Analysing the effects variable injection and exhaust valve timing have on a two-stroke diesel engine

Bachelor thesis in marine engineering

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Sammanfattning

En tvåtakts dieselmotor är den i särklass vanligaste framdrivningsprincipen för stora fartyg. På grund av motorns storlek, förbrukar den en väldigt stor mängd bränsle per dag. Så därför, bara genom att förbättra effektiviteten på motorn marginellt, finns det stor potential att spara både på miljön och minska bränslekostnaderna. Genom att ställa in bränsleinsprutningen och avgasventilens öppning och stängningstider kan man åstadkomma en bättre effektivitet på varierande laster.

Det undersöktes om det var möjligt att köra motorn över på olika delar utav motorns varvtal/last diagram och hur motorns parametrar behövde injusteras för att kunna utföra detta. I rapporten undersöktes även den specifika bränsleförbrukningen och hur utsläppen av kväveoxider påverkades.

Metoden för att utföra detta bestod ut av en MATLAB kod som involverar en validerad dimensionslös två takts motor. I simuleringarna för att hitta den bästa bränsleförbrukningen justerades de olika timingarna och antagande om hur mycket luftflöde motorn erfordrade gjordes också på motorns maximala varvtalskurva.

Resultaten visar att med ett variabelt system kan besparingar upp till 1.8 % göras på bränsleförbrukningen men på lägre laster blev denna siffran lägre. För kväveoxider, sågs en reducering på 1.3 % i utsläpp. Att köra motorn på maximala vridmomentskurvan och propeller design kurvan krävde en senare stängningstid på avgasventilen och en tidigare insprutning utav bränslet. För att kunna köra motorn på maximala varvtalskurvan krävdes även här en senare stängning utav avgasventilen men även en senare insprutning utav bränslet.

Nyckelord: Tvåtakt, diesel, tuning, optimering, variable ventil timing

Abstract

The two-stroke marine diesel engine is the most common propulsion system in large ships. They consume a large amount of fuel per day and therefore by improving the engine efficiency just slightly will make a difference in costs of fuel oil and in the environmental department. By tuning the engine with variable valve and exhaust timing, so it gets the highest efficiency over various loads, is one way to accomplice that.

It was investigated if the engine would be able to run over the entire engine speed/load diagram and how the engine needed to be tuned for that. It was also investigated how the specific fuel oil consumption and nitrogen oxide emissions would be affected.

The simulations involved changing different timings and flows of the engine and to be able to run at the speed limit, assumptions of how much mass flow of air the engine would need were done. All the simulations were done with a MATLAB code of a two stroke zero-dimensional engine which is validated.

The results show that with a variable timing setup compared to a fixed one could save up to 1.8 % fuel in terms of specific fuel oil consumption and with a lower load the savings in of specific fuel oil consumption were less. In nitrogen oxides emissions the possible reduction could be as high as 1.3 %. It also showed that to run on the torque limit curve and propeller design curve a later exhaust valve closing and earlier start of injection were preferred but on the speed limit curve it was necessary to get a later exhaust valve closing but also a later start of injection.

Keywords: Two-stroke, diesel, tuning, optimizing, variable valve timing

Preface

We would like to thank our supervisor Ulrik Larsen for all the support in this thesis, Anders Olsson for interesting discussions and Stacy Mahon for helping us with the English language.

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Nomenclature

ABDC	*After bottom dead center*
AFR	*Air/fuel ratio*
BDC	*Bottom dead center*
CAD	*Crank angle degrees*
CI	*Compression ignition*
cSt	*Centistoke*
EVC	*Exhaust valve closing*
EVO	*Exhaust valve opening*
EOI	*End of injection*
FMEP	*Friction mean effective pressure*
g/kWh	*Gram per kilowatt hour*
MCR	*Maximum continuous rating*
NOx	*Nitrogen oxides*
Pcomp	*Compression pressure*
PM	*Particulate matter*
PMax	*Maximum compression pressure*
SFOC	*Specific fuel oil consumption*
SOI	*Start of injection*
SOx	*Sulphur oxides*
TDC	*Top dead center*

1 Introduction

Today the world is depends heavily on the diesel engine. It is used as prime mover for transportation of goods worldwide, produces power to the worlds equipment and in combination with a generator it also produces electricity (Verschaeren et.al., 2014). Even though alternative fuels and alternative propulsion systems are tested on ships, the fossil fuels will still be the most common fuel for propulsion for the upcoming ten years at least.

The diesel engine's disadvantage is that it is a big contributor to environmental pollution, mostly sulphur oxides (SOx), nitrogen oxides (NOx) and particulate matter (PM) and that is because the air and fuel is not premixed before entering the combustion chamber and also the short mixing time inside the combustion chamber (Lloyd, Cackette, 2001). The environmental effect the shipping industry has is a subject that is discussed more frequently today. The combustion engine burns fuel to produce mechanical energy, when the fuel is burnt exhaust emissions are produced. The harmful exhaust emissions need to be kept to a minimum and ever since 1997, the pollutant emission have been regulated by MARPOL (Prevention of Air Pollution from Ships, n.d). The revised MARPOL Annex VI regulates the SOx, NOx and PM emissions. These emissions increase the risk of cancer on humans and animals, visibility degradation, contributes to the greenhouse effect and acid rain etc (Lloyd & Cackette, 2011). Studies show that the shipping industry alone stands for about 15 % of the worlds nitrogen oxides emissions and 3-7 % of the worlds sulphur oxide emissions (Neef, n.d). With such a large impact on the environment the engines need to be optimized to keep the emissions as low as possible. Today the latest regulation for NOx is the Tier III which is applied to ships built after 1st January 2016 and they have to run inside certain areas called emission control areas (ECA). The Tier III NOx emissions got reduced by over 75 % compared to the Tier II so lately the regulations become more strict.

The two-stroke diesel is the most common engine used as propulsion plant in large ships due to its high efficiency and reliability (Mondejar et. al., 2018). Even though the diesel engine has been around for over a century, engine designers still find ways to optimize it and thus lower fuel oil consumption and emissions. There are different ways to reduce the emissions either by an after-treatment system or in the engine design. After-treatment systems can reduce the emissions to a large extent by making the NOx react with costly chemicals, but the disadvantages are that the systems are large, require maintenance and are also expensive. (Wang et.al., 2005)

One way to adjust the amount of NOx emissions produced is to have a variable injection timing system. An early injection will provide less soot but higher NOx emissions due to the fact that cylinder pressure and temperature will be higher (Raeie, Emami, Sadaghiyani, 2014). Another possible way is to advance the exhaust valve closing (EVC) and by doing so, the effective compression stroke will be lowered and therefore less pressure and temperature will build up inside the cylinder.

This thesis will investigate how the engine is affected in terms of pressures, SFOC, NOx and other performance parameters by changing the exhaust valve and injection timings.

1.1 Purpose

The purpose is to find out how variable valve timing on the exhaust valve and variable injection timing effects the specific fuel oil consumption and relevant engine parameters on some which points in the engine speed/load diagram

1.2 Research question

How are the valve and injection timings adjusted, to be able to run the engine on different curves in the engine speed/load diagram.

How much difference in terms of SFOC and NOx are there between traditional fixed exhaust valve timing and injection timing compared to the variable injection and exhaust valve timing?

1.3 Delimitations

This thesis will only look at the low speed two-stroke diesel engine because the four-stroke engine uses a different principle and that would be too comprehensive to include given the time frame of this thesis. Will only look at 100 and 75 % load of the engine and in terms of environmental affect only NOx will be looked at.

2 Background

In the sections 2.1 and 2.2 in the background chapter the history and the basic operating principle for a two-stroke compression ignition engine is described. Later on, it is described how the engine is affected by different quality of fuel, timings and temperatures and other engine parameters.

2.1 Introducing the diesel engine

Rudolf Diesel demonstrated the first working prototype combustion engine using the diesel cycle in 1897, it was a one-cylinder engine and ran at 172 revolutions per minute (RPM) and produced 14.7 kW. The specific fuel consumption (SFOC) of the engine was 312 g/kWh and had a thermal efficiency of 26.2 %. The most popular engine of that time was the steam engine which only had a thermal efficiency of about 10 % so Rudolf's combustion engine invention had more than doubled the efficiency (Jääskeläinen, 2013) It did not take many years for the diesel engine to be used as propulsion plants in ships and in 1910 the ordinary steam engine started to be replaced by the much better efficiency four stroke diesel engine. (Latarche, 2017)

The first turbocharged two-stroke crosshead diesel engine as we know today was installed as propulsion system in 1952. The crosshead design allows the engine to be built with a very long stroke which makes it possible to burn more fuel per cycle due to more air trapped inside the cylinder and more power can be produced. The two stroke crosshead engines are not very sensitive to lower grade fuels as the four-stroke diesel engine is. (Dragsted, 2013)

2.2 Working principle for the two-stroke diesel engine

The two-stroke crosshead diesel completes the full combustion cycle in one revolution of the crankshaft. It converts the chemical energy in the fuel to rotating mechanical energy.

1. Piston is moving towards the top dead center (TDC) and starts to compress the air trapped in the cylinder when the exhaust valve closes. This is called the compression stroke.

2. A few crank angle degrees (CAD) before the piston reaches the TDC fuel is injected into the cylinder, due to the heat from compression the fuel injected will self-ignite and put work on the piston and press it downwards. This is called the expansion stroke.

3. About 110° after TDC the exhaust valve will open and let the exhaust gases out from the cylinder.

4. When the piston reaches the lower part of the cylinder it will reveal and open the scavenging ports and compressed air from the turbocharger will enter the cylinder and flush out the remaining exhaust gasses and fresh air will enter. As the piston starts to move towards TDC again the scavenging ports will close, and compression will occur again when the exhaust valve closes. (The basics The two stroke cycle n.d.)

