



National Technical University of Athens

Laboratory of Marine Engineering

SIMULATION OF THE TRANSIENT OPERATION OF A LARGE TWO-STROKE MARINE DIESEL ENGINE EQUIPPED WITH AN EXHAUST GAS RECIRCULATION SYSTEM (EGR) FOR NO_x REDUCTION

DIPLOMA THESIS

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Abstract

Large two-stroke marine diesel engines cannot meet the strict Tier III NO_x emission limits, imposed by the IMO, with in-engine modifications. A possible solution that has been tested over the last years, in order to comply with the new regulations, is to implement an Exhaust Gas Recirculation (EGR) system.

High operational costs of tests in real marine diesel engines, as well as the limited number of available vessels that operate with an EGR system installed, make an engine simulation model that can predict the engine's performance with EGR, a valuable tool.

In the present Thesis, a zero-dimensional engine model was developed, using the thermodynamic engine performance prediction code MOTHER of the NTUA Laboratory of Marine Engineering. The model of a low-speed two-stoke diesel engine (6S70MC-C), that had already been setup for Tier II operation (without an EGR system), was used. All the extra components that were necessary in order to simulate the engine's operation with EGR, such as the exhaust gas blower, the heat exchanger for EGR cooling, the EGR throttle valve etc. were added to the model and calibrated accordingly. The shop test reports of a slightly bigger engine (6S80ME-C) were used in order to qualitatively assess the model's accuracy, along with the results from the research that was conducted on a testbed engine (4T50ME-X).

A ship hull model and a 4-quadrant propeller model were also used, in order to have a realistic representation of the entire propulsion arrangement, while studying the engine's transient behaviour. Simulations were carried out both at steady-state and transient operational conditions.

The simulation results proved to be in good agreement with the research results found in literature. More specifically, reduced in-cylinder temperatures during combustion and a lower fuel burning rate are predicted by the model during steady-state operation. The oxygen content is reduced to acceptable levels and the rest of the thermodynamic parameters (p_{max} , T_{scav} , p_{scav} , T_{exh} , p_{exh} etc.) present a similar trend to the one's in the available shop test and research results that were available.

Three different transient loading scenarios were carried out for the model. In the first one, the engine accelerates from 50% to 75% load, with and without the EGR system active. Results are plotted into common diagrams, in order to compare the two operational conditions. The model is able to predict the reduction of the O_2 concentration into the scavenge receiver, along with the negative O_2 peak due to the turbocharger lag. During the entire simulation, the acceleration capabilities of the engine remain unaffected from the presence of the exhaust gas into the scavenging manifold. In the second and third scenario, an EGR start and stop sequence are carried out respectively, both at 50% engine load. The behaviour of the engine for a varying EGR rate is studied and in extent, the response of the turbocharger unit. Extremely high EGR rates, that lead to the formation of undesired black smoke, should be avoided.

Περίληψη

Οι μεγάλοι δίχρονοι ναυτικοί κινητήρες diesel, δεν είναι σε θέση να ικανοποιήσουν τα αυστηρά κριτήρια Tier III για τις εκπομπές NO_x, τα οποία τίθενται από τον Διεθνή Οργανισμό Ναυτιλίας, αν δεν υποστούν κάποια τροποποίηση στη λειτουργία τους. Μία πιθανή λύση για την ικανοποίηση των νέων κανονισμών, η οποία έχει δοκιμαστεί κατά τη διάρκεια των τελευταίων χρόνων, είναι η εφαρμογή ενός συστήματος ανακυκλοφορίας καυσαερίων (EGR).

Τα υψηλά λειτουργικά κόστη για την εκτέλεση πειραμάτων σε πραγματικούς ναυτικούς κινητήρες diesel, σε συνδυασμό με τον περιορισμένο αριθμό πλοίων που έχουν ενσωματωμένο ένα σύστημα EGR, καθιστά τη δημιουργία ενός μοντέλου που θα μπορεί να προβλέψει την συμπεριφορά της μηχανής με EGR ένα απαραίτητο σχεδιαστικό εργαλείο.

Στην παρούσα Διπλωματική, αξιοποιήσαμε τον θερμοδυναμικό κώδικα MOTHER, του Εργαστηρίου Ναυτικής Μηχανολογίας, για να δημιουργήσουμε ένα μοντέλο μηδενικών διαστάσεων, που θα προβλέπει τη συμπεριφορά της μηχανής. Το μοντέλο που χρησιμοποιήθηκε ήταν ενός αργόστροφου δίχρονου ναυτικού κινητήρα diesel (6S70MC-C), που είχε προηγουμένως ρυθμιστεί για Tier II λειτουργία (χωρίς EGR). Τα επιπλέον στοιχεία που ήταν απαραίτητα για την προσομοίωση της λειτουργίας με EGR, όπως το EGR blower, το ψυγείο για την ψύξη των επανακυκλοφορούμενων καυσαερίων, η ρυθμιστική βαλβίδα κτλ., προστέθηκαν στον μοντέλο και ρυθμίστηκαν καταλλήλως. Για τον ποιοτικό έλεγχο της ακρίβειας του μοντέλου, αξιοποιήθηκαν τα διαθέσιμα shop test μιας λίγο μεγαλύτερης μηχανής (6S80ME-C), καθώς και αποτελέσματα που προέκυψαν από δοκιμές σε μία πειραματική μηχανή (4T50ME-X).

Επιπλέον, ένα μοντέλο αντίστασης της γάστρας του πλοίου και ένα μοντέλο προπέλας τεσσάρων τεταρτημόριων αξιοποιήθηκαν, προκειμένου να έχουμε μία ρεαλιστική αναπαράσταση ολόκληρου του συστήματος πρόωσης, καθώς μελετάμε τη μεταβατική συμπεριφορά της μηχανής. Προσομοιώσεις πραγματοποιήθηκαν τόσο υπό συνθήκες σταθερού φορτίου, όσο και υπό μεταβαλλόμενου.

Τα αποτελέσματα των προσομοιώσεων δείχνουν να είναι σε συμφωνία με αυτά που βρέθηκαν στην αντίστοιχη βιβλιογραφία. Συγκεκριμένα, η αναμενόμενη μείωση των θερμοκρασιών εντός του κυλίνδρου, καθώς και ο μειωμένος ρυθμός καύσης, προβλέπονται με επιτυχία από το μοντέλο κατά τη διάρκεια των σταθερών φορτίσεων. Το οξυγόνο μειώνεται σε αποδεκτά επίπεδα, ενώ οι υπόλοιπες θερμοδυναμικές παράμετροι (pmax, Tscav, pscav, Texh, pexh κτλ.) παρουσιάζουν συμπεριφορά παρόμοια με αυτήν των διαθέσιμων shop tests. Κατά τη διάρκεια λειτουργίας με μεταβαλλόμενο φορτίο, τα αποτελέσματα φανερώνουν ότι το turbocharger lag είναι εντονότερο όταν η μηχανή λειτουργεί με EGR. Η αποδιδόμενη ισχύς της μηχανής δεν επηρεάζεται από την παρουσία του EGR.

Τρεις διαφορετικές προσομοιώσεις μεταβαλλόμενου φορτίου πραγματοποιήθηκαν. Στην πρώτη, η μηχανή επιταχύνει από 50% στο 75% του φορτίου, με και χωρίς το σύστημα EGR. Τα αποτελέσματα παρουσιάζονται σε κοινά διαγράμματα για εύκολη σύγκριση. Το μοντέλο προβλέπει ικανοποιητικά την μείωση της περιεκτικότητας του οξυγόνου στο δοχείο αέρα, καθώς και την απότομη μείωσή του κατά την επιτάχυνση, λόγω του φαινομένου turbocharger lag. Καθ' όλη την προσομοίωση, η ικανότητα επιτάχυνσης της μηχανής παραμένει ανεπηρέαστη από την παρουσία του καυσαερίου στο δοχείο αέρα. Στη δεύτερη και τρίτη προσομοίωση, μία διαδικασία εκκίνησης και αντίστοιχα απενεργοποίησης του συστήματος EGR εξετάζονται, στο 50% του φορτίου. Η συμπεριφορά της μηχανής για μεταβαλλόμενα ποσοστά EGR εξετάζεται, και κατ' επέκταση η απόκριση του υπερπληρωτή. Υπερβολικά υψηλά ποσοστά EGR, που οδηγούν στο σχηματισμό μαύρου καπνού, πρέπει να αποφεύγονται.

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Chapter 1: Introduction

1.1 Introduction

Compliance with the strict emission limits, imposed by the regulations of the International Maritime Organisation (IMO), has been a constant challenge for marine diesel engine designers during the last decades. The latest Tier III regulations, introduced for ships with a keel laying date on or after January 1st 2016, that sail inside Emission Control Areas (ECAs), demand an 80% reduction of NO_x emissions, compared to Tier I regulations. These limits cannot be achieved with internal engine optimisation alone, thus new emissions control technologies must be adopted.

The present Thesis focuses on the study of the Exhaust Gas Recirculation (EGR) system on a large twostroke marine diesel engine. During EGR operation, part of the exhaust gas is recirculated back to the scavenge receiver, after a cooling and cleaning process. Due to the increased heat capacity and the reduced oxygen concentration of the charge mixture, a reduction in the peak cylinder temperature can be observed and thus, the new NO_x limits can be achieved. Although this method has been successfully used in the automotive industry for years, its application in the marine industry poses certain challenges that must be further investigated. The need for a positive scavenging pressure between the exhaust and the scavenging manifold, in order to achieve uniflow scavenging, as well as the use of Heavy Fuel Oil (HFO) with high sulphur content, are two of the main challenges. An electrically driven blower is incorporated into the EGR path to overcome the pressure difference between the manifolds, while SO_x scrubbers are also installed to clean the exhaust gas.

In the present Thesis, the engine model was developed in the in-house engine simulation tool MOtor THERmodynamics. The initial model of the engine was already setup for Tier II operation in a previous work. In this study, the EGR components were added and the engine model was recalibrated to correspond to EGR operation. A ship hull model along with a propeller model were then attached to the engine model, in order to have a realistic representation of the load imposed to the engine during transient conditions. Finally, a number of transient simulations were performed.

1.2 Motivation of the Thesis

Although much research can be found in the literature about automotive engine models with EGR, like in [1], [2], [3], [4] and [5], on the contrary, the literature for EGR systems in low-speed two-stroke marine diesel engines is quite limited. EGR systems for this kind of engines are still in development, with only a few vessels currently operating with EGR, making operational data scarce to find. Furthermore, extensive testing of an EGR system in engines this large, involves a high financial cost, mainly due to the fuel cost associated with their size. For the aforementioned reasons, an engine simulation model with an EGR system integrated is an important tool for engine development, in order to predict the performance of an engine under EGR operation. This is especially true for a model that can also predict the transient behaviour of the engine. Some of the research done is presented in [6], [7], [8] and [9].

1.3 Structure of the Thesis

- **Chapter 1:** An initial description of the general problem that will be studied in the Thesis is presented. The motivation behind the project and the aim of the study are also mentioned.
- **Chapter 2:** A reference to the legislation imposing the strict emissions limits is made. Furthermore, the mechanisms that characterise the formation of NO_x during the combustion process are briefly explained. Finally, a number of technologies that have been incorporated over the years to reduce NO_x emissions are presented, with more details given on the various EGR systems.
- **Chapter 3:** A brief presentation of the software used for the engine model is given. The submodels that describe the various engine processes (combustion, heat transfer etc.) are explained, along with the extra components that were added to the model to simulate the operation with EGR. After that, the calibration of the model during steady-state operation and the simulation results are presented and compared with the available literature.
- **Chapter 4:** The additional ship hull and propeller models, that were necessary in order to have a realistic representation of the propulsion arrangement during transient loading, are presented. A brief explanation on the calculation of the propeller's inertia and the speed governor's setup is also presented.
- **Chapter 5:** The results during the transient simulations are included in this chapter. Three different transient simulations were carried out. An engine acceleration from 50% engine load to 75%, with and without EGR, and an EGR start and stop sequence respectively, at 50% engine load.
- **Chapter 6:** The results drawn from this Thesis are discussed and recommendations for future scientific work are proposed.

Chapter 2: Theoretical Background

In recent years, since the impact of the internal combustion engines emissions on human health and climate change became apparent, engine designers have had to address the challenge of tightening controls on exhaust gas emissions, imposed by regional, national and international authorities. Various methods in order to regulate the harmful pollutants of the exhaust gas emissions have been incorporated, depending on the operational principles of each engine type. In spark ignition (SI) engines, which are mostly used in the automotive industry (passenger cars) and in light-duty applications, the main pollutants contained in the exhaust gas are nitrogen oxides (NO_x) , unburned hydrocarbons (HC) and carbon monoxide (CO). Stoichiometric air to fuel ratio in SI engines enables the utilization of a Three-Way Catalytic converter (TWC), which is capable of reducing all of the aforementioned emissions simultaneously and efficiently. More specifically, a TWC combines oxygen (O_2) with unburned hydrocarbons (HC) and carbon monoxide (CO) to produce water (H_2O) and carbon dioxide (CO₂), while at the same time reduces nitrogen oxides (NO_x) to nitrogen (N₂), water (H₂O) and/or carbon dioxide (CO₂). These reactions occur most efficiently when the engine operates in a very narrow window, close to the stoichiometric air to fuel ratio and therefore, TWCs can only be used in SI engines. Contrary to the SI engines, where the homogeneous mixture of air and fuel and the uniform propagation of flame due to the spark plug initiation ensure complete combustion, in diesel engines, the injection of fuel into the already present air result in a heterogeneous mixture which ignites at multiple areas in the cylinder. As a result, in locally rich areas, where there is not enough oxygen to burn the fuel, carbon is left behind in the form of soot. Furthermore, due to the high compression rates, which are common in diesel engines, high incylinder peak temperatures and thus high NO_x emissions are also to be expected. Because the problem of controlling NO_x in diesel exhaust is more complicated, different approaches are required. Two proven technologies that are used today in order to reduce the NO_x emissions of diesel engines are the Exhaust Gas Recirculation (EGR) and the Selective Catalytic Reduction (SCR), both of which will be discussed later on in this Chapter. It is to be noted that, although these two technologies have already been extensively tested and used in heavy-duty engines and power plants, their application in the low speed, two-stroke marine diesel engines is still being tested, as the operation of these engines with Heavy Fuel Oil (HFO) poses certain challenges.



Figure 2.1: Typical exhaust emissions from a modern low-speed diesel engine [10]

2.1 Legislation on Emissions

Since the Industrial Revolution began, fossil fuels have been an integral part of our society. At the beginning, coal was the dominant fuel, used to power the first commercially successful steam engines. That was until the early decades of the 20th century, when the first internal combustion engines made their appearance and the use of coal gradually gave way to the use of petroleum products such as gasoline and diesel oil. From electricity generation, heating and industrial production, to transportation of humans and goods, whether it is by land, air or sea, to this day, fossil fuels remain the largest contributors to energy consumption. However, during the last decades society has come to realize the grave impact that fossil fuels have on the environment and on our health. All of the aforementioned activities require the burning of large quantities of fossil fuels, resulting in the emission of harmful elements in the environment, such as carbon monoxide (CO) and carbon dioxide (CO₂), nitrogen oxides (NO_x), sulphur oxides (SO_x), particulate matter (PM) and unburned hydrocarbons (HC), which are the cause for a series of various problems. The greenhouse effect, caused by the excessive concentration of atmospheric CO_2 that results in global warming, respiratory problems or acid rain caused by sulphur dioxide (SO₂) and oxides of nitrogen (NO_x), as well as haze (PM) and smog, produced when HC react with nitric oxides (NO_x) to form ozone, are only some of the problems that must be confronted. Therefore, authorities worldwide soon realized that strict regulatory limits concerning the control of the exhaust emissions must be applied, both at a national (USA coasts) and at an international (EU) level. These regulatory limits are set by different organizations and vary according to the sector that they are applied (automotive industry, marine industry etc.) and the operational principles of each engine (spark ignition, diesel).

