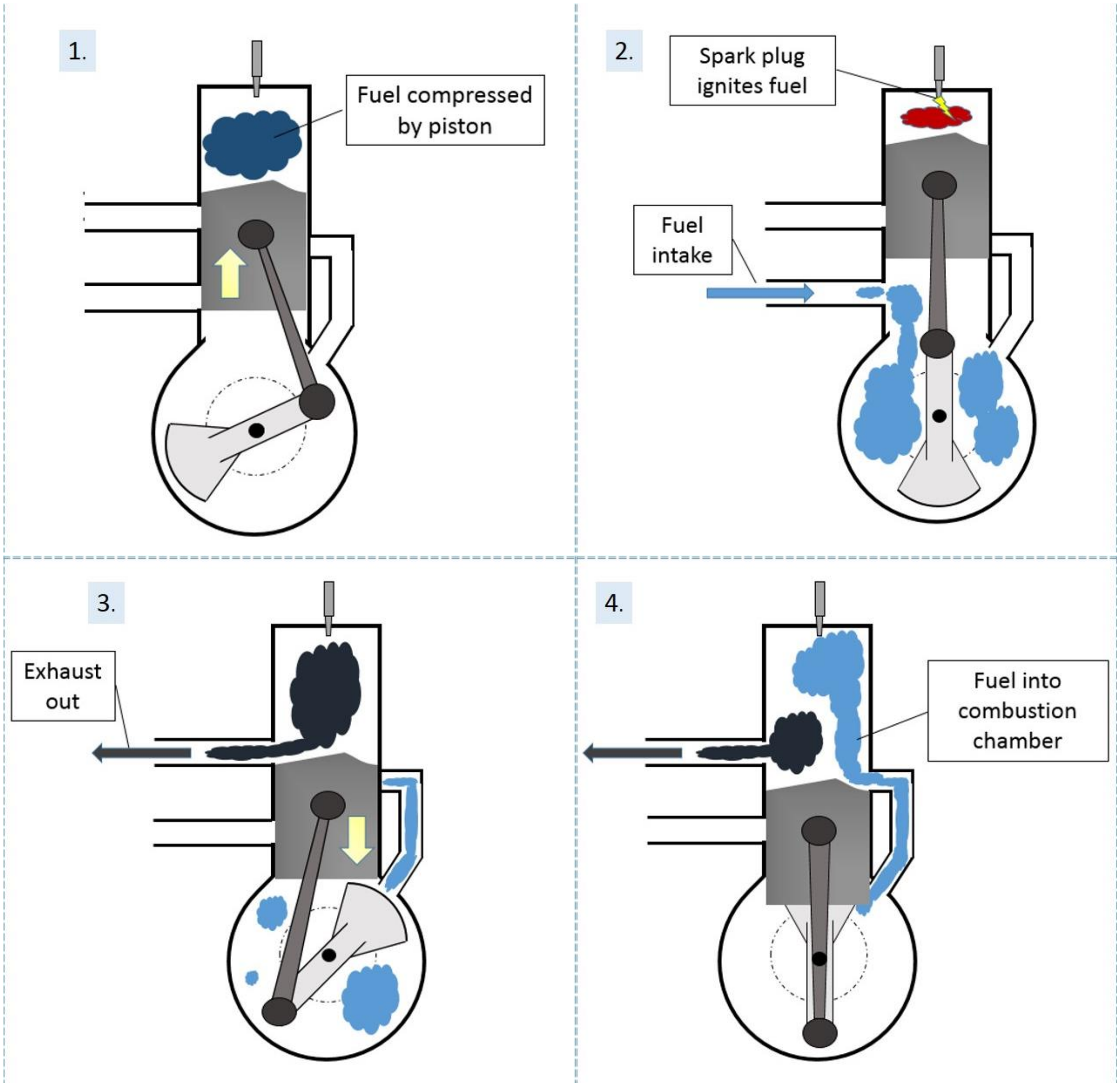


Figure 7: Two-stroke principle

2. When the piston is at top dead center, air (if it's a diesel engine) or a mix of fuel and air (in a petrol engine) flows into a chamber bellow the piston. Above the piston compressed fuel and air is ignited by a spark plug, or diesel is injected into compressed air and ignites.

3. The piston moves down and reveal the exhaust outlet. Exhaust starts flowing out of the cylinder.

4. When the piston is at bottom dead center, a transfer port between the cylinder and the chamber bellow the piston is revealed. Air and fuel fills the cylinder while the remainder of the exhaust is pushed out.

1. The fresh fuel and air gets compressed as the piston moves back up to top dead center. The cycles repeats from step one.

4.2. Heat exchangers

Heat exchangers are actually passive devices where fluids run through and exchanges heat due to differences in temperature. Many different shapes and configurations are available, but the basic principle that apply to all are that heat is transferred through a wall from a hot to a colder fluid.

The energy consuming part of a heat exchanger is to circulate auxiliary fluid through it. This is of course done by pumps. Calculations still has to be done for the heat exchanger in order to know how much liquid needs to be pumped. If the primary fluid has to be heated to high temperatures, the auxiliary fluid has to be applied energy from e.g. a boiler.

All heat exchangers in this thesis will be considered as ideal with counter flowing mediums. Counter flow means that the cold medium enters at the side where the hot medium exits. The two mediums flow through the heat exchanger in opposite directions. Figure 8 illustrates counter flow compared to same flow.[11]

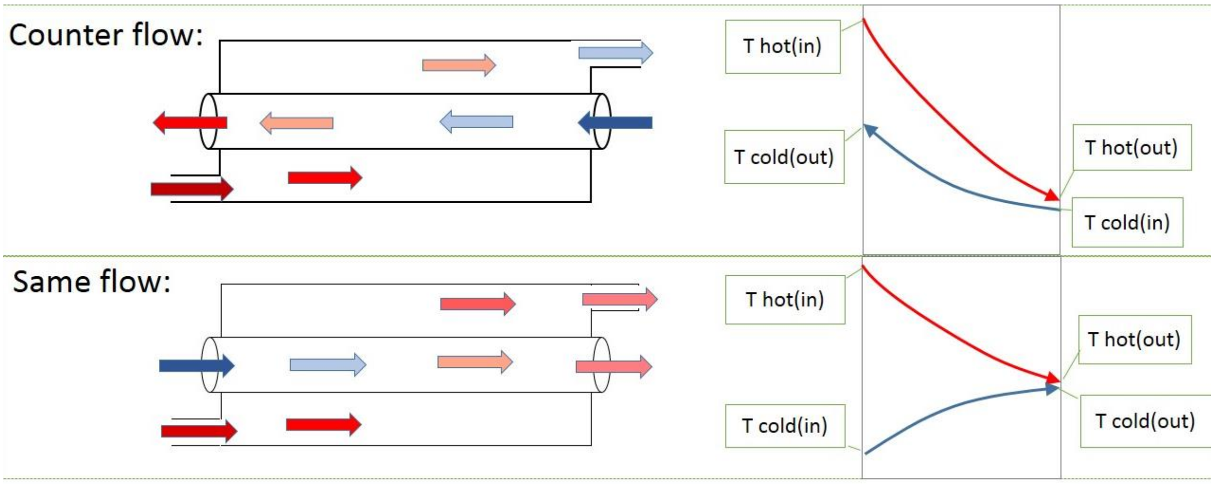


Figure 8:Counter flow vs. parallel flow in a heat exchanger

This flow pattern enables the most efficient heat exchanging since it allows the hot fluid to have a lower temperature at the outlet than the cold fluid at its outlet. With the two fluids flowing in the same direction, the hot fluid cannot be colder than the cold fluid at the outlet, since there has to be a difference in temperature for heat to be transferred.

When a heat exchanger is said to be ideal, it means that the hot fluid at its outlet can reach the same temperature as the cold fluid entering the heat exchanger. In addition, all the heat taken out of the hot fluid is received by the cold fluid. Said in other words, the heat exchanger has a 100 % efficiency.

Terms for heat exchangers equations:

- All heat exchangers will be regarded as ideal with counter flow.
- Specific heat capacity is regarded as constant through the heat exchanger.

Equations for ideal counter flow heat exchangers:

The law of conservation of energy states that the change in total energy (ΔE_{system}) during a period of time inside a system, is equal to the energy of the mass entering (E_{in}), minus the energy of the mass exiting (E_{out}), plus the heat exchanged (Q) with the surroundings, and the work (W) done by, or to, it.[8]

$$\boldsymbol{\Delta E_{System} = E_{in} - E_{out} + Q + W} \qquad \text{(C)}$$

When we apply this to a heat exchanger where changes in energy are mostly caused by changes of internal energy, energy (E) can be replaced by internal energy (U) .

$$\Delta U_{System} = U_{in} - U_{out} + Q + W \qquad \text{(C)}$$

Where ΔU_{System} is accumulated internal energy in the system during a period of time. $U_{in/out}$ is the internal energy of the fluids flowing in and out of the system.

Electrochemical work (W_e) and volume changing work can be neglected. That leaves shaft work (W_s) and flow work (W_f). Since there is no moving parts inside the heat exchanger, the only type of work left in the equation is W_f. The equation for flow work is:

$$W_f = (pV)_{in} - (pV)_{out}$$

This expression is inserted to equation C.

$$\Delta U_{Systems} = U_{in} - U_{out} + (pV)_{in} - (pV)_{out} + Q$$

Enthalpy is the sum of internal energy and pressure multiplied with volume.

$$H = U + p * V \quad \rightarrow \quad \Delta H = \Delta U + \Delta(pV)$$

$$H_{in} - H_{Out} = U_{in} + (pV)_{in} - U_{out} - (pV)_{out}$$

Internal energy and flow work is substituted with enthalpy in equation C.

$$\Delta U_{system} = H_{in} - H_{out} + Q$$

Mass flows in and out of the heat exchanger are equal and constant. There are no accumulation and $\Delta U_{System} = 0$.

$$Q = H_{out} - H_{in}$$

Specific enthalpy, enthalpy per kg mass, is introduced.

$$Q = m * (h_{out} - h_{in})$$

To translate equation C to expressed heat rate ($\dot{Q}$) in Watt, the mass is replaced by the mass flow rate ($\dot{m}_c$).

$$\dot{Q} = \dot{m} * (h_{out} - h_{in})$$

$$h = T * C_p$$

$$\dot{Q} = \dot{m} * \left(C_{p,out} * T_{out} - C_{p,in} * T_{in}\right)$$

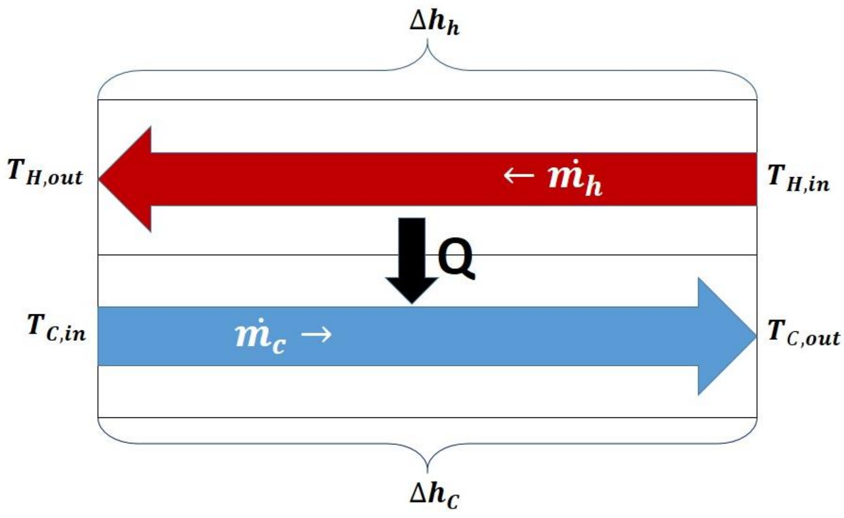


Figure 9: Energy transfer in a heat exchanger

Energy balance for the cold fluid in a heat exchanger:

If the cold side of the heat exchanger is regarded as the system, the hot as the surroundings and the specific heat capacity is assumed to be constant, Q for the cold side is:

$$\dot{Q} = \dot{m}_c C_{p,c}(T_{C,out} - T_{C,in}) \tag{1}$$

$Where:$
$\dot{Q}\ (J/s = W) = Heat\ transfer\ rate\ from\ hot\ to\ cold\ side$
$\dot{m}_c(kg/s) = Mass\ flow\ of\ cold\ fluid$
$C_{p,c}\ (J/kg\ K) = Specific\ heat\ capacity$
$T_{c,ut}(K) = Temperature\ of\ the\ cold\ fluid\ at\ the\ outlet$
$T_{c,inn}(K) = Temperature\ of\ the\ cold\ fluid\ at\ the\ inlet$

Equation 1 can be expressed by change in enthalpy. This is useful when we have evaporation at constant temperature, or change in enthalpy can be read from a diagram.

$$\dot{Q} = \dot{m}_c \Delta h_c \tag{2}$$

$\Delta h\ (J/kg) = Change\ in\ specific\ enthalpy\ from\ T_{C,out}\ to\ T_{C,inn}$

Energy balance for the hot fluid in a heat exchanger:

When the heat exchanger is regarded as ideal, no heat goes in and out of the heat exchanger's boundary. All the heat added to the cold fluid must come from the hot fluid and vice versa. If the specific heat capacity is constant, $\dot{Q}$ for the hot side is:

$$-\dot{Q} = \dot{m}_h C_{p,h}(T_{H,out} - T_{H,in})$$

$$\dot{Q} = \dot{m}_h C_{p,h} (T_{H,in} - T_{H,out}) \tag{3}$$

$Where$:
$\dot{m}_h (kg/s) = Mass\ flow\ of\ hot\ fluid$
$C_{p,h}\ (J/kg\ K) = Specific\ heat\ capacity$
$T_{h,ut}(K) = Temperature\ of\ the\ hot\ fluid\ at\ the\ outlet$
$T_{h,inn}(K) = Temperature\ of\ the\ cold\ fluid\ at\ the\ inlet$

Equation 3 can be expressed by change in enthalpy. This is useful when we have condensation at constant temperature, or change in enthalpy can be read from a diagram.

$$\dot{Q} = \dot{m}_h \Delta h_c \tag{4}$$

$\Delta h\ (J/kg) = Change\ in\ specific\ enthalpy\ from\ T_{H,in}\ to\ T_{H,out}$

Logarithmic mean temperature difference:

is a number that describes the overall temperature difference that drives the heat transfer from hot to cold fluid. This number is especially useful when heat exchangers are dimensioned.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left({\Delta T_1}/{\Delta T_2}\right)} \tag{5}$$

$Where$:

$\Delta T_{lm} = Log\ mean\ temperature\ difference$

$\Delta T_1 = T_{H,in} - T_{C,out}$
$\Delta T_2 = T_{H,out} - T_{C,in}$

Required surface area in the heat exchanger is expressed as:

$$A = \frac{\dot{Q}}{U\ \Delta T_{lm}} \tag{6}$$

$Where$:

$A = Surface\ area\ (m^2)$

$U = Heat\ transfer\ coefficient\ (W/m^2\ K)$

4.3. Pumps

As earlier mentioned, the most energy consuming part of these systems is to pump fuel and water. Energy consumption from a pump is determined by how much the pressure needs to be increased from the suction side to the discharge side, and the flow rate of fluid to be pressurized. Pressure at the suction side is determined by the height from the fluids source to the pump, and the pressure losses in the pipes. Pressure at the discharge side equals required pressure of the liquid at the target plus the pressure required to lift the liquid there and the pressure losses between the two points.

Most of the pumps are centrifugal pumps. Centrifugal pumps are counted as reliable and durable. They can handle a wide range of flow rates and pressures. The pumps characteristics are mostly determined by their impeller.

Terms for pumpequations:

- The pumping process is regarded as adiabatic, meaning no heat is transferred in or out of the fluids.
- Fluids being pumped are regarded as incompressible.

