

Exploring Dilution Strategies and Prechamber Ignition on a Marine Engine Retrofitted to Methanol SI Operation

Matthijs Van Duyse, Sebastiaan Malfait

Student numbers: 02009437, 02010321

Supervisor: Prof. dr. ir. Sebastian Verhelst

Counsellor: Ir. Quinten Dejaegere

Master's dissertation submitted in order to obtain the degree of Master of Science in Electromechanical Engineering - Mechanical Energy Engineering

Academic year 2024-2025

This master's dissertation is part of an exam. Any comments formulated by the assessment committee during the oral presentation of the master's dissertation are not included in this text.

The authors give permission to make this master dissertation available for consultation and to copy parts of this master dissertation for personal use. In all cases of other use, the copyright terms have to be respected, in particular with regard to the obligation to state explicitly the source when quoting results from this master dissertation.

Ghent, May 2025
Matthijs Van Duyse and Sebastiaan Malfait

Preface and Acknowledgments

This master's dissertation is the result of a challenging, but deeply rewarding year in which we expanded our knowledge of sustainable combustion technologies and experimental engine research. We, Matthijs and Sebastiaan, hope that our work adds valuable insight to the ongoing transition toward greener marine propulsion systems. The growing need for sustainable alternative fuels in the heavy-duty sector immediately made the relevance of our research clear, and we were highly motivated to contribute to this pressing global challenge.

However, this journey was not one we undertook alone. The intense months of development, experimentation, and analysis were not only made possible, but also truly fulfilling thanks to the guidance, support, and encouragement we received from many people.

First and foremost, we would like to express our sincere gratitude to our counsellor, Ir. Quinten Dejaegere. Quinten, your contagious enthusiasm and technical expertise were instrumental during the course of the year. Your feedback was always constructive and your door was always open, which helped us to remain confident even when challenges accumulated.

We are also deeply grateful to our supervisor, Prof. Dr. Ir. Sebastian Verhelst, for allowing us to work on a topic that we were truly passionate about. Your courses and feedback gave us both the technical foundation and the critical thinking tools needed to pursue this research with autonomy. The balance between academic rigor and real-world application that you instilled in us will serve us well beyond this project.

Furthermore, we would like to thank Koen Chielens and Bart Van Daele for their expertise in combustion engines and for their willingness to share practical experience with us. Your hands-on knowledge helped bridge the gap between theory and practice. Special thanks also go to Frederik Martens, whose support during electrical challenges was essential to keep our test setup running smoothly. Moreover, we want to thank all members of the Sustainable Thermo-Fluid Energy Systems (STFES) research group at Ghent University. The collaborative environment and shared passion for innovation made this experience even more memorable.

A sincere thank you goes towards our families and loved ones, whose unwavering support and encouragement have been a constant source of inspiration and motivation throughout this journey.

Finally, the authors would like to express their gratitude towards one another for the smooth and pleasant collaboration throughout the project. It was an enjoyable and rewarding experience to work together towards a common goal.

To achieve great things, two things are needed: a plan and not quite enough time.

Leonard Bernstein

Exploring Dilution Strategies and Prechamber Ignition on a Marine Engine Retrofitted to Methanol SI Operation

Matthijs Van Duyse & Sebastiaan Malfait

Supervisor: Prof. dr. ir. Sebastian Verhelst

Counsellor: Ir. Quinten Dejaegere

Master's dissertation submitted in order to obtain the academic degree of
Master of Science in Electromechanical Engineering

Ghent University

Faculty of Engineering and Architecture

Departement of Electromechanical, Systems and Metal Engineering

Research group of Sustainable Thermo-Fluid Energy Systems (STFES)

Academic year 2024–2025

The heavy-duty (HD) transport sector is under increasing pressure to improve its environmental impact. While diesel engines remain the dominant technology due to their high efficiency and robustness, their emissions, particularly in terms of CO₂, NO_x, and particulate matter, are unsustainable. Methanol emerges as a promising alternative fuel, offering high energy density, favorable combustion characteristics, carbon-neutral potential and compatibility with existing infrastructure. Instead of replacing the entire fleet, retrofitting existing compression ignition (CI) diesel engines to operate in spark ignition (SI) mode on methanol presents a practical and scalable solution. However, this approach introduces new technical challenges, especially the increased knock sensitivity due to high compression ratios and large bore sizes, typical of marine engines.