2.3 Emissions from combustion process

Combustion of fossil fuel generates exhaust emissions such as nitrogen (N2), carbon dioxides (CO2), water (H20), oxygen (O2) and pollution emissions which include carbon monoxide

(CO), hydrocarbons (HC), nitrogen oxides (NOx) and particulate matter (PM). Although the pollutant emissions are less than 1 % of the exhaust gases, they must be reduced to absolute minimum because of environmental and health effects.

In theory, an ideal combustion reaction of diesel fuel would only generate CO_2 and H_2O. (Prasad, Bella, 2011). But in reality, it is not possible to obtain this due to numerous things such as (air/fuel ratio, injection timing, turbulence in the combustion chamber, shape of the combustion chamber, combustion temperature and all the other important running parameters.) (Reşitoɫlu, Altinişik, Keskin, 2015).

Nitrogen oxides (NOx)
The air that enters the combustion chamber consists of mostly oxygen and nitrogen, when the fuel is injected in the combustion chamber it will start to combust due to the high temperature of the compressed air. If the temperature doesn't exceed 1600°C the nitrogen in the air and the oxygen in the combustion chamber will not react with each other. However, if any part of the gasses in the cylinder becomes higher than 1600°C then the heat will cause the nitrogen and oxygen to react with each other and NOx emissions will be produced.

There are three major factors to the NOx formation, maximum temperature in the cylinder, residence time and the oxygen concentration. The combustion has its peak temperature in the early stages when the piston is close to TDC, and for every 100°C increase the NOx emissions will increase three times. (Reşitoɫlu, Altinişik, Keskin, 2015). It is therefore possible to lower the NOx emissions by carefully tuning the engine to get lower internal temperatures.

2.4 Important engine parameters

2.4.1 Fuel

The particulate matter (PM) emissions from the combustion process of an engine are determined by mainly two different factors, which fuel and lubrication oil that is burnt and also the design and running conditions of the engine. These PM emissions consist of several of different compounds, such as carbon, sulphate, inorganic compounds of different metals. (Moldanová et. al., 2013)

2.4.1.1 Viscosity

To maintain a good atomization of the fuel spray injected by the nozzle, the viscosity is frequently gauged. The unit is centistoke (cSt), the greater the cSt value is, the thicker and more viscous the fuel is. For the two stroke crosshead engines, a normal value of the viscosity is below 15 cSt. To maintain the right viscosity, the temperature of the fuel is adjusted, by either cooling or heating up the fuel. (Kuiken, 2012)

2.4.1.2 Calculated Carbon Aromaticity Index (CCAI)

Due to the high demand and low access of fuel, power plants and ships are forced to use fuel of varying quality. This is due to the increased request of light fuels. Which is why the introduction of using the Calculated Carbon Aromaticity Index (CCAI). This was introduced, to present the ignition quality of the fuel. Heavy fuel oils with long aromatic compounds tend to have inferior characteristics as longer ignition delay. (Kuiken, 2012)

2.4.2 Injection timing

According to Kuiken (2012), the best way to reduce NO_x is to use a delayed injection valve opening timing. Later injection timing reduces the combustion pressure. Less combustion pressure has a limiting effect to the amount of NO_x formation due to the reduced temperature and pressure.

When increasing the duration of the injection, by extend the injection time by 10 % with delayed injection valve closing reduces the formation of NO_X by 5 % (kuiken, 2012).
By prolong the injection time, the maximum cylinder pressure (Pmax) is reduced.

2.4.3 Exhaust valve timing

The two stroke crosshead diesel engines with uniflow scavenging is fitted with one exhaust valve in the cylinder head (Kuiken, 2012). The exhaust valve in the MAN 12K98ME is electrohydraulic controlled and thereby it is possible to freely adjust the opening and closing timing of the exhaust valve. (MAN Marine Engines & Systems. 2014).

2.4.4 Compression ratio

The compression ratio (r_c) is the ratio that determines the relationship between the cylinder volume when the piston is positioned in BDC relative to the cylinder volume when the piston is positioned in TDC. (Heywood, 1988).

$$r_c = \frac{maximum\ cylinder\ volume}{minimum\ cylinder\ volume} = \frac{v_d + v_c}{v_c}$$

Where V_d equals to the volume above the piston when in BDC and V_c is the volume above the piston when it is positioned in TDC

2.4.5 Effective compression ratio

The effective compression ratio ($\varepsilon_{effective}$) is a more realistic way to find out how much the gasses inside the cylinder are compressed, since it takes into account when the compression actually starts. Although the piston is moving toward TDC, compression of the gasses will not begin until the exhaust valve and scavenge ports are closed. Which means that, unlike the compression ratio, the effective compression ratio is affected by the travelled distance of the piston after BDC where the closing of the exhaust valve occurs. (Kuiken, k. 2012)

$$\varepsilon_{effective} = \frac{(1-k)\ v_d + v_c}{v_c}$$

Here k is the relation between V_d and the total volume between when the piston is in BDC and when piston is in the position where the exhaust valve closes, V_d and v_c is the same as in chapter 2.4.4.

By changing the effective compression ratio by either using a more advanced or delayed exhaust valve timing, it is possible to maintain different compression pressures.

2.4.6 Compression pressure (Pcomp)

When the piston moves towards TDC, the air trapped in the cylinder will increase in both temperature and pressure due to the decreasing volume. The harder the air is compressed the bigger will the rise in pressure and temperature be. This physical property is important for the function of the engine because the fuel needs the heat from the compression work to ignite. (kuiken, 2012)

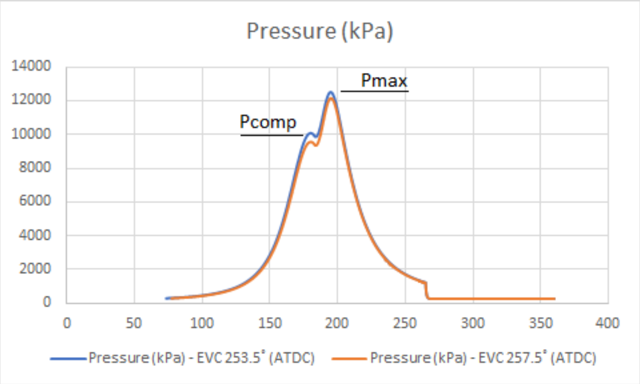

Figure 1: Pressure diagram (Pcomp)

Figure 2 illustrates the pressures in the cylinder during one engine cycle. There are two different running scenarios, the blue curve has an EVC timing of 253.5˚ and the orange one has a later EVC timing, at 257.5˚.

As shown in figure 2, by closing the exhaust valve later, the compression work will begin later. When the compression stroke begins later, less air will be trapped in the cylinder, which result in a lower compression pressure (Pcomp) due to the lower effective compression ratio.

2.4.7 Maximum cylinder pressure (Pmax)

When the combustion starts, the pressure and temperature are increasing in the cylinder, the highest pressure the combustion reaches is then the Pmax. Since the combustion raises the pressure that is before the combustion occur, the Pmax is affected by the Pcomp, which is illustrated in Figure 2.

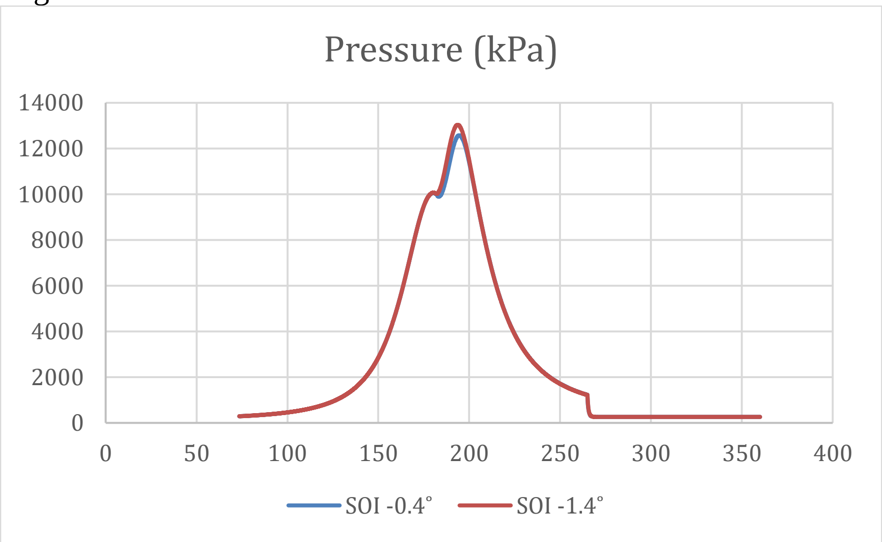

Figure 2: Pressure diagram (Pmax)

Figure 3 illustrates the pressure in the cylinder during one revolution. The start of injection (SOI) was the only parameter that was changed. The red curve is representing 1˚ advanced SOI.

Unlike the Pcomp, the Pmax is also affected by the injection timing and the amount of injected fuel. As illustrated in Figure 3, earlier SOI increasing the Pmax.

2.4.8 Air/Fuel ratio (AFR)

One important engine running parameter is the ratio between the amount of injected fuel and the quantity of trapped air.

$$AFR = \frac{Air}{fuel} ratio \left(\frac{A}{F}\right) = \frac{\dot{m}_a}{\dot{m}_f}$$

Where the mass flow rate of air $\dot{m}_a$ is divided by the mass flow rate of fuel $\dot{m}_f$. The normal air/ fuel ratio operating range for combustion ignition (CI) engines fuelled by diesel is:

$18 \leq A/F \leq 70$ $(0.14 \leq F/A \leq 0.056)$ (Heywood, 1988)

If the AFR value is low, it means that the combustion process appears with a rich amount of fuel relation to quantity of air. This can result in an excessive amount of exhaust smoke and low efficiency due to that the engine will not be able to combust all the fuel.