Equations for pump power

When fluids are regarded as incompressible the reversible (ideal) shaft work (W_s) is:[8]

$$W_s = \int_{p_1}^{p_2} V\,dp = (p_2 - p_1)V$$

The required power from a pump with 100% efficiency is from her referred to as hydraulic power (P_h). Power is equal to work per second. Hydraulic pump power is:

$$P_h = \frac{W_s}{s} = \frac{(p_2 - p_1)\,V}{s}$$

$$\boldsymbol{P_h = (p_2 - p_1)\,\dot{V}} \tag{7}$$

$Where:$

$p_2 = \ Pressure\ at\ the\ discharge\ side\ of\ the\ pump\ \left({}^{N}/_{m^2}\right)$

$p_1 = Pressure\ at\ the\ suction\ side\ of\ the\ pump\ ({}^{N}/_{m^2})$

$\dot{V} = Volumetric\ flow\ rate\ ({}^{m^3}/_{s})$

Pressure can be expressed by the weight of a column of fluid. This is what is called pump head.

$$p = \rho * g * h$$

$Where$:
$\rho = the\ density\ of\ the\ fluid\ \left({}^{kg}/_{m^3}\right)$
$h = the\ height\ of\ the\ column\ (m)$
$g = gravitational\ acceleration\ ({}^{m}/_{s^2})$

Since volume flow times density is equal to mass flow, hydraulic pump power expressed as:

$$P_h = \Delta h * \rho * g * \dot{V}$$

$$\boldsymbol{P_h = \Delta h * g * \dot{m}} \tag{8}$$

$Where$:
$\Delta h = Height\ difference\ between\ the\ target\ and\ the\ source\ of\ the\ fluid\ being\ pumped\ (m)$
$\dot{m} = Mass\ flow\ rate\ ({}^{kg}/_{s})$

In any pump there will be some loss in effect due to friction, leakage and other factors. The electric power is the effect that has to be put in to the pump in order to produce the required hydraulic effect. The efficiency (η) of the pump is useful power to do work on the fluid (P_h), divided by the work put in by electric power (P_e).

$$\eta = {}^{P_h}/_{P_e} \tag{9}$$

5. Current technologies

In this chapter the functionality of these different options for ECA compliance is explained. The purpose is to give readers a basis for understanding and evaluating the methods used in the analysis.

The main focus is on characteristics of the fuel used, the process flow and the purpose of different components. [These are general explanations, so details like piping arrangements, exact placement of equipment, valves, and number of pumps in parallel is not included.]

5.1. LNG

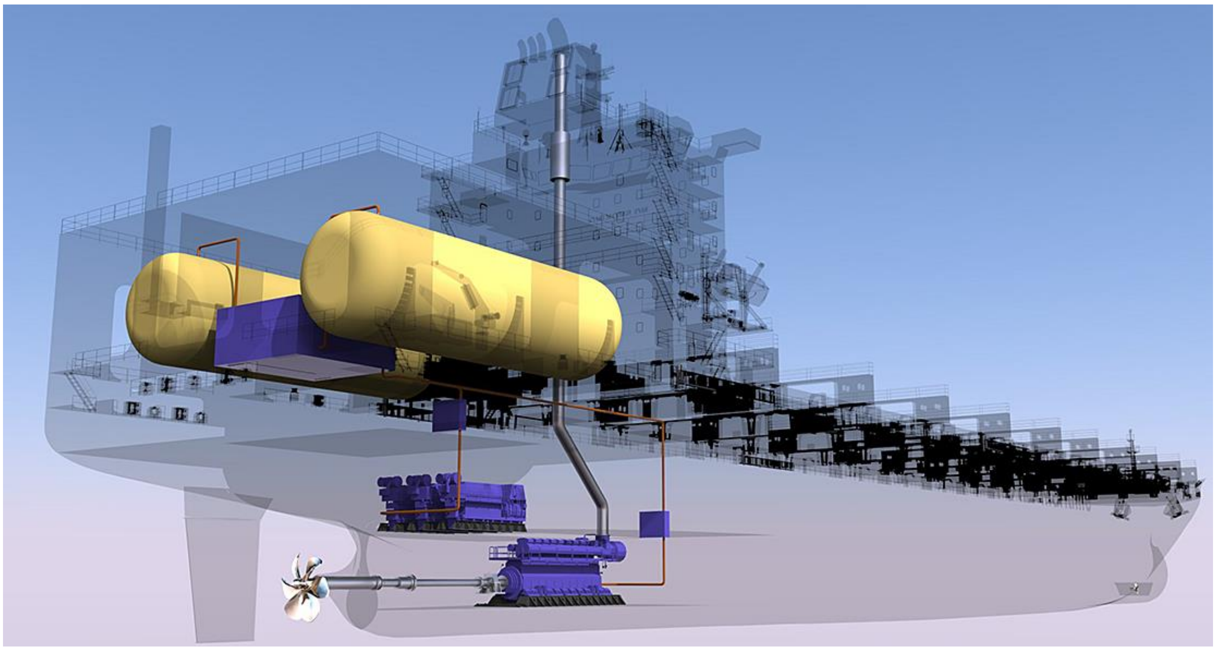


Figure 10: LNG Propulsion system.[12]

5.1.1. What is LNG

LNG is natural gas, primarily methane, cooled down to a liquid state at approximately -163℃. The volume of natural gas decreases about 600 times when it is liquefied, illustrated in Figure 11, giving it an energy density high enough to be a practical energy carrier on a vessel. Before it is injected to the combustion chamber of the engine, the LNG is heated up and vaporized back to a gaseous stage. LNG is a term more describing to how the fuel is stored, than a fuel itself. Table 8 lists some key properties and the composition of natural gas and LNG.

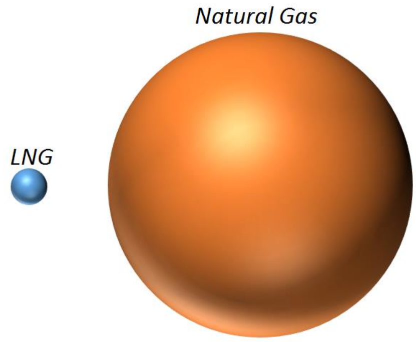


Figure 11: Volume of LNG and Natural Gas

Table 8: Key properties of NG and LNG.[6]

NATURAL GAS PHYSICAL PROPERTIES	VALUE	UNIT
CARBON CONTENT	≈ 75	$\%\ Mass$
C/H RATIO	≈ 25	-
DENSITY GAS (AT $\mathbf{273\ °K, 1\ atm}$)	$0{,}7 - 0{,}9$	kg/m^3
DENSITY LIQUID (AT $\mathbf{111{,}6\ °K, 1\ atm}$)	$410 - 500$	kg/m^3
CETANE NUMBER	≤ 135	
AUTO-IGNITION TEMPERATURE	810	$°K$
LOWER HEATING VALUE	49,2	MJ/kg
ENERGY DENSITY	$20 - 25$	$MJ/Liter\ LNG$
IGNITION LIMITS	$3 - 15$	$\%\ volume\ in\ air$
BOILING POINT ($\mathbf{at\ 1\ atm}$)	111,6	$°K$
NATURAL GAS COMPOSITION		
METHANE	70 – 90	%
PROPANE	0 – 20	%
CARBON DIOXIDE	0 – 8	%
OXYGEN	0 – 0.2	%
NITROGEN	0 – 5	%
HYDROGEN SULPHIDE	0 – 5	%
RARE GASES	Traces	%

5.1.2. Dual-fuel engines

The LNG systems reviewed in this thesis deliver natural gas to a dual fuel two-stroke engine, operating on either high- or low pressure. These are chosen because they are amongst the newest on the market, and there has recently been debates in the media between different parties claiming that either high or low pressure LNG is future for two-stroke gas engines.[13]

A dual-fuel engine works by combusting natural gas and a small portion of diesel called pilot fuel. [13] In the low pressure engines gas is injected to the engine at 5-7 bars, compressed by the piston, before the pilot diesel is injected and starts the combustion. In the high pressure engine, air is compresses, before the pilot diesel is injected and self-ignites. Gas is injected at up to 300 bar and burn instantaneously.

5.1.3. The fuel supply system

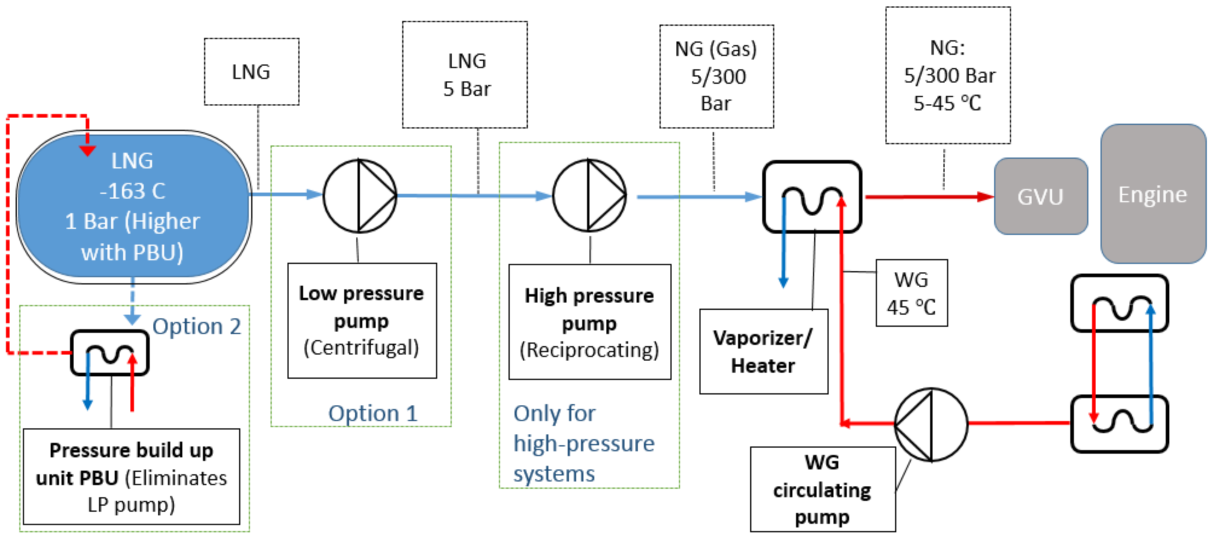


Figure 12: LNG, Flow chart

1. **LNG tank:** The LNG is stored in a well-insulated tank. Normally a double walled vacuum insulated tank. This keep the unwanted evaporation of LNG to a minimal amount.

2. **Low pressure pump:** A cryogenic[d] low pressure pump transports LNG from the tank to the fuel vaporizer, or to the high pressure pump if the engine is a high pressure type.

[d] Cyogenic refers to temperatures below - 160 C.

In a low pressure system this pump have to pressurize the fuel to engine specifications, which are between 5 and 10 bar. The pump placement varies from each manufacturer and each vessel. Some prefer placing the pump inside the LNG tank and some place it outside. The placement has little impact on the mechanical work required by the pump. These are centrifugal pumps.

3. **Pressure Buildup Unit:** Another way of transporting LNG out of the storage tank is to use a PBU instead of a pump. LNG is drained from the bottom of the tank, vaporized in the PBU and brought back to the top of the tank as gas. [14] The pressure inside the tank increases enough to push LNG to the fuel vaporizer without a pump. An advantage of using a PBU is the elimination of the pump and a simpler setup. However, the use of a PBU lead increases the design pressure, hence price of the tank.

4. **High pressure pump:** A high pressure pump is required when the engine type is high pressure. This pump increases the pressure of the LNG to between 250-300 bars.

   The pump type used for this application is cryogenic reciprocating pumps. A reciprocating pump works by pushing a piston in and out from the fluid. When the piston moves away from the chamber, the one way valve at the inlet opens and fluid is sucked into the chamber. The piston moves towards the chamber and increases the pressure in the fluid which is let out of the valve at the outlet when the pressure is high enough. Reciprocating pumps are suitable for high pressure applications at lower flow rates.

5. **Vaporizer:** LNG, still close to cryogenic temperatures, enters a heat exchanger where it is heated up by a water-glycol mixture until it is completely vaporized and have the temperature required for the engine type.[15] The water glycol mixture are heated by a secondary loop of either seawater or water heated by the engine.

6. **Gas Valve Unit:** This device are named differently by each manufacturer, but they serve the same purpose. One is to precisely adjust gas pressure to the engines current load. The other is to shut down gas supply to the engine in case of an emergency.[14]

7. **Pilot fuel:** A small portion of diesel is injected at pressures up to 2000 bar.[16] This diesel self-ignites when entering the cylinder and starts the combustion of the natural gas.

5.2. Wet scrubber

A wet SOx scrubber is a device that cleans SOx from the exhaust by showering it with alkali water.[17] As mentioned in the introduction chapter, there are three types of wet scrubbers; Open loop which uses seawater for scrubbing, closed loop that uses fresh water added with a chemical, and hybrid that alternates between open and closed loop mode.

There are the main stages/components in all types of wet scrubbers: [7]

- A scrubber unit where exhaust gas flow through and gets intimately mixed with alkali water. The water reacts and forms new substances with most of the SOx. Dirty wash water[e] is drained at the bottom of the scrubber unit, while clean exhaust is discharged through the top. To minimize water vapor in the exhaust, the outlet of the unit has a demister. Scrubber units are dimensioned based on MW installed power of the engines connected to it. Both size and layout of the scrubber unit are the same whether the system is closed loop, open loop or hybrid.
- A treatment plant for cleaning the wash water. The wash water coming from the scrubber unit is too contaminated to be discharged directly overboard. Wash water is separated from sludge and cleansed until it is pure enough to be discharged.
- A facility for handling and storing sludge. Sludge from a scrubber cannot be discharged at sea, but must be stored and disposed properly at shore.
- If the scrubber is of an integrated type, which means that several combustion units are connected to is, an exhaust fan to reduce the backpressure is included.
- A emission measuring system to monitor if SOx emission levels are equal to a ECA compliant fuel.

[e] Wash water is a term for water that has been used in a scrubber.

5.2.1. Open loop

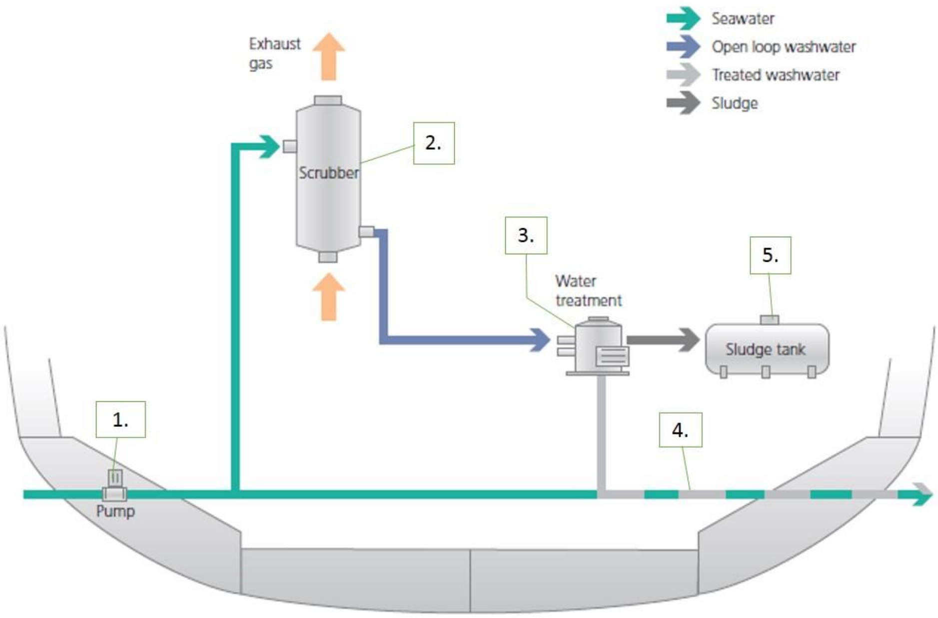


Figure 13: Open loop scrubber.[7]

1. Water is pumped directly from the sea crest to the scrubber unit. The working pressure of the pump depends on the required pressure for the scrubbing water and the scrubber unit's height above the seabed. An open loop scrubber has a scrubbing water flow rate of approximately $45\ m^3/MWh$. [7] The pump has increase pressure high enough to lift water from the seabed to the scrubber unit, and to spray water at the required velocity.