In the automotive industry, the various tiers of the Euro legislation have been the regulatory limit of the exhaust gas emissions for light-duty and heavy-duty engines. Euro I, which came into force in 1992, set the first limits for CO, HC, PM and NO_x emissions, followed by Euro II in 1996 and Euro III in 2000. Up to Euro III legislation, the emission limits could be met with refined engine technology, like increased fuel injection pressure or later injection timing. However, when Euro IV regulations were introduced in 2005, these measures alone were not enough to satisfy the new emission limits [11]. As a result, new technologies were introduced in order to comply with the regulations. One of them was EGR, which was introduced as a way to lower the engine's NO_x emissions, combined with a Diesel Particulate Filter (DPF) to lower the PM emissions. Another one, was to lower the PM emissions with early injection timing and use a SCR post-treatment system to lower the NO_x emissions. These methods became even more compulsory when Euro V came into force and soon after, Euro VI, imposing emissions limits as low as 0.46 [g/kWh] for NO_x emissions and 0.01 [g/kWh] for PM (Figure 2.2).



Figure 2.2: NO_x and PM legislation for heavy-duty engines in Europe [11]

HC and CO emissions are also regulated, but in diesel engines their limits can easily be achieved due to the lean mixture operation (high air-fuel ratio). Similar regulations are also into force in the USA, by the Environmental Protection Agency (EPA).

International shipping is responsible for the transportation of more than 80 per cent of global trade, being one of the most efficient and cost-effective methods of transporting goods [12]. Since it is an international industry, it can only operate effectively if the regulations and standards are also implemented on an international level. For that reason, the regulatory framework according to which the maritime sector operates, is developed and maintained by the International Maritime Organization (IMO). IMO is an agency of the United Nations that adopts international rules and standards which are then implemented and enforced by governments all around the world. Prevention of air pollution from ships has been one of the main concerns of the IMO during the recent years. MARPOL 73/78 Annex VI, that was first adopted in 1997, is responsible for regulating the main air pollutants contained in ships exhaust gas, including SO_x and NO_x. The regulations also specify certain areas around the world, like the US coasts and the Baltic Sea, where even stricter limits apply for the aforementioned emissions. These are called Emissions Control Areas (ECAs). SO_x emissions are mainly reduced by burning fuels that have a low sulphur content. The current global limit of sulphur concentration in fuel is set at 3.5% (m/m) and it is expected to drop at 0.5% by 2020. Inside the Sulphur Emission Control Areas (SECAs), the limit is set at 0.1% and will remain so in the near future. Alternative measures for SO_x reduction, especially inside the SECAs, include the use of scrubbers or any other technological method that can effectively meet the requirements. Figure 2.3 shows the permissible SO_x limits throughout the years.



Figure 2.3: MARPOL Annex VI SO_x emission limits [13]

On the other hand, NO_x emission limits are regulated according to the rated speed of the engine, as well as the ship's construction date. Tier I was first introduced for ships with a keel laying date from January 1st 2000, followed by Tier II regulations for vessels constructed after January 1st 2011. The most recent Tier III regulations apply for vessels with a keel laying date from January 1st 2016, but only when sailing inside the NO_x Emissions Control Areas (NECAs), i.e. North American coasts, parts of Canada and the Caribbean Sea, setting the NO_x emission limit at 3.4 [g/kWh]. Outside these areas, Tier II regulations are still into force, with a limit of 14.4 [g/kWh]. Figure 2.3 shows the permissible NO_x limits, depending on the engine rated speed. Overall, low speed marine diesel engines have more lenient NO_x limits, compared to medium and high-speed engines.



Figure 2.4: MARPOL Annex VI NO_x emission limits [14]

While Tier II regulation limits can be achieved with internal engine optimization, the strict emission limits imposed by Tier III regulations (about 80% lower emission levels with respect to the Tier I limit) require the implementation of new emissions control technologies, such as the EGR and SCR.

2.2 NO_x Formation Mechanisms

Understanding the NO_x formation mechanisms during the duration of the combustion is an important step, when designing technologies to reduce their emission. In conventional diesel combustion, NO_x formation is strongly related to the adiabatic flame temperature, with the formation rate increasing exponentially with temperature [15]. Furthermore, oxygen concentration inside the cylinder, as well as residence time in high temperature, are also factors that influence the NO_x formation [16], as it can be seen on Figure 2.5.



Figure 2.5: Effect of the intake oxygen concentration (a) and the stoichiometric flame temperature (b) on the fuel-specific NO_x emissions of a diesel engine [16]

Nitrogen oxide (NO_x) formation takes place in the high-temperature burned gas regions inside the cylinder, through chemical reactions involving nitrogen and oxygen atoms and molecules that do not attain chemical equilibrium [15]. After the combustion ends and during the expansion stroke, the reactions involving NO_x freeze due to the exhaust gases cooling, resulting in NO_x concentration levels much higher than the ones corresponding to equilibrium at exhaust conditions. NO_x mainly comprise of two basic constituents, nitric oxides (NO) and nitrogen dioxides (NO₂). Inside the engine cylinder, the predominant oxide produced is the nitric oxide (NO), although until the exhaust gas is emitted to the atmosphere, most of the NO will have been furthered oxidized into NO₂. The principal source of NO formation is the oxidation of atmospheric nitrogen. The formation of NO_x, and more specifically the formation of NO, can be attributed to the following major reaction mechanisms.

Thermal NO Formation (Zel'dovich Mechanism)

The thermal formation mechanism is considered to be the dominant source of NO emissions in conventional diesel combustion. It was firstly introduced by Zel'dovich in [17], who proposed equations (2.1) and (2.2), and was later complemented by Lavoie's work [18], who introduced the third equation (2.3).

$$0 + N_2 \rightleftharpoons N + N0 \tag{2.1}$$

$$N + O_2 \rightleftharpoons O + NO \tag{2.2}$$

$$N + OH \rightleftharpoons H + NO \tag{2.3}$$

Out of these three reactions, the one described by equation (2.1) is much slower than the other two, as it requires a relatively high activation energy (318 [kJ/mol]) to break the N₂ triple bond, thus (2.1) will be the rate-determining step. Due to the high activation energy needed, the whole mechanism becomes relevant at high temperatures (T > 1800 [K]), leading to the name "thermal" [19].

Prompt NO Formation (Fenimore Mechanism)

This mechanism was first proposed by Fenimore, in his work [20]. According to [20], NO was found to form in the primary reaction zone of the flame, under fuel-rich conditions and in the presence of hydrocarbon radicals (CH, C), which can react in many different ways. These radicals react with molecular nitrogen (N₂) to from hydrocyanic acid (HCN) and monoatomic nitrogen (N), according to equation (2.4), as well as cyanide (CN), according to equation (2.5).

$$CH + N_2 \rightleftharpoons HCN + N$$
 (2.4)

$$C_2 + N_2 \rightleftharpoons 2CN \tag{2.5}$$

Afterwards, the N atoms could form NO through reactions such as (2.2) and (2.3). CN might also as well yield NO [20]. The term "prompt" derives from the low activation energy required for these reactions, which already take place at temperatures of about 1000 [K]. For flame temperatures less than about 2125 [K] (see also Figure 2.5 (b), $10^4/T_f > 4.71$), the Zel'dovich mechanism isn't relevant, and the major source

of NO emission is the Fenimore mechanism [16]. In diesel combustion engines, about 5% - 10% of NO_x are produced from the Fenimore mechanism and 90% - 95% from the Zel'dovich mechanism [21].

Nitrous NO Formation

Although this mechanism is negligible in conventional diesel engines, it becomes important when EGR is applied to the engine, reducing the peak in-cylinder temperatures and leading to lean air-fuel mixtures, which results in suppression of the thermal and prompt mechanisms respectively. This mechanism was proposed by Wolfrum, in his work [22], and suggests that N_2O is formed from N_2 and O, along with a stabilizing molecule M, so that N_2O and not NO is produced, according to the equation (2.6).

$$N_2 + 0 + M \rightleftharpoons N_2 0 + M \tag{2.6}$$

N₂O is then further oxidized into NO, according to the equation (2.7).

$$N_2 0 + 0 \rightleftharpoons 2N0 \tag{2.7}$$

Due to the three-body reaction (2.6), the formation of N_2O is favoured by high pressures. It can also become important at low temperatures, due to its low activation energy (76 [kJ/mol]), as explained in [19].

Fuel NO Formation

It has been observed that NO can be also formed from nitrogen that is bound to the fuel oil. During the combustion, the fuel-bound nitrogen is released as a free radical, which can ultimately form free N_2 or NO, at temperatures around 1100 [K] [21].

2.3 NO_x Reduction Methods

Various methods have been proposed since the first regulations for NO_x emissions reduction appeared. According to [14], all relevant new engines can be modified by internal methods, in order to comply with the Tier II regulation limits for NO_x . Modifications varying from new fuel system components, like plunger/barrel and fuel valve nozzles (number and size of spray holes), to adjustments in the combustion chamber volume with piston rod shims, the scavenge air pressure and the exhaust cam profile have been incorporated. The new electronic engine offered even more possibilities for emission control, with the introduction of the electronically controlled fuel injection, exhaust valve actuation and turbocharger control via a high-pressure hydraulic oil system, along with a combination of the aforementioned design changes of the combustion chamber components.

With the introduction of the strict Tier III limits though, the need for new NO_x reduction techniques arose. Further bellow, a number of methods for NO_x reduction that have been developed and tested throughout the years, others more and others less successful, are presented. These methods can be divided into two basic categories:

- In-Cylinder NO_x Reduction, aiming to reduce the amount of NO_x formed during the combustion process, generally by reducing the adiabatic flame temperature.
- Exhaust After-Treatment Systems for NO_x Reduction, designed to remove NO_x from the exhaust gas by downstream cleaning techniques, mostly through a catalytic converter that converts NO_x to non-harmful gases.

2.3.1 In-Cylinder NO_x Reduction

The in-cylinder NO_x reduction strategies comprise of a number of technologies that all have the same common goal; to reduce the adiabatic flame temperature, which is the basic parameter that leads to high thermal NO_x emissions. These technologies can be further divided into three categories, with respect to their operational principle:

- Cooling of the flame:
 - Direct Water Injection
 - Water-Fuel Emulsion
- Reduction of the reactant temperature:
 - Miller Valve Timing
 - Charge Air Cooling
 - Reduced Compression Ratio
- Change in charge composition:
 - Charge Air Humidification
 - Nitrogen Enrichment
 - Exhaust Gas Recirculation (EGR)

The aforementioned techniques are briefly explained bellow, with more focus being given on EGR, which is the main subject of the present study.

Direct Water Injection (DWI)

As a way of reducing the flame's temperature, water can be directly introduced into the combustion chamber either through separate nozzles or by the segregated injection of water and fuel from the same nozzle. Its operational principle is that when the added water in the cylinder chamber is evaporated during the combustion, it steals a portion of the emitted heat, resulting in a NO_x reduction. The water injection takes place before the injection of fuel, leading to a cooler combustion chamber and thus lower NO_x formation, and stops when the fuel is injected so that the ignition and combustion are not disturbed. The water is handled by a fully independent, common rail delivery system, under electronic control. Before introduced into the cylinder, the water is filtered in order to remove all solid particles. DWI enables the

water to be injected in the cylinder at the right time in order to obtain the greatest NO_x reduction, making it one of the most efficient methods of reducing NO_x emissions. In addition, the space requirements for the DWI equipment are minimal, facilitating retrofit applications in existing engines. On the other hand, the extra injector for the water and the required redesign of the combustion chamber are major changes that make DWI a very costly option. Special attention must be also given to the logistics of providing sufficient fresh water onboard.

According to tests done by Wärtsilä in medium-speed engines [10], for a 0.4-0.7:1 water to fuel ratio, a 50-60% reduction of NO_x emissions can be achieved, without adversely affecting power output or engine components. With 70% water to fuel ratio, NO_x emissions could be reduced to around 50% lower than the Tier I limit.

Water-Fuel Emulsion (FWE)

Similarly to the DWI method, the FWE strategy is based on the principle that part of the combustion heat will be used to evaporate the water instead for work, reducing the in-cylinder peak temperature. This time though, instead of directly injecting the water into the combustion chamber, fresh water is mixed with fuel onboard the ship (or sometimes offline in batches) to form an emulsion, suitable for injection into the cylinder. Except from NO_x reduction, the fast water evaporation during the emulsion's injection also leads to better mixing and increases the soot oxidation, resulting in a reduction of soot after combustion as well. According to research done by MAN Diesel & Turbo on a number of vessels and test engines [14, 10], the FWE method showed a significant reduction in NO_x emissions, with a limited increase in fuel oil consumption. Tests for the two-stroke engines showed that a 10% reduction in NO_x can be expected for every 10% water added.

Certain difficulties that arise by the implementation of FWE are, the need to redesign the fuel injection nozzle in order to be adapted to the increased quantity of liquid injected (fuel + water), as well as the fact that the fuel-water emulsion is very unstable and will be separated without the use of proper additives. Furthermore, the portion of water added is limited by the maximum delivery capacity of the fuel pump and the viscosity of the final mixture, along with the degree of heating required to lower that viscosity for easier injection. Nevertheless, according to [23], the NO_x reduction achieved by FWE is higher compared to the DWI method, for the same amount of water.

Miller Valve Timing

The Miller cycle was first introduced in the four-stroke diesel engines, along with the development of turbocharging, as a way to increase the power density of some engines. Its operational principle involves the early closure of the inlet valve, before the piston reaches the Bottom Dead Centre (BDC), leading to a reduction of the cylinder pressure as the piston continues to move downwards and thus, a reduction of the trapped air's temperature. Lower charge temperature at the beginning of the cycle results in a reduced peak temperature during combustion, meaning lower NO_x formation. Except from the NO_x emission reduction, Miller cycle simultaneously improves the fuel consumption as, for the same cylinder pressure levels, the wall heat losses are reduced due to the lower temperatures and the higher air-fuel ratio. For that reason, Miller cycle is applied to practically every modern engine, with at least a moderate valve timing.

A major issue that had to be addressed with this technique was that, due to the shorter inlet valve opening duration, the charge air must be compressed to a higher pressure than that required for the engine cycle, leading to increased demands for a better turbocharging system. High-efficiency turbochargers with increased pressure ratios and new turbocharging techniques, like the two-stage turbocharging systems, compensate for these power output losses and ensure that the quantity of the intake air, and thus the engine performance and efficiency, remains unaffected. This means, though, an increase in cost and complexity of the whole system. Certain problems may also arise during low load operation or at the engine start-up of the engine (ignitability problems), due to the low efficiency of the turbocharger at these conditions. These problems have been dealt with by introducing the electronic engine and the Variable Valve Actuation (VVA), that ensures a smooth start-up and low load operation.