2.4.9 Specific fuel oil consumption (SFOC)

Combustion engines convert chemical energy stored in the fuel to work on the crankshaft. Due to the efficiency of the engine, all the energy that is injected with the fuel is not converted to power on the crankshaft. To get the specific fuel oil consumption, the mass flow of fuel ($\dot{m}_f$) is divided with the power output (P)

$$sfoc = \frac{\dot{m}_f}{P}$$

The specific fuel oil consumption is then used, to compare the ability between different engines to convert fuel oil to power on the crankshaft. The lower the SFOC value is, the more energy from the fuel is converted to mechanical energy.

2.4.10 Scavenge pressure

The pressure difference between the exhaust gas manifold and the scavenge air manifold forces the scavenging process to occur. This pressure difference is maintained by the scavenging air compressor. To achieve a satisfactory scavenging process, the pressure inlet air pressure must be high enough to exceed the backpressure in the exhaust manifold by a certain margin.

2.4.11 Exhaust temperature

When the combustion process occurs, the temperature in the cylinder can reach 2000˚C. Then the gases later pass the exhaust valve before entering the turbine, the exhaust gases are approximately 325-600˚C. (Kuiken, 2012)

The exhaust temperature is under influence of the engine running parameters and the engine design.

2.4.12 Friction

In a combustion engine there will always be friction losses and can be divided into three major subjects, the piston cylinder unit, the crankshaft and the pumps and then there is also some minor losses from gear drive and the valvetrain. The piston cylinder unit losses are about 30-40 % of the total frictional losses and in this category losses from piston rings, piston skirt and connecting rods are all taken into account for. (Krishnan, 2014)

3 Method

To be able to answer the research question a MATLAB code by the means of a zero-dimensional engine model was used. The first step was to analyze the operational envelope for the engine and from there it was decided that the torque limit curve, heavy propeller curve (engine design curve), light propeller curve (propeller design curve) and speed limit curve were the most interesting specific points to use for comparison between a fixed valve/injection system and a variable valve/injection system (see Fig. 4).

Two different load settings of the engine were investigated, 100 % and 75 %. For a more realistic scenario the turbocharger had to be scaled and optimized for a certain load and rpm, which means that when the load later on were to be changed the turbochargers properties were exactly the same, and that was done at the specified engine MCR. The mass flow of air and fuel for both of the reference points, which is the blue dots in Figure 2, were set to be as engine manufacturer instructed. To find the injection valve and exhaust valve timings to match the given power output from the specified engine MCR the Pmax, Pcomp, SFOC and NOx were the objectives to be as close as the engine manufacturer instructed, with the two first mentioned to be the priority. To determine the rpm for the propeller design curve, torque limit curve and speed limit curve the engine/load diagram seen in Figure 2 were used in combination with the formula below.

$$P = c * n^3$$

P is for power output, c is a constant, and n is the rpm.

For each percent increase or decrease in engine rpm at the same engine power, the mass flow of air the compressor should provide to the engine, was either increased or decreased the same percentage to simulate that the engine needs more mass flow of air at higher rpms and vice versa. When mapping the engine for optimization of SFOC on the different curves, the EVO were unchanged in relation to the reference to that degree that the engine model was possible to function. The optimization of the engine involved changing EVC, SOI, EOI and mass flow of fuel and air.

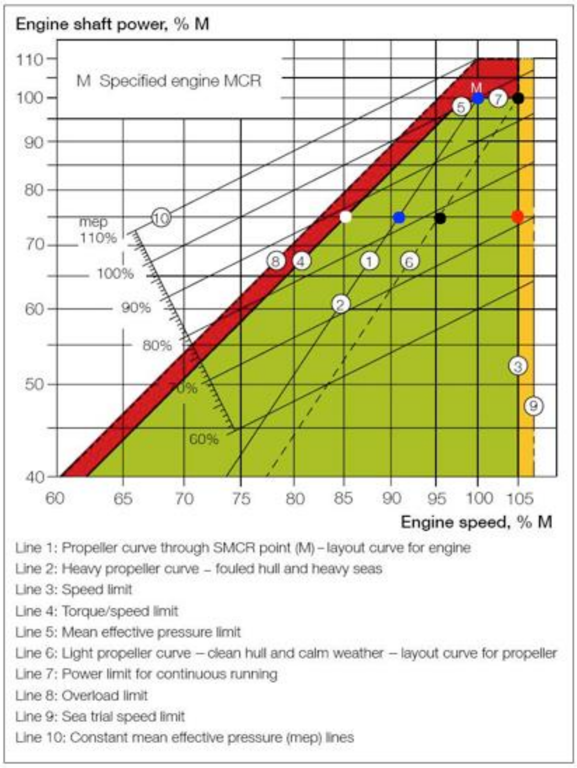

Figure 3: Engine load and speed diagram (source: MAN Diesel & Turbo)

3.1 Describing of the engine model

The engine model which was used were created by Fabio Scappin, Sigurður H. Stefansson, Fredrik Haglind, Anders Andreasen and Ulrik Larsen. The basics of how the model operates is described below but for more detailed information it can be found in the authors paper of the engine model (Scapping, Stefansson, Haglind, Andreasen, Larsen, 2015) and in the report by (Larsen, Pierobon, Baldi, Haglind, Ivarsson, 2015)

3.1.1 Thermodynamic properties of the engine model

The thermodynamic states, for each of the stages that the engine passes (compression, combustion, expansion and blow down) is defined by a couple of differential equations. The thermodynamic states include an energy balance, a mass balance, an equation of state, heat loss correlation and heat release correlation. The general energy balance is as following:

$$\frac{dU}{dt} = \frac{dW}{dt} + \frac{dQhr}{dt} - \frac{dQcl}{dt} + \dot{m}i * hi - \dot{m}o * ho$$

This formula shows that any changes in work (W), heat (Q), time (t) and specific enthalpy (h) will affect the internal energy (U). The ṁ is for mass flow rate and i, o, cl, hr is short for in, out, cylinder losses and heat release.

To determine an approximation value of internal energy, enthalpy and specific heats correlations by Gyftopaulos and Baretta has been used. The correlations can be applied in the temperature range from 300-4000K and therefore it can be applied in the combustion engine. It also assumes that it is ideal gas conditions. When using the ideal gas conditions, studies show that the result is a higher temperature during combustion compared to an experimental value. And as a result of this the NOx formation will be affected significantly. So, to compensate for this error another equation of state (EOS) has been implemented in the model. The EOS that has been used is the Redlich-Kwong EOS and the mathematical model was created by Danov and Gupta and can be further explored in their study (Danov, Gupta, 2001)

3.1.2 Heat losses

The heat losses are extremely complex to calculate in a zero-dimension model since there are so many actions happening during a limited time. For example, variation in volume, turbulence, gas exchange between exhaust and scavenge air, spreading of the flame front and radiation, but it also one of the most important factors that influences the temperature and pressure over the entire engine cycle. The heat transfer can vary up to five MW/m^2 for each centimeter and a difference of temperature in the tenth of a millisecond can be as much as ten MW/m^2. Therefore, a base equation has been setup as following.

$$Qw = \int_{cycle} \sum_{i} U * Ai(T - Ti) * d\theta$$

U is the average heat transfer coefficient, Ai and Ti are the area of the i'th surface and θ is crank angles. In this model the equation above is applied to all surfaces and it is assumed all surfaces get the same heat variation. The heat transfer coefficient that has been used is the Woschni correlation which was developed in 1967 for internal combustion engines. Even though it is an old equation it is still extensively used, and its' validity is well confirmed. For further information what the Woschni correlation heat transfer coefficient is affected by it is explained by (Larsen, Haglind, Gabrielli, Elmegaard (2014).

3.1.3 Combustion

The diesel combustion process can be divided into three phases, pre-mixed combustion, diffusion combustion mixture-controlled and diffusion combustion reaction-kinetically controlled.

- Pre-mixed combustion- The fuel that are injected during the ignition delay makes a mixture with the air inside the combustion chamber which is well mixed, easy to ignite and it burns quickly. During this phase the pressure increase is the highest and the highest pressure is expected to occur, but it is not certain due to the injection timing might alter the pressure peak.

- Diffusion combustion mixture-controlled- This process goes on during the main combustion phase and it affects the pollutant formation. How well the air and fuel are mixed is what controls the combustion rate. When the maximum temperature in the combustion chamber is obtained then the main combustion phase is over.

- Diffusion combustion reaction-kinetically- At the time this phase develops, the pressure and temperature at the flame front have decreased so much, that in comparison to the simultaneously progressing mixture process the chemical reactions is considered as slow. During this phase the still unburnt fuel and intermediate products will be burned here (Merker, Schwarz, Stiesch, Otto, 2014).

Ignition delay

The ignition delay is the time between start of injection (SOI) and when combustion starts. To be able to simulate this, a model which was presented by Hardenberg and Hase has been used (Heywood, 1988).

$$\Delta\theta id = (0{,}36 + 0{,}22\bar{v})\exp\left[Ea(\frac{1}{RTcomp} - \frac{1}{17900})(\frac{21.2}{Pcomp - 12.4})^{0,63}\right]$$

This equation gives us the ignition delay in terms of crank angles degrees, $\bar{v}$ is the mean piston speed, Ea is the activation energy, R is the universal gas constant, id is for ignition delay and comp is for compression. The activation energy is determined by the following formula.

$$Ea = \frac{618{,}840}{CN + 25}$$

CN is the cetane number in the fuel, by having the cetane number taken in consideration when getting the activation energy, it enables the possibility that the model can take fuel type into account also to a certain degree.