2. Sea water and exhaust is mixed inside the scrubber unit. Seawater is sprayed onto packed beds which provide increased surface area and reaction time, making the process more effective. Some scrubbers have three mixing stages and some have two. SOx is dissolved and ionized to bisulphite and sulphite, which is further oxidized to sulphate.[7]

3. Solid matter and liquid carbonate is separated from the water in the water treatment unit. Wash water is pumped, or drained by gravity, through a cyclonic separator.[7] Baffles inside the separator makes the wash water rotate, which separates heavier particles from water. Heavy particles (sludge) are drained to the sludge tank.

4. Wash water from the water treatment unit is mixed with seawater to reduce its acidity to the required level before being discharged to the ocean. The reason for mixing wash water with the same seawater it is being discharged into, is that many ports have pH limits for water leaving the vessel.

5. Sludge from the water treatment cannot be discharged into the ocean, hence has to be stored in the sludge tank until proper disposal at shore is possible.

5.2.2. Closed loop

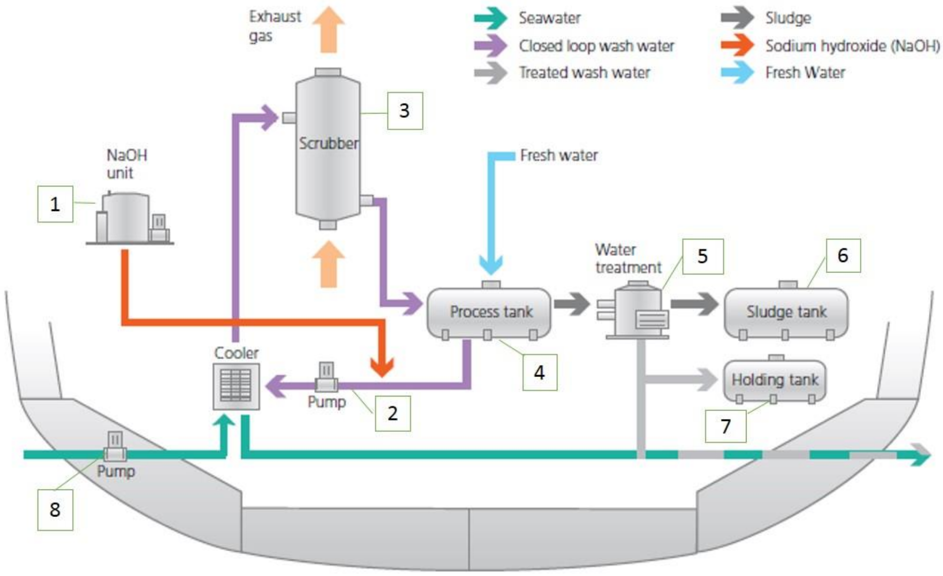


Figure 14: Closed loop scrubber.[7]

1. In a closed loop scrubber fresh water is pH-regulated with an alkali chemical, often Sodium Hydroxide (NaOH), before the being used as a scrubbing medium. Wash water get neutralized in the scrubbing process so NaHO has to be added constantly to the water loop. The NaHO-unit has to functions, storing and dosing of the NaHO.[18]

2. Scrubbing water is pumped to the scrubber unit by the circulation pump (indicated by number 2). Fresh water dosed with NaOH is a more effective scrubbing water than seawater which allows for a lower flow rate of scrubbing water. The scrubbing water flow rate in a closed loop scrubber is approximately 20 m^3/MWh.[7]

3. The scrubber unit is identical to the scrubber unit used in an open loop scrubber. SOx from the exhaust and scrubbing water reacts and forms sodium bisulphite, sulphite and sulphate. [7]

4. Wash water is drained from the scrubber unit into the process tank/reservoir.[17] To avoid an accumulation of reaction byproducts in the water loop, a small portion of the wash water is extracted from the tank while the same amount of fresh water is added. Water is then drained from the process tank back to the scrubbing water loop.

5. Water taken out from the loop is separated from heavy particles in the wash water treatment unit. This unit is a mechanical centrifugal separator. Sludge is sent to the sludge tank, whilst cleansed water is discharged to the sea or stored in the holding tank.

6. Sludge is stored in the sludge tank until proper disposal at shore is possible.

7. If a vessel is in areas with a zero discharge policy, cleaned wash water can be stored in the holding tank. The size of the holding tank is determined by how long zero discharge mode is required.

8. Water will heat up after being in contact with the exhaust in the scrubber. To minimize water vapor in the exhaust and reduce the freshwater consumption, scrubbing water is cooled in a heat exchanger before going back into the scrubber unit.[17] Seawater is pumped through the other side of the heat exchanger and serve as the cooling medium, before being discharged to the ocean.

5.2.3. Hybrid

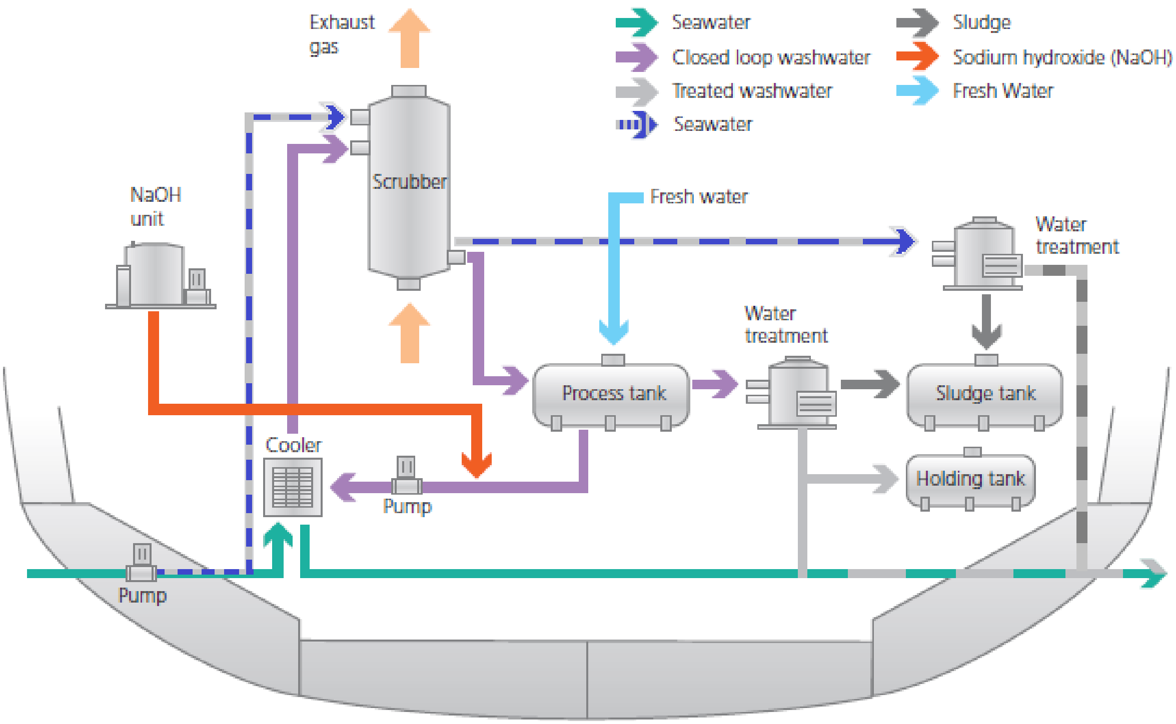


Figure 15: Hybrid scrubber. ECGs guide. Lloyds. P. 19

The hybrid scrubber has the option to switch between open- and closed loop mode. It has the same components as a closed loop scrubber, with the exception of some piping and an extra water treatment unit.[7]

Open loop mode: When running in open loop mode the pump at the seabed delivers seawater to the scrubber unit, and seawater for diluting the wash water before discharge.[7]

The amount of wash water is much higher in open loop mode. Wash water is cleaned in a water treatment unit designed for high flow rates. Both modes store sludge in the same sludge tank. Wash water is diluted with seawater before discharge.

Closed loop mode: When switching over to closed loop the stream of seawater to the scrubber unit is shut off while the closed loop circulation pump starts pumping water from the process tank. The pump at the seabed is now supplying cold seawater to the heat exchanger where closed loop scrubbing water is cooled down.[7]

Wash water from the scrubber unit is lead to a set of pipes going to the process tank. The stages from the process tank are the same as in a closed loop scrubber.

5.3. HFO

HFO are made of a large proportion of the residue from the distillation of crude oil. It is a byproduct created in the process of separating other lighter fuels, which makes it a very cheap fuel. This is the main reason for its popularity as a marine fuel.

HFO are not an official fuel grade, but a common nominator for all the residual fuels. Characteristics of can be read in appendix II. All the fuel grades that start with “RM” are HFOs. The number following “RM” are the fuel grade’s maximum viscosity. Usually higher viscosity means lower price.

As Figure 16 bellow shows, there is a lot of heating in all stages of a HFO supply system. Because of its high pour point[f] and viscosity, HFO has to be heated before it is possible to pump it.

[f] The pour point of a liquid is the temperature where a liquid loses loses its flow characteristict and can not be pumped.

Mode of operation:

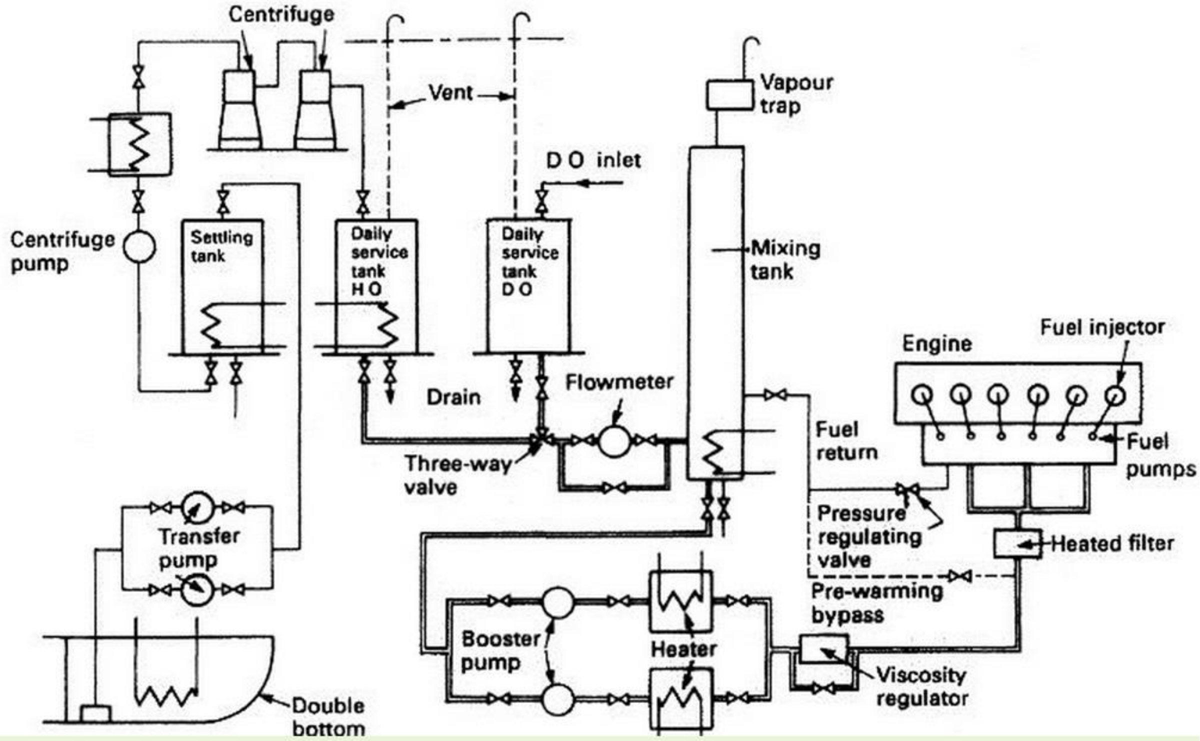


Figure 16: Fuel oil system for a diesel engine.[19]

1. HFO is stored in a fuel tank normally located in the bottom of the vessel. The fuel is heated to and kept at a temperature approximately 10 °C above the pour point.[20]
2. The fuel is pumped to the settling tank and heated to between 75-90 °C.[19] Separation of solids and water from HFO starts here by fluctuation. Water and sludge are drained from the bottom and stored in a sludge tank.
3. The fuel is pumped and further heated before entering the centrifuge for further purification. An electrical engine rotates the centrifuge which makes HFO and other components separate due to difference in centrifugal force.
4. After the centrifuge, HFO is filled into the daily service tank. Here the fuel is kept at 75-90 °C. This tank serves as a backup in case of shut down of fuel supply, and provide additional settling of the fuel.[21]
5. The booster pump, fuel heater and viscosity regulator makes up the booster unit. In the viscosity regulator, measurements of the fuels viscosity are compared to the required viscosity for the engine.[20] If the viscosity is to low, the heater is regulated to provide more heating and vice versa.
6. Remaining particles are removed in the fuel filter.
7. A high pressure pump driven by the camshaft increases the pressure of the fuel to required levels. Inlet pressure is engine specific, but ranges between 1000 and 2000 bar, according to engine approval engineers at DNV GL.[16]

5.4. MGO

Low sulphur fuel is defined as a fuel with a sulphur content low enough to compile to the sulphur limits in effect. Maximum sulphur content allowed is therefore varying in different areas and changes over time when new limits come into effect.[22] Marine gas oil (MGO) and marine diesel oil (MDO) can have a sulphur content below 0.1 % m/m, and can be used in engines designed for HFO. MGO, MDO and HFO are not official fuel grades, but can be translated in to the fuel grades DMA, DMB and RMx respectively. Details about these fuel grades can be read in appendix II.

Since this thesis are reviewing methods to compile to the sulphur limits taking effect in 2015 in ECAs, MGO and MDO with a sulphur content of maximum 0,1 % m/m will be used in this thesis. And will be referred to simply as low sulphur diesel.

While HFO is a residue from the distillation of crude oil, low sulphur diesel is made from the fractions with a lower boiling point.

Low sulphur fuels has a higher net specific energy than HFO, which results in a reduction in specific fuel consumption of about 5 %.[4]

Obstacles related to a HFO to MGO switch.

Even though these fuels can used in the an engine designed for HFO, there are some problems related to switching between high- and low-sulphur fuels. Viscosity of low sulphur fuels are significantly lower than HFO. Lower viscosity than the fuel delivery system is designed for can cause increased leakage of fuel, especially in pumps, and higher flow through injection valves, which again lead to reduced and less optimized fuel supply to the engines.

Lower viscosity reduces the fuels lubricity. Pumps that relay on lubricity from the medium being pumped, and other components that relay on lubrication from the fuel oil, will have more wear and tear when using low sulphur fuels.

To solve this there can be installed a separate set of pumps and nozzles, designed to handle lower viscosities. Since low sulphur fuels normally will be used only when a ship is inside an ECA, this solution can be unnecessary complex and costly. Another option is to install a cooler, that lower the fuels temperature which increases its viscosity.

Lower sulphur levels means less acidic sulphur products are created in the combustion process. Cylinder lubrication oil contains alkaline additives to neutralize the sulphuric acids created when HFO is combusted. Whit less acid, the lubrication oil should be switch to one containing less alkaline additives.