Because of the shorter inlet valve opening duration, the application of Miller cycle was mostly limited to medium-speed or large-bore engines, where there is sufficient time for the cylinder to be filled [19]. In the case of two-stroke engines, which cannot utilize the Miller cycle due to the lack of inlet valves, a variable exhaust valve closing can be applied, through an electronic control system [24].

Charge Air Cooling

A common practice for all turbocharged diesel engines, is to place an air cooler after the turbocharger's compressor, which reduces the inlet air temperature. Apart from the increase of the engine's power density (lower temperature means higher air density, thus a bigger quantity of air enters the cylinder), the cooler also has a beneficiary effect on the NO_x emissions. It is a simple and relatively cheap way to reduce NO_x emissions, but it is heavily dependent on the environmental temperature, as the intake air cannot be cooled further than the ambient temperature. This is especially problematic in case of marine applications, where the environmental temperature of the engine room is relatively high.

Reduced Compression Ratio

The compression ratio is one of the most basic parameters that characterizes the combustion and thus the engine's power output and emissions. A reduction of the compression ratio means a reduction of the effective compression stroke of the piston, resulting in a lower pressure at the end of the stroke (Top Dead Centre) and thus a lower temperature. Although this is beneficial for the NO_x emissions, this method is better to be avoided, as it also reduces the engine's thermodynamic efficiency, which is intrinsically linked with the compression ratio. Lower thermodynamic efficiency means an increase in the SFOC.

Charge Air Humidification

One of the techniques that has been tested by both MAN Diesel & Turbo and Wärtsilä, was the introduction of water into the combustion chamber by humidifying the scavenge air. Water is injected and evaporated into the intake air, resulting in an increased humidity and heat capacity, due to the higher heat capacity of water ($c_p = 4.181 \text{ [kJ/kg·K]}$ at 25°C) compared to that of the dry air ($c_p = 1.005 \text{ [kJ/kg·K]}$ at 25°C). This increase leads to higher in-cylinder peak temperatures and thus a reduction of NO_x emissions.

Wärtsilä's Combustion Air Saturation System (CASS) utilizes special injection nozzles that introduce the water directly into the charge air stream, right after the turbocharger, in the form of very small droplets. Given the environment that these droplets are injected to (temperatures higher than 200°C), they evaporate quickly, resulting in a charge air with a humidity of around 60 [g/kg] of air. According to Wärtsilä's experiments, this amount of water can lead to NO_x levels lower than 3 [g/kWh], with no substantial increase in the fuel consumption and no rise in material temperatures [10].

MAN Diesel & Turbo on the other hand, introduced the Scavenge Air Moisturising (SAM) system, which is partly based on reducing the oxygen content of the cylinder charge (dilution of the charge air with water vapour), and partly on increasing the heat capacity of the charge due to the addition of the vapour. The water vapour is produced onboard from sea water, using the engine's heat sources. The system comprises of three stages, a sea water injection stage, where sea water is injected in order to saturate and cool the hot air stream coming from the compressor, and two fresh water stages which only act as cleaning stages, removing any salt which passes with the air due to the sea water stage. All the water needed for humidification of the charge air is provided exclusively by the sea water. A basic advantage of the system is that it utilises the engine's own heat sources (i.e. the engine coolant and exhaust gas heat) to heat the charge air and increase its capacity to absorb moisture. Other notable advantages are the low operational cost due to the use of untreated sea water, the high availability and a cleaner combustion procedure with reduced deposits on the cylinder and the turbine, due to the presence of water vapour. Finally, the increased mass flow in the exhaust stream due to the evaporated water means that, part of the exhaust gas can be bypassed, and its energy can be supplied to an additional power turbine, leading to lower operational costs and an overall improvement of the engine's efficiency. Special attention must be given to the amount of water introduced into the cylinder, as too much water can be harmful for the cylinder's condition or the compressor's wheels [14, 10].

Nitrogen Enrichment

Except from water, dilution of the charge air can be also achieved with the use of inert gasses, such as nitrogen. Nitrogen reduces part of the charge air oxygen, leading to a lower combustion rate and lower peak temperatures, thus reducing the amount of NO_x formed during the combustion. A major disadvantage is the need of an extra supply of nitrogen or alternatively, an oxygen filter.

Exhaust Gas Recirculation (EGR)

In the automotive industry, EGR has been a well-established method for NO_x reduction for a long time. Similarly to the addition of an inert gas for air dilution, like nitrogen, the EGR system utilises the engine's exhaust gas, which is a mixture of inert gases (N_2 , CO_2 , H_2O) and excess air, and mixes it with the charge air [25]. During EGR operation, part of the exhaust gas is recirculated back in the scavenging manifold, after a cooling and cleaning process. The reduction in NO_x formation derives partly from the reduction of the cylinder's oxygen concentration, which results in a slower combustion rate, and partly from the introduction of water and carbon dioxide into the cylinder, with higher heat capacities compared to that of the oxygen. For these reasons, a lower in-cylinder peak temperature is to be expected and thus, a reduction in NO_x emissions. This reduction is almost linear to the ratio of the recirculated exhaust gas [26].

A major advantage of the EGR system compared to other dilution methods, is that the inert gas is readily available in the form of exhaust gas, whereas in nitrogen enrichment or water injection, additional equipment is required. Moreover, compared to the SCR, EGR systems are more suitable for low-load operation and manoeuvring in costal and harbour areas [14], which is essential considering that Tier III NO_x limits apply close to coastal areas. On the other hand, the implementation of EGR results in an increase in soot emissions, due to the reduction of the cylinder's available oxygen to oxidize soot, as well as a small increase of carbon monoxide (CO) emissions, due to the lower local air-fuel ratios inside the cylinder. Additional problems that arise when EGR is implemented on low-speed, two-stroke marine diesel engines are, the handling of corrosive components in the exhaust gas, the preservation of the cylinder's scavenging and the allowing of switch-ability. More specifically, marine diesel engines burn HFO, with higher sulphur concentration compared to distillate fuels. For that reason, some kind of cleaning of the exhaust gas is required before it is recirculated back to the scavenging manifold, to avoid fouling and damaging the air cooler and the receiver components. The installation of a pre-scrubber and a scrubber ensure the cleaning of the exhaust gas. In order to preserve the cylinder's scavenging (usually uniflow in case of marine engines), the scavenging manifold's pressure must always be higher compared to that of the exhaust manifold. To introduce the exhaust gas back to the scavenge air receiver, an electrically driven blower must be installed, enabling the exhaust gas to overcome the pressure difference between the manifolds [14]. Finally, the easy switch-ability is of particular importance due to the fact that, for the engine's operation outside of the ECAs, IMO Tier II limits still remain in force and thus, EGR operation is not required.

At this point, it must be noted that large diesel engines, like the one examined in the present study, usually operate with rather high values of air-fuel equivalence ratio λ , in order to reduce the thermal load on the combustion chamber's components (the excess air has a thermal heat capacity which reduces the cycle temperatures) and achieve a short and efficient combustion, with low PM emissions. For that reason, considerably higher EGR rates are required to accomplish the same reduction in NO_x emissions, compared to smaller engines [25]. The EGR rate is the amount of exhaust gas, referred to the total intake flow (intake air + recirculated exhaust gas). For an easy evaluation of the EGR system's effectiveness, the oxygen concentration is the intake air is used, instead of the EGR rate (Figure 2.6).



Figure 2.6: O_2 concentration as a function of λ and EGR rate [25]

For low-speed engines, especially those with electronic control of the exhaust valve, Wärtsilä proposes the water-cooled residual gas technique (WaCoReG), which is a combination for EGR and DWI. More specifically, with appropriate control of the exhaust valve, part of the exhaust gas is left in the cylinder after the combustion. This would normally lead to an increased thermal load on the combustion chamber's components, and to inferior combustion. These drawbacks can be avoided though, if the remaining exhaust gas is cooled by direct injection of water in the cylinder, which will reduce the temperature levels [10].

On the other hand, MAN Diesel & Turbo has been experimenting on the EGR layout presented in Figure 2.7. The installation of a wet scrubber, along with an appropriate water treatment system, is of paramount importance in order to keep the scavenge air clean and protect the combustion chamber components. MAN also confirmed that EGR can be applied in Tier II operation, in order to obtain significantly improved SFOC, especially at part load operation [27].



Figure 2.7: Schematic diagram of the EGR system developed by MAN [10]

The EGR setups that have been developed and applied over the years for marine engines, will be discussed in detail on Chapter 2.4.

2.3.2 Exhaust After-Treatment Systems For NO_x Reduction

Apart from the in-cylinder reduction strategies described, a number of after-treatment NO_x reduction techniques have been also developed throughout the years. Though an expensive solution, after-treatment systems can effectively reduce NO_x emissions to levels even as low as 98%, through the use of appropriate catalysts. They can replace the in-cylinder technologies or they can be used in combination, for a further reduction and cleaning process of the exhaust gas. The most common method that has been used in the marine industry is the Selective Catalytic Reduction (SCR)

Selective Catalytic Reduction (SCR)

Selective Catalytic Reduction (SCR) systems were first implemented in land-based power plant stations in the late seventies. The first marine application took place in 1989 by MAN Diesel & Turbo [28]. During SCR operation, the exhaust gas is sprayed with ammonia before entering the SCR catalytic reactor. Into the reactor, NO_x is reduced catalytically to harmless nitrogen (N₂) and water (H₂O) through a chemical reaction, with the help of the reactant (ammonia). From the various reactants that are available today, the safest to store is urea. For that reason, the ammonia is introduced into the system in the form of aqueous urea ((NH₂)₂CO_(aq)), which decomposes to ammonia and carbon dioxide when injected into the exhaust gas, according to equation (2.8).

$$(NH_2)_2 CO_{(aq)} + H_2 O \to 2NH_3 + CO_2$$
 (2.8)

After the injection, the mixture of exhaust gas and urea passes through the SCR catalyst, which is a reactor that consists of blocks with a large number of channels (Figure 2.8), providing a large surface area in which the catalytic process takes place.



Figure 2.8: SCR reactor operational principle [29]

 NO_x is decomposed to N_2 and H_2O , according to the following equations [26].

$$4NO + 4NH_3 + O_2 \to 4N_2 + 6H_2O \tag{2.9}$$

$$2NO + NO_2 + 4NH_3 \to 4N_2 + 6H_2O \tag{2.10}$$

$$2NO_2 + 4NH_3 + O_2 \to 3N_2 + 6H_2O \tag{2.11}$$

Before the exhaust gas is mixed with the urea, first it passes through an oxidizer catalyst that oxidises the CO and any unburned HC, as well as a particulate filter to remove the soot.

The catalytic conversion rate of the SCR system is heavily dependent on the amount of urea injected into the exhaust gas. Increased doses of urea result in high conversion rates. Nevertheless, excessive doses must be avoided, as it can cause ammonia slip downstream of the reactor (unreacted ammonia), which is detrimental both for the system's operation and the operational cost [10]. The doses usually derive from the engine's speed and load and are controlled by a process computer, which calculates the NH₃ feed rate based on testbed results [14]. Additionally, in order for the SCR system to work efficiently, the catalyst must operate in a specific temperature window. If the temperature is too high, NH₃ will burn rather than

react with the NO and NO₂ (for temperatures approaching 500°C). Furthermore, high temperatures result in increased sulphur trioxide (SO₃) formation, which reacts with water to create sulphuric acid (H₂SO₄), that can be highly corrosive. On the other hand, at low temperatures, the H₂SO₄ is neutralised by ammonia. This results to the formation of ABS (Ammonium Bisulphate, NH₄HSO₄), a sticky product that accumulates and clogs the SCR elements. The lower limit of the temperature that the SCR system must operate is dictated by the sulphur content in the fuel (Figure 2.9). For engines operating with HFO, the ideal SCR inlet temperature is between 330-350°C, while for low sulphur fuel oils ($\leq 0.1\%$), 310°C would be sufficient. This is also affected by the pressure of the exhaust gas, hence the operational load of the engine [26].



Figure 2.9: Required lower temperature limits for SCR related to sulphur content and exhaust gas pressure [26]

A typical example of a SCR setup that has been proposed and used by MAN Diesel & Turbo in marine diesel engines can be seen on Figure 2.10.



Figure 2.10: High Pressure (HP) SCR system [26]

Due to the high energy efficiency of the two-stroke diesel engines, the temperature of the exhaust gas after the turbocharger is relatively low, varying from 230-260°C, depending on the engine load and the ambient conditions [28]. These temperatures are too low for the SCR system to work efficiently and for that reason, the SCR is placed on the high-pressure side of the turbocharger, right after the exhaust manifold. On Figure 2.10, the vaporiser, where the urea is injected into the exhaust gas, is placed after the exhaust manifold, followed by the SCR reactor. When the engine operates in Tier II mode, the SCR system is cut off through the Reactor Sealing Valve (RSV) and the Reactor Throttle Valve (RTV). The Reactor Bypass Valve (RBV) is open and the exhaust gas is led directly to the turbocharger. An Exhaust Gas Bypass valve (EGB) is also present, to avoid overspeeding of the turbocharger during high load operation. When the engine operates in Tier III, the RSV and RTV are open and the RBV stays closed. At low loads operation (approximately below 50%), even if the SCR is placed before the turbocharger, the temperatures are still too low for efficient operation. For that reason, a low method has been developed, in order to increase the exhaust gas temperatures. This is achieved with the Cylinder Bypass Valve (CBV), where part of the intake air is bypassed from the scavenging manifold to the turbine inlet. By reducing the quantity of the air in the cylinder, the result is a richer combustion (for a fixed amount of fuel) and thus, higher exhaust gas temperatures. At the same time, the flow through the turbocharger remains almost unchanged, meaning that the scavenge air pressure is maintained, and the combustion remains unaffected. Finally, due to the fact that the reactor and the vaporiser introduce a significant heat capacity and thermal delay between the exhaust manifold and the turbine, the installation of high performing auxiliary blowers is required, to counteract any thermal instability.

Overall, SCR systems are a powerful NO_x reduction technology, that has been proven to reduce NO_x emissions even as low as 98% on stationary power plants. However, when introduced in the marine industry, some complications and limitations can arise. Retrofit installation in existing ships has proven to be extremely difficult, due to the limited space in the engine room. This problem is more easily solved on new buildings. In addition, the danger of ammonia slip (small presence of ammonia in the exhaust gas after the catalyst) limits the systems efficiency to 90-95% NO_x reduction [14]. The system's inability to work efficiently below certain temperatures leads to certain emission control problems during cold start. Finally, it is one of the most expensive solutions for NO_x reduction, mainly due to the need for constant supply of urea.

2.4 EGR Systems

Throughout the years, various EGR systems have been developed and tested in order to comply with Tier III regulations. These systems can be divided into two major categories, the internal and the external EGR.

In internal EGR systems, part of the exhaust gas is retained inside the cylinder with appropriate valve timing. Although no extra components are needed to achieve this method, the effect on NO_x reduction is relatively low, due to the fact that the retained exhaust gas cannot be cooled. Furthermore, an unacceptable increase in SFOC and exhaust gas temperature can be observed after a certain EGR rate (about 20%). For this reason, this method is not implemented, as it doesn't fulfil the Tier III emission requirements [25].