Heat release

The function used for simulation of the heat release is a reworked version of the one Ivan Wiebe came up with in 1962. The reworked version gives a better prediction for diesel engines due to the fact that it takes both the premixed and diffusion-controlled phases into account, the equations are listed below (Miyamoto, Chikahisa, Murayama, Sawyer, 1985).

$$\frac{dQ}{\omega dt} = 6.9\frac{Qpc}{\Delta\theta pc}(Mpc + 1)(\frac{\theta}{\Delta\theta pc})^{Mpc}\,exp\left[-6.9(\frac{\theta}{\Delta\theta pc})^{Mpc+1}\right]$$
$$+6.9\frac{Qdc}{\Delta\theta dc}(Mdc + 1)(\frac{\theta}{\Delta\theta dc})^{Mdc}\,exp\left[-6.9(\frac{\theta}{\Delta\theta dc})^{Mdc+1}\right]$$

Qpc is the heat released during the premixed phase, Qdc is the heat released during the diffusion-controlled phase. $\Delta\theta$ is the duration of combustion expressed in CAD, ω is angular velocity, M is a parameter for the shape of combustion chamber.

Two-zone combustion model

To simulate how the combustion occurs over time is complex, the solution used here is a so called two-zone combustion model. The two zones consist of the reaction zone which the combustion contains and the other is the unburnt gas zone containing the remaining fuel and gasses. The reason for the two-zone combustion model is to get a better prediction of the engines SFOC and NOx emission with altering injection timings and amount. The mass ratio of oxygen to fuel is kept as a constant over the entire combustion event

Combustion species

To be able to do a prediction of the NOx formation the gas composition that is trapped inside the cylinder would have to be as realistic as possible. The gas model consists of 11 elements and those are O2, N2, CO2, H2O, H, H2, N, NO, O, OH and CO. Although the model can predict the NOx formation it does not take particulate matter or sulphur content in the fuel into consideration, so an improvement to NOx formation could lead to a deterioration in PM.

3.2 Nitrogen oxides formation and modelling

NOx is the common term that includes both nitrogen monoxides and nitrogen dioxides. The combustion engine produces both these elements but mainly nitrogen monoxides. The formation of NO during combustion is complex but to simplify it the NO is divided into four, fuel NO, prompt NO, thermal NO and through N20.

- Prompt NO- This kind of NO formation is not used in the model due to that the prompt NO from a diesel engine combustion is so small that it does not need to be taken into account (Goldsworthy, 2003) and due to that, it is not used in the model.

- Fuel bound nitrogen- When running a diesel engine on HFO the fuel bound nitrogen could contribute as much as 10 % of the total NOx production (Geist, Holtbecker, 1997). In other fuels than the residual fuels such as the HFO, the amount of produced fuel bound nitrogen is negligible, so when running on HFO a 10 % increase should be added to the total calculated NOx emissions.

- Thermal NO- This is by far the most important NO producer in a combustion engine, the thermal NO is produced under high temperatures from the atmospheric nitrogen and oxygen. The model which were used to determine the thermal NO is the extended version of the Zeldovich mechanism. The reaction equations are valid for combustion which occurs around the stoichiometric area, with the equation followed below.

$$N2 + O \leftrightarrow NO + N$$

$$N + O2 \leftrightarrow NO + O$$

$$N + OH \leftrightarrow NI + H$$

A quick look at the formulas shows that the formation of NO in the second and third equation is depended on the amount of N in the first equation and therefore the first equation limits the other two. For an even further detailed explanation look at (Larsen, Haglind, Gabrielli, Elmegaard (2014).

3.3 Friction

The friction model that has been used in the model is a model created by Chen and Flynn and it is a simple function but well tested.

$$FMEP = 0.137 + \frac{Pmax}{200} + 162\bar{v}$$

It can be seen that the friction is settled by the maximum pressure and the piston speed ($\bar{v}$).

3.4 Turbocharger

The power estimation required for the compressor is determined by the scavenge air volume flow, compressor inlet pressure, compressor outlet pressure and the isentropic efficiency which is found by the means of a compressor map provided by ABB for the ABB A175. A certain amount of the exhaust gas mass flow to the turbine is always bypassed, to the turbine exhaust, at any load. For the thermodynamic properties of the inlet and exhaust air a method involving NIST Refprop software was used (Lemmon, Huber, McLinden, 2010).

4 Results and analysis

This chapter will present how the engine is tuned by the means of fuel flow, air flow, exhaust valve timing and injection timing when changing through different design curves over the engine envelope. It is also presented how a fixed setting compared to a variable injection and valve timing setting would influence the engines internal pressures, fuel consumption, NOx emissions and other important engine parameters.

The graphs shown in chapter 4.1.1 and 4.2.1 illustrates the difference, expressed in percentage, for NOx emissions, Pmax, Pcomp, SFOC, differential pressure between scavenge air and exhaust air (ΔP Co/Ti), mass of trapped cylinder air, maximum gas temperature and mass of injected fuel. The reference point is when the engine runs at the engine design curve at the indicated load. The variable and fixed settings is when the engine runs on either the propeller design curve, speed limit curve or the torque limit curve at the indicated power.

4.1 Comparison between engine design curve and propeller design curve at 100 % load

- REF= 100 % load running on the engine design curve
- A= 100 % load running on the propeller design curve with fixed injection and valve timings.
- B= 100 % load running on the propeller design curve with variable injection and valve timings.

Table 1. Shows the settings (REF) required for the zero-dimensional model to keep the same Pmax and Pcomp as instructed by the manufacturer at 100 % load. This is also the reference point for the presented graphs in section 4.1.1

	REF	**A**	**B**
RPM *(REVOLUTIONS/MIN)*	104	109.2	109.2
SOI *(CA˚ BTDC)*	-1.2	-1.2	-1.4
EOI *(CA˚ ATDC)*	24.0	24.0	19.0
EVO *(CA˚ ATDC)*	85.0	85.0	85.0
EVC *(CA˚ ATDC)*	254.1	254.1	258.5
SCAVENGE PRESSURE *(BAR)*	3.7	3.7	3.7
AIR *(KG/S)*	17.1	17.9	17.9
FUEL *(KG/S)*	0.275	0.273	0.268
AFR LOCAL	22.9	24.4	23.9
AFR GLOBAL	61.9	65.4	66.6

4.1.1 Graphs to compare the results

As seen in figure 5, when running on the propeller design curve at 100 % load, a reduction of NOx emissions is accomplished by 8.1 % with the fixed setting and 8.5 % with a variable system.

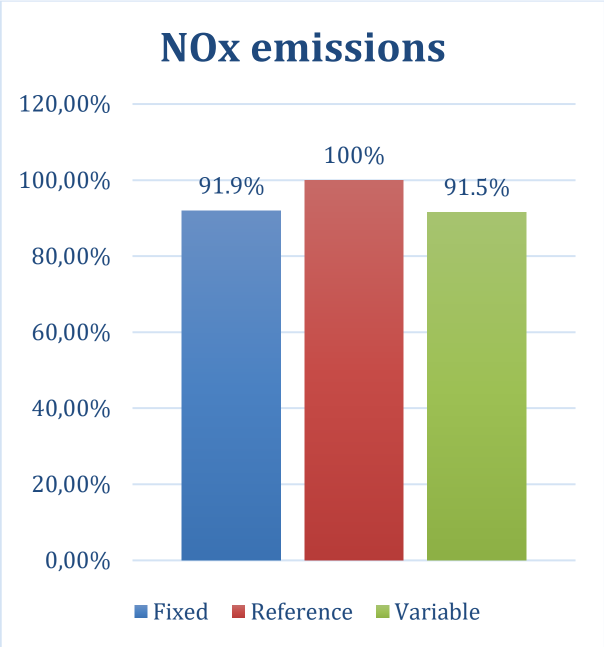

Figure 4: Diagram of the change in NOx emissions at the different curves at 100 % load.

When running on the propeller design curve at 100 % load, a reduction of NOx emissions is accomplished by 8.1 % with the fixed setting and 8.5 % with a variable system. So, the reduction in NOx emissions is approximately the same in both of the cases. The reason for this is due to the lowered maximum cylinder pressure which leads to that the peak temperature of the burning gasses will not be as high and less NOx is formatted.

Figure 6 shows that the variable system indicate a reduction of 4.9 % and the fixed 2.4 % in terms of maximum cylinder pressure.

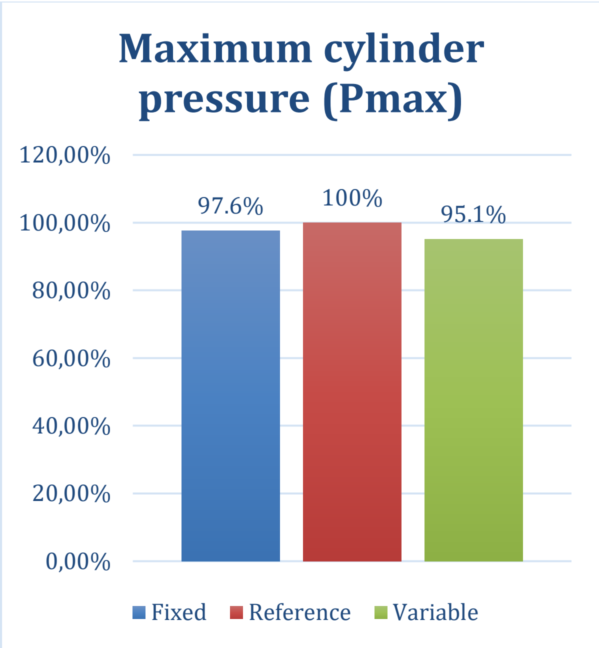

Figure 5: Diagram that shows change in terms of maximum cylinder pressure, at 100 % load.