Mode of operation

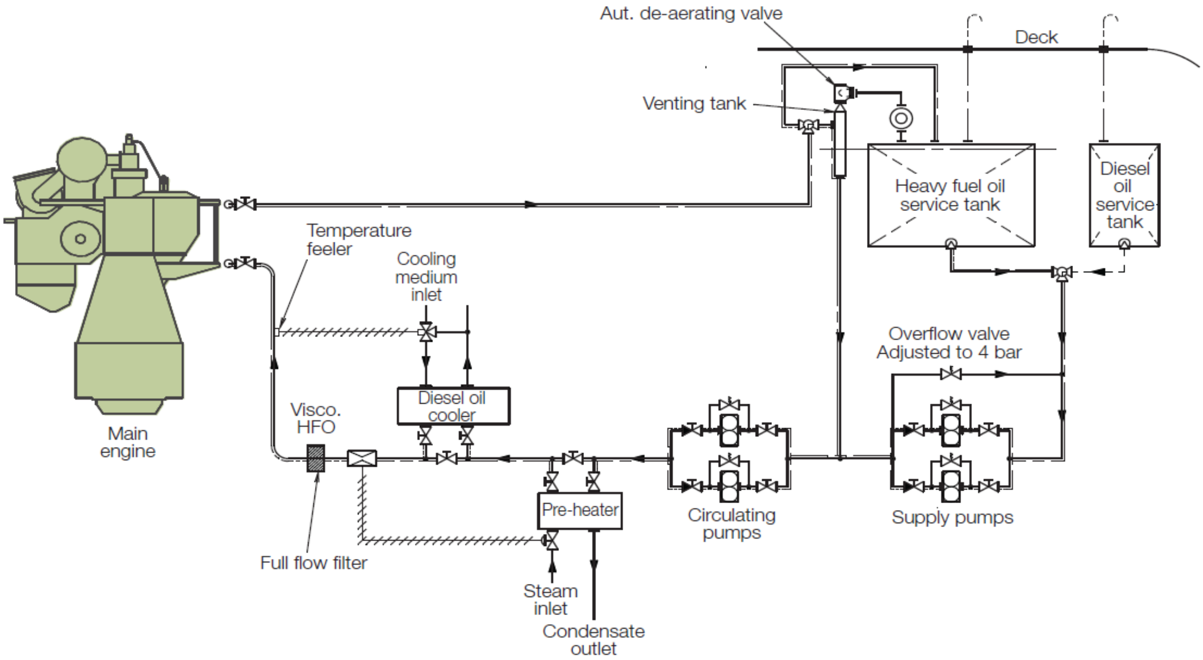


Figure 17: Low sulphur fuel supply with a diesel cooler.[23]

Figure 17 show a fuel supply system set up to deliver both HFO and low sulphur fuels to the engine. This is MAN Diesel and Turbo's recommended placement of the diesel cooler. They note that the cooler can be placed differently if desired. The illustration starts from the service tanks.

1. Low sulphur fuel is stored in a separate tank. Because the fuel have a lower pour point than HFO the storage tanks do not require heating like the HFO tanks. Tank size are determined by the expected sailing time in ECA.

2. HFO supply is gradually decreased while the flow of low sulphur fuel increases. In this scenario a cooler is used, and both fuels uses the same pumps and flow through the same pipes. A very specific change over procedures has to be made in cooperation whit the engine manufacturer, as equipment can easily be damaged in the process. Rapid variations in the fuels temperature will cause faster thermal expansions and contractions. A rule of thumb is that the maximum temperature gradient should not exceed 2 degrees per minute. The relation between fuel temperature and the ratio of HFO and low sulphur diesel, has to be closely monitored to make sure the viscosity does not increase or decrease more than allowed.[22] Its recommended that the engine load is decreased to 24-40 %.[23]

3. Pre heaters are shut off, or set to lower heat, while a diesel oil cooler is turned on. Most marine equipment requires a minimum viscosity of 2 cSt. The cooling duty required will depend on the fuel bunkered as viscosity and other properties will vary between each bunkering. A viscometer will regulate the flow of cooling liquid.[22]

4. Similar to HFO, a high pressure pump driven by the camshaft increases the pressure of the fuel to required levels. Inlet pressure ranges between 1000 and 2000 bar, according to engine approval engineers at DNV GL.[16]

6. Analyses of energy consumption

In all these systems, what consumes most energy is to heat the fuel, transport, and pressurize it. There are a lot of other energy consuming parts such as exhaust fans, air compressors, pumps for circulating lubricating oil and engine cooling. This thesis is about comparing the systems, so these parts mentioned above will not be included since they are assumed to be of a similar magnitude.

Assumptions for pumps:

Since this analysis does not include pressure losses in pipes and most equipment, pumps that do work at different stages on the same flow of fluid, will be regarded as one unit. This unit must increase the pressure of the fluid from the pressure at the suction side of the first pump, to the discharge pressure of the last pump.

Pump efficiency of 0,75 will be used for all pumps. Efficiency is dependent of flow rate and pressure and are different for each pump, so this must be taken into consideration when evaluating the results. This efficiency is an assumption.

Assumptions for heat exchangers:

Where there is heating at several stages, the heaters are regarded as one unit, heating the fuel from the initial temperature to the required supply temperature. This can be done because the temperature increase due to work done in pumps and friction in equipment, and temperature decrease due to heat losses in the pipes and tanks, are neglected.

In this thesis, there are three heating/cooling sources. Seawater, cooling water after the engine, and steam heated by the exhaust from the engine. These may be the auxiliary fluid in a fuel heat exchanger, or run through a secondary heat exchanger and heat the auxiliary fluid for the fuel heat exchanger.

- The seawater is assumed to have a temperature of 20 ℃.
- Cooling water after the engine is assumed to be 85 ℃. This is the cooling water temperature after a two-stroke engine of similar size used in one of DNV GL's projects.[24]
- The steam heated from the engine exhaust is assumed to be saturated at 8 bar and 170 ℃.

To simplify, if one of these heat sources are in a secondary loop, it is assumed that the auxiliary fluid reaches the same temperature after the secondary heat exchanger.

The pressure loss over a heat exchanger is assumed to be **1 bar**. This is the maximum pressure loss quoted for a heat exchanger in a scrubber product guide.[25]

6.1. Low pressure LNG

The following calculations will be based on a five cylinder Wärtsilä RT-flex50DF engine. This engine is one of the newest low pressure two-stroke gas engines in the market. Table 9 list some key specifications for this engine. The engines output and break specific fuel consumption (BSFC) used are at maximum continuous rating, where the engine is most effective.[26] Brake specific fuel consumption apply for LNG with a lower calorific value of 50 MJ/kg.

Table 9: Wärtsilä RT-flex50DF, technical information[26]

Wärtsilä RT-flex50DF		
Cylinder bore	mm	500
Piston stroke	mm	2050
Cylinders		5
Speed	rpm	99-124
Mean effective pressure at R1	bar	17,3
Stroke /Bore	-	4,10
Output (at 124 rpm, R1)	kW	7200
BSFC Gas (at 124 rpm, R1)	g/kWh	140,8
BSFC Pilot fuel (at 124 rpm, R1)	g/kWh	2
Gas supply pressure[15]	bar	16
Gas supply temperature[15]	℃	5-45

Flow rate:

Mass flow of LNG to the engine is found by multiplying BSFC with the corresponding output power.

$$\dot{m} = BSFC * P = 140{,}8\ g/kWh * 7200\ kW = 1\ 013\ 760\ g/h$$

$$= 1013760\ g/h * 1/(3600\ s/h * 1000\ g/kg)\ =\ 0{,}2816\ kg\ /s$$

The density of LNG varies depending on the composition. LNG whit a density of 480 kg/m^3 will be used in this calculation.

The volumetric flow rate (Q) is calculated:

$$\dot{V} = \dot{m}/\ \rho\ = \frac{0{,}2816 kg/s}{480\ kg/m^3}\ =\ 5{,}87\ *\ 10^{-4}\ m^3/s$$

6.1.1. Low pressure pump

From equation 8 we can find the power consumption of an ideal pump, delivering the required mass flow of fuel at the required pressure.

In their product webpage, Wärtsilä states that pressure of the gas supplied to the engine is less than 16 bar.[26] However, in a DNV GL project involving a RT-flex 50DF engine, supply pressure of the gas before the GVU was exactly16 bar.[15] The same value will be used in this analysis.

In a real scenario there would be some pressure losses in the pipes, and there might be a height difference between the LNG tank and pump affecting the total head.. Since this engine is not yet installed on a vessel in operation, a piping arrangement cannot be obtained, and pressure loss due to friction in pipes must be ignored. Increased or decreased head due to height difference between the LNG tank and engine is also ignored for the same reasons.

The system in this example uses a pump instead of a PBU, so there is no gauge pressure in the LNG tank. Tank pressure is assumed to be 1 bar.

The pressure increase (Δp) that has to be applied by the pump is:

$$\Delta p = p_{at\ GVU} - p_{tank} = 16\ bar - 1\ bar = 15\ bar$$
$$= 1\ 500\ 000\ N/m^2$$

Hydraulic power (P_h) from the pump:

$$P_h = \Delta p * \dot{V} = 1\ 500\ 000 \frac{N}{m^2} * 5{,}87 * 10^{-4} \frac{m^3}{s} = 880{,}5\ W$$
$$\approx 0{,}9\ kW$$

Electric power (P_e):

Electric power is found using equation 9.

$$P_e = \frac{0{,}9\ kW}{0{,}75}$$

$$\underline{= 1{,}2\ kW}$$

6.1.2. Vaporizer and heater

Fuel side:

Since the composition of LNG is variable, there is quite difficult to find a reliable data on heat capacities, heat of vaporization and other thermodynamicall data at different temperatures. To simplify these calculation, LNG will be considered as pure methane in analysis for the vaporizer and heater.

The liquid temperature would increase from the tank to the pump, inside the pump, and in the pipes before the vaporizer, due to friction and heating from the surrounding air. This rise in temperature is assumed to have little impact on the total heating requirements, and will be neglected. LNG enters the vaporizer at a pressure of 16 bar and a temperature of –163 °C, and exits as saturated gas at the same pressure . The saturated gas is further heated to the engines requirements in the heater. Since the heater and vaporizer works the same way, uses the same type of heating media, and do not have a pump between them, they can be considered as one unit. The gas feed temperature is set to the maximum, 45 °C.

The required increase in enthalpy is the difference between the enthalpy at –163 °C and 45 °C, at 16 bar. This is read from a pressure-enthalpy diagram for methane (appendix 3).

$$h\,[16\,bar, -163℃] \approx -910\frac{kJ}{kg}, \qquad h\,[16\,bar, 45℃] \approx 30\frac{kJ}{kg}$$

$$\Delta h\left(\frac{kJ}{kg}\right) = 30\frac{kJ}{kg} - \left(-910\frac{kJ}{kg}\right) = 940\,kJ/kg$$

When the mass flow rate and the difference in enthalpy is known, heat rate required to vaporize LNG and increase the gas temperature, can be calculated using formula (2).

$$\dot{Q} = \dot{m}_c \Delta h = 0{,}2816\frac{kg}{s} * 940\frac{kJ}{kg} = 264{,}7\,kW$$

Auxiliary side:

Most vaporizers for this application uses a water-glycol mixture as the heating media.[27] Data on a LNG heat exchanger, used in a DNV GL project, states that gas leaving the heat exchanger is 5 °C colder than the glycol-water entering.[28] Water-glycol has to enter the heat exchanger at 50 °C. Temperature of the water–glycol out of the exchanger is assumed to be 0 °C, to ensure there is no freezing in case of a pump failure.

$$T_1 = 50\,℃, \qquad T_2 = 0\,℃$$

The composition of water and glycol is not specified, so it is assumed to be a 50 % volume solution, which has the following properties:[29]

$$C_p = 3{,}4\ \frac{kJ}{kg\ °C}, \qquad (At\ the\ midle\ temperature\ of\ 25\ °C)$$

$$\rho = 1077\frac{kg}{m^3}, \qquad (At\ the\ midle\ temperature\ of\ 25\ °C)$$

Using equation 3 the required flow rate of water-glycol can be found.

$$Q = \dot{m}_h C_{p,h}\left(T_{H,in} - T_{H,out}\right)$$

$$264{,}7\ kW = \dot{m}_h * 3{,}4\ \frac{kJ}{kg\ °C} * (50\ °C - 0\ °C)$$

$$\dot{m}_h = 1{,}55\ kg/s$$

$$\dot{V} = \frac{1{,}55\ kg/s}{1077\ kg/m^3} = 1{,}44 * 10^{-3}\ m^3/s$$

As defined in the terms for calculations, the pressure loss in a heat exchanger is 1 bar. The water-glycol goes through the heat exchanger where LNG is vaporized, and a secondary heat exchanger where water-glycol is reheated from engine cooling water. The total pressure drop for the water glycol loop is 2 bar.

Hydraulic power and electric power for the water-glycol circulating pump is found:

$$P_h = \Delta p * \dot{V} = 200\ 000\frac{N}{m^2} * 1{,}44 * 10^{-3}\frac{m^3}{s}$$
$$288\ W \approx 0{,}3\ kW$$

$$P_e = \frac{0{,}3\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 0{,}4\ kW}}$$

6.1.3 Total power

The total electric pumping power as a percentage of the engine power, is found.

$$\frac{P_{e,total}}{P_{engine}} * 100\ \% = \frac{P_{e,LNG} + P_{e,WG}}{P_{engine}} * 100\%$$

$$= \frac{1{,}2\ kW + 0{,}4\ kW}{7200\ kW} * 100\ \%$$

$$= \mathbf{0{,}029}\ \%$$

6.2. High-pressure

The following calculations is based on a five cylinder MAN B&W G45ME-C9-GI two-stroke, high-pressure engine.[30 p. 47] This specific engine is chosen because it is the model with a maximum output closest to the Wärtsilä low-pressure engine reviewed in the previous chapter.

Table 10: MAN B&W G45ME-C9-GI, Technical information.[30]

MAN B&W G45ME-C9-GI		
Piston stroke	mm	2250
Speed	rpm	87-111
Mean effective pressure at L1	bar	21
Output (at 111 rpm, L1)	kW	6950
BSFC Gas (at 111 rpm, L1)	g/kWh	140
BSFC Pilot fuel (at 111 rpm, L1)	g/kWh	5,1
Gas supply pressure[31]	bar	300
Gas supply temperature[31]	℃	45

Mass flow:

Mass flow of LNG to the engine is found by multiplying BSFC with the corresponding output power.

$$\dot{m} = BSFC * P = 140\ g/kWh * 6\ 950\ kW = 973\ 000\,\frac{g}{h}$$

$$= 973\ 000\ g/h * 1/\ (3\ 600\ s/h * 1\ 000\ g/kg) = \ 0{,}27\ kg\ /s$$

LNG with the same properties as in the low-pressure engine will be used. The volumetric flow rate is calculated:

$$\dot{V} = \dot{m}/\rho = \frac{0{,}27\,kg/s}{480\,kg/m^3} = 5{,}625 * 10^{-4}\,m^3/s$$

6.2.1. Fuel pumps

High-pressure fuel delivery systems are often set up with a low- and high pressure pump. Fuel is transported out of the tank by a low pressure pump, while the high pressure pump boost the pressure up to approximately 300 bar. To simplify the calculations, these two pumps will be treated as one.

The LNG tank have no PBU and the tank pressure is assumed to be 1 bar.

The pressure increase (Δp) that has to be applied by the pump is:

$$\Delta p = p_{at\,GVU} - p_{tank} = 300\,bar - 1\,bar = 299\,bar$$
$$= 29\,900\,000\,N/m^2$$

Hydraulic power (P_h) from the pump:

$$P_h = \Delta p * Q = 29\,900\,000\,\frac{N}{m^2} * 5{,}625 * 10^{-4}\,\frac{m^3}{s}$$
$$= 16\,818{,}75\,W = 16{,}8\,kW$$

Electric power (P_S):

$$P_S = \frac{16{,}8\,kW}{0{,}75}$$

$$\underline{= \mathbf{22{,}4\,kW}}$$

6.2.2. Vaporizer and Heater

LNG side:

Since this is the only system where there are a very high pressure increase before the heat exchanger, the rise in rise in enthalpy due to pump work is included. In the other system this temperature change is low enough to be neglected. Increase in enthalpy can be found by a variation of the conservation law (Equation C).[8] As earlier mentioned, pumps are regarded adiabatic, Q =0. As found in the analysis of low pressure LNG, enthalpy at 1 bar and -163 ℃ is *- 910 kJ/kg.*

$$\dot{m}(h_2 - h_1) = \dot{W} = P$$

$$h_2 = \frac{P}{\dot{m}} + h_1 = \frac{22{,}4\ kW}{0{,}27 kg/s} - 910\frac{kJ}{kg} = -830\ kJ/kg$$

The temperature at 300 bar and a enthalpy of -830 kJ/kg is read to approximately - *155* °C in appendix 3.