In case of external EGR, the exhaust gas exits the engine's cylinders and is recirculated back to the scavenging manifold. This provides a much higher NO_x reduction potential, as the exhaust gas can now be cooled. Additional equipment is required in order to cool and recirculate the exhaust gas. The external EGR systems can be further divided into two categories, the Low Pressure EGR (LP-EGR) and the High Pressure EGR (HP-EGR).

LP-EGR (Long Route or Semi-Short Route) systems recirculate the exhaust gas at ambient pressure, i.e. downstream the turbine. Since the exhaust gas has been expanded in the turbine, it is merged and mixed with the intake air before the compressor stage. For single-stage turbocharging, the gas is recirculated before the only compressor in the system (Long Route), while for two-stage turbocharging, the gas is recirculated after the low-pressure compressor and before the high-pressure one (Semi-Short Route), as presented in Figure 2.11. The system has a relatively simple design and operation principle. No additional pump devices are required for exhaust gas recirculation, as the function of the EGR pump is assumed by the HP compressor. On the other hand, as the exhaust gas passes through the compressor, there is a high risk of fouling on the compressor's blades, causing a faster efficiency decrease over time. If HFO is used, even erosion and corrosion may occur. Furthermore, a failure in the EGR path involving the main compressor would also impair normal engine operation without EGR. For these reasons, the HP-EGR appears to be a more suitable choice for large marine engines, burning low quality fuel.



Figure 2.11: Semi-Short Route EGR [25]

HP-EGR (Short-Route) systems recirculate the exhaust gas at boost pressure, i.e. upstream the turbocharger. This means that a pumping device is required, in order to overcome the pressure difference between the scavenging and exhaust manifolds and lead the exhaust gas downstream the compressor. These systems have increased complexity and cost, due to the number of extra components required for the recirculation (pump, cooler, valves etc.), but offer a wide operational range. At the same time, a potential failure in the EGR components doesn't obstruct the engine's normal operation. The role of the pump could be undertaken either by a small turbocharger or by an electrically driven blower. In case of the turbocharger, part of the exhaust gas is used in order to drive the turbine. This exhaust gas represents an energetic loss, hence a reduction in turbocharging efficiency is to be expected (Figure 2.12). For large two-stroke engines that require high EGR rates (between 30% and 40%), the penalty on turbocharging efficiency is too high, making this solution not viable. It is more suited for four-stroke applications, where the required EGR rates are lower [25].



Figure 2.12: Turbocharging efficiency with EGR turbocharger operation [25]

MAN Diesel & Turbo found in their experiments that HP-EGR with the assistance of an electrically driven EGR blower would be the most suitable solution for large two-stroke marine diesel engines [14]. To keep the installation's dimensions in an acceptable range, high-speed blowers must be considered, which have a high cost. Despite the high cost and complexity, the system now has a great flexibility since the blower can be controlled independently of the engine speed. Higher turbocharging efficiency is also to be expected, as no additional exhaust gas is required to drive the gas turbine. According to MAN [26], two different matching methods are used for the EGR system, depending on the bore size and the number of turbochargers:

- EGR with bypass, for engines of bore 70 [cm] or less and with only one turbocharger.
- EGR with turbocharger cut-out, for engines of bore 80 [cm] or greater with two or more turbochargers.

EGR With Bypass

The EGR with bypass layout is presented in Figure 2.13. The system composes of two paths, the main path, which leads the intake air through the turbocharger's compressor and air cooler, and the EGR path, which has the capacity to lead up to 40% of the exhaust gas through the pre-spray and the EGR unit to mix with the intake air of the main path.

When the engine operates in Tier II mode, only the main path is active. The EGR path's Shut-off Valve (SOV) and Blower Throttle Valve (BTV) are closed, as well as the Cylinder Bypass Valve (CBV). The Exhaust Gas Bypass valve (EGB) is fully open at high loads and partly open at lower loads in order to ensure that turbocharger overspeeding will be avoided. It is to be noted that for smaller engines with a bore of 40 [cm] or less, in high loads, the EGB is closed and the EGR path is activated instead. This way, an improved SFOC is to be expected [27], while the restriction on the turbocharger speed is still achieved, as part of the flow is directed at the EGR path.

In Tier III mode operation, the EGR path is activated by opening the SOV and BTV. Part of the exhaust gas is led through the pre-spray and spray units, also known as pre-scrubber and scrubber. The role of the scrubber is to clean the exhaust gas before introducing it into the scavenging manifold, in order to avoid corrosion due to the high concentration of SO₂. This is a common problem for engines that burn HFO

with high sulphur content. Inside the scrubbers, sea water or fresh water is sprayed into the exhaust gas. When SO_x come in contact with the water, they form sulphuric acid (H₂SO₄), which is then drained from the EGR path along with the water and sent to the Water Treatment Unit (WTU). The WTU removes the accumulated particles and the H₂SO₄ and sends the cleaned water back to the scrubbers.



Figure 2.13: EGR with bypass layout [26]

An EGR cooler is also placed, usually between the pre-scrubber and the scrubber, which is externally regulated to ensure that the recirculated exhaust gas will have the same temperature as the scavenging manifold, independently of the flow. A Water Mist Catcher must be installed after the cooler, to ensure that any condensed water in the exhaust gas will be removed from the system. Finally, the EGR blower is needed to overcome the pressure difference between the manifolds. Two blowers are usually installed in parallel, as they allow a wider mass flow operational range. When they are both running, they do so at the same speed, in order to avoid back flows. The EGR rate is controlled by both the EGR blower and the BTV. The EGB valve is closed at this mode but, due to the reduced flow in the turbocharger's turbine, the CBV is open to increase the scavenging pressure and thus, reduce the SFOC [26].

Stainless steel is used in the EGR blower and cooler, in order to deal with corrosion challenges.

EGR With Turbocharger Cut-Out

The layout for the cut-out setup is presented in Figure 2.14. The system is similar to the bypass setup, although now there is a third path, that of the smaller, cut-out turbocharger. When the engine operates without EGR, the main path and the cut-out path are open, the later through the Turbine Cut-Out Valve (TCV) and the Compressor Cut-Out Valve (CCV). The main path can lead up to 70% of the intake air through the cooler and the WMC, while the cut-out path leads up to 40% of the air through the EGR unit and the balance pipe, in order to mix with the intake air of the main path and enter the scavenging

manifold. The Blower Bypass Valve is open, while the BTV remains closed. In this case, the EGR cooler works as a normal scavenge air cooler and the scrubber trays are emptied of water, so that the air passes through an empty scrubber. The CBV is kept close in this mode.



Figure 2.14: EGR with turbocharger cut-out layout [26]

Before entering the EGR operational mode, there is an intermediate mode where the engine still operates with Tier II compliance, but with the smaller turbocharger cut out through the TCV, CCV and BBV. This step happens in low loads and is necessary in order to avoid overspeeding of the main turbocharger. It also works as a fall-back mode, in case the EGR system shuts down.

During EGR operation, the EGR path is open through the SOV and BTV. The scrubbers are now in operation and the cut-out path remains closed, in order to compensate the reduce exhaust gas amount that is now directed to the EGR path. The CBV is partially open at low loads.

Chapter 3: Engine Model

A model is a mathematical representation of a natural process. In case of internal combustion engines, an engine model is designed in order to simulate the different processes that take place inside of a real engine, through various mathematical models and equations. A proper engine model is of great importance to any engine designer, as it can be used to determine the engine's performance in any situation, while erasing the high cost of an actual test engine. In the present study, the model of the engine was developed in the in-house engine simulation tool MOtor THERmodynamics (MOTHER), which has already been implemented successfully in studies like [30] and [31].

3.1 MOTHER Simulation Program

Mathematical engine simulation models are divided into 2 main groups, the fluid-dynamics-based models and the thermodynamics-based models. Fluid-dynamics-based models are also called multidimensional models, due to the fact that the equations for the conservation of mass, chemical species and energy, apply at any location within the engine's cylinder or manifolds, and at any time. Thus, both spatial coordinates and time are necessary in order to accurately describe the system, and the equations applied are partial differential equations. These models provide us with detailed information about the spatial properties of the fluid, but their solution requires a lot of computational data and time, making them nonpractical for studying the effects that changes on the design and operational variables will have on the engine's performance and emissions. In case of thermodynamic-based models on the other hand, also known as zero-dimensional models, the only parameter required to describe the system is time, resulting in the use of ordinary differential equations. These models are based on the thermodynamic analysis of the cylinder contents, and the First Law of Thermodynamics is applied to the various engine cylinders and manifolds, which are considered as open systems.

MOTHER is a comprehensive thermodynamic engine performance prediction code which falls under the category of zero-dimensional or control volume simulation models. It considers the engine as a series of interconnected volumes via valves or ports. Spatial uniformity of fluid properties and constant rate of change of parameters ("quasi-steady") is assumed in each control volume and at any given time step. It is also assumed that work, heat and mass transfer take place across the boundaries of each control volume. The program has been under development for a number of years in the Laboratory of Marine Engineering (LME), and is capable of predicting the performance of an engine under both steady state and transient conditions.

The four governing equations that are applied in any control volume are presented below:

$$\dot{T} = f(\dot{U}, \dot{H}, \dot{\varphi}, \dot{Q}, \dot{W}) \tag{3.1}$$

$$\dot{m} = f(P, T, g, R, A_{flow}, C_d) \tag{3.2}$$

$$\dot{m} = \sum \dot{m}_j \tag{3.3}$$

$$P = f(m, R, T, V) \tag{3.4}$$

Equation (3.1) describes the non-steady flow of energy inside the control volume, manipulated so as to express the rate of change of temperature \dot{T} , in terms of other parameters. The rate of change of the internal energy \dot{U} and enthalpy \dot{H} of the working fluid is obtained by reference to thermodynamic property data for air/fuel mixtures. The rate of change of equivalence ratio $\dot{\phi}$, is obtained by summing the air and fuel exchanges. The rate of change of heat \dot{Q} , depends on the heat released by combustion and the heat lost due to heat transfer. The rate of change of work \dot{W} , depends on the rate of change of the control volume based on the engine geometry, as well as the instantaneous pressure.

Equation (3.2) gives us the mass flow between interconnected volumes, which depends on the instantaneous pressure P, temperature T and mixture properties in each volume, as well as the geometry dependent flow area A_{flow} and discharge coefficient C_d of the restrictions between the volumes (valves, ports etc.).

At the end of each computational step, a summation of mass exchanges is made for each control volume, given by equation (3.3), which expresses the conservation of mass.

Finally, the equation of state (3.4) is used to determine the instantaneous pressure *P*, based on the actual volume *V*, mass *m*, temperature *T* and fluid properties.

The resulting set of coupled differential equations is solved numerically for all the control volumes of the system, with a typical resolution of one degree crank angle or smaller (for more precision). Various submodels are also included, corresponding to the different processes inside the engine's cylinders or other subsystems, like the turbocharger. Typical examples are a combustion model, a heat transfer model, a friction model etc.

The philosophy of MOTHER is based on the concept of Basic Engineering Elements (BEE), which allows the representation of a wide range of complex configurations, like a diesel engine, with a limited number of simple elements. MOTHER offers a number of BEE, such as thermodynamic elements, divided into flow receivers (cylinders, plenums) and flow controllers (valves, compressors, turbines), mechanical elements (crankshaft, shaft loads etc.) and controller elements (speed governor, PID controllers).

In Table 3.1, all the available BEE that MOTHER provides for the modelling of any engineering configuration are presented. Detailed information about each one and their operation can be found in [32].

| Thermodyna | mic Elements | Mechanical Elements | Control Elements |
|----------------|------------------|---------------------|-------------------------|
| Flow Receivers | Flow Controllers | | |
| Cylinder | Valve | Crank Shaft | Speed Governor |
| Plenum | Heat Exchanger | Shaft | PID Controller |
| Fixed Fluid | Compressor | Shaft Load | |
| | Turbine | Clutch | |
| | | Gear Box | |

Table 3.1: BEE available in MOTHER

3.2 Tier II Engine Model

In the present chapter, a brief description of the most important subsystems, along with their mathematical models, is provided, to give a better understanding of the various processes that are influenced by the implementation of an EGR system on the engine. The engine modelled in the present study is a MAN B&W 6S70MC-C mechanically controlled engine, equipped with two turbochargers. Engine particulars are summarized in Table 3.2.

| <i>G i i i i i i i i i i</i> | | | | | |
|------------------------------|--------------------|--|--|--|--|
| MAN B&W 6S70MC-C | | | | | |
| No. of Cylinders | 6 | | | | |
| Bore | 700 mm | | | | |
| Stroke | 2800 mm | | | | |
| Compression Ratio | 17.57 | | | | |
| Power/cylinder | 3110 kW | | | | |
| Engine Speed | 91 rpm | | | | |
| Mean Effective Pressure | 19 bar | | | | |
| Firing Order | 1-5-3-4-2-6 | | | | |
| Turbocharger Unit(s) | $2 \times ABB T/C$ | | | | |

Table 3.2: Engine characteristics

The various components that constitute the engine model are:

- Cylinders
- Scavenge and Exhaust Manifolds
- Turbocharger
- EGR Components (only active in Tier III operation)

The outside environment is regarded as a fixed fluid element, with constant pressure, temperature and chemical composition. The working mixture consist of 11 gas species plus the fuel. These are O_2 , N_2 , H_2O , H, H_2 , N, NO, O, OH and CO.

Cylinder Model

In MOTHER simulation program, three major sub-systems are interconnected to form the cylinder unit. These are the actual cylinder, along with its mathematical models that describe the various in-cylinder processes (combustion, heat transfer etc.), the inlet ports and the exhaust valves.

For an overall simulation code like MOTHER, single zone thermodynamic models are an indicated solution in order to describe the combustion process, due to their simplicity. In single zone models, the cylinder charge is assumed to be a homogenous mixture of ideal gases at all times, whose instantaneous state can be described by its pressure, temperature and equivalence ratio. Fuel is added to the cylinder during the combustion period to increase the energy and fuel-air ratio of the mixture. The fuel is considered to burn instantaneously and its effect on ignition delay is ignored.

In the present study, the model used to describe the combustion process was developed by Woschni and Anisits. It is based on an S-curve general model and is described via the mass fraction of fuel burnt x_b , as

$$x_b = \frac{m_b}{m_{tot}} = 1 - e^{-a \cdot \left(\frac{\theta - \theta_0}{\Delta \theta_b}\right)^{m+1}}$$
(3.5)

where m_b [kg] is the burnt fuel, m_{tot} [kg] the total fuel injected into the cylinder, θ [deg] the crank shaft angle θ_0 , [deg] the crank shaft angle at the start of combustion, $\Delta \theta_b$ [deg] the total duration of combustion and a, m are adjustable parameters which fix the shape of the S-curve. The basis of their approach was to assume that the combustion characteristics are known for a reference condition and based on them, any other condition can be defined. It is to be noted that for the calculation of θ_0 , the ignition delay is also considered. Detailed information for the calculation of each combustion parameter can be found in [32].

A friction model is also applied in order to calculate the total engine losses, expressed as friction mean effective pressure (fmep). The model was developed by Mc Auly et al. and assumes that the total losses vary linearly with the peak pressure P_{max} and the piston speed V_p , according to the following equation

$$fmep = k_1 + k_2 \cdot P_{max} + k_3 \cdot V_p \tag{3.6}$$

where k_1 [Pa], k_2 , k_3 [Pa/m/s] are constants that differ for each engine.