Even though, the variable system has an 0.2 CAD earlier injection, as seen in Figure 6, which would increase the maximum cylinder pressure, it has the lowest pressure, and that is due to the EVC is retarded and that results in less air will be trapped in the cylinder that can be compressed which can be seen in both Figure 5 and Figure 8. The mass of injected fuel is also lowered in both cases and that is the main contributor to the lowering of the Pmax in the fixed system case.

As seen in Figure 7, whilst the fixed setting still obtains the same compression pressure as the reference point the variable system has a decrease of 6.7 % in pressure and that is directly connected to the retard in EVC by 4.4 CAD

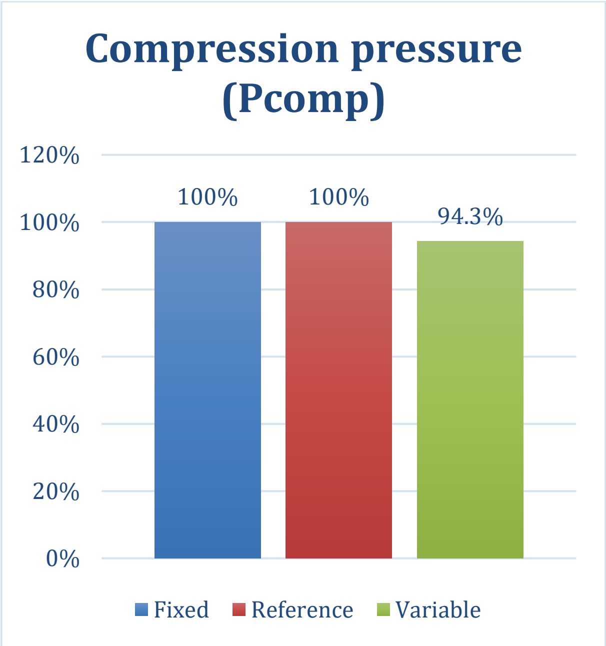

Figure 6: This diagram shows the change in compression pressure, at 100 % load.

Since the closing of the exhaust valve is retarded, the piston will push some of the gasses that are contained inside the cylinder out through the exhaust valve which will lead to less mass of air trapped in the cylinder and then the compression pressure will be lower.

Figure 8 shows that the SFOC is 1.8 % lower in the variable system compared to the fixed and both of the systems have a reduction in SFOC.

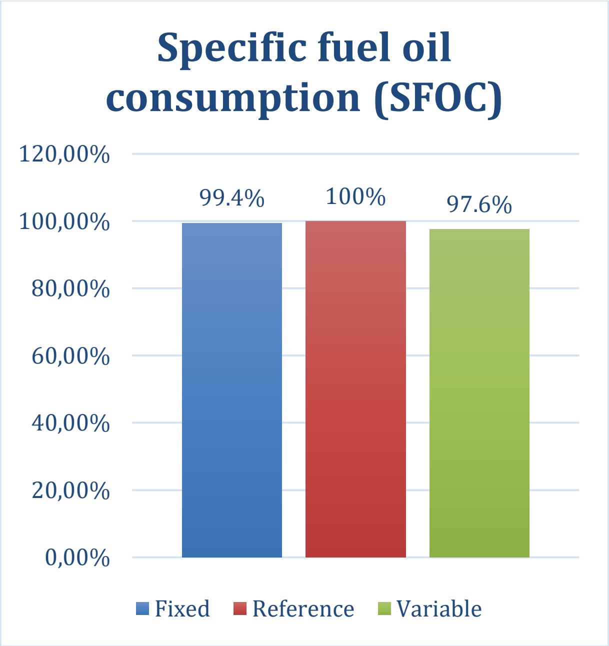

Figure 7: In this diagram, the change in terms of SFOC at 100 % load is illustrated.

The explanation to the reduced SFOC when the speed is increased, as seen in Figure 8, is that the engine in the reference point is mapped for as the engine manufacturers recommendations for Pmax and Pcomp instead of the best SFOC.

Figure 9 shows that the delta-pressure is decreasing by almost 90 % for the fixed system and approximately 66 % for the variable system. The scavenge pressure is a fixed variable in both the engine design and propeller design curve so it is the exhaust backpressure that has increased.

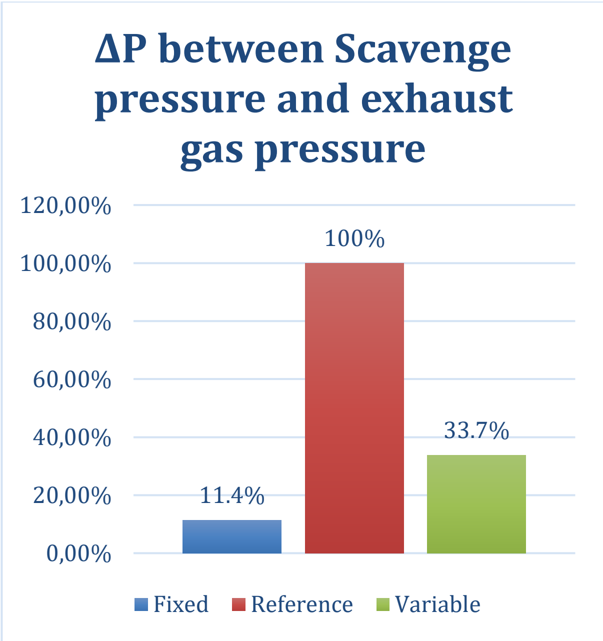

Figure 8: These staples shows the Pressure difference between exhaust and inlet air, at 100 % load.

This staple diagram shows the pressure difference between scavenge air pressure and exhaust gas backpressure. The delta-pressure between these two pressures are the driving force for the scavenging process in combination with the mass flow of the air from the compressor.

There are two possible factors that explain why the exhaust backpressure increases here, the first one is due to the increase in mass flow of air from the compressor and the second is because of the timing of the exhaust valve. If the mass flow of air increases, which is the case in the propeller design curve, the amount of trapped air inside the cylinder also does. With more mass of air inside the cylinder more exhaust gasses will be produced hence the higher exhaust back pressure. The timing of the exhaust also determines the backpressure in the exhaust, a retarded EVC leads to less mass of trapped air and therefore also lower Pcomp and Pmax which is the case in the variable system. So, when the exhaust valve opens the exhaust gasses will have a lower pressure since the EVO is unchanged. That is the reason why the variable system has lower exhaust backpressure compared to the fixed system. The downside of having a later EVC though is that the power output will be less.

As seen in figure 10, The variable system shows a 3.7 % decrease in trapped air but since the mass of injected fuel also is less the AFR stays within range. In the fixed system a 1 % increase is seen due to the mass flow of air is higher.

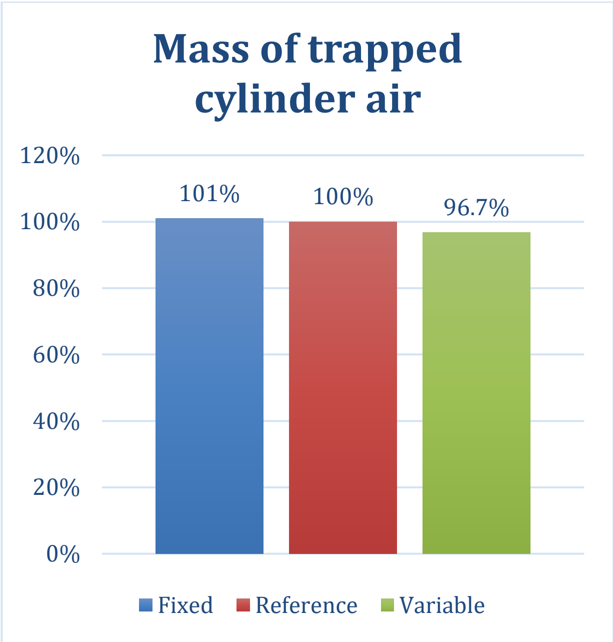

Figure 9**:** This figure shows the mass of trapped cylinder air, at 100 % load.

The amount of mass of trapped air is an indication on how much fuel it is possible to burn, the more air that is trapped, the more fuel can be burned in each combustion stage.

Figure 11 is a diagram that shows the highest temperature of the total gas mass inside the cylinder, but it should not be confused with the peak temperature of the combustion.

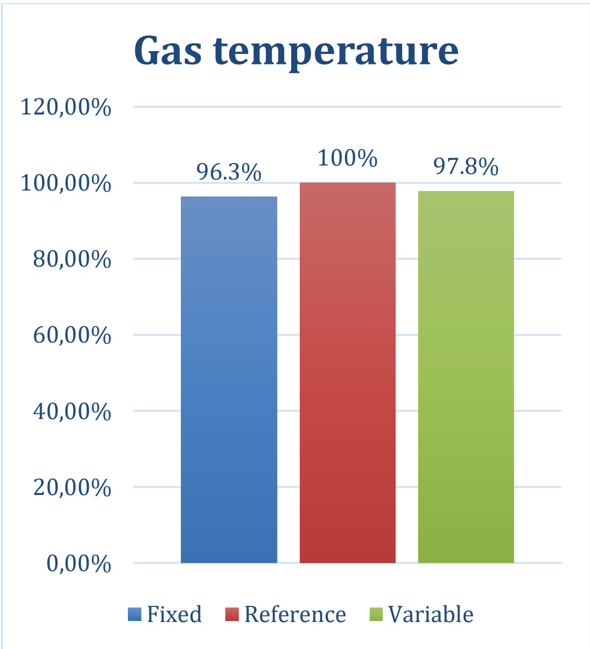

Figure 10**:** Diagram that displays the gas temperature at different speeds, at 100 % load.

As seen in Figure 11, the variable system has less fuel injected and lower Pmax and Pcomp because of the later EVC than the fixed system which should result in a reduction in gas temperature. But since that is compensated for with an earlier SOI and also a shorter injection period which both results in higher temperatures and that is the reason for the variable system to have a higher gas temperature than the fixed system as shown in Figure 11.