The vaporizer and heater are considered to be identical to those in the low-pressure system. Since LNG at 300 bar is above the critical point, the LNG cannot actually be vaporized, but must be heated from supercritical liquid to supercritical fluid at 300 bar. Enthalpy difference of methane at -155 °C and 45 °C, at 300 bar is read from appendix 3.

$$h\ [300\ bar, -155°C] \approx -830\frac{kJ}{kg},$$

$$h\ [300\ bar, 45°C] \approx -160\frac{kJ}{kg}$$

$$\Delta h\ \left(\frac{kJ}{kg}\right) = -160\frac{kJ}{kg} - \left(-830\frac{kJ}{kg}\right) = 670\ kJ/kg$$

$$\dot{Q} = \dot{m}_c \Delta_{vap} H = 0{,}27\ \frac{kg}{s} * 670\frac{kJ}{kg} = 180{,}9\ kW$$

Auxiliary side:

This heat exchanger is assumed to be identical to the one used for low pressure LNG, and has the same properties.

$$T_1 = 50\ °C, \qquad T_2 = 0\ °C$$

$$C_p = 3{,}4\ \frac{kJ}{kg\ °C}, \qquad (At\ the\ midle\ temperature\ of\ 25\ °C)$$

$$\rho = 1077\frac{kg}{m^3}, \qquad (At\ the\ midle\ temperature\ of\ 25\ °C)$$

Using equation 3 the required flow rate of water-glycol is found:

$$Q = \dot{m}_h C_{p,h}\left(T_{H,in} - T_{H,out}\right)$$

$$180{,}9\ kW = \dot{m}_h * 3{,}4\ \frac{kJ}{kg\ °C} * (50\ °C - 0\ °C)$$

$$\dot{m}_h = 1{,}06\ kg/s$$

$$\dot{V} = \frac{1{,}06\ kg/s}{1077\ kg/m^3} = 1 * 10^{-3}\ m^3/s$$

The total pressure drop for the water glycol loop is 2 bar.

Hydraulic power and electric power for the water-glycol circulating pump is found:

$$P_h = \Delta p * \dot{V} \ = 200\ 000 \frac{N}{m^2} * 1 * 10^{-3} \frac{m^3}{s}$$
$$200\ W = 0{,}2\ kW$$

$$P_e = \frac{0{,}2\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 0,3\ kW}}$$

6.1.3 Total power

The total electric pumping power as a percentage of the engine power, is found:

$$\frac{P_{e,total}}{P_{engine}} * 100\ \% = \frac{P_{e,LNG} + P_{e,WG}}{P_{engine}} * 100\%$$

$$= \frac{22{,}4\ kW + 0{,}3\ kW}{7200\ kW} * 100\ \%$$

$$\underline{\boldsymbol{= 0,32\ \%}}$$

6.3. Scrubber

Since most of parameters in a scrubber is expressed as function of kWh engine power, and sulphur % in the fuel, there is no need to select a engine for this application. For the ease of comparison, the output of the engine is set to 7 MW. The fuel has a sulphur content of 3,5 %.

These scrubbers only clean the exhaust from one engine, hence they are not integrated scrubbers. Integrated scrubbers must have an exhaust fan to reduce backpressure on the engines. This fan draws approximately 0,4 % of the engine power being scrubbed.[25]

6.4. Open Loop Scrubber

In the open loop scrubber the most energy consuming job is to pump water from the seabed to the scrubber tower. Layout of the ship has a big impact of the work required from the pump as the scrubbing tower is normally placed in the upper part of the ship. There will be done calculations for three scenarios where the scrubbing tower is placed 10, 20 and 30 meters above the seabed.

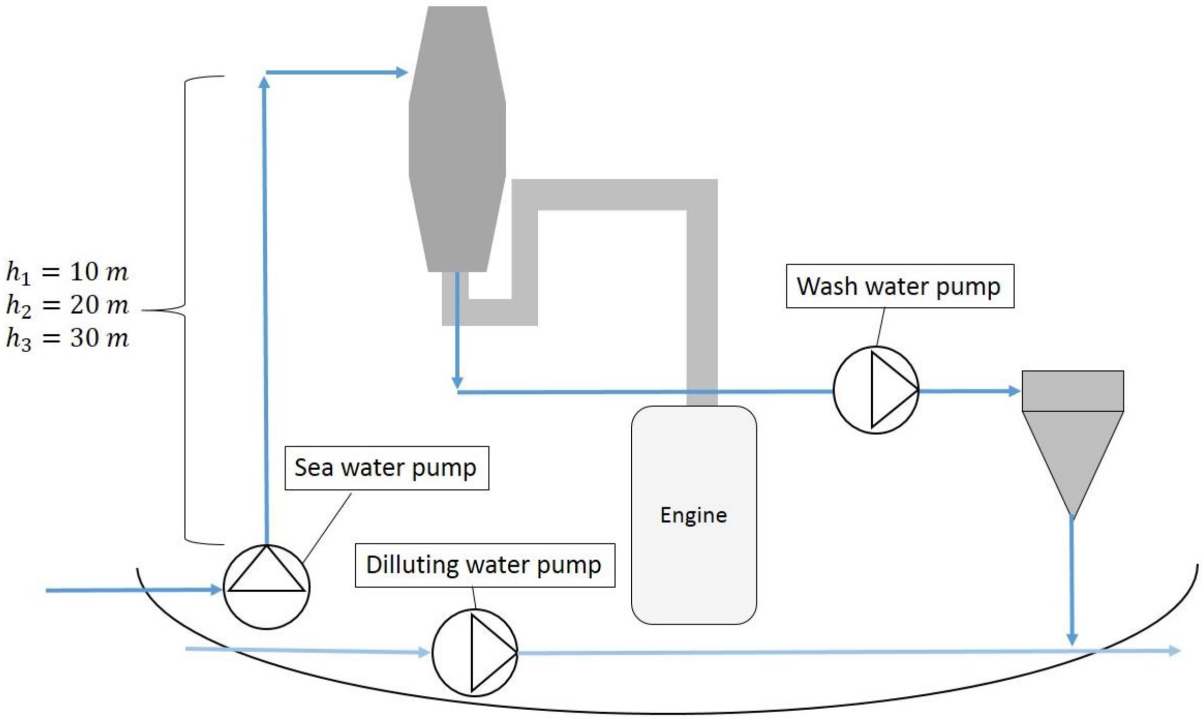


Figure 18: Energy consuming parts of an open loop scrubber

6.4.1. Sea water pump

The seawater pump in an open loop scrubber has to supply a volumetric flow to the scrubber unit of approximately 45 m^3/MWh.[7] Volumetric flow rate when the engine produces an output of 7 MW is:

$$\dot{V} = 45\ m^3/MWh * 7\ MW = 315\, m^3/h = 0{,}0875\, m^3/s$$

Scrubbing water is taken from the sea crest, and the pump is placed at the same height, which means the pressure at the suction p_1 side is 1 atm. or approximately 1 bar. Due to lack of detailed piping arrangements pressure losses are neglected. That means the pressure at the pump side has to be large enough to lift the water up to the scrubber unit, and for the scrubbing water to have the required pressure at the inlet of the scrubber unit.

Data on required pressure of wash water at the scrubber unit's inlet is very limited in open literature. In a pilot project where a Hamworthy – Krystallon scrubber where installed on one of Holland America Line's vessels, the pressure under normal conditions were measured to 7 bars.[32] Due to the lack of other data, this pressure will be used in the following calculations. This number may only apply for this exact vessel and the scrubber that it was measured on.

Seawater is regarded to have the same density as freshwater.

Using equation 8, an expression for hydraulic pump power at variable scrubber heights (h_x) is found.

$$\boldsymbol{P_h = \Delta p * \dot{V}}$$

$$\dot{V} = 0{,}0875\, m^3/s$$

$$\Delta p = \rho * g * h + p_3 - p_1$$

$$= 1\,000\,\frac{kg}{m^3} * 9{,}81\,\frac{m}{s^2} * h_x\, m + 700\,000\,\frac{N}{m^2} - 100\,000\,\frac{N}{m^2}$$

$$= h_x * 9\,810\,\frac{N}{m^2} + 700\,000\,\frac{N}{m^2} - 100\,000\,\frac{N}{m^2}$$

$$P_h = \left(h_x * 9\,810\,\frac{N}{m^2} + 700\,000\,\frac{N}{m^2} - 100\,000\,\frac{N}{m^2}\right)$$
$$* 0{,}0875\, m^3/s$$

Hydraulic and electric power is found for the scenarios where the scrubber is places 10, 20 and 30 meters above the seabed:

- $\boldsymbol{h = 10\ m}$:

$$P_h = \left(10 * 9\,810\ \frac{N}{m^2} + 700\,000\frac{N}{m^2} - 100\,000\frac{N}{m^2}\right) * 0{,}0875\frac{m^3}{s}$$

$$= 61\,084\ W$$

$$\underline{\approx 61{,}1\ kW}$$

$$P_e = \frac{61{,}1 kW}{0{,}75}$$

$$\underline{\boldsymbol{= 81,5\ kW}}$$

- $\boldsymbol{h = 20\ m}$:

$$P_h = \left(20 * 9\,810\ \frac{N}{m^2} + 700\,000\frac{N}{m^2} - 100\,000\frac{N}{m^2}\right) * 0{,}0875\frac{m^3}{s}$$

$$= 69\,668$$

$$\underline{\approx 69{,}7\ kW}$$

$$P_e = \frac{69{,}7\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 92,9\ kW}}$$

- $\boldsymbol{h = 30\ m}$:

$$P_h = \left(30 * 9\,810\ \frac{N}{m^2} + 700\,000\frac{N}{m^2} - 100\,000\frac{N}{m^2}\right) * 0{,}0875\frac{m^3}{s}$$

$$= 78\,251\ W$$

$$\underline{\approx 78{,}3\ kW}$$

$$P_e = \frac{78{,}3\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 104,4\ kW}}$$

6.4.2. Diluting water pump

The flow of diluting water is about the same as the scrubbing water. Volumetric flow rate of diluting water is $0{,}0875\ m^3/s$. In the above mentioned pilot project, pressure of diluting water was measured to 2 bars.[32] Water is pumped from the seabed at atmospheric pressure, approximately 1 bar.

Ideal hydraulic pump power is found:

$$\Delta p = 200\ 000 \frac{N}{m^2} - 100\ 000\ \frac{N}{m^2} = \ 100\ 000\ \frac{N}{m^2}$$

$$Q = \ 0{,}0875\ m^3/s$$

Hydraulic and electric power is found:

$$P_h \ = \ 100\ 000\ \frac{N}{m^2} = \ 0{,}0875 \frac{m^3}{s}$$
$$\underline{= 8{,}75\ kW}$$

$$P_e = \frac{8{,}75 kW}{0{,}75}$$

$$\underline{\boldsymbol{= 11{,}7\ kW}}$$

6.4.3. Wash water pump

The wash water pump must pressurize the wash water sufficient to go through the hydro cyclone and be discharged to sea. The hydro cyclone in the open loop scrubber is a passive device, and do not consume any energy by itself. From one DNV GL "LNG ready" project, it is found that this pump have a head of 35 meters.[33] When seawater is assumed to have the same density as freshwater, this correspond to:

$$\Delta p = \ 35\ m * 9810 \frac{N}{m^3} = \ 343\ 350 \frac{N}{m^2}$$

The flow rate of scrubbing water is the same as into the scrubber. The very small amount that vaporizes with the exhaust is neglected. $\dot{V} = \ 0{,}0875\, {m^3}/{s}$.

Hydraulic and electric pump power is found:

$$P_h = 343\ 350 \frac{N}{m^2} * 0{,}0875 \frac{m^3}{s} = 30\ 043\ W$$
$$\approx 30\ kW$$

$$P_e = \frac{30\ kW}{0{,}75}$$
$$\underline{= \mathbf{40\ kW}}$$

6.4.4. Total pumping power

The total electric pumping power as a percentage of the engine power, is found for the lowest and highest placement of the scrubber unit.

Scrubber unit placed 10 meters above the waterline:

$$\frac{P_{e,total}}{P_{engine}} * 100\ \% = \frac{P_{e,sea\ w} + P_{e,dil} + P_{e,wash}}{P_{engine}} * 100\%$$
$$= \frac{81{,}5\ kW + 11{,}7\ kW + 40{,}0\ kW}{7000\ kW}$$
$$\underline{= \mathbf{1{,}90}\ \%}$$

Scrubber unit placed 30 meters above the waterline:

$$\frac{P_{e,total}}{P_{engine}} * 100\ \% = \frac{P_{e,sea\ w} + P_{e,dil} + P_{e,wash}}{P_{engine}} * 100\%$$
$$\frac{104{,}4\ kW + 11{,}7\ kW + 40{,}0\ kW}{7000\ kW}$$
$$\underline{= \mathbf{2{,}23}\ \%}$$

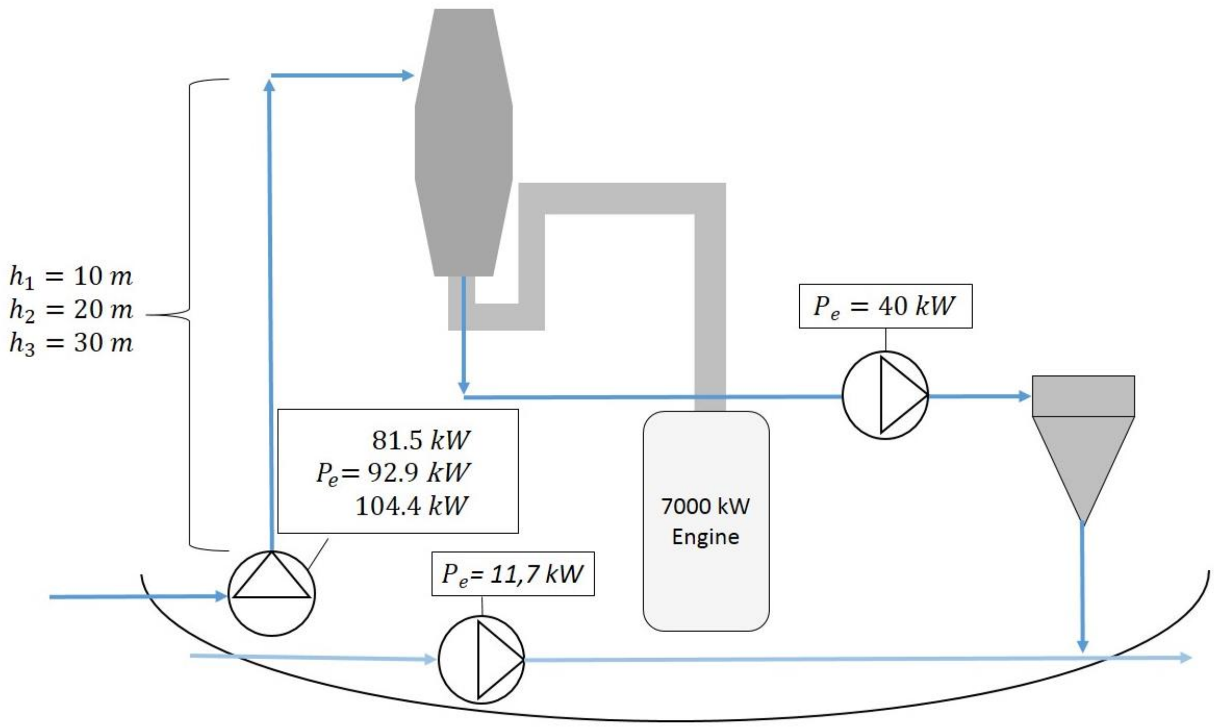


Figure 19: Energy consumption - Open loop scrubber

6.5. Closed Loop scrubber

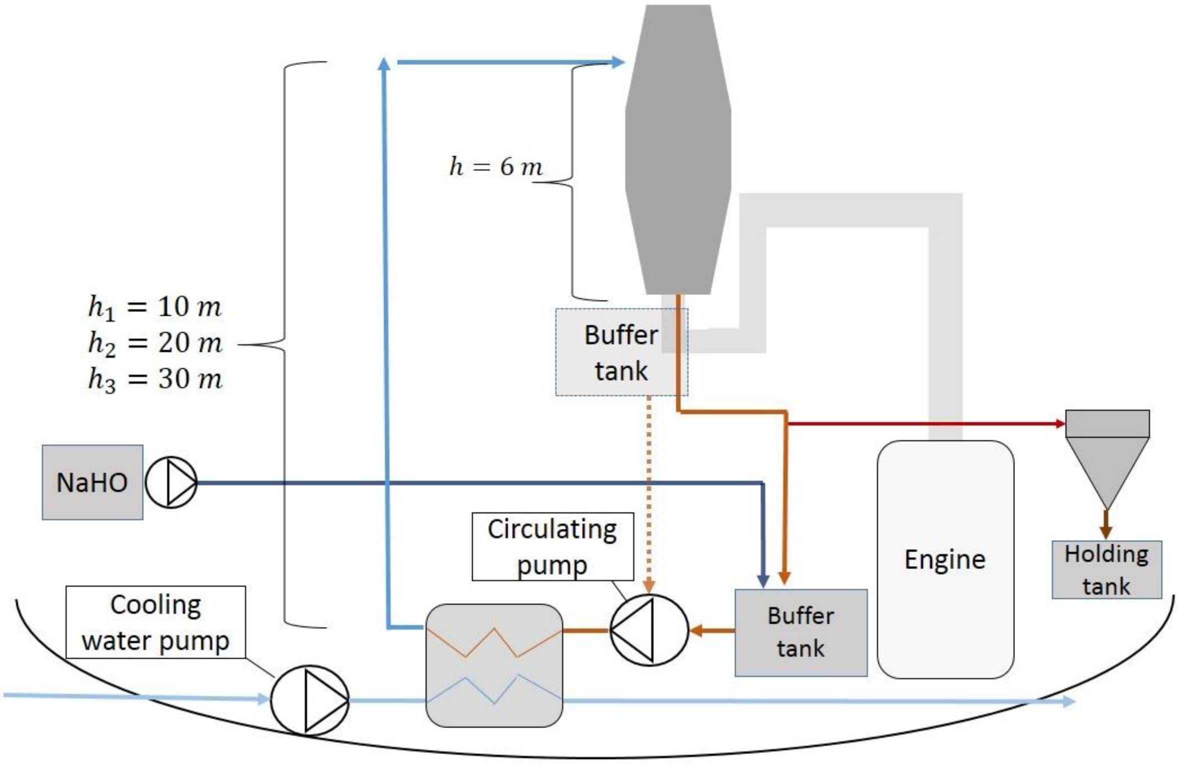


Figure 20: Closed loop scrubber - process flow

Scrubbing water flow rate:

The specific flow rate of circulating scrubbing water is approximately $24\ m^3/MWh$.[34]. Volumetric flow rate when the engine produces an output of 7 MW is:

$$\dot{V} = 24\ m^3/MWh * 7\ MW = 168\ m^3/h = 0{,}0466\ m^3/s$$

6.5.1. Circulation pump

In a product guide for scrubbers from Wärtsilä, it is recommended that the heat exchanger is placed below the water line.[17]

This means that the scrubbing water has to be lifted from the seabed to the scrubbing unit, the same height as in the open loop scrubber. The same placements of the scrubber unit are used, meaning the height between the scrubber and the seabed is 10 m, 20 m and 30 m.

However, if the buffer tank is placed above the seabed, the suction side of the pump will have a pressure from the weight of the water between the buffer tank and the pump. There will be done calculations for two placements of the buffer tank. One where the buffer tank is placed right below the scrubber unit, and the head is equal the height of the unit independent of the distance between the unit and the seabed, and one where the tank is placed at the seabed.

A scrubber unit sized for a 7 MW engine, has the water inlet approximately 6 meters above the drain.

$$h_1 = 10\ m, \qquad h_2 = 20\ m, \qquad \mathrm{h}_3 = 30\ m, \qquad \mathrm{h}_4 = 6\ m$$

Since the scrubber units are the same type for open and closed loop, the same required injection pressure of 7 bars will be used. Pressure in the buffer tank is assumed to be 1 bar. An expression for hydraulic pump power at variable scrubber heights (h_x) is found using equation 8.

$$\boldsymbol{P_h = \Delta p * \dot{V}}$$

$$\Delta p = \rho * g * h + p_2 - p_1$$

$$= 1\,000 \frac{kg}{m^3} * 9{,}81 \frac{m}{s^2} * h_x\ m + 700\,000 \frac{N}{m^2} - 100\,000 \frac{N}{m^2}$$

$$= h_x * 9\,810\ \frac{N}{m^2} + 700\,000 \frac{N}{m^2} - 100\,000 \frac{N}{m^2}$$

$$P_h = \left(h_x * 9\,810\ \frac{N}{m^2} + 700\,000 \frac{N}{m^2} - 100\,000 \frac{N}{m^2}\right) * 0{,}0466 m^3/s$$

Hydraulic and electric power is found for the different lifting heights.

- $\boldsymbol{h = 10\ m}$:

$$P_h = \left(10\ * 9\,810\ \frac{N}{m^2} + 700\,000 \frac{N}{m^2} - 100\,000 \frac{N}{m^2}\right) * 0{,}0466 \frac{m^3}{s}$$

$$= 32\,531\ W$$

$$\underline{\approx 32{,}5\ kW}$$

$$P_e = \frac{32{,}5\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 43{,}3\ kW}}$$

- $\boldsymbol{h = 20\ m}$:

$$P_h = \left(20 * 9\,810\ \frac{N}{m^2} + 700\,000 \frac{N}{m^2} - 100\,000 \frac{N}{m^2}\right) * 0{,}0466 \frac{m^3}{s}$$

$$= 37\,103\ W$$

$$\underline{\approx 37{,}1\ kW}$$

$$P_e = \frac{37{,}1\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 49{,}5\ kW}}$$

- $\boldsymbol{h = 30\ m}$:

$$P_h = \left(30 * 9\,810\ \frac{N}{m^2} + 700\,000\frac{N}{m^2} - 100\,000\frac{N}{m^2}\right) * 0{,}0466\frac{m^3}{s}$$

$$= 41\,674\ W$$

$$\underline{\approx 41{,}7kW}$$

$$P_e = \frac{41{,}7\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 55{,}6\ kW}}$$

- $\boldsymbol{h = 6\ m}$:

$$P_h = \left(6 * 9\,810\ \frac{N}{m^2} + 700\,000\frac{N}{m^2} - 100\,000\frac{N}{m^2}\right) * 0{,}0466\frac{m^3}{s}$$

$$= 30\,703\ W$$

$$\underline{\approx 30{,}7kW}$$

$$P_e = \frac{30{,}7\ kW}{0{,}75}$$

$$\underline{\boldsymbol{= 40{,}9\ kW}}$$

6.5.2. Cooling water pump

The specific flow rate of cooling water is approximately $30\ m^3/MWh$.[33] With the 7 MW engine, the flow rate is:

$$\dot{V} = 30\ m^3/MWh * 7\ MW = 210\ m^3/h = 0{,}0583\ m^3/s$$

The pressure drop through the heat exchanger is set to 1 bar. That means the pump has to increase the pressure from 1 bar atmospheric pressure at the sea chest, to 2 bar discharge pressure.

$$P_h = 100\,000 \frac{N}{m^2} * 0{,}0583 \frac{m^3}{s} = 5830\, W$$

$$\underline{\approx 5{,}8\, kW}$$

$$P_e = \frac{5{,}8\, kW}{0{,}75}$$

$$\underline{\boldsymbol{= 7{,}7\, kW}}$$

6.5.3. NaOH

From figure 3.2 in Wärtsilä's environmental product guide it can be read that the 50 % $NaOH$ consumption is approximately 140 l/h with a 7 MW engine using 3,5% sulphur fuel.[17] This flow rate is so low that the pumps energy consumption can be neglected, but the NaOH consumption should still be noted, as it is a quite expensive chemical.

NaOH consumption per MWh is:

$$Consumption\ of\ 50\ \%\ NaOH = \frac{140\ {}^{l}/_{h}}{7MW}$$

$$\underline{\boldsymbol{= 20\ l/MWh}}$$

6.5.4. Other consumers

The bleed of treatment unit, illustrated in Figure 20 as a hydro cyclone, is an active energy consuming component in the closed loop scrubber. In a DNV GL project, a scrubber supplier estimated its power to be 3 kW for an 7700 kW engine.[35] It is quite complicated to calculate the required power to run this bleed of treatment unit, so the estimate used in the DNV GL project is interpolated to the 7000 kW engine used in this analysis.

$$Power\ for\ bleed\ of\ treatment\ \text{unit} = 3 * \frac{7000}{7700} = 2{,}7\ kW$$

There are also some energy consumption from miscellaneous units such as venting fans for tanks, sludge pumps and monitoring systems. The flow through these are low compared to the scrubbing and cooling water flow, and will be neglected. This is also because similar components are neglected in the other systems.

6.5.5. Total electric power

The total electric power as a percentage of the engine power, is found for the best and worst case scenario for placement of equipment.

Buffer tank placed below the scrubber unit:

$$\frac{P_{S,total}}{P_{engine}} * 100\ \% = \frac{40{,}9\ kW + 7{,}7\ kW + 2{,}7}{7000\ kW} * 100\% = \ 0{,}88\ \%$$

Buffer tank placed at sea level and scrubber unit is 30 meters above sea level:

$$\frac{P_{S,total}}{P_{engine}} * 100\ \% = \frac{55{,}6\ kW + 7{,}7\ kW + 2{,}7}{7000\ kW} * 100\% = \ 1{,}09\ \%$$

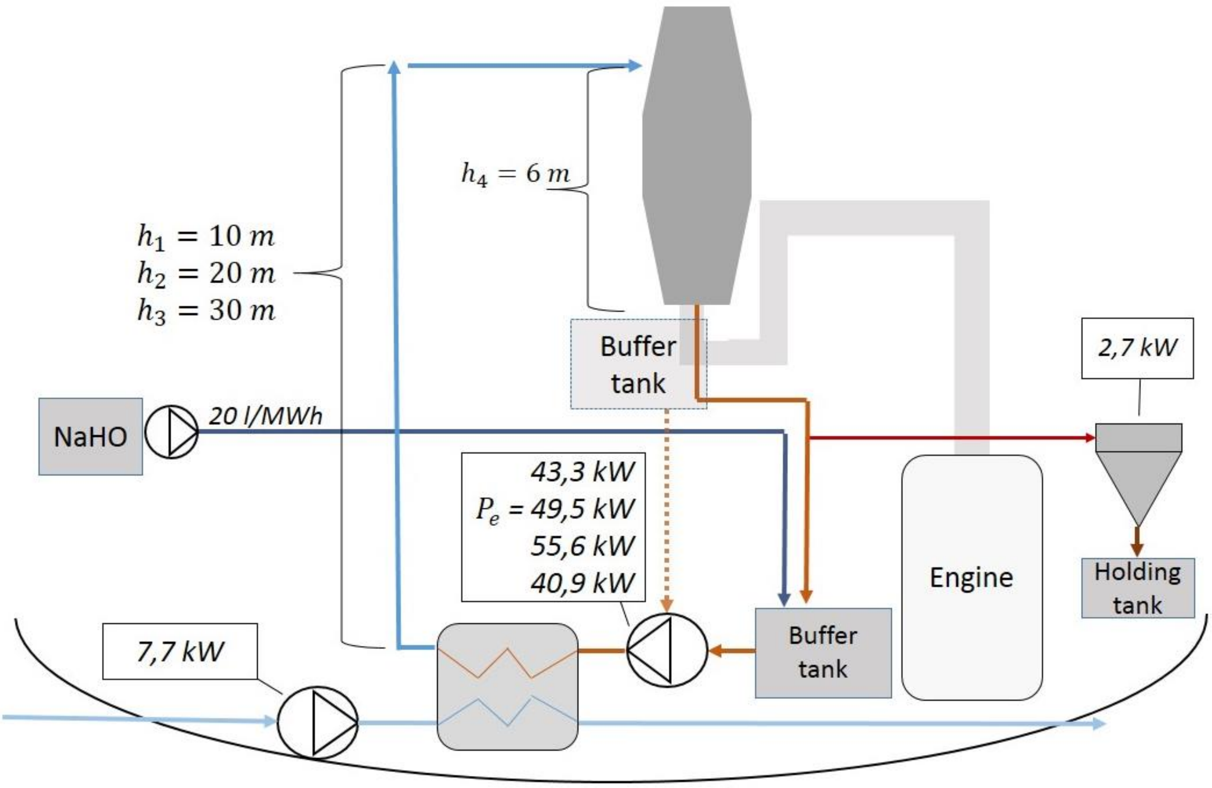


Figure 21: Consumption of open loop scrubber

6.6. MGO

Table 11: MAN B&W G45ME-C9-GI, Diesel mode, Technical information.[30]

MAN B&W G45ME-C9-GI (Diesel mode)		
Piston stroke	mm	2250
Speed	rpm	87-111
Mean effective pressure at L1	bar	21
Output (at 111 rpm, L1)	kW	6950
BSFC Diesel (at 111 rpm, L1)	g/kWh	170
Fuel supply pressure[16]	bar	1000-2000

This engine is the same used for high pressure LNG, but the specifications are for operation on diesel fuel.

Fuel flow rate:

Mass flow of MGO to the engine is found by multiplying BSFC with the corresponding output power.

$$\dot{m} = BSFC * P$$
$$= 170\ g/kWh * 6950\ kW * \frac{1}{3600\ s/h * 1000\ g/kg}$$
$$= 0{,}3282\ kg/s$$

MGO has a density of approximately 840 kg/m^3.[36]

$$\dot{V} = \dot{m}/\rho = \frac{0{,}3282\ kg/s}{840\ kg/m^3} = 3{,}9071 * 10^{-4}\ m^3/s$$

6.6.1. Fuel pumps

To cover for the big range of fuel injection pressure, there will be analysis for two scenarios. One where the pressure is 1000 bar, and one where it is 2000 bar. All the pumps is regarded as one unit increasing the pressure from 1 bar at the fuel tank to the injection pressure.

1000 bar:

Hydraulic pump power and Electric pump power is found:

$$P_h = \Delta p * \dot{V} = 100\ 000\ 000 \frac{N}{m^2} * 3{,}9071 * 10^{-4} \frac{m^3}{s}$$
$$= 39\ 071\ W \approx 39{,}1\ kW$$

$$P_e = \frac{39{,}1\ kW}{0{,}75}$$
$$\underline{\mathbf{= 52{,}1\ kW}}$$

2000 bar:

Hydraulic pump power and Electric pump power is found:

$$P_h = \Delta p * \dot{V} = 200\ 000\ 000 \frac{N}{m^2} * 3{,}9071 * 10^{-4} \frac{m^3}{s}$$
$$= 78\ 142\ W \approx 78{,}1\ kW$$

$$P_e = \frac{78{,}1\ kW}{0{,}75}$$

$$\underline{= \mathbf{104{,}1\ kW}}$$

6.6.2. Heating

As mentioned previously, the MGO has to be cooled down to approximately 35 ℃ for it to have a viscosity complying to the engine and components requirements.[23] The temperature of the fuel before the heat exchanger is quite difficult to estimate since it depends on the surrounding temperature, the fuel tanks layout and the temperature of pipes and pumps. In Man Diesel and Turbo's guide for operation on low-sulphur fuels, an illustration show that the temperature is 40 ℃ after the centrifuge.[23] Temperature difference is the same when using K or ℃, so the inlet and outlet temperature does not have to be converted.

The specific heat of fuels are normally not measured in fuel samples, and reliable data for each fuel grade is difficult to find. Engineeringtoolbox.com has a general estimate for all fuel oil of 1.67-2.09 kJ/kgK. The middle value of 1,9 kJ/kgK is used for this example.

Equation 3 is used to find how much heat has to be transferred from the fuel to the cooling medium:

$$\boldsymbol{\dot{Q} = \dot{m}_h C_{p,h}(T_{H,in} - T_{H,out})}$$

$$T_{H,in} = 40\ ℃, \qquad T_{H,out} = 35\ ℃, \qquad C_{p,h} = 1{,}9\ kJ/kg\ K,$$
$$\dot{m}_h = 0{,}3282\ kg/s$$

$$\dot{Q} = 0{,}3282\frac{kg}{s} * 1{,}9\frac{kJ}{kg\ K} * (40℃ - 35℃) = 3{,}12\ kW$$

Since heat added to the fuel is defined as positive, the heating requirement is:

$$\underline{\dot{Q} = -3{,}12\ kW}$$

A quick estimate using the same procedure as for HP and LP LNG, gives that the pumping power is approximately 10 Watt. Pumping of coolant is neglected.