A heat transfer model from the gas to the cylinder walls and from each wall to the coolant is also attached to the cylinder model. The instantaneous heat fluxes q from the gas to each cylinder wall (cylinder head, piston crown, upper and lower part of the liner) are calculated at each step of the simulation, according to the equation

$$q = h \cdot A \cdot \left(T_{gas} - T_{wall}\right) \tag{3.7}$$

where $h [kW/m^2K]$ is the gas-cylinder instantaneous spatial average heat transfer coefficient, $A [m^2]$ the respective cylinder part wall gas side area, $T_{gas} [K]$ the instantaneous cylinder gas temperature and T_{wall} [K] the respective cylinder part wall surface temperature. The heat transfer coefficient h is calculated based on Woschni's equation, as

$$h = 0.00326 \cdot B^{-0.2} \cdot P^{0.8} \cdot T_{gas}^{-0.55} \cdot w^{0.8}$$
(3.8)

were B [m] is the cylinder bore, P [bar] the cylinder gas pressure and w [m/s] the average cylinder gas velocity.

A full heat transfer wall to coolant model is used for the cylinder, where the equivalent thermal circuit of each cylinder wall is considered, in order to calculate the cylinder wall surface temperatures (Figure 3.1).

Heat is transferred from the gas to the coolant, through each cylinder wall part. The energy balance in each part can be written as

$$Q = \frac{T_{wall} - T_{cool}}{R_{wall} - R_{cool}}$$
(3.9)

where T_{cool} [K] is the temperature of the cooling medium corresponding to each cylinder part, R_{wall} [K/kW] is the thermal resistance of the corresponding cylinder part and R_{cool} [K/kW] is the thermal resistance of the cooling medium, which can be calculated as

$$R_{cool} = \frac{1}{h_{cool} \cdot A_{cool}} \tag{3.10}$$

In (3.10), h_{cool} [kW/m²K] is the heat transfer coefficient of the cooling medium and A_{cool} [m²] the cooling medium side area.



Figure 3.1: Equivalent thermal circuits of cylinder wall parts [32]

A different approach is used for each cylinder part. The cylinder head is considered as a flat plate of equivalent surface area (excluding the valve face area). The liner is modelled as cylindrical duct, divided into two parts, the upper and the lower part. The piston is also modelled as two parts, the piston crown, which is considered as a flat plate and the piston skirt, which is a cylindrical duct.

As far as the inlet ports and the exhaust valves are concerned, a certain operational profile about their opening and closing is loaded into MOTHER (Figure 3.2). No heat transfer from the gas to the coolant takes place as the scavenge air passes through the inlet ports, while a full thermal model is applied in the case of the exhaust valves, as the exhaust gas passes through them. The exhaust valve is divided into the face, the back, the seat, the stem and the core, and it is assumed that each one of them has a uniform temperature, which does not vary during one engine cycle. The higher exhaust valve effective area corresponds to higher engine loads.



Figure 3.2: Inlet ports (a) and exhaust valves (b) operational profile

Scavenge and Exhaust Manifolds

The two manifolds are modelled as flow receivers. The flow from the two compressors (during Tier II operation) or from one compressor and the EGR path end up in the scavenge receiver, before entering the cylinders, while the exhaust gas produced during the combustion in the cylinders, ends in the exhaust manifold.

A similar approach as to the one presented for the engine's cylinder is applied for the gas to plenum walls heat transfer model. The calculation of the instantaneous heat flux q is calculated based on equation (3.7), and the heat transfer coefficient of the plenum gas h_{gas} [kW/m²K] can be calculated by a Nusselt-Reynolds-Prandtl relationship, as

$$Nu = C_1 \cdot Re^{C_2} \cdot Pr^{C_3} \tag{3.11}$$

where $C_1 = 0.021 - 0.023$ for intake plenums

 $C_1 = 0.035 - 0.045$ for exhaust plenums $C_2 = 0.75 - 0.85$ $C_3 = 0.3 - 0.5$

For the wall to coolant heat transfer, a simple model is utilised, in which each plenum is regarded as a cylindrical duct and its surface temperature is assumed to be between the temperature of the cooling medium and the adiabatic temperature of the wall. The latter is a theoretical temperature, corresponding to the temperature of the wall if no heat was transferred from the gas to the plenum wall (adiabatic case).

The pressure and the temperature at each manifold are calculated and used in order to calibrate the model (see Chapter 3.4). In order to ensure uniflow scavenging of the cylinder at all times, it is important that the pressure in the scavenging manifold is higher to the pressure in the exhaust manifold. In case of the scavenge receiver, it is also important to calculate the oxygen content during EGR operation, for an easy evaluation of the EGR system's effectiveness (see also Figure 2.6).

Turbocharger

The turbocharger can be modelled as a three-piece system, comprising of two flow controllers, i.e. the compressor and the turbine, mechanically connected with a shaft for power transmission. The power produced by the expansion of the exhaust gas in the turbine is used to drive the compressor and lead the compressed air into the scavenging manifold. Right after each compressor, an air cooler is installed in order to reduce the density of the inlet air. Two auxiliary blowers are also installed, to ensure sufficient cylinder scavenging during engine start and low load operation.

Compressor

To predict the performance of the compressor, a digital representation of its performance map is used. This is obtained experimentally, under steady-state conditions and is usually provided by the manufacturer. The compressor map represents the compressor speed and efficiency as functions of the compressor corrected air mass (or volumetric) flow, with respect to the pressure ratio (pressure downstream of the compressor to the pressure upstream). If an air cooler is installed after the compressor, the pressure drop of the air is taken into consideration. The compressor map used in the present study is depicted in Figure 3.3.



Figure 3.3: Compressor performance map

The stable operational range of the compressor is limited by the surge line at low flow rates, to avoid instability phenomena, and by the choke line at high flow rates, to avoid choking of the flow while exiting the compressor's diffuser. The performance map is given in terms of corrected quantities, as far as the rotational speed N_{corr} and the volumetric flow \dot{V}_{corr} are concerned. These are calculated based on the following equations

$$N_{corr} = N \cdot \sqrt{\frac{T_{ref}}{T_{in}}} \tag{3.12}$$

$$\dot{V}_{corr} = \dot{V} \cdot \sqrt{\frac{T_{in}}{T_{ref}}}$$
(3.13)

where N [RPM] and \dot{V} [m³/s] are the actual compressor rotational speed and volumetric flow respectively, T_{ref} [K] is the reference temperature, taken as 298 [K] and T_{in} [K] is the compressor's inlet temperature.

The gas to wall and wall to coolant heat transfer models used for the compressor, are similar to the ones described for the exhaust and scavenging manifolds.

<u>Turbine</u>

Similarly to the case of the compressor, a digital representation of an experimentally obtained efficiency map and swallowing capacity map, provided by the manufacturer, are used to predict the performance of the turbine. The efficiency map provides a plot of the turbine's efficiency against the turbine's expansion ratio (the ratio between the pressure upstream of the turbine to the pressure downstream), while the swallowing capacity map consists of the turbine mass flow plotted against the expansion ratio. The two maps are presented in Figure 3.4 and Figure 3.5.



Figure 3.4: Turbine efficiency map



Figure 3.5: Turbine swallowing capacity map

As in the compressor's case, the gas to wall and wall to coolant heat transfer models are similar to the ones presented for the scavenge and exhaust manifolds.

3.3 EGR Components

Based on the layout proposed by the engine maker (MAN Energy Solutions) for two-stroke diesel engines with two turbochargers (Figure 2.14), a similar configuration was developed in MOTHER (Figure 3.6), by adding all the extra components necessary to the already existing model of the engine. During EGR operation, the second turbocharger is deactivated, as part of the exhaust gas mass flow is now directed to the EGR path. Furthermore, for simplification purposes, the scrubbers and the WMC were omitted from the engine model since they had little effect on the thermodynamics and their pressure drop could be considered negligible [33].



Figure 3.6: Engine structure and EGR configuration in MOTHER

Blower Throttle Valve (BTV)

The BTV is the system's component that controls the mass flow of the exhaust gas passing through the EGR path, thus the EGR rate, defined as

$$EGR = \frac{\mathbf{m}_{egr}}{\dot{\mathbf{m}}_{cyl}} \times 100 \tag{3.14}$$

where \dot{m}_{egr} [kg/s] is the exhaust mass flow passing through the BTV and \dot{m}_{cyl} [kg/s] is the total exhaust mass flow exiting the engine's cylinders.

The flow through the BTV may be analysed on a quasi-steady basis. It is assumed that the flow is one dimensional, and any secondary flow effects (boundary layer separation, friction etc.) are taken into account through the introduction of an empirically based discharge coefficient. The instantaneous mass flow rate m [kg/s] through the valve, is a function of the pressure ratio, i.e. the pressure in each flow receiver on either side of the valve. It can be calculated by applying the following equation

$$\dot{\mathbf{m}} = C_d A_v P_u \sqrt{\frac{2\gamma}{\gamma - 1} \frac{1}{R T_u} \left[\left(\frac{P_d}{P_u}\right)^{\frac{2}{\gamma}} - \left(\frac{P_d}{P_u}\right)^{\frac{(\gamma+1)}{\gamma}} \right]}$$
(3.15)

where C_d is the valve discharge coefficient, A_v [m²] the geometric valve flow area, P_u [N/m²] the gas pressure upstream of the valve, P_d [N/m²] the gas pressure downstream of the valve, T_u [K] the gas temperature upstream of the valve, R [J/kgK] the gas constant and γ the specific heat ratio.

The BTV was modelled as a one-way valve with variable opening, which changes depending on the system's need on recirculated gas. The valve opening is regulated for each steady-state load case (see also Chapter 3.4), in order to provide an adequate exhaust gas mass flow into the scavenge receiver. The oxygen concentration in the scavenge receiver works as an indicator on whether the EGR system is effective or not (compared to the available data from the engine's shop tests). The gas to wall heat transfer model is calculated based on the Nusselt-Reynolds-Prandtl equation, while no wall to coolant heat transfer model is considered.

BTV Outlet and EGR Blower Outlet Plenums

According to [32], connection of two or more flow controllers in series is prohibited. A flow controller must always be connected on either side to a flow receiver. For that reason, the BTV outlet and EGR blower outlet plenums are incorporated into the layout, in order to connect the BTV, the EGR blower and the EGR cooler. The gas to wall and wall to coolant heat transfer models are similar to the ones presented for the scavenge and exhaust manifolds and represent the heat losses throughout the EGR path.

EGR Cooler

The EGR cooler's role is to lower the recirculated exhaust gas temperature so it will be similar with the charge air temperature in the scavenge receiver. The gas temperature after the cooler can be calculated by the following equation

$$T_{out} = (1 - \varepsilon)T_{in} + \varepsilon T_{C,in} \tag{3.16}$$

In equation (3.16), T_{in} [K] and $T_{C,in}$ [K] are the temperatures of the gas and the coolant respectively, when entering the EGR cooler and ε is the effectiveness of the cooler, given by

$$\varepsilon = C_1 + C_2 \dot{\mathbf{m}}_{HE} + \dot{\mathbf{m}}_{HE}^2 \tag{3.17}$$

where \dot{m}_{HE} [kg/s] is the gas mass flow rate through the cooler (heat exchanger) and C_1 , C_2 , C_3 are constants. In the present study, the effectiveness of the EGR cooler is considered constant and equal to 0.94.

EGR Blower

In order to overcome the pressure difference between the exhaust and scavenging manifold, a blower was added to the model. The EGR blower was modelled as a simple compressor in MOTHER. The heat transfer models (gas to wall and wall to coolant) were configured similarly to the turbocharger's compressor.

To properly simulate the operation of the blower, a performance map covering its operational range is essential. This kind of maps are usually provided by the manufacturer and consist of various diffuser positions with a unique speed line for each position. In the present study, one iso-speed line of the map was provided, for various diffuser positions ranging from 0 to 12. Based on the available datasets, the diffuser position was kept constant in all of them, with a constant value of 100%, corresponding to diffuser position 12, according to the equation

$$u_{diff}[\%] = \frac{diff_i - diff_{min}}{diff_{max} - diff_{min}} = \frac{diff_i - 0}{12 - 0}$$
(3.18)

The map can be either expressed in the dimensionless space, using the Flow Coefficient Φ and the Head Coefficient Ψ , or through the volumetric flow and the pressure ratio.

The dimensionless parameters Φ and Ψ can be calculated according to the following equations [1]

$$\Phi = \frac{Q}{N \pi R^3} \tag{3.19}$$

$$\Psi = \frac{2 c_p T (\Pi^{\frac{\gamma-1}{\gamma}} - 1)}{(N R)^2}$$
(3.20)

where Q [m³/s] is the volumetric flow rate through the blower, N [rad/s] the blower angular speed, R [m] the blower's blade radius, c_p [kJ/kgK] the specific heat capacity of the air, T [K] the ambient temperature and Π the pressure ratio over the blower.

Since the data for only one constant speed curve were provided by the manufacturer, the rest of the blower's map had to be extrapolated. In order to predict the blower's performance in various rotational speeds, the affinity laws, as described in [34], were applied. Since the impeller size is always constant, the following equations can be used

$$\Phi_i = \Phi_{ref} \quad \rightarrow \quad Q_i = \frac{N_i}{N_{ref}} Q_{ref}$$
(3.21)

$$\Psi_i = \Psi_{ref} \quad \rightarrow \quad \Pi_i = 1 + \left(\Pi_{ref} - 1\right) \left(\frac{N_i}{N_{ref}}\right)^2$$
(3.22)

where the index *ref* represents the already known parameters for the provided iso-speed line and the index *i* represents the parameters for any other operational speed of the blower.

Based on the aforementioned equations, the blower's map was created using Matlab Software (Figure 3.7).

As it has already been mentioned, the EGR blower is electrically driven by a high-speed generator. In order to simulate the operation of the generator that drives the blower in MOTHER, two extra BEE are incorporated. These are a shaft load and a shaft element, that connects the load to the EGR blower. Since the role of the generator is to produce and provide power to the EGR blower, the shaft load acquires negative torque values throughout the simulation. In MOTHER simulation code, negative torque values denote that the shaft load produces power, instead of absorbing it. Thus, the operation of the generator can be simulated accordingly.



Volumetric Flow Q (m³/s)

Figure 3.7: EGR blower performance map

3.4 Steady-State Model Calibration

The initial model of the engine was setup for Tier II operation, without an EGR system. After the EGR components were added to the model, the model was recalibrated for EGR operation. The calibration was carried out for four different engine loads, at 100%, 90%, 75% and 50% of the engine's load. The shop test reports from a different, slightly bigger engine (6S80ME) were available and used to qualitatively assess the model's accuracy. Furthermore, results from the research done by the engine manufacturer MAN Energy Solutions [27] were also used. According to these results, operation of the engine with EGR generally leads to a reduced combustion rate due to the reduced oxygen, as well as an increased burning time (Figure 3.8). The results in Figure 3.8 correspond to the 75% load case scenario and are presented as a reference for validation. The results of the MOTHER simulation code seem to be in agreement with the findings of MAN.