Part I of this dissertation presents a comprehensive literature review. Chapter 1 examines the environmental and operational challenges facing the HD industry and introduces methanol as a promising alternative fuel. Chapter 2 focuses on the properties of methanol and discusses technical implications of retrofitting CI engines for SI operation. Finally, Chapter 3 provides a critical assessment of the underlying mechanisms affecting the performance and emissions. This chapter also identifies potential strategies to overcome the limitations of methanol use in high-load SI applications.

Part II presents the experimental research conducted. Chapter 4 details the experimental setup and retrofitting process. Chapter 5 analyzes the effects of dilution strategies, lean operation and exhaust gas recirculation, on engine performance and emissions. Chapter 6 introduces a passive prechamber ignition system aimed at improving combustion stability under high dilution. Finally, Chapter 7 consolidates the findings with reference to diesel performance benchmarks, while Chapter 8 outlines future research directions.

Keywords: High-speed marine engine, Heavy-duty, Retrofit, Methanol, High compression ratio, Large bore, Lean operation, Exhaust Gas Recirculation, Passive prechamber

EXPLORING DILUTION STRATEGIES AND PRECHAMBER IGNITION ON A MARINE ENGINE RETROFITTED TO METHANOL SI OPERATION

Matthijs Van Duyse & Sebastiaan Malfait

Supervisor: Prof. dr. ir. Sebastian Verhelst

Counsellor: ir. Quinten Dejaegere

Department of Electromechanical, Systems and Metal Engineering
Ghent University

Sint-Pietersnieuwstraat 41, B9000 Gent, Belgium

E-mail: matthijs.vanduyse@ugent.be & sebastiaan.malfait@ugent.be

ABSTRACT

To meet climate objectives, the heavy-duty (HD) industry must transition away from fossil fuels. Methanol is a promising alternative, with practical implementation possible via retrofitting diesel compression ignition (CI) engines for spark ignition (SI) operation. However, the inherently high compression ratio (CR) and large bore of such engines elevate knock sensitivity, limiting performance and efficiency. This study investigates the effect of lean operation and exhaust gas recirculation (EGR) dilution on a retrofitted methanol-fueled marine SI engine, and explores the potential of a passive prechamber ignition to improve combustion stability. Both dilution strategies improved brake thermal efficiency (BTE), but also narrowed the stable operating range. At 10.6 bar BMEP, BTE increased from 37.24% (stoichiometric) to 39.25% under 18.2% EGR and wide-open throttle (WOT), approaching the original diesel BTE of 41.42%. Significant reductions in nitric oxide (NO) and carbon monoxide (CO) emissions were also observed. However, EGR dilution preserves a stoichiometric exhaust, being fully compatible with a three-way catalyst (TWC), avoiding the need for extremely complex and costly aftertreatment systems, thereby emerging as the more favorable dilution strategy. A passive prechamber showed promise for enhancing stability in highly diluted conditions, thereby expanding the dilution potential and improving BTE. A complete design was developed to support future experimental validation.

INTRODUCTION

In alignment with the climate objectives set by the Paris Agreement [1], the heavy-duty industry, responsible for 9% of global greenhouse gas (GHG) emissions

[2], must significantly reduce its carbon footprint. This sector primarily relies on diesel-powered internal combustion engines (ICEs) due to their high efficiency, energy density, and reliable performance under demanding conditions. However, diesel combustion produces carbon dioxide (CO_2), a major GHG, along with nitrogen oxides (NO_x) and particulate matter (PM), which degrade air quality and harm health [3]. Moreover, sea freight demand is expected to triple, and trucking to double by 2050 [2], further pressuring the HD industry to achieve carbon neutrality.