6.6.3. Total power

Pumping power as percent of engine output is found for the two injection pressures:

$$\frac{P_{S,total}}{P_{engine}} * 100\ \%$$

1000 bar:

$$\frac{52{,}1\ kW}{6950kW} * 100\%$$

$$\underline{= \mathbf{0{,}75\ \%}}$$

2000 bar:

$$\frac{104{,}1\ kW}{6950kW} * 100\% =$$

$$\underline{\mathbf{1{,}50\ \%}}$$

6.7. HFO

The engine used for MGO in the previous chapter will also be used in the analyses for HFO. As stated in MAN Diesel & Turbos marine engine guide, where the data in Table 11 is taken from, the BSFC is given that the fuel used has a lower calorific value (LCV) of 42,7 MJ/kg. This correspond to MGO. LCV of a fuel will vary some for each time is bunkered, but 40,5 MJ/kg is the standard value used for HFO at DNV GL.

BSFC can be converted to apply for HFO by multiplying it with the LCV of MGO and dividing it with the LCV for HFO. This engine's consumption of HFO is found:

$$\mathrm{BSFC}_{HFO} = BSFC_{MGO} * \frac{\mathrm{LCV}_{MGO}}{\mathrm{LCV}_{HFO}} = 170\ g/kWh * \frac{42{,}7}{40{,}5}$$

$$= 179{,}23\ g/kWh$$

Flow rate:

Mass flow of LNG to the engine is found by multiplying BSFC with the corresponding output power.

$$\dot{m} = BSFC * P$$
$$= 179{,}23\ g/kWh * 6950\ kW * 1/\left(3600\,\frac{s}{h} * 1000\,\frac{g}{kg}\right)$$
$$= 0{,}3460\ kg/s$$

The fuel grade RMG380 is used for this example. Appendix 4 show that maximum density of RMG380 is 991 kg/m^3 at 15 ℃. As mentioned in the chapter about current technologies, HFO is kept at a higher temperature where the density would be lower. Reliable sources for the density at a different temperature is not found, so the maximum density at 15 ℃ is used to calculate the volumetric flow rate ($\dot{V}$). It has to be mentioned that this will give a volumetric flow rate lower than what's realistic , hence a lower hydraulic power from the pump.

$$\dot{V} = \dot{m}/\rho = \frac{0{,}3460 kg/s}{991\ kg/m^3} = 3{,}491 * 10^{-4}\ m^3/s$$

6.7.1. Fuel pumps

The injection pressure for HFO is approximately the same as for MGO. There are done two scenarios here as well, one for 1000 bar and one for 2000 bar.

1000 bar:

Hydraulic pump power and Electric pump power is found:

$$P_h = \Delta p * \dot{V} = 100\,000\,000\,\frac{N}{m^2} * 3{,}491 * 10^{-4}\,\frac{m^3}{s}$$
$$= 34\,910\ W \approx 34{,}9\ kW$$

$$P_e = \frac{34{,}9\ kW}{0{,}75}$$

$$\underline{\mathbf{= 46{,}5\ kW}}$$

2000 bar:

Hydraulic pump power and Electric pump power is found:

$$P_h = \Delta p * \dot{V} = 200\,000\,000 \frac{N}{m^2} * 3{,}491 * 10^{-4} \frac{m^3}{s}$$
$$= 69\,820\ W \approx 69{,}8\ kW$$

$$P_e = \frac{69{,}8\ kW}{0{,}75}$$

$$\underline{= \mathbf{93{,}1\ kW}}$$

6.7.2. Heating

Fuel side:

Heating of the fuel tanks will not be included since it depend on tank material, tank design and ambient temperature, variables that are ship specific.

As mentioned in the current technologies chapter, the HFO is normally stored at 10 ℃ above the pour point which is max 30 °C for RMG380. Fuel pumped from the storage tank has a temperature of 40 °C.

In a HFO supply system there are several stages of heating, but for this purpose these will be regarded as one unit heating the HFO from storage temperature to required temperature before the engine. Recommended viscosity after the heater is between 10-15 cSt, and MAN D&T allow up to 20 cSt.[37] The middle value of 15 cSt is chosen. The heating chart in appendix III shows that a fuel with a viscosity of 380 cSt at 50 °C has to be heated to approximately 130 °C to have a viscosity of 15 cSt after the heater.

$$T_{C,in} = 50\ ℃, \qquad T_{C,out} = 130\ ℃$$

The specific heat capacity used for MGO is also used for HFO.

$$C_{p,c} = 1{,}9\ kJ/kgK$$

Required heat transfer to the fuel is calculated using equation 1:

$$\dot{Q} = 0{,}3460 \frac{kg}{s} * 1{,}9 \frac{kJ}{kg℃} * (130\ ℃ - 50\ ℃) = 52{,}6\ kW$$

Auxiliary side:

The fuel is heated with water vapor, saturated at 8 bar. At the outlet, the steam is completely condensed to liquid water, and the heat of evaporation is transferred to the HFO. There is a pressure drop of 1 bar at the heat exchanger where HFO is heated, and 1 bar where the water is vaporized again by the engine exhaust. This scenario is simplified by saying that the circulating pump is placed at the liquid side of the loop.

The latent heat of evaporation for saturated steam at 8 bar is:[38]

$$\Delta h = 2046\ kJ/kg$$

Required mass flow of steam is found:

$$Q = \dot{m}_h \Delta h$$

$$52{,}6\ kW = \dot{m}_h * 2046\ kJ/kg$$

$$\dot{m}_h = 0{,}026\ kg/s$$

All liquids in this thesis are assumed to be incompressible, which means the density of the pressurized water is 1000 kg/m^3. The volume flow of saturated water is found.

$$\dot{V} = \frac{0{,}026\ kg/s}{1000\ kg/m^3} = 2{,}6 * 10^{-5}\ m^3/s$$

Power for the circulating pump is found:

$$P_h = \Delta p * \dot{V} \ = 200\ 000 \frac{N}{m^2} * 2{,}6 * 10^{-5} \frac{m^3}{s}$$
$$5{,}2\ W$$

This seems incredible low for heating the HFO, so an alternative procedure is tried. The water is assumed to be saturated liquids at both sides of the pump. The variation of equation C, saying that shaft work in an adiabatic process is equal to change in enthalpy is used. The change in specific enthalpy from saturated liquid water at 7 bar to saturated liquid water at 9 bars is:[38]

$$\Delta h = 742{,}6 \frac{kJ}{kg} - 697{,}1 \frac{kJ}{kg} = 45{,}5 \frac{kJ}{kg}$$

$$P_h = 0{,}026 \frac{kg}{s} * 45{,}5 \frac{kJ}{kg} = 1{,}2\ kW$$

$$P_e = \frac{1{,}2\ kW}{0{,}75}$$

$$\underline{= \mathbf{1{,}6}\ \boldsymbol{kW}}$$

This value is assumed to be more realistic, and will be used as the electric pump power to heat the HFO.

6.7.3. Total power

Pumping power as percent of engine output is found for the two injection pressures:

$$\frac{P_{S,total}}{P_{engine}} * 100\ \%$$

1000 bar:

$$\frac{46{,}5\ kW + 1{,}6\ kW}{6950kW} * 100\%$$

$$\underline{= \mathbf{0{,}69}\ \%}$$

2000 bar:

$$\frac{93{,}1\ kW + 1{,}6\ kW}{6950kW} * 100\%$$

$$\underline{= \mathbf{1{,}36}\ \%}$$

6.8. Results and comparison

Table 12: Comparison of energy consumption

			HFO	MGO	OL Scrubber	CL Scrubber	HP LNG	LP LNG
Engine output	*kW*		6950	6950	7000	7000	6950	7200
Pump power	*kW*	Highest	48,1	52,1	133,2 (+ HFO)	51,3 (+ HFO)	22,7	1,6
		Lowest	94,7	104,1	156,1 +(HFO)	66,0 (+HFO)	22,7	1,6
	% of engine load	Highest	0,70%	0,75%	1,90% (+ HFO)	0,88% (+ HFO)	0,32%	0,03%
		Lowest	1,36 %	1,50%	2,23% (+HFO)	1,09 % (+HFO)	0,32%	0,03%
Heating fuel	*kW*		52,6	-3,1	= HFO	= HFO	180,9	264,7

Important notes to Table 12:

- The heating of fuel is not direct energy consumption, but the rate of energy exchanged between the auxiliary fluid and the fuel. However, if a fuel requires much heating or cooling, it also requires a high flow of auxiliary fluid. This increases the energy consumption from the auxiliary pump. If the fuel has to be heated to high temperatures, the heating medium has to be warmed up somehow, e.g. by a boiler. More details about heating can be read in chapter 4.2. and chapter 6.
- All energy consumption in a scrubber are in addition to the energy consumption related to HFO.
- The scrubber pump consumption has two values. These values are respectively the best and worst case scenario for placement of the scrubber unit.
- Heating of HFO does not include heating of the fuel storage tank.

Table 13: Possible energy consumption not included.

System:	Possible energy consumers
HFO	• Heating of fuel tanks. • Boiling of steam for heating the fuel during a cold start. • The high viscosity of HFO will give substantially higher pressure losses. • Centrifuging of the fuel.
MGO	• Centrifuging. • Higher cooling requirements during changeover of fuel.
Open Loop Scrubber	• Sludge pump.
Closed Loop Scrubber	• Heating of NaHO tank. • Pumping of NaHO. • Ventilation pumps for different tanks.
HP and LP LNG	• Pumping of pilot fuel.

7. Discussion

In this chapter the flaws and strengths of the chosen approach for this analysis are discussed. Initially the assignment was to find the energy consumption for all components excluding the engine. The purpose was for DNV GL Maritime advisory have more insight to compare the advantages and disadvantages of these systems.

While the energy consumptions of the systems have been estimated as expected, uncertainties and imprecisions need to be accounted for.

Regarding all systems:

An important weakness in this thesis is the lack of precise estimates for the actual energy required to pump the auxiliary fluid through the heat exchanger and to warm it up when it acts as a heat source.

All the engines were set to 100% load. It has been noticed since that the LP LNG engine is most efficient at this load, while the other engines are most efficient around 85 % load. This will favor the results for LP LNG over the other technologies.

Pumps:

The results for the pumps hydraulic power are fairly accurate. There are some uncertainties regarding the required pressure increase for some of the systems.

The pressure of scrubbing water entering the scrubber unit may also vary some between the different models. This information was hard to access and was only found in one source. However with so many scrubber suppliers and scrubber sizes, a large number of energy consumption estimations would have been needed to capture the wide range of systems. The choice made in this work was to select a scrubber size that was tailored to the engine considered.

Injection pressure of both high and low pressure LNG has been verified by several sources.

In a real scenario pressure losses in pipes and over different equipment would increase the required work by the pump. These would be higher for HFO that has a higher viscosity than the other mediums being pumped.

If these systems were installed on a ship, there would be an additional positive or negative pump head due to height difference between equipment. This

Actual shaft power will be quite different in a real scenario. The pump efficiency used in this thesis is assumed, and the same is used for all pumps. Efficiency will vary from pump to pump, and depends on the flow rate and pressure.

Combining the pumps of LNG, MGO, HFO simplifies and compresses the analyses. It might be misleading in regards to HFO, where the fuel is pumped to settling tanks in several stages.

Work flow and project planning:

Some aspects of this assignment was underestimated. The magnitude of the work was bigger than expected. Seen in retrospect more precise results could have been produced if the assignment was limited to include fewer of these systems, or one specific component for all systems.

An error in the planning phase that had large consequences was to assume that all required input would be easy and quick to find. Some of the required input are simply not measured, and cannot be found online , also because most of the systems presented in this thesis are not fully mature and have only been installed on few ships if not none. Other information exists but are confidential and needs to be supplied by the manufacturers. While several suppliers were contacted only few data were actually obtained. The scrubber and LNG markets for ships being rather new, it is still highly competitive and suppliers were not ready to share detailed information.

This could have been solved better if the analyses was begun sooner in the process. First when the equations were written it became clear what information were missing. If this was known earlier, emails to manufacturers and experts from DNV GL could have been sent earlier. Other work could have been done while waiting for responses, and responses that have not come yet might have come in time. It would have been clear what information that was accessible and not, and the thesis could have been restructured more accordingly.

8. Conclusion

The aim of this thesis was to do a system-level comparison of energy consumption for maritime SOx reduction technologies.

Some parts of the fuel supply system for HFO are not included, and the cases which the analyzes are based on have some degree of uncertainty. Energy consumption for the components expected to consume most of the energy was found and compared.

I believe the approach used for this analysis will give quite precise results for LNG, Scrubbers and MFO if the assumed values is substituted with verified values. More components must be included for a precise analysis of HFO.

8.1. Results

The results of the analysis show that the fuel supply system for low pressure LNG has close to no energy consumption. High pressure LNG has a slightly higher consumption, but this is still just a small fraction of the engines output power. This estimation is assumed to be quite precise, as these technologies have few parts and most energy consumption are from one pump.

MGO and HFO have consumption ranging from 0,7 % to 1,5 % of the engine power. Almost all of this comes from the high pressure pump. This wide interval is due to the big variation of diesel injection pressure. HFO is assumed to have a larger energy consumption than found in this thesis since many components are excluded. HFO is also assumed to have gained most from pressure losses not being included, since it have higher viscosity than the other fluids and viscosity increases pressure losses.

The scrubbers are consuming more energy than the other ECA compliance options, and open loop scrubbers consumes the most. By them self the open and closed loop scrubbers had a pumping power per cent of engine power cleaned of respectively 1,9-2,23 and 0,88-1,09 %. In addition a vessel using scrubbers have the power consumption of the HFO fuel system. Injection pressure of the scrubbing water entering the scrubber unit was only found in one source. This pressure of 7 bar accounted for a large part of the power required for the scrubbing water pumps. If this pressure appears to be lower, it will reduce the pump power substantially.

8.2. Recommendations

In order to validate the results and reduce the degree of uncertainty, the following steps are recommended:

- Determine each pumps efficiency.
- Substitute the fuel consumption for all engines except the low pressure LNG engine, with the fuel consumption at 85 % load.
- Find out the pressure losses over the different heat exchanger.
- Determine what auxiliary fluid runs through the heat exchanger, their inlet temperature and their outlet temperature. This will allow for a precise calculation of auxiliary fluid flow rate and resulting circulating pump power.
- Include pumping of pilot fuel for LNG. Even though the pilot fuel consumption is low, this might add some because of the high pressures involved.

8.3. Future work

After the above mentioned steps to validate the results are done, the obvious next step for this work would be to link these findings to actual increased operational expenses. OPEX is essentially the driving factor when choosing between these options. A starting point for this could be to set a required engine output. Add to the increased energy consumption and find the difference in fuel consumption. Together with the fuel prices, this can indicate the increase in OPEX.

When the two-stroke high and low pressure engines gets installed on operating ships, it might be possible to obtain detailed drawings and real measured values, which allow for more thorough analysis of these.

A comparison of all pollution from the ECA compliance options should be done. While this thesis only regards technologies reducing SOx, there are also emissions of e.g. NOx and CO2 from combustion engines. A possible new thesis could be to compare the total emission of greenhouse gasses from the high and low pressure LNG engines. There currently media debates regarding the methane slips from Wärtsilä new low pressure two-stroke engine.[13] As methane is a stronger greenhouse gas than CO2, it might be an interesting study.