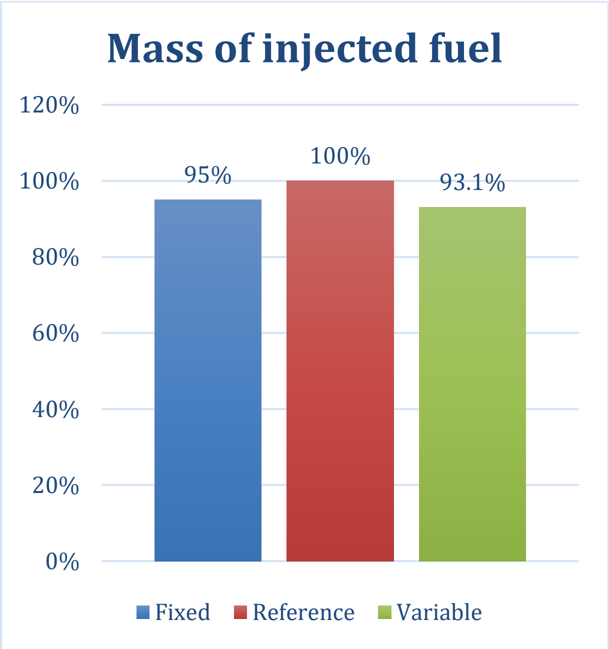

Figure 11**:** This diagram illustrates how much fuel that is injected per power cycle of the engine**,** at 100 % load.

As seen in Figure 12, the total amount of fuel injected is thereby affected by the rpm, also if the power output is kept as a constant. Since the rpm is changed in the propeller design curves but power output is the same the mass of injected fuel per cycle is lowered.

4.2 Comparison at 75 % load

- REF= 75 % load running at the engine design curve
- C= 75 % load running at the propeller design curve with fixed injection and valve timing.
- D= 75 % load running at the propeller design curve with variable injection and valve timing.
- E= 75 % load running at the torque limit curve with fixed injection and valve timing.
- F= 75 % load running at the torque limit curve with variable injection and valve timing.
- G= 75 % load running at the speed limit curve with fixed injection and valve timing.
- H= 75 % load running at the speed limit curve with variable injection and valve timing.

Table 2. Shows the settings (REF) required for the zero-dimensional model to keep the same Pmax and Pcomp as instructed by the manufacturer at 75 % load. This is also the reference point for the presented graphs in section 4.2.1

	REF	**C**	**D**	**E**	**F**	**G**	**H**
RPM *(REVOLUTIONS/MIN)*	94.5	99.2	99.2	88.4	88.4	109.2	109.2
SOI *(CA˚ BTDC)*	-0.4	-0.4	-1.3	-0.4	-1.0	N/A	0.0
EOI *(CA˚ ATDC)*	19.0	19.0	19.0	19.0	19.0	N/A	45
EVO *(CA˚ ATDC)*	87.0	87.0	87.0	87.0	87.0	N/A	83
EVC *(CA˚ ATDC)*	253.5	253.5	256.7	253.5	258.5	N/A	256.5
SCAVENGE PRESSURE *(BAR)*	2.9	2.9	2.9	2.9	2.9	N/A	2.9
AIR *(KG/S)*	12.9	13.5	13.5	12.0	12.0	N/A	14.9
FUEL *(KG/S)*	0.199	0.199	0.196	0.202	0.2	N/A	0.218
AFR LOCAL	23.0	24.4	24.1	20.9	20.5	N/A	24.1
AFR GLOBAL	64.4	67.9	69.0	59.6	60.2	N/A	68.2

4.2.1 *Graphs to compare the results*

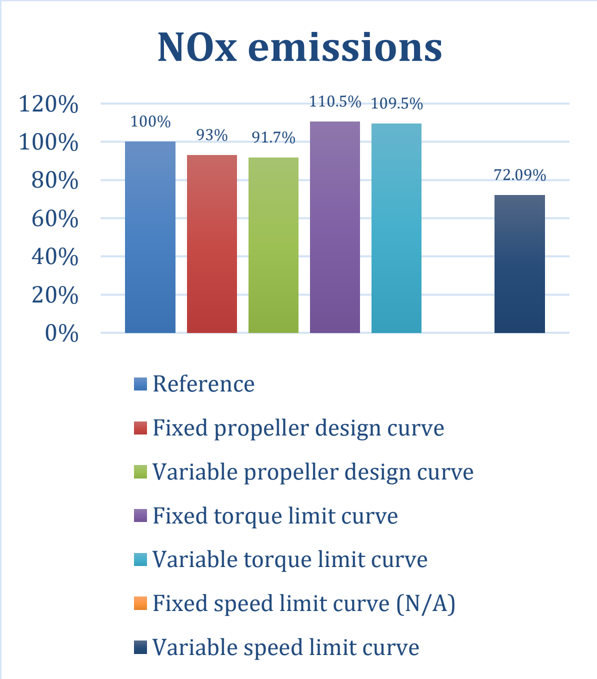

Figure 12: NOx emissions at 75 % load

The NOx emissions were reduced with increased engine speed. When running on high speed and the same load, temperatures and pressures was quite low. On the contrary, the highest amount of NOx emission, was detected at the Torque limit curve with fixed timing. This was due to the high temperature and pressure, which was a result of the reduced cooling effect of the scavenge air, caused by the lowered scavenge air flow. The speed of the engine effected the NOx far more than any other parameters did.

As shown in Figure 14, at 75 % power the variable speed limit curve had the lowest maximum cylinder pressure of all the 7 cases. This was due to the delayed start of injection and the retarded exhaust valve closing as seen in Table 2.

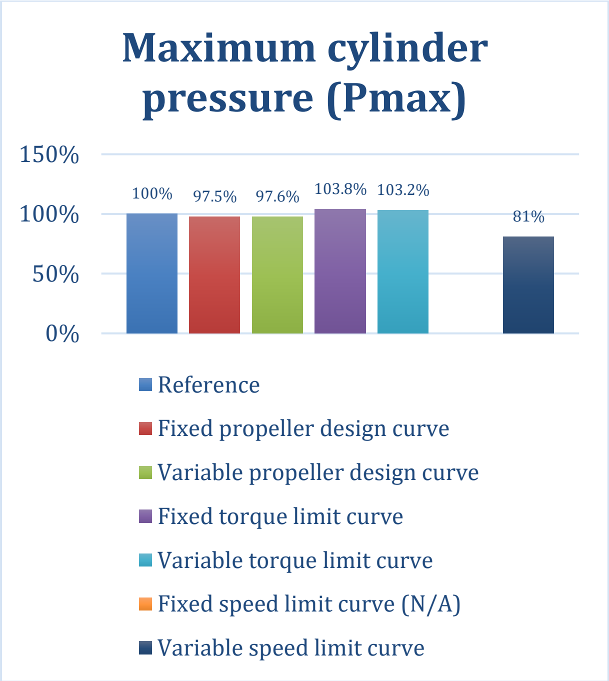

Figure 13: Maximum cylinder pressure at 75 % load.

It was shown that, when decreasing the speed of the engine, the maximum cylinder pressure was increased (Fig 14).
The high (Pmax) on the torque limit curve is explained by the high combustion temperature. When the temperature is increased the pressure is also increased since the air is trapped in the cylinder.

Figure 15 shows that, when applying the variable exhaust valve timing, the compression pressure was decreased up to 4 % below the reference

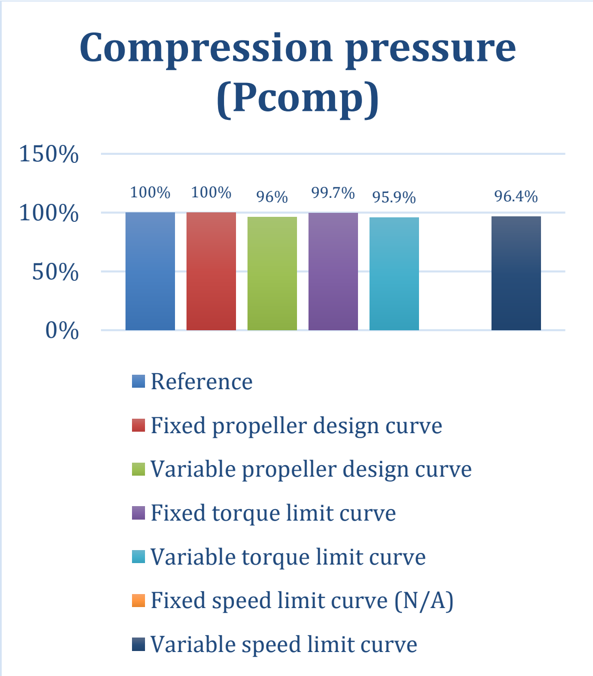

Figure 14: Compression pressure att 75 % load.

The compression pressure was almost the same in all cases with fixed timing as for the reference point. This is explained by that the parameters that effects the compression pressure was not changed..

As seen in Figure 16, when using the variable timing of the fuel injector and exhaust valve at the propeller design curve, it was shown that the specific fuel consumption was reduced by approximately 2 % from the reference point.