Figure 3.8: Fuel Burning Rate [kg/s] (a) and Cumulative Fuel Burnt [kg] (b) with and without EGR operation for 75% load on a 4T50ME-X engine [27]

In order to capture the engine's behaviour under the new operational conditions (EGR), the cylinder's combustion model, as well as the heat transfer model, had to be modified. During the steady-state EGR calibration trials, it was concluded that the parameter a of the equation (3.5) is crucial in regulating the intensity of the combustion rate. Therefore, a perturbation analysis was performed to identify the influence of this parameter in combustion and, as a result, a proper value was selected for each operational load case with EGR, to simulate the new combustion profile and minimize errors. The same procedure was followed for the case of the heat transfer model, where a proper value for the heat transfer factor of each cylinder component was chosen, in order to accurately simulate the increased heat capacity of the charge in the cylinder, due to the presence of the exhaust gas. After the model's recalibration, these changes are visible in the fuel burning rate and cumulative fuel burnt diagrams, for all of the load cases. Regarding the in-cylinder temperature variation, EGR operation results in peak temperature below 1800 K, leading to a significant thermal NO_x reduction [35].

The average oxygen molar concentration $X_{O,scav}$, is included in the following tables, in order to monitor its reduction. Furthermore, according to [27], the reduced combustion rate and the increased combustion duration result in an increased specific fuel oil consumption, which is also in agreement with the predicted results from the present model. Other thermodynamic parameters, such as the maximum cylinder pressure (p_{max}), the turbocharger's rotational speed ($N_{T/C}$) and the exhaust and scavenging manifold's pressure and temperature are also presented and compared with the available measured values from the shop trials. Looking at the values of the model's parameters during EGR operation, it is observed that their deviation from the values during normal engine operation follow the same trend compared to the parameters in the shop tests and, to some extent, this deviation is similar. It is to be noted that the temperature in the scavenge receiver during EGR operation is similar to the one during normal operation, due to the presence of the EGR cooler. The reduction of the relative air-fuel ratio, due to the reduction of the $X_{O,scav}$ in the scavenge receiver, is also visible.

Further bellow, the results of the parameter calibration for each one of the aforementioned load cases is presented, along with the fuel burning rate and the cumulative fuel burnt profiles, as well as the variation of the in-cylinder temperature.

| 10 | 00% | p_{max} | SFOC | N _{T/C} | T_{exh} | p_{exh} | T_{scav} | p_{scav} | X _{O,scav} | λ (Relative AFR) |
|------------|----------|-----------|---------|------------------|-----------|-----------|------------|------------|---------------------|--------------------------|
| | 70 /0 | [bar] | [g/kWh] | [RPM] | [K] | [bar] | [K] | [bar] | [-] | [-] |
| Simulation | No EGR | 150.7 | 180.7 | 15660 | 634 | 3.56 | 313 | 3.87 | 0.210 | 2.139 |
| Model | EGR | 146.5 | 187.9 | 16865 | 674 | 4.00 | 312 | 4.17 | 0.177 | 2.097 |
| Widdei | Diff [%] | -2.80 | 3.96 | 7.69 | 6.40 | 12.38 | -0.23 | 7.70 | -15.72 | -1.99 |
| Shop Tests | Diff [%] | -1.34 | 1.73 | 4.94 | 2.21 | 10.46 | 0.00 | 10.18 | -15.52 | - |

100% Load Case

Table 3.3: Model parameters for the 100% load case



Figure 3.9: Fuel Burning Rate [kg/s] with and without EGR operation, 100% load



Figure 3.10: Cumulative Fuel Burnt [kg] with and without EGR operation, 100% load



Figure 3.11: In-Cylinder Temperature Variation [K] with and without EGR operation, 100% load

| 0 | 00/ | p_{max} | SFOC | N _{T/C} | T_{exh} | p_{exh} | T_{scav} | p_{scav} | X _{O,scav} | λ (Relative AFR) |
|------------|----------|-----------|---------|------------------|-----------|-----------|------------|------------|---------------------|--------------------------|
| 2 | 070 | [bar] | [g/kWh] | [RPM] | [K] | [bar] | [K] | [bar] | [-] | [-] |
| Simulation | No EGR | 150.6 | 178.4 | 14897 | 608 | 3.25 | 308 | 3.56 | 0.210 | 2.186 |
| Model | EGR | 146.8 | 183.5 | 15561 | 617 | 3.58 | 310 | 3.80 | 0.174 | 2.174 |
| Model | Diff [%] | -2.54 | 2.88 | 4.46 | 1.54 | 9.96 | 0.78 | 6.68 | -17.22 | -0.54 |
| Shop Tests | Diff [%] | -0.88 | 1.53 | 5.56 | 1.22 | 13.26 | 0.00 | 10.95 | -16.59 | - |

Table 3.4: Model parameters for the 90% load case

90% Load Case



Figure 3.12: Fuel Burning Rate [kg/s] with and without EGR operation, 90% load



Figure 3.13: Cumulative Fuel Burnt [kg] with and without EGR operation, 90% load



Figure 3.14: In-Cylinder Temperature Variation [K] with and without EGR operation, 90% load

75% Load Case

Table 3.5: Model parameters for the 75% load case

| 7 | 5% | p _{max} | SFOC | N _{T/C} | T_{exh} | pexh | T _{scav} | p _{scav} | X _{O,scav} | λ (Relative AFR) |
|------------|----------|------------------|---------|------------------|-----------|-------|-------------------|-------------------|---------------------|--------------------------|
| / | 570 | [bar] | [g/kWh] | [RPM] | [K] | [bar] | [K] | [bar] | [-] | [-] |
| Simulation | No EGR | 135.5 | 178.5 | 13881 | 586 | 2.83 | 305 | 3.13 | 0.210 | 2.200 |
| Model | EGR | 131.2 | 180.7 | 14364 | 584 | 3.05 | 309 | 3.31 | 0.174 | 2.199 |
| WIOdel | Diff [%] | -3.12 | 1.25 | 3.48 | -0.30 | 7.58 | 1.45 | 5.73 | -17.17 | -0.07 |
| Shop Tests | Diff [%] | -1.92 | 0.50 | 8.52 | -1.82 | 16.93 | 0.32 | 17.57 | -17.11 | - |



Figure 3.15: Fuel Burning Rate [kg/s] with and without EGR operation, 75% load



Figure 3.16: Cumulative Fuel Burnt [kg] with and without EGR operation, 75% load



Figure 3.17: In-Cylinder Temperature Variation [K] with and without EGR operation, 75% load

50% Load Case

| 5 | 004 | p _{max} | SFOC | N _{T/C} | T_{exh} | pexh | T _{scav} | p_{scav} | X _{O,scav} | λ (Relative AFR) |
|------------|----------|------------------|---------|------------------|-----------|-------|-------------------|------------|---------------------|--------------------------|
| | 070 | [bar] | [g/kWh] | [RPM] | [K] | [bar] | [K] | [bar] | [-] | [-] |
| Simulation | No EGR | 105.8 | 180.5 | 11467 | 561 | 2.05 | 300 | 2.29 | 0.210 | 2.151 |
| Model | EGR | 101.2 | 184.7 | 11858 | 556 | 2.18 | 308 | 2.40 | 0.174 | 2.113 |
| Widdei | Diff [%] | -4.35 | 2.31 | 3.41 | -0.84 | 6.10 | 2.66 | 4.78 | -17.06 | -1.76 |
| Shop Tests | Diff [%] | -0.88 | 0.74 | 13.68 | 0.14 | 18.41 | 0.33 | 19.59 | -17.70 | - |

Table 3.6: Model parameters for the 50% load case



Figure 3.18: Fuel Burning Rate [kg/s] with and without EGR operation, 50% load



Figure 3.19: Cumulative Fuel Burnt [kg] with and without EGR operation, 50% load



Figure 3.20: In-Cylinder Temperature Variation [K] with and without EGR operation, 50% load

At this point, it is important to point out that the available shop tests used for the calibration correspond to an electronic engine. In these engines, the electronic control of the exhaust valves opening and closing, gives the engine designer the freedom to set various exhaust valve operational profiles depending on the engine load, something which is not possible in the case of mechanical engines. As a result, some thermodynamic parameters can have significant variations compared to the present model, which is applied on a mechanical engine, in order to examine EGR retrofit possibilities in existing engines. It can be seen from the tables presented, that the overall behaviour is in accordance to the available data and literature.

Chapter 4: Ship Hull and Propeller Models

In order to have a realistic representation of the load imposed to the engine during transient loading, a ship hull model and a propeller model were developed. In the following chapters, a detailed description of each model and the way they interact with the engine model is given.

4.1 Propeller Model

The propeller produces the required thrust for the ship to move. It is directly connected to the marine engine through a crankshaft, which is responsible for the transmission of torque produced by the engine to the propeller. A propeller model is thus necessary, in order to estimate the variation of the propeller thrust and torque during transient operation. There are two main types of propellers installed in ships, Fixed Pitch Propellers (FPP) and Controllable Pitch Propellers (CPP). Ordinary merchant ships that don't require a high degree of manoeuvrability (like the oil tanker used in the present study), usually employ FPP, which are less expensive and have a lower risk of presenting problems during operation. In the present study, the propeller model that was developed was based on the results of the Wageningen B-Screw Series, as presented in [36] and [37].

The most common way of operating a propeller of a conventional merchant ship is with positive rotational speed and inflow velocity. Nonetheless, there are also times when the propeller must produce thrust in the reverse direction, in order for the ship to decelerate or move astern, meaning a negative rotational speed and inflow velocity must also be considered. When studying the response of the propeller during a transient condition, the model has to encompass all the aforementioned operating conditions. For that reason, in the present Thesis, the four-quadrant propeller performance model was used, enabling the simulation of any operating condition of the propeller. The key element, which is applied in the specific model, to enable the distinction between positive and negative rotational and advance speed, is the advance angle β , estimated at 70% of the propeller radius, as

$$\beta = \arctan\left(\frac{V_A}{0.7 \cdot \pi \cdot n \cdot D_P}\right) \tag{4.1}$$

Where V_A [m/s] is the effective wake velocity, n [rps] is the rotational speed of the propeller and D_P [m] is the diameter of the propeller.

The reason that the use of this model is preferred over the conventional propeller performance models, is the advantage the advance angle β provides. More specifically, simulations that require the prediction of the propeller's performance in more than one quadrant, cannot be based on the use of the advance coefficient $J = V_A / (n \cdot D_P)$ (conventional models), as it can take an infinite value when the propeller's rotational speed *n* equals zero. On the other hand, for zero values of the propeller's rotational speed, the advance angle β is equal to 90° or 270°, depending on the direction of the advance speed V_A . As a result, by using the advance angle β instead of the advance coefficient *J*, we can efficiently describe all four propeller operating conditions, by relating them to the four quadrants (Table 4.1).

| | - | | |
|-----------------|-------------------------|-----------------|-------------------------------------|
| Quadrant | Rotational Speed | Advance Speed | Advance Angle |
| 1 st | Ahead/Positive | Ahead/Positive | $0^\circ \le \beta \le 90^\circ$ |
| 2 nd | Astern/Negative | Ahead/Positive | $90^\circ \le \beta \le 180^\circ$ |
| 3 rd | Astern/Negative | Astern/Negative | $180^\circ \le \beta \le 270^\circ$ |
| 4 th | Ahead/Positive | Astern/Negative | $270^\circ \le \beta \le 360^\circ$ |

Table 4.1: Four quadrant operation

As it can be seen from Table 4.1, each quadrant represents a potential combination of the propeller's rotational speed and advance speed. The first quadrant corresponds to the most common combination of a positive rotational and inflow speed. In the second quadrant, the inflow velocity is positive, but the rotational speed is negative, which means that the ship decelerates and the propeller produces a negative thrust. The third quadrant represents the operating condition of a ship moving astern, where both the rotational and the advance speed are negative. Finally, in the fourth quadrant, the ship is moving astern but the propeller produces positive thrust. All of the above can be summed up in Figure 4.1



Figure 4.1: Four quadrants of propeller's operation [38]

In order to calculate the advance angle β through equation (4.1), the effective wake velocity V_A must first be determined. The presence of the wake in the aft part of the ship, which is practically the local disturbance created in the water flow due to the rotation of the propeller, constitutes a common problem for self-propelled ships. It's a complex hydrodynamic problem, but a relatively simple approach is to assume that the propeller of the self-propelled ship operates in an effective wake velocity V_A , which is also called speed of advance and is lower than the ship's speed V_{SHIP} . This is justifiable, as the ship's boundary layer around the hull blocks part of the flow to the propeller, resulting in an inhomogeneous velocity field [39]. This difference in speed, which is expressed as a percentage of V_{SHIP} , is the wake fraction coefficient w and so

$$V_A = (1 - w) \cdot V_{SHIP} \tag{4.2}$$

According to [38], an accurate estimation of w has been the center of interest for a lot of researchers. Some of the most popular methods are van Lammeren's diagrams, based on the single parameter of the vertical prismatic coefficient and used only as a first approximation, Harvald's curves, based on the ship's geometric characteristics and the hull form, the equation of Schoenherr, also based on the ship's geometric characteristics and the characteristics of the propeller and finally the regression formulae presented by Holtrop and Mennen for single- and twin-screw vessels, derived from the results of model tests over a comparatively wide range of hull forms. From the above methods, the last one was considered to give the most precise results and was implemented in the present Thesis. According to [40], the wake fraction coefficient for single-screw vessels can be calculated using the equation

$$w = C_9 \cdot (1 + 0.015 \cdot C_{STERN}) \cdot [(1 + k) \cdot C_F + C_A] \cdot \frac{L_{WL}}{T_A}$$

$$\cdot \left\{ 0.050776 + 0.93405 \cdot C_{11} \cdot \frac{(1 + k) \cdot C_F + C_A}{1.315 - 1.45 \cdot C_P + 0.0225 \cdot lcb} \right\} + 0.27915$$

$$\cdot (1 + 0.015 \cdot C_{STERN}) \cdot \sqrt{\frac{B}{L_{WL} \cdot (1.315 - 1.45 \cdot C_P + 0.0225 \cdot lcb)}} + C_{19}$$

$$\cdot (1 + 0.015 \cdot C_{STERN})$$
(4.3)

In equation (4.3), C_9 , C_{11} and C_{19} are coefficients which are calculated according to [40], C_{STERN} is a constant depending on the shape of the sections in the stern area and *lcb* the longitudinal center of buoyancy written as a percentage of L_{WL} , calculated from amidships. In addition, L_{WL} [m], B [m] and T_A [m] refer to the length along the waterline, the breadth and the after draught of the ship respectively. Furthermore, C_P is the prismatic coefficient and (1 + k) is the hull form factor of the vessel. Finally, C_F is the coefficient of frictional resistance of the ship as defined in [41] and C_A is the correlation allowance coefficient between the model and the ship.

According to [42], typical values of w for single screw ships range between 0.20 and 0.45.

After calculating β and V_A , the propeller model provides an estimation of the propeller's effective thrust T_{pr} and torque Q_{pr} , through the following equations [43]

$$T_{pr} = \frac{\pi}{8} \cdot \rho \cdot C_T^* \cdot V_r^2 \cdot D_P^2 \tag{4.4}$$

$$Q_{pr} = \frac{\pi}{8} \cdot \rho \cdot C_Q^* \cdot V_r^2 \cdot D_P^3$$
(4.5)

where ρ [kg/m³] is the density of water and C_T^* , C_Q^* are the non-dimensional thrust and torque propeller coefficients. V_r [m/s] is the relative advance velocity and is calculated as

$$V_r = \sqrt{V_A^2 + (0.7 \cdot \pi \cdot n \cdot D_P)^2}$$
(4.6)

Equations (4.4) and (4.5) require the calculation of the C_T^* and C_Q^* coefficients. These coefficients describe the propeller's thrust and torque characteristics in the four-quadrants and can be based on periodic functions of the advance angle β , obtained from experiments [36, 37]. The results of these experiments were grouped together in diagrams, for a specific number of blades and expanded area ratio and for various pitch to diameter ratios P/D (Figure 4.2).