To address this, a sustainable energy carrier must be identified. Battery-electric systems and gaseous fuels (e.g. hydrogen or methane) are hindered by low energy density and complex storage requirements, making them impractical for HD applications [4] [5]. In contrast, methanol stands out as a viable solution, offering high energy density, liquid storage, compatibility with existing infrastructure, and the potential of a closed carbon cycle when produced from renewable sources [4].

Methanol offers several advantageous properties for combustion applications, including a high octane number, high laminar flame speed (LFS), and high heat of vaporization [6]. The high octane number enhances knock resistance, while the high LFS accelerates combustion, improving thermal efficiency and stability. Additionally, the high heat of vaporization provides a cooling effect within the cylinder, further reducing the risk of knock [7].

As replacing the entire fleet with new methanol-compatible engines is impractical, retrofitting existing diesel compression ignition (CI) engines to operate on methanol in spark ignition (SI) mode offers a feasible pathway.

The primary challenge however for methanol SI operation is knock. This issue is especially problematic

Table 1: Engine specifications of the baseline Volvo Penta D7C-B TA.

Volvo Penta D7C-B TA	
Engine layout	4-stroke 6 cylinder in-line
Aspiration	Turbocharged with intercooler
Compression ratio	17.6
Bore / stroke	108 mm / 130 mm
Displacement	7.15 l
Maximum torque / speed	904 Nm / 1500 RPM
Rated power / speed	195 kW / 2300 RPM

due to the large bore diameter and high compression ratio (CR) when retrofitted from diesel CI operation, as this prolongs the residence time of the end gas at high temperatures.

This study explores the performance of methanol SI operation on a marine engine retrofitted from diesel CI. The first objective is to evaluate the impact of dilution strategies, specifically lean combustion and exhaust gas recirculation (EGR), on engine performance and emissions. The second objective is an exploratory investigation of a passive prechamber, aimed at enhancing combustion stability, which could enable higher dilution levels and further improve efficiency.

EXPERIMENTAL SETUP

The baseline diesel engine utilized in this dissertation is the Volvo Penta D7C-B TA, a six cylinder in-line four-stroke marine engine with TA indicating turbo aftercooler. It features a bore diameter of 108 mm and a CR of 17.6. Additional engine specifications are provided in Table 1.

The engine was initially retrofitted for Diesel-Methanol Dual-Fuel (DMDF) operation by integrating a methanol supply system featuring a multi-point injection (MPI) setup and an additional Engine Control Unit (ECU). To enable pure methanol SI operation, the original diesel injectors were replaced with a dedicated ignition system, consisting of spark plugs, ignition coils, and ignition leads.

Over time, the engine setup was enhanced with additional components, including a wastegate, heat exchanger, three-way valve, and an EGR loop. A complete schematic of the engine setup is shown in Figure 1. For this dissertation, only the EGR loop will be utilized.

MEASUREMENT METHODOLOGY

All measurement campaigns were conducted at 1500 RPM, the speed at which the baseline engine achieves maximum torque, across three load conditions: low (3.5 bar BMEP), medium (7.1 bar BMEP), and high load (10.6 bar BMEP). For each operating condi-

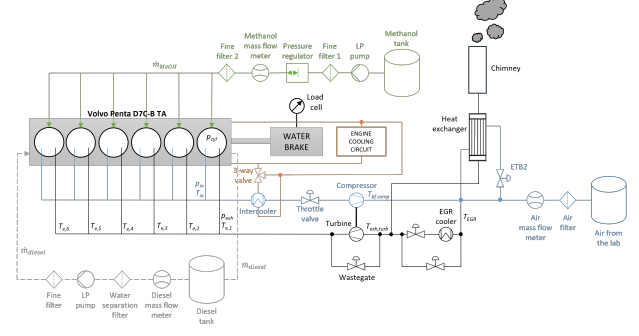


Figure 1: Schematic overview of retrofitted SI setup, adapted from [8].

tion, 200 consecutive in-cylinder pressure cycles were recorded to ensure statistical relevance.