9. References

Written sources

4. The International Council on Combustion Engines (CIMAC). (2013). Guideline for the operation of marine engines on low sulphur diesel. CIMAC Technical database. 28 pp.
6. Christos Chryssakis, S. S., Peter N. Hoffmann, Tomas Tronstad, Gerd Petra Haugom. (2012). Alternative fuels for maritime applications. DNV GL. 78 pp.
8. Skogestad, S. (2009). *Prosessteknikk: masse- og energibalanser*. Trondheim: Tapir akademisk forl. 387 pp.978-82-519-2457-3
11. Ignatowitz, E. (1999). *Prosesskjemi*. Oslo: Yrkesopplæring. 454 pp.82-585-0954-3
14. Sören Karlsson, T. H., Marko Kuusisaari. (2013). The Wärtsilä Gas Valve Unit enclosed design (GVU-ED™) for marine applications. *In detail, Wärtsilä Technical Journal*: 42-45.
17. Wärtsilä. (2013). *Wärtsilä enviromental product guide*. 2 ed. p. 54.
18. Reynolds, K. J. (2011). Exhaust Gas Cleaning Systems Selection Guide: The golsten associates. 154 pp.
20. Totten, G. E., Westbrook, S. R. & Shah, R. J. (2003). *Fuels and lubricants handbook: technology, properties, performance, and testing*. West Conshohocken, PA: ASTM International. 1087 pp.978-0-8031-2096-9
21. Ford, M. C. (2012). A master's guide to using fuel oil onboard ships. The Standard P&I Club,: Charles Taylor Consulting. 68 pp.
22. DNV GL. Low sulphur fuels. Properties and associated challenges. DNV GL. 12 pp.
27. WCN FGES/ Barry Yang. (2012). LNG FUEL SUPPLY SYSTEM,: Wärtsilä. 50 pp.
30. Turbo, M. D. (2014). *Marine Engines, IMO Tier II*. p. 144.
32. Holland America Line. (2010). Sea water scrubber technology demonstration project on the MS Zaandam. Final Report to the U.S Environmental Protection Agency. 14 pp.
34. Letnes, M. (2013). *Wärtsilä exhaust gas cleaning* Available at: http://www.wartsila.com/ (accessed: 15.02).

Web pages

1. International Maritime Organization (IMO). *Sulphur oxides (SOx) – Regulation 14*. Available at: http://www.imo.org (accessed: 21.01).
2. United States Enviromental Protection Agency. *Sulfur Dioxide, Health*: United States Enviromental Protection Agency. Available at: http://www.epa.gov (accessed: 21.01).
3. Blikom, L. P. (2011). *LNG for Greener Shipping in North America*. Available at: http://blogs.dnvgl.com/lng (accessed: 02.02).
5. Bunkerworld. *Latest prices*. Available at: http://www.bunkerworld.com/prices/ (accessed: 30.04).
7. Lloyds register. (2012). *Understanding exhaust gas treatment systems - Guidance for shipowners and operators*. Available at: http://www.lr.org/.
9. Biezen, M. v. (2013). *The Otto Cycle*. Physics - Thermodynamics (14:14). Available at: youtube.com (accessed: 03.05).
10. Manteufel, R. (2014). *Derivation of thermal efficiency for Diesel Cycle (2)*. Available at: www.youtube.com (accessed: 01.05).
12. Piellisch, R. (2013). *Tote LNG propulsion system*. HHP Insight. Available at: http://hhpinsight.com/.

13. Stensvold, T. (2014). *LNG-produsentene dypt uenige om totaktsmotorer*. Teknisk Ukeblad. Available at: http://www.tu.no/industri/2014/02/24/lng-produsentene-dypt-uenige-om-totaktsmotorer.
16. Strøm, T. O. A. (2014). *Email correspondence* (28.03).
19. Machinery Spaces. (2010). *The fuel oil system for a diesel engine* Available at: http://www.machineryspaces.com/.
23. Man Diesel & Turbo. (2010). *Operation on low-sulphur fuels, MAN B&W Two-stroke engines*. Available at: http://www.mandieselturbo.com/1015012/Press/Publications/Technical-Papers/Marine-Power/Low-Speed/Operation-on-Low-Sulphur-Fuels.html.
26. Wärtsilä. (2014). *Wärtsilä RT-flex50DF*. Available at: http://www.wartsila.com (accessed: 20.01).
29. The engineering toolBox. *Ethylene Glycol Heat-Transfer Fluid*. Available at: http://www.engineeringtoolbox.com/.
31. Laursen, R. S. (2012). *Dual Fuel ME-GI Engine, Performance and the economy*: MAN Diesel & Turbo. Available at: http://www.ngvaeurope.eu/images/MAN-Rene-Sejer.pdf.
36. Caltex Australia. (2011). *Marine Gas Oil (diesel)*. Available at: http://www.caltex.com.au (accessed: 01.05).
37. Man Diesel & Turbo. *Guidelines for Fuels and Lubes Purchasing, Operation on Heavy Residual Fuels*. Available at: http://www.mandiesel.com (accessed: 28.04).
38. The engineering Toolboox. *Properties of Saturated Steam*. Available at: http://www.engineeringtoolbox.com/.

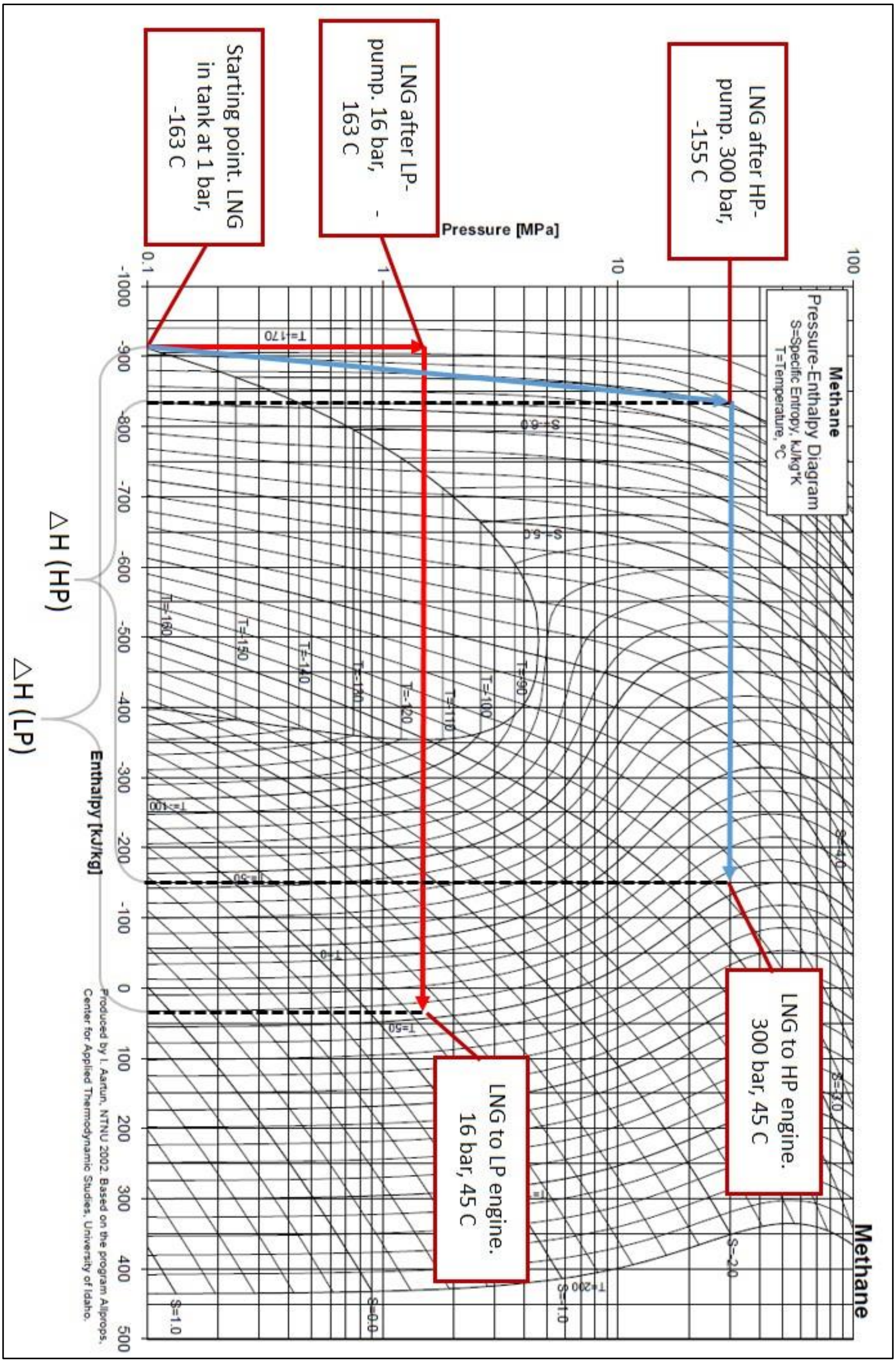
Confidential material

15. LNG. Confidential source nr.1 project: LNG Ready.
24. LNG. Confidential source nr.2 project: LNG Ready.
25. Scrubber. Confidential source nr.3
28. LNG. Confidential source nr.4. project: LNG Ready.
33. LNG. Confidential source nr.5 project: LNG vs. Scrubber.
35. LNG. Confidential source nr.6 project: LNG Ready.

Appendix 1: List of figures

Appendix 2: List of tables

Appendix 3: Pressure enthalpy diagram, Methane



Appendix 4: ISO 8217 Fuel Standard

DNV MANAGING RISK

ISO 8217 Fuel Standard, Third Edition 2005

For marine distillate fuels and for marine residual fuels.

MARINE DISTILLATE FUELS

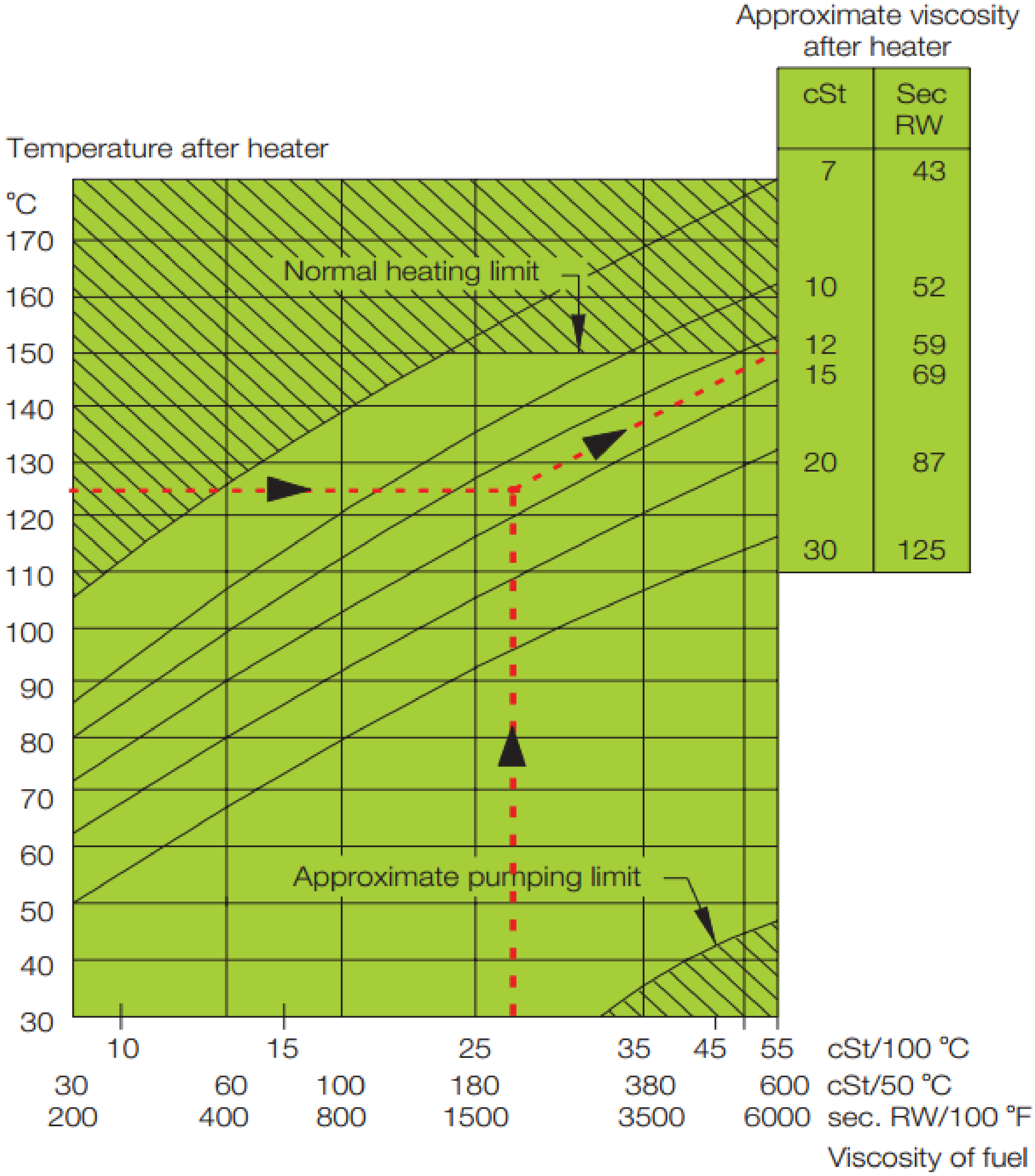
Parameter	Unit	Limit	DMX	DMA	DMB	DMC
Density at 15 °C	kg/m³	Max	-	890.0	900.0	920.0
Viscosity at 40 ° C	mm²/s	Max	5.5	6.0	11.0	14.0
Viscosity at 40 °C	mm²/s	Min	1.4	1.5	-	-
Micro Carbon Residue at 10% Residue	% m/m	Max	0.30	0.30	-	-
Micro Carbon Residue	% m/m	Max	-	-	0.30	2.50
Water	% V/V	Max	-	-	0.3	0.3
Sulfur [c]	% (m/m)	Max	1.0	1.5	2.0	2.0
Total Sediment Existent	% m/m	Max	-	-	0.10	0.10
Ash	% m/m	Max	0.01	0.01	0.01	0.05
Vanadium	mg/kg	Max	-	-	-	100
Aluminium + Silicon	mg/kg	Max	-	-	-	25
Flash point	°C	Min	43	60	60	60
Pour point, Summer	°C	Max	-	0	6	6
Pour point, Winter	°C	Max	-	-6	0	0
Cloud point	°C	Max	-16	-	-	-
Calculated Cetane Index		Min	45	40	35	-
Appearance			Clear & Bright		-	-
Zinc [d]	mg/kg	Max	-		-	15
Phosphorus [d]	mg/kg	Max	-		-	15
Calcium [d]	mg/kg	Max	-		-	30
[c]	A sulfur limit of 1.5% m/m will apply in SOx Emission Control Areas designated by the International Maritime Organization, when its relevant Protocol comes into force. There may be local variations					
[d]	The Fuel shall be free of ULO. A Fuel is considered to be free of ULO if one or more of the elements are below the limits.All three elements shall exceed the limits before deemed to contain ULO.					

MARINE RESIDUAL FUELS

Parameter	Unit	Limit	RMA 30	RMB 30	RMD 80	RME 180	RMF 180	RMG 380	RMH 380	RMK 380	RMH 700	RMK 700
Density at 15 °C	kg/m³	Max	960.0	975.0	980.0	991.0		991.0		1010.0	991.0	1010.0
Viscosity at 50°C	mm²/s	Max	30.0		80.0	180.0		380.0			700	
Water	% V/V	Max	0.5		0.5	0.5		0.5			0.5	
Micro Carbon Residue	% m/m	Max	10		14	15	20	18	22		22	
Sulfur [c]	% m/m	Max	3.5		4.00	4.50		4.50			4.50	
Ash	% m/m	Max	0.10		0.10	0.10	0.15	0.15	0.15		0.15	
Vanadium	mg/kg	Max	150		350	200	500	300	600		600	
Flash point	°C	Min	60		60	60		60			60	
Pour point, Summer	°C	Max	6	24	30	30		30			30	
Pour point, Winter	°C	Max	0	24	30	30		30			30	
Aluminium + Silicon	mg/kg	Max	80		80	80		80			80	
Total Sediment,Potential	% m/m	Max	0.10		0.10	0.10		0.10			0.10	
Zinc [d]	mg/kg	Max	15									
Phosphorus [d]	mg/kg	Max	15									
Calcium [d]	mg/kg	Max	30									
[c]	A sulfur limit of 1.5% m/m will apply in SOx Emission Control Areas designated by the International Maritime Organization, when its relevant Protocol comes into force. There may be local variations.											
[d]	The Fuel shall be free of ULO. A Fuel is considered to be free of ULO if one or more of the elements are below the limits.All three elements shall exceed the limits before deemed to contain ULO.											

Source: ISO 8217 Third Edition 2005-11-01
Petroleum products - Fuels (class F) - Specifications of marine fuels

Appendix 5: Heating table for marine fuels



Source: Man Diesel and Turbo: Guidelines for Fuels and Lubes Purchasing Operation on Heavy Residual Fuels. Page 13.

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