Figure 4.2: B4-70 series 4-quadrant thrust and torque coefficients for various P/D [43]

Roddy, in his work [43], expressed these coefficients as periodic functions of the advance angle β , over a range of $0^{\circ} \le \beta \le 360^{\circ}$, by applying a Fourier series representation, as it is shown below

$$C_T^* = \frac{1}{100} \sum_{k=1}^{N} \{A_k \cdot \cos(k \cdot \beta) + B_k \cdot \sin(k \cdot \beta)\}$$
(4.7)

$$C_Q^* = -\frac{1}{1000} \sum_{k=1}^N \{A_k \cdot \cos(k \cdot \beta) + B_k \cdot \sin(k \cdot \beta)\}$$
(4.8)

The A_k and B_k are the Fourier coefficients that can also be found in Roddy's work [43] and similarly to the diagrams, they are grouped together in tables, corresponding to specific propeller designs (number of blades, expanded area ratio and pitch to diameter ratio P/D) for a number of calculation points N. In this case, N=30.

The main characteristics of the propeller that was employed in the present study, are presented in Table 4.2, while the resulting values of equations (4.7) and (4.8), for a span of $0^{\circ} \le \beta \le 360^{\circ}$, are presented in Figure 4.3. It must be noted that, since the *P/D* ratio of our propeller doesn't correspond to any of the *P/D* for which the Fourier coefficients A_k and B_k were calculated, coefficients C_T^* and C_Q^* in Figure 4.3 are calculated through linear interpolation between the values of C_T^* and C_Q^* for *P/D* = 0.6 and *P/D* = 0.8.

| Туре | Wageningen B-Series | | | |
|--|------------------------|--|--|--|
| Model | B4-70 | | | |
| No. of Blades (z) | 4 | | | |
| Diameter (D_P) | 7.74 m | | | |
| Pitch (P) | 6.00 m | | | |
| Expanded Area Ratio (EAR) | 0.70 | | | |
| Pitch - Diameter Ratio (P/D) | 0.78 | | | |
| Propeller Material Density ($ ho_P$) | 7600 kg/m ³ | | | |

 Table 4.2: Propeller characteristics



Figure 4.3: C_T^* and C_Q^* coefficients for the B4-70 series with P/D = 0.78

With all the necessary parameters calculated, equations (4.4) and (4.5) enable us to estimate the propeller's thrust and torque at each computational time step of the simulation code.

4.2 Propeller Shaft Dynamics and Inertia

Since the present study focuses on the behaviour of the engine during transient conditions, an integral part of our analysis is the calculation of the engine's angular acceleration. For an FPP which is directly connected to the engine through the crankshaft, the angular acceleration can be calculated as

$$\frac{d\omega}{dt} = \frac{Q_{eng} - Q_{pr}}{I_{dry} + I_E + I_{eng}}$$
(4.9)

The torque absorbed by the propeller Q_{pr} is calculated according to (4.5), while the torque produced by the engine Q_{eng} is calculated by MOTHER, for each computational step. At this point, the inertia of the rotating objects must also be calculated.

The total inertia of the engine I_{eng} was not available. Hence, it was calculated based on a similar engine from Wärtsilä (W6X72), using the program GTD. The propeller contributes to the shafting system's total polar moment of inertia with two terms, the dry polar moment of inertia I_{dry} , referring to the propeller rotating in the air and the entrained water inertia or added polar moment of inertia I_E , referring to the propeller being immersed and rotating in water.

Dry Propeller Inertia

By definition, the polar mass moment of inertia of any rotating body is the sum of all elemental masses multiplied by the square of their distance from a reference axis. Due to the propeller's complex geometry though, this definition is not a particularly helpful approach for the calculation of the dry propeller inertia. In the present Thesis, to overcome that problem, a Simpson's numerical integration was implemented in order to get an accurate estimation of the dry inertia's value. More specifically, the method proposed in [38] was followed, in which each blade of the propeller is divided into an odd number of blade sections. In this case, the blades were divided into 9 sections, given as a percentage of the propeller's radius R (x = r/R), with a range from 0.2 to 1 (the reference axis being the propeller's axis of rotation). It must be noted that the first section begins at 20% of the radius R, as the part of the propeller from the axis of rotation up to 0.2R is occupied by the hub, whose inertia will be calculated separately.

Each of the above blade sections is further divided into 11 vertical sections x_c along the chord's length c (10 intervals of $0.1 \cdot c$) and for each vertical section the thickness $t(x_c)$ was measured. Afterwards, a Simpson's integration was also applied in order to calculate the section's total area A_x , according to the formula

$$A_x = \frac{0.2 \cdot c \cdot \Sigma_0}{3} \tag{4.10}$$

Where *c* is the chord length of each blade section and Σ_0 is the resulting sum after multiplying each vertical section's thickness t(x_c) with the Simpson multiplier SM (Figure 4.4).



Figure 4.4: Blade thickness measurements [38]

Consequently, when the total area of each of the 9 blade sections is calculated, we can apply an overall Simpson integration according to the following table.

| x=r/R | A _x | Simpson Multipliers | $A_x \cdot SM$ | 1 st moment arm | A _x · SM · 1 st MA | 2 st moment arm | $A_x \cdot SM \cdot 1^{st}$ MA $\cdot 2^{nd}$ MA |
|----------------|-----------------|------------------------|---------------------|----------------------------------|---|----------------------------------|---|
| [1] | [2] | [3] | [4] =[2]·[3] | [5] | [6] =[5]·[4] | [7] | [8] =[7]·[6] |
| 1.0 | A _{x1} | 0.5 | 0.5·A _{x1} | 0 | 0 | 0 | 0 |
| x ₂ | A _{x2} | 2 | 2.A _{x2} | 1 | 2·A _{x2} | 1 | 2·A _{x2} |
| X ₃ | A _{x3} | 1 | A _{x3} | 2 | 2·A _{x3} | 2 | 4.Ax3 |
| X4 | A _{x4} | 2 | 2·A _{x4} | 3 | 6∙A _{x4} | 3 | 18·A _{x4} |
| X5 | A _{x5} | 1 | A _{x5} | 4 | 4·A _{x5} | 4 | 16·A _{x5} |
| X ₆ | A _{x6} | 2 | 2·A _{x6} | 5 | 10·A _{×6} | 5 | 50·A _{x6} |
| X7 | A _{x7} | 1 | A _{x7} | 6 | 6∙A _{x7} | 6 | 36·A _{x7} |
| X ₈ | A _{x8} | 2 | 2.A** | 7 | 14·A _{x8} | 7 | 98·A _{x8} |
| X9 | A _{x9} | 0.5 | 0.5·A _{x9} | 8 | 4·A _{x9} | 8 | 32·A _{x9} |
| | | | Σ1 | | Σ2 | | Σ3 |

Table 4.3: Simpson integration for blade's moment of inertia

By following the instructions of Table 4.3, we can calculate the values of three sums Σ_1 , Σ_2 and Σ_3 , which are used in the following equations [38]

$$I_{tip} = \frac{2}{3} \cdot \left(\frac{\frac{D_P}{2} - r_h}{8}\right)^3 \cdot \Sigma_3$$
(4.11)

$$l_{ct_{tip}} = \frac{\Sigma_2}{\Sigma_1} \cdot \frac{\frac{D_P}{2} - r_h}{8}$$
(4.12)

$$l_{ct_{shaft}} = \frac{D_P}{2} - l_{ct_{tip}} \tag{4.13}$$

Equation (4.11) calculates the volume inertia of each propeller blade, about the blade tip. Equation (4.12) gives us the distance of the centroid of the blade from the tip of the blade, while equation (4.13) calculates the distance of the centroid of the blade from the shaft. In these equations, r_h stands for the radius of the propeller hub at its middle length.

The volume inertia from equation (4.11) is calculated about the propeller tip, so the Steiner theorem must be applied, in order to transfer the calculated value from the blade tip to the shaft, according to equation (4.14)

$$I_{blade} = \rho_P \cdot \left\{ I_{tip} - l_{ct_{tip}}^2 \cdot \left(\frac{\frac{D_p}{2} - r_h}{8} \cdot \frac{2}{3} \cdot \Sigma_1 \right) + l_{ct_{shaft}}^2 \cdot \left(\frac{\frac{D_p}{2} - r_h}{8} \cdot \frac{2}{3} \cdot \Sigma_1 \right) \right\}$$
(4.14)

Or it can be rewritten as

$$I_{blade} = \rho_P \cdot \left\{ I_{tip} - \left(\frac{\frac{D_P}{2} - r_h}{8} \cdot \frac{2}{3} \cdot \Sigma_1 \right) \cdot \left(l_{ct_{tip}}^2 - l_{ct_{shaft}}^2 \right) \right\}$$
(4.15)

The inertia of the hub is calculated separately. According to [36], the hub's diameter at its middle length is given as a portion of the propeller's diameter and more specifically, $D_{hub,m} = 0.167 \cdot D_P$. Considering the fact that the propeller's hub shape is that of a truncated cone, then its polar mass moment of inertia can be calculated using the following equation:

$$I_{hub} = \frac{3}{10} \cdot \frac{r_{h1}^5 - r_{h2}^5}{r_{h1}^3 - r_{h2}^3} \cdot V_{hub} \cdot \rho_P$$
(4.16)

where r_{h1} [m], r_{h2} [m] is the radius of truncated cone at the base and the peak respectively, while V_{hub} [m³] is the volume of the cone, calculated as

$$V_{hub} = \frac{\pi}{3} \cdot (r_{h1}^2 + r_{h2}^2 + r_{h1} \cdot r_{h2}) \cdot l_{hub}$$
(4.17)

In (4.17), l_{hub} [m] is the height of the cone.

To sum up, the dry polar moment of inertia of the propeller is equal to

$$I_{dry} = I_{hub} + z \cdot I_{blade} \tag{4.18}$$

Water Entrained Propeller Inertia

Due to the propeller's immersion and rotation in water instead of plain air, an increase in its polar mass moment of inertia is to be expected. This augmentation, which is caused by the propeller's interaction with the surrounding water, cannot be calculated as easily and precisely as the dry propeller inertia and can vary significantly. It must not be ignored in our calculations though, as it constitutes a significant proportion of the total propeller' inertia.

In the present Thesis, for the calculation of the added propeller inertia the empirical equation produced by Schwanecke was applied [44], which states that

$$I_E = C_{IE} \cdot \rho \cdot D_P^{-5} \tag{4.19}$$

where ρ is the sea water density and C_{IE} is a coefficient, calculated as

$$C_{IE} = \frac{0.0703 \cdot \left(\frac{P}{D}\right) \cdot EAR^2}{\pi \cdot z} \tag{4.20}$$

Total Propeller Inertia

Having calculated both the dry and the added propeller inertia, the total inertia of the propeller can be given as

$$I_{propeller} = I_{dry} + I_E \tag{4.21}$$

All the necessary parameters for these calculations (the blade's thickness $t(x_c)$ in each vertical section, the chord's length *c* etc.), were acquired by using HydroComp's program "PropCad", in order to produce a Wageningen B-Series propeller with a specific *P*/*D*.

4.3 Ship Model

According to equation (4.2), for the calculation of the advance speed V_A , the speed of the ship V_{SHIP} must first be calculated. V_{SHIP} can be calculated at every computational step, by applying Newton's second law of motion, in the following form

$$\dot{V}_{SHIP} \cdot (M_{SHIP} + M_{ADDED}) = (1 - t) \cdot T_{pr} - R_{tot}$$
(4.22)

In (4.22), \dot{V}_{SHIP} [m/s²] is the acceleration of the ship, M_{SHIP} [kg] is the total mass of the vessel, M_{ADDED} [kg] is the added mass that must be taken into consideration due to the surrounding fluid, T_{pr} is the thrust produced by the propeller (calculated according to (4.4)) and R_{tot} is the ship's total resistance. Finally, t is called the thrust deduction factor and it takes into account the reduction of the propeller's effective thrust, due to the distortion that the propeller's rotation creates in the ship's wake.

For this particular model, all the necessary data concerning the main characteristics of the vessel, were available from the sea-trials of an oil tanker (Table 4.4). Other parameters that were not available in the sea-trials, such as the hull coefficients (C_B , C_M , C_P and C_{WP}), the wetted surface (S) etc., were calculated by using equations found in [45]

| Length Between Perpendiculars (L_{BP}) | 264.00 m |
|--|-------------|
| Breadth (B) | 48.00 m |
| Draft at Midship (T) | 15.985 m |
| Depth (<i>D</i>) | 23.10 m |
| Displacement (Δ) | 169797.50 t |

Table 4.4: Vessel characteristics

Ship's Total Mass

When calculating the total mass of the vessel, we must keep in mind that apart from the vessel's actual mass M_{SHIP} , we also have to calculate the added mass M_{ADDED} due to the fluid being deflected during the acceleration of the ship. The ship's mass is equal to its displacement, taken from the sea trials. In general, the calculation of M_{ADDED} is a very complicated hydrodynamics problem, which usually requires the application of complex numerical methods. For simplicity, it can be considered as a fraction of the ship's total mass. In this Thesis, M_{ADDED} is considered equal to 10% of M_{SHIP} .

Thrust Deduction Factor

The presence of the propeller on the after part of the ship leads to a reduction of the local pressure field and as a result to an increase of the ship's total resistance (alternatively a decrease in the propeller's effective thrust), in comparison to the one measured in the towing experiment. Thus, it is important to have an accurate estimation of the thrust deduction factor t.

Due to the absence of a resistance and propulsion model for the provided vessel, we have to resort to an estimation of the thrust deduction factor using the equation given by [40]

$$t = \frac{0.25014 \cdot (B/L_{WL})^{0.28956} \cdot (\sqrt{B \cdot T}/D)^{0.2624}}{(1 - C_P + 0.0225 \cdot lcb)^{0.01762}} + 0.0015 \cdot C_{STERN}$$
(4.23)

where L_{WL} [m] is the length of ship along the waterline, which according to [46] can be calculated as $L_{WL} = 1.01L_{BP}$. Furthermore, C_P is the prismatic coefficient, C_{STERN} a constant depending on the shape of the ship's sections (for normal section shape $C_{STERN} = 0$) and *lcb* the longitudinal position of the center of buoyancy, calculated forward of $0.5L_{WL}$ and given as a percentage of L_{WL} .

According to [39], common values for *t* range between 0.1 and 0.3.

Ship's Total Resistance

Similarly to the case of the thrust deduction factor, an accurate estimation of the ship's total resistance would require to perform an experiment with an appropriate ship model, calculate its resistance and then use specific correlation factors to relate it to the resistance of the actual ship. Due to the lack of a model as such though, the estimation of R_{tot} was based on experimental methods and more specifically on the one proposed in [40, 47]. This is a power prediction method, that was based on a regression analysis of random model and full-scale test data and it has been adjusted in order to have a wide range of application.