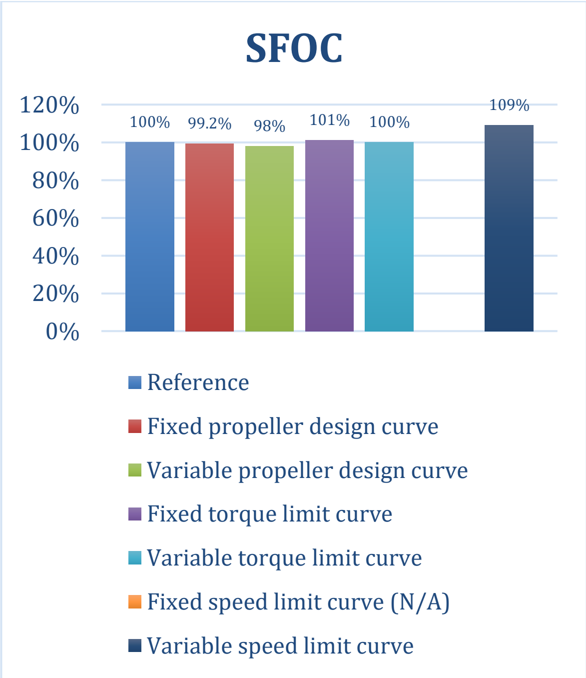

Figure 15: The chart shows the specific fuel oil consumption at 75 % load.

When running on the variable speed limit curve, the SOI had to be delayed reducing the power output, and EVC had to be delayed maintaining the correct exhaust pressure and flow. This was due to the high amount of scavenge air flow caused by the high engine speed. To maintain the high, scavenge air flow that was required, the fuel needed to be injected very late to achieve as much as exhaust as possible relation to the power output.

As described further in Figure 9, Figure 17 shows how much greater the scavenge air pressure was compared to the exhaust gas back pressure. When running on the speed limit curve, the delta pressure was as low as 15.8 % of the reference pressure

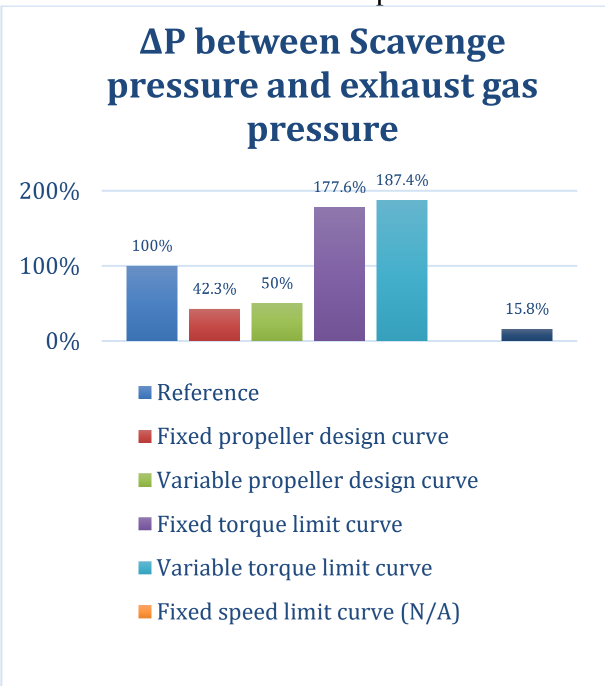

Figure 16: Pressure difference between exhaust and inlet air.

The low delta pressure might be a problem, because it effects the scavenging process negatively. To the contrary, the torque limit curve showed very high delta pressure. This was explained by the high-power output relation to the scavenge air flow.

In Figure 18 the mass of trapped air illustrated and it is telling how much air it is in the cylinder after both exhaust valve and scavenge port are closed during the compression stroke. As said earlier in Figure 10, the trapped air determines how much fuel that can be burned with a sufficient AFR

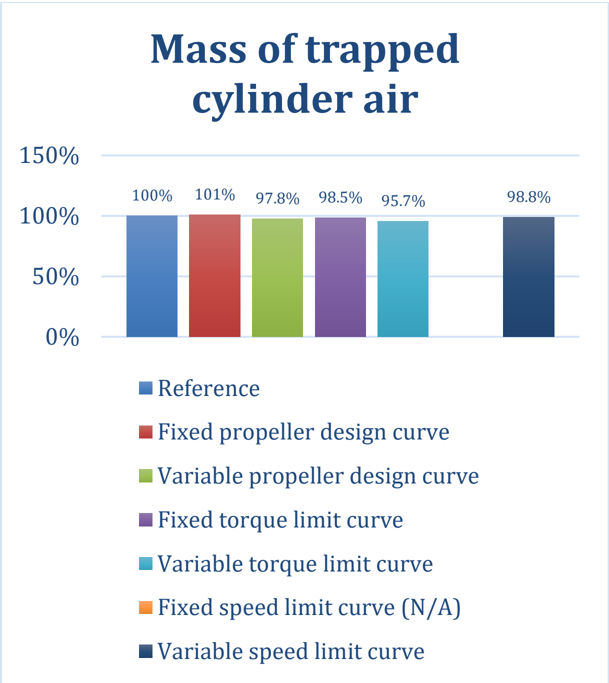

Figure 17: Mass of trapped air in the cylinder at 75% load.

. Between the different curves, the mass of trapped air did not depart to much compared to the reference point. But in fact, the mass flow of trapped air is increased by increased engine speed if the mass of trapped air is still the same. Which explain why the power output increase with the increase in speed.

Figure 19 shows the gas temperature at 75% load, which is further explained in Figure 11. The variable propeller curve has an higher maximum gas temperature than the fixed propeller design curve. This is explained by the 0.9 degrees earlier injection timing on the fixed curve.

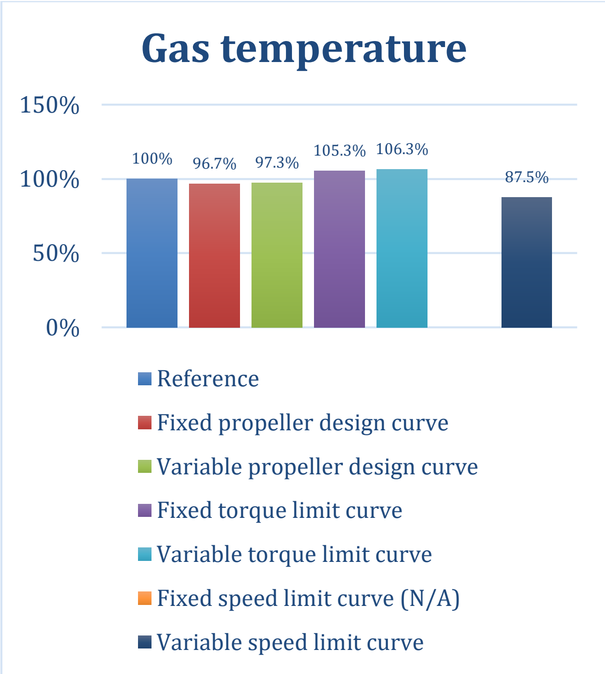

Figure 18: Gas temperature at 75 % load.

The earlier timing result in an higher Pmax which led to an increased exhaust gas temperature.

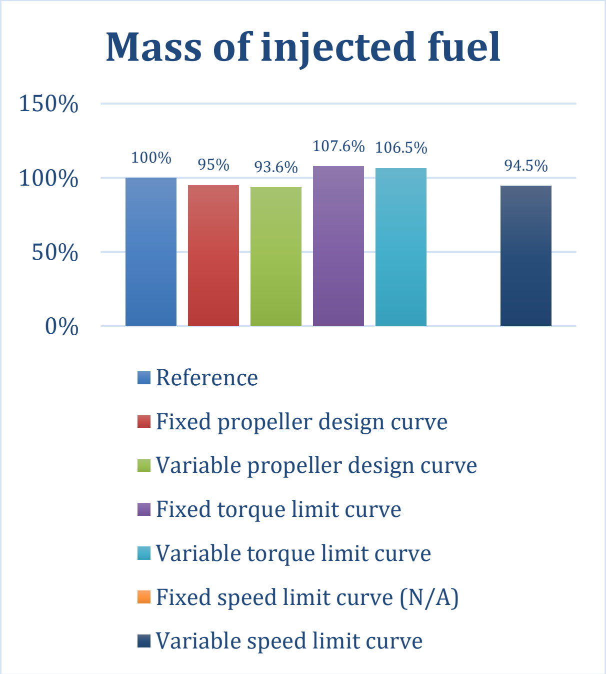

Figure 19: Mass of injected fuel at 75 % load.

As described in Figure 12, the mass of injected fuel is presented as mass per revolution. Which explain why the values are decreasing with increased engine speed, because less air has to be injected per revolution to maintain the power output. It should be mentioned, that due to this, the mass of injected air is not linear with the specific fuel oil consumption.

5 Discussion

5.1 Result discussion

For the 100 % load a fixed timing setting shows a 1.8 % increase in SFOC compared to a variable system which is an increase of 2.99g/kWh. There is not a big change of the NOx emissions but there still is a 0.4 % difference where the variable setup is lower than the fixed system. Worth to notice is the change in pressure difference between scavenge air and exhaust backpressure. The variable system has decreased 66.7 % from the reference value and the fixed system another 22 %. We could only speculate what kind of effect this has in reality, but it is safe to say that a greater delta pressure would lead to a higher gas exchange rate when the scavenge air ports open up. If the delta pressure were to be zero or higher in the exhaust manifold, then the gasses would reflux.

For the 75 % load while running on the propeller design curve the fixed timing settings NOx emissions is 1.3 % higher than the variable one. The difference between SFOC is 1.2 % which is about 2.3g/kWh. The delta pressure difference trends are the same here as for the 100 % load but not quite that extreme.

When going in the lower rpms to the torque limit curve the results get reversed, the NOx emissions, SFOC and delta pressures all increase in comparison to the reference. The fixed system produced 1 % more NOx emissions and SFOC compared to the variable one. The 1 % in SFOC equals about 2.3g/kWh. The interesting part here is that the delta pressure also increases by 77.6 % for the fixe and 87.4 % for the variable in relation to the reference. That means, since the scavenging pressure is kept as a constant, that the exhaust pressure has decreased therefor is the scavenging process really good when running at the torque limit curve.