According to this method, the ship's total resistance R_{tot} can be estimated using the following equation

$$R_{tot} = (1+k_1) \cdot R_F + R_{APP} + R_W + R_B + R_{TR} + R_A$$
(4.24)

As it can be seen, R_{tot} is divided into six different components. These are the frictional resistance R_F , the resistance due to the ship's appendages R_{APP} , the wave making and wave breaking resistance R_W , the additional pressure resistance of bulbous bow near the water surface R_B and due to the immersed transom stern R_{TR} and finally the model-ship correlation resistance R_A .

A detailed methodology on how to calculate every component in equation (4.24) is given in the work of Holtrop [40, 47]. After the ship's total resistance has been calculated, the acceleration of the vessel for a specific computational step can be calculated from equation (4.22).

4.4 Speed Governor

Except from the thermodynamic and mechanical elements, MOTHER also enables us to incorporate control elements, such as a speed governor. The incorporation of a speed governor is necessary, since ungoverned diesel engine transient simulations cannot converge, unless active control is imposed on the fuel mass injected in the cylinders per cycle. Hence, the role of the governor is to stabilise the engine speed around a speed setpoint in the operating region. Its use is of great importance, especially during engine operation in heavy weather, when large variations of the engine's speed take place and if not stabilised, they can lead to serious damage.

The major operational principle of the governor is based on negative feedback, which in the present study is the engine crankshaft speed feedback (Figure 4.5). The negative feedback, closed loop control can improve the system's performance and robustness.



Figure 4.5: Conceptual block diagram of governor's operational principle [32]

In MOTHER simulation platform, the speed governor also includes a core PID controller. The output of the controller is given either in the Laplace s-domain (equation (4.25)), or in the time t-domain (equation (4.26)), as follows

$$x(s) = K \cdot \left\{ K_P \cdot e(s) + \frac{K_I}{s} \cdot e(s) + K_D \cdot s \cdot e(s) \right\}$$
(4.25)

$$x(t) = K \cdot \left\{ K_P \cdot e(t) + K_I \cdot \int_{t_0}^t e(\chi) d\chi + K_D \cdot \frac{d}{dt} e(t) \right\}$$
(4.26)

In equation (4.26), e(t) is the speed error, defined as

$$e(t) = N_{ord}(t) - N(t)$$
 (4.27)

where $N_{ord}(t)$ is the speed setpoint and N(t) is the actual speed of the crankshaft.

The output x of the PID controller is the fuel rack position, which takes values between 0 and 1, and directly affects the fuel mass injected into the engine cylinders per cycle. Furthermore, after performing an initial transient simulation with the governor, the P and I constants (K_P and K_I) were adjusted, through a trial-and-error process, in order to achieve a smooth transition during the following transient simulations.

Finally, MOTHER allows the application of certain limiters for the rack position. These limiters are usually provided by the engine manufacturer, as parameterised functions of one or more engine operating variables, and ensure a safe engine operation and avoidance of overspeeding. In the present study, two limiters were applied to the rack position, the first in respect to the crankshaft rotational speed and the second in respect to the pressure in the scavenge receiver.

Chapter 5: Transient Simulations

After developing the ship hull model and the propeller model, and with the addition and calibration of the speed governor for fuel control, the entire layout (Figure 5.1) can successfully simulate transient loading operation of the engine.



Figure 5.1: Complete simulation layout for transient operation

In the present Thesis, based on the work of Llamas [6], [7], [8] and [9], whose research also focused on transient simulations during low-load engine operation, three different transient simulations were studied:

- An acceleration simulation from 50% to 75% engine load
- An EGR start simulation at 50% engine load
- An EGR stop simulation at 50% engine load

5.1 Engine Acceleration from 50% to 75% Load

In this simulation, the engine's acceleration from 50% (72.2 [RPM]) to 75% (82.7 [RPM]) load, with an EGR system integrated, is studied. At first, the engine operates at 50% load for a short period of time, in order to eliminate any initial disturbances that may affect the acceleration. At 150 [sec], a command is given to the governor, through a step function (ordered speed), to accelerate from 72.2 [RPM] to 82.7 [RPM]. By altering the fuel rack position, the governor controls the fuel injected into the cylinder and the engine accelerates. After a short period of time (~30 [sec]), the new engine speed is achieved.

In Figure 5.2, the operational and thermodynamic parameters of the engine during EGR operation are presented and compared with the ones during normal engine operation (Tier II operational mode). In



Figure 5.3, the effects of EGR on the relative air-fuel ratio and on oxygen concentration in the scavenging manifold are given.

Figure 5.2: Operational and thermodynamic parameters of the engine with and without EGR, during acceleration



Figure 5.3: Relative air-fuel ratio and oxygen concentration of the engine with and without EGR, during acceleration

EGR is an efficient method of reducing the NO_x emissions to acceptable levels, without affecting the power output of the engine [10]. Indeed, as it can be seen on Figure 5.2, during the whole transient operation with EGR, the engine load concurs with the load during normal (Tier II) operation. The same can also be said about the engine rotational speed. With a proper governor calibration, a smooth transition to higher loads can be achieved, without any overspeeding.

During the acceleration, a temporary drop of the oxygen concentration in the scavenging manifold can be observed. This can be attributed to the fact that, in order for the engine to accelerate, more fuel is injected into the cylinders, resulting in exhaust gas richer in CO_2 . Part of the richer exhaust gas is recirculated back to the scavenging manifold by the blower. On the other hand, the turbocharger requires some time before perceiving this change, sending more air to the scavenging manifold (turbocharger lag). That is why, for this duration (approximately 30 [sec]), a reduction in the oxygen concentration is observed. When the turbocharger adapts to the new operational conditions, the oxygen concentration returns to normal levels, corresponding to the 75% load operation. Severe negative O_2 peaks in the scavenge receiver, that lead to the formation of undesired black smoke, should be avoided, through better EGR controlling [48].

Turbocharger lag is a common phenomenon that is present in marine diesel engines and derives from the slow turbocharger response during transient loading, due to the large volumes of the scavenge and exhaust receivers (it takes more time for pressure to build up). The results of turbocharger lag are clearly visible in the relative air-fuel ratio alteration. During EGR operation, turbocharger lag is slightly more intense, meaning that the variation of the thermodynamic parameters happens at a slower rate.

Due to the introduction of the CO_2 into the cylinders and the resulting richer combustion, compared to Tier II operation (lower λ values), higher exhaust gas temperatures are to be expected and thus, an increase in the turbocharger rotational speed as well.

A temporary overspeed of the EGR blower during the acceleration, results in the appearance of an intermediate peak in the EGR mass flow rate. In total, the increased mass flow rate on the EGR path during acceleration is to be expected, since the increase in the turbocharger rotational speed results in an increase in the air mass flow rate through the compressor. Thus, more recirculated exhaust gas is required, to maintain the same levels of oxygen concentration in the scavenge receiver.

Finally, during the whole process, the pressure of the exhaust gas is retained lower than the pressure in the scavenging manifold, ensuring that uniflow scavenging is possible.

5.2 EGR Start at 50% Load

In this simulation, the engine's load was kept constant at 50% and an EGR system start was attempted. Initially, the engine operates at Tier II - T/C Cut-Out mode, meaning that the second turbocharger is deactivated and the EGR path remains closed, though the BTV valve. At 150 [sec], the BTV begins to gradually open for approximately 90 [sec], until the desired EGR mass flow rate and thus, the desired reduction in oxygen concentration, is achieved.

The regulation of the EGR mass flow rate is achieved through a combination of the BTV opening and the EGR blower rotational speed. During the simulation, the rotational speed of the EGR blower is increased along with the valve opening, in order to direct a larger part of the exhaust gas flow to the EGR path.

The operation of the blower begins during the Tier II - T/C Cut-Out mode, before the BTV opens, in order to avoid any back-flow issues in the EGR path and ensure a smooth transition to Tier III - EGR operation.

During the gradual opening of the BTV, an increasing mass flow though the EGR path can be observed, with a corollary decrease of the exhaust gas mass flow directed to the turbocharger.

Introduction of increasing mass of exhaust gas into the engine's cylinders further results into a reduction of the relative air-fuel ratio, and of the oxygen concentration, which is replaced with CO_2 . At low oxygen concentrations, EGR systems have a great potential for NO_x reduction. On the other hand, the reduced oxygen availability during high EGR rates limits the amount of fuel that can be burned, without producing an undesired black smoke. Thus, an upper limit is set on the EGR rate. Formation of black smoke appears to be more intense at lower engine loads and can reduce the vessel's manoeuvrability inside the NECAs [9].

Although a richer combustion takes place (due to the reduction of the relative air-fuel ratio), at the same time, less exhaust gas mass flow is directed to the turbine, resulting in a lower turbocharger rotational speed. Ultimately, the turbocharger's reduced rotational speed has a bigger impact on the engine's operation and thus, a reduced boost pressure in the scavenging manifold can be observed. Simultaneously, lower exhaust gas temperatures are also observed (due to the reduced air mass flow in the cylinders), which are partially balanced by the richer combustion process.



Figure 5.4: Operational and thermodynamic parameters of the engine during EGR start, at 50% load



Figure 5.5: Relative air-fuel ratio and oxygen concentration of the engine during EGR start, at 50% load

5.3 EGR Stop at 50% Load

An EGR stop sequence simulation was also tested in the present model, in order to study the behaviour of the engine while the EGR is deactivated. This can happen either when the vessel leaves a NECA, or if a malfunction in the EGR paths appears and the engine falls back to Tier II – T/C Cut-Out mode operation.

Initially, the engine operates at Tier III – EGR mode, meaning that the second turbocharger is deactivated and the EGR path is open, though the BTV valve. At 150 [sec], the BTV begins to gradually close for approximately 90 [sec], until the EGR mass flow rate is nullified, and only the first turbocharger is in operation.

Similarly to the case of EGR start, the regulation of the EGR mass flow rate is achieved through a combination of the BTV opening and the EGR blower rotational speed. During the simulation, the rotational speed of the EGR blower is decreased along with the valve opening, until the EGR path is completely closed. Results of the simulation are summarized in Figures 5.6 and 5.7.

A similar engine behaviour as the one presented during EGR start can be observed in this simulation, only in reverse.

During the gradual closing of the BTV, the mass flow though the EGR path is slowly reduced until nullified, while the exhaust gas mass flow through the turbocharger increases, until it reaches the initial value of normal engine operation. At the same time, when the EGR path closes, the blower operates close to its minimum rotational speed and after a while, it is practically deactivated.

The gradual reduction of the exhaust gas mass flow into the scavenge receiver results into an increase of the relative air-fuel ratio and of the oxygen concentration. The appearance of black smoke during the transient loading is not a threat during EGR stop, as the oxygen mass concentration returns to normal atmospheric levels.

Increased mass flow to the turbocharger after EGR deactivation, results in a higher rotational speed, which in effect leads to a higher boost pressure in the scavenging manifold.



Figure 5.6: Operational and thermodynamic parameters of the engine during EGR stop, at 50% load



Figure 5.7: Relative air-fuel ratio and oxygen concentration of the engine during EGR stop, at 50% load

Chapter 6: Conclusions

6.1 Discussion

The aim of the present Thesis was to develop a simulation model that will be able to predict the performance of a large two-stroke marine diesel engine, with an EGR system integrated, during steady-state and transient operational conditions.

In this respect, a zero-dimensional thermodynamic model of such an engine was developed, using the engine performance prediction code MOTHER. The recalibration of the model for the new operational conditions (EGR activated), was carried out for a number of steady-state load cases, through alterations on the combustion and heat transfer model. Available data from literature and the shop test results from a slightly bigger engine were used to qualitatively assess the model's accuracy.

The main findings of the present work are summarised as follows:

- After the model's recalibration for EGR operation, the engine's behaviour seems to be in agreement with the associated literature. The reduced combustion ratio and the increased combustion duration that MAN B&W predicts in its findings can be clearly seen in the Fuel Burning Rate and Cumulative Fuel Burnt diagrams, presented in Chapter 3.4. Furthermore, the peak in-cylinder temperatures are reduced bellow the critical temperature of 1800 [K], above which thermal NO_x formation increases exponentially. Finally, the model's thermodynamic parameters (exhaust and scavenge pressure and temperature, peak pressure etc.) follow the same trend as the corresponding values in the shop test results of the similar EGR engine. Hence, it can be concluded that MOTHER software is able to simulate a large two-stroke marine diesel engine equipped with an EGR system.
- The developed model also provides the capability to study the transient behaviour of these engines, with and without EGR. Maintaining a high EGR rate during engine acceleration at low loads is a complicated problem, especially due to the turbocharger lag on large engines. It can be observed that EGR does not affect the acceleration capabilities of the engine for these specific loads (the load curves in Figure 5.2 are identical). This is achieved through appropriate control of the fuel rack position by the speed governor. The oxygen content is reduced to acceptable levels and a slight increase on the exhaust gas temperature can be observed, due to the richer combustion.
- During the EGR start simulation, a decrease in the turbocharger's rotational speed can be observed, along with a reduction of the boost pressure. The BTV opening is gradually regulated during a certain period of time, to ensure a smooth behaviour during the increase of EGR rate. The same applies for the EGR blower rotational speed. Oxygen mass concentration is reduced from its normal atmospheric value to a lower level, leading to reduced NO_x formation.

• In order to simulate the entire ship propulsion system, models for the propeller and the vessel's hull resistance, were also implemented in the simulation code. They were developed based on the modelling approach of Holtrop [40, 47] and they define the engine's rotational speed and the total resistance that the engine must overcome, at each computational time-step.

6.2 Recommendations for Future Research

The study of the transient performance of a marine diesel engine, equipped with an EGR system, is quite complicated. Not many simulation models, that can accurately the engine's behaviour, exist in the current literature, as measurement data for validation are scarce. Some recommendations for future research can be:

- An in-depth research on how the turbocharger lag affects the operational behaviour of the engine, especially at engine start and low load operation, and an adaptation into the current model.
- During low load operation of the engine, there is a specific rotational speed range, where torsional vibrations that are harmful for the crankshaft are produced (Barred Speed Range BSR). Continuous operation inside this range must be avoided. Swift passage through this speed range is necessary for smooth engine operation and can be achieved by utilising technologies such as MAN's Dynamic Limiter Function (DLF). On the other hand, increased turbocharger lag, caused by the implementation of high EGR rates, makes this swift transition more difficult. A combined model that incorporates both the EGR and BSR technologies is proposed, in order to study the effects that increased EGR rates may have in this swift transition.
- Oxygen sensor measurements in EGR models contain inherent delays that can cause a simple PI governor to perform poorly during fast load transients [9]. Hence, better EGR controllers, that can handle these acceleration scenarios more appropriately and that don't limit the vessel's manoeuvrability in coastal areas, must be implemented in the present model.
- Contrary to after-treatment methods like SCR, where the switch off for the entire configuration, when the vessel doesn't sail inside a NECA, can be easily achieved without influencing the engine run, EGR switch off causes a series of problems, mainly affecting the turbocharger's operation. The use of a Variable Geometry Turbocharger (VGT) is proposed and should be incorporated in the model.

6.3 Publication

D. Vlon, M. Foteinos and N. Kyrtatos, "Simulation of the Transient Operation of a Large Two-Stroke Marine Diesel Engine Equipped with an Exhaust Gas Recirculation System (EGR) for NO_x Reduction," *Proceedings of the Hellenic Institute of Marine Technology (HIMT) Annual Meeting 2018.*

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