From the results, it is seen that the fixed timing settings for 75 % load running at the speed limit curve is not possible to run, hence the N/A in the diagrams. The increase in rpm made the engine produce too much power output, as an attempt to decrease the power output the mass flow of fuel was lowered to such a low point that there was not enough energy in the exhaust gasses to provide the power required by the compressor. By that point the engine produced about 700kW higher power output than the reference point. So, the conclusion is that it is not possible to run engine giving these circumstances. Though by adjusting the valve and injection timings make it possible to reduce the power output and run the engine, but with an increase in SFOC with 9 % and a huge decrease in NOx emissions of 27.9 %.

It was hard to find information about tuning methods for these types of engines. Therefore, more general knowledge of what each engine parameter effect, was considered to achieve the results. By then, experiment with different timings, to achieve the goal of low specific fuel oil consumption was made. When optimizing an engine in purpose to reduce the fuel oil consumption, there are several different parameters which may deteriorate. For example, when reducing the mass flow of fuel, it tends to be to small amount of exhaust gases to run the turbine on the scavenge air compressor. This problem was solved by changing the injection and exhaust valve timing.

Then the rpm was increased from the engine design curve to the speed limit curve, the pressure difference between the compressor output compared to the turbine inlet was dropped far below the reference value. This could be a problem, because this pressure difference is important for the quality of the scavenging process. Since, it is mainly this pressure difference that forces the scavenge air to flow in to the cylinder and pushing out the exhaust gases during the gas exchange in the scavenging process.

5.2 Method discussion

In this thesis a validated MATLAB code of the MAN 12k98ME was used. All processes that take place in an engine are very complex. In the code, there are calculations for most of these processes, which makes the code perform well. But of course, although if the equations were the most accurate available at the time when the code was made it is extremely difficult to create a mathematical code to illustrate all the physical actions in a combustion engine.

6 Conclusions

6.1 Answers to the research questions

Tuning an engine is extremely difficult and each engine is different from another in terms of settings. In this report it shows, to get an optimized engine while running on both the torque limit curve and propeller design curve it requires a later EVC and an earlier SOI. The engine settings to run on the speed limit curve is quite different from the engine design curve, since the engine produces more output when the rpm is increased it had to be limited somehow. By reducing the fuel flow would also lead to less exhaust gasses and the compressor would not get enough energy to be driven. As well, on the speed limit curve the EVC is late and the SOI is also later, the injection time is almost doubled which resulted in a major decrease in Pmax and NOx emissions but with a lot lower SFOC since the efficiency went down.

The fuel and NOx emissions show a decrease with a variable system. The savings is bigger while running on 100 % load than 75 % load in terms of SFOC and vice versa in 75 % load. In the 100 % load there is a reduction of 1.8 % in SFOC and 0.4 % in NOx while in the 75 % it is a reduction of 1.2 % in SFOC and 1.3 % in NOx emissions while running with a variable system.

6.2 Further studies

It would be interesting to see how the results would be on the 50 % load and lower since engines operates in that area also. Since the results rely on a mathematical code, which is calculated to simulate the reality, it would also be interesting to see how these results would compare to a real engine even though that would be hard to perform.

7 References

Danov, S. N., & Gupta, A. K. (2001). Influence of Imperfections in Working Media on Diesel Engine Indicator Process. *Journal of Engineering for Gas Turbines and Power*, *123*(1), 231. https://doi.org/10.1115/1.1339986

Dragsted, J. (n.d) The first 50 years of turbocharged 2-stroke, crosshead, marine diesel engines. [PDF]. Available at: http://www.jicef.org/japanese/cimac/pdf/Dragsted_History_Booklet_2013.pdf

Geist, M. a, & Holtbecker, R. (1997). *Marine Diesel NOx Reduction Technique - A New Sulzer Diesel Ltd Approach -. SAE Technical Paper 970321*. https://doi.org/10.4271/970321

Goldsworthy, L. (2003). Reduced kinetics schemes for oxides of nitrogen emissions from a slow-speed marine diesel engine. *Energy and Fuels*, *17*(2), 450–456. https://doi.org/10.1021/ef020172c

Hariram, V., & Vagesh Shangar, R. (2015). Influence of compression ratio on combustion and performance characteristics of direct injection compression ignition engine. *Alexandria Engineering Journal*, *54*(4), 807–814. https://doi.org/10.1016/j.aej.2015.06.007

Heywood, J. B. (1988). *Internal Combustion Engine Fundamentals*. *McGrawHill series in mechanical engineering* (Vol. 21). https://2k9meduettaxila.files.wordpress.com/2012/09/internal-combustion-engines-fundamentals-heywood.pdf

Jääskeläinen, H. (2013, November). *Early History of the Diesel Engine*. Retrieved from Dieselnet: https://www.dieselnet.com/tech/diesel_history.php

Krishnan, A. (2015). Simulation of an engine friction strip test. *Department of applied mechanics division of combustion*. (master's thesis 2014;39)

Kuiken, K. (2012). *DIESEL ENGINES 1: for propulsiun and power plants* . Onnen: Target Global Energy Training.

Kuiken, K. (2012). *Diesel engines 2: for ship propulsion and power plants* (2. Ed.). Onnen, Netherlands: Target Global Energy Training.

Larsen, U., Pierobon, L., Baldi, F., Haglind, F., & Ivarsson, A. (2015). Development of a model for the prediction of the fuel consumption and nitrogen oxides emission trade-off for large ships. *Energy*, *80*, 545–555. http://doi.org/10.1016/j.energy.2014.12.009

Larsen, U., Haglind, F., Gabrielii, C., & Elmegaard, B. (2014). *Design and modelling of innovative machinery systems for large ships*. DTU Mechanical Engineering. (DCAMM Special Report; No. S176).

Latarche, M. (2017, July 13). *Power, Propulsion & Turbochargers*. Retrieved from Shipinsight: https://shipinsight.com/ships-diesel-engines-a-brief-history/

Lemmon, E. W., Huber, M. L., & McLinden, M. O. (2010). NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties (REFPROP), Version 9.0. *Physical and Chemical Properties …*. https://doi.org/10.1234/12345678

Lloyd, A. C., & Cackette, T. A. (2001). Diesel Engines: Environmental Impact and Control. *Journal of the Air & Waste Management Association*, *51*(6), 809–847. https://doi.org/10.1080/10473289.2001.10464315

Merker, G., Schwarz, C., Stiesch, G., & Otto, F. (2014). *Simulating Combustion: Simulation of combustion and pollutant formation for engine-development*. *Springer*. https://doi.org/10.1007/s13398-014-0173-7.2

Miyamoto, N., Chikahisa, T., Murayama, T., & Sawyer, R. (1985). DESCRIPTION AND ANALYSIS OF DIESEL ENGINE RATE OF COMBUSTION AND PERFORMANCE USING WIEBE'S FUNCTIONS. *Proceedings - Society of Automotive Engineers*, 77–88.

Moldanová, J., Fridell, E., Winnes, H., Holmin-Fridell, S., Boman, J., Jedynska, A., … Niessner, R. (2013). Physical and chemical characterisation of PM emissions from two ships operating in European emission control areas. *Atmospheric Measurement Techniques*, *6*(12), 3577–3596. https://doi.org/10.5194/amt-6-3577-2013

Mondejar, M. E., Andreasen, J.G., Pierobon, L., Larsen, U., Thern, M., Hagling, F. (2018) A review of the use of organic Rankine cycle power systems for maritime applications. *Renewable and Sustainable Energy Reviews*. https://doi.org/10.1016/j.rser.2018.03.074

Neef, D. (n.d) The Development of a Global Maritime Emissions Inventory Using Electronic Monitoring and Reporting Techniques. [PDF] Mystic. Available at: https://www3.epa.gov/ttnchie1/conference/ei18/session1/neef.pdf.

Prasad, R., & Bella, V. R. (2011). A Review on Diesel Soot Emission, its Effect and Control. *BULLETIN OF CHEMICAL REACTION ENGINEERING AND CATALYSIS*, *5*(2). https://doi.org/10.9767/bcrec.5.2.794.69-86

Prevention of Air Pollution from Ships. (n.d). Retrieved from http://www.imo.org/en/OurWork/environment/pollutionprevention/airpollution/pages/air-pollution.aspx

Raeie, N., Emami, S., & Karimi Sadaghiyani, O. (2014). Effects of injection timing, before and after top dead center on the propulsion and power in a diesel engine. *Propulsion and Power Research*, *3*(2), 59–67. https://doi.org/10.1016/j.jppr.2014.06.001

Reşitoılu, I. A., Altinişik, K., & Keskin, A. (2015). The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems. *Clean Technologies and Environmental Policy*. https://doi.org/10.1007/s10098-014-0793-9

Scappin, F., Stefansson, S. H., Haglind, F., Andreasen, A., & Larsen, U. (2012). Validation of a zero-dimensional model for prediction of NOxand engine performance for electronically controlled marine two-stroke diesel engines. *Applied Thermal Engineering*, *37*, 344–352. http://doi.org/10.1016/j.applthermaleng.2011.11.047

The basics the two-stroke cycle. (n.d) Retrieved from: http://www.marinediesels.info/Basics/the_2_stroke_engine_explanation.html

Verschaeren, R., Schaepdryver, W., Serruys, T., Bastiaen, M., Vervaeke, L., & Verhelst, S. (2014). Experimental study of NOx reduction on a medium speed heavy duty diesel engine by the application of EGR (exhaust gas recirculation) and Miller timing. *Energy*, *76*(x), 614–621. https://doi.org/10.1016/j.energy.2014.08.059

Wang, Y., Zeng, S., Huang, J., He, Y., Huang, X., Lin, L., & Li, S. (2005). Experimental investigation of applying Miller cycle to reduce NOxemission from diesel engine. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, *219*(8), 631–638. https://doi.org/10.1243/095765005X31289