The first phase focused on stoichiometric operation, where the impact of spark timing was evaluated across all load levels. Spark timing was advanced until the Knock Limited Spark Advance (KLSA) was reached, defined as the point at which light knock was audibly detected, followed by a 1° CA retardation.

Next, the effect of lean operation was assessed. At each load, the air-fuel mixture was incrementally leaned in steps of 0.1 in the air-fuel equivalence ratio (λ) until the stability limit was exceeded. This limit was determined using the coefficient of variance (COV) of the indicated mean effective pressure (IMEP) exceeding the threshold of 3%. For each λ value and load, an ignition timing sweep was performed.

Subsequently, the impact of EGR was investigated. Exhaust gases were recirculated, achieving an EGR rate calculated using Equation 1. At stoichiometric combustion, the EGR rate was incremented in steps of 5% initially, and in 2.5% near the EGR limit, until combustion stability was lost. At each EGR rate and load, an ignition sweep was performed.

$$\text{EGR}(\%) = \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{EGR}} + \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}} \cdot 100 \quad (1)$$

Finally, an exploratory measurement series was conducted of operating the engine at wide open throttle (WOT), motivated by the observation made that the throttle valve approached a nearly fully open position at the dilution limit during high-load EGR experiments.

RESULTS AND DISCUSSION

A Unveiling the impact of dilution

A.1 Operational limits

Figure 2 illustrates the variation in combustion stability as a function of spark timing across the different engine loads under stoichiometric operation. A general trend is evident: advancing the spark timing reduces cycle-to-cycle variability as a result of shorter

combustion durations associated with earlier ignition. However, at higher loads, the number of stable operating points diminishes as the knock limit becomes the dominant constraint. This limit shifts more significantly than the stability limit, thereby reducing the range of stable spark timing.

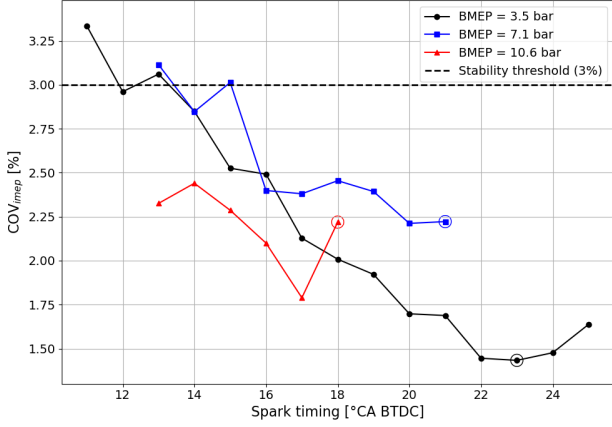


Figure 2: Effect of spark timing on COV_{IMEP} for different loads (BMEP) under stoichiometric conditions, with the spark timing corresponding to peak BTE highlighted.

To enable a fair comparison between EGR and lean operation, the dilution effect of a lean mixture can be converted to an equivalent air dilution percentage using Equation 2. AFR_{stoich} is the stoichiometric air-to-fuel ratio for methanol, being equal to 6.5.

$$\%Air\ Dilution = \frac{\lambda - 1}{\lambda + \frac{1}{AFR_{stoich}}} \times 100 \quad (2)$$

Table 2 shows the calculated air dilution percentages corresponding to several λ values.

Table 2: Air Dilution Percentage for Various λ Values.

λ	Air Dilution [%]
1.1	$\frac{1.1-1}{1.1+\frac{1}{6.5}} \times 100 = 7.98$
1.2	$\frac{1.2-1}{1.2+\frac{1}{6.5}} \times 100 = 14.77$
1.3	$\frac{1.3-1}{1.3+\frac{1}{6.5}} \times 100 = 20.63$
1.4	$\frac{1.4-1}{1.4+\frac{1}{6.5}} \times 100 = 25.74$
1.5	$\frac{1.5-1}{1.5+\frac{1}{6.5}} \times 100 = 30.23$

Figure 3 and Figure 4 illustrate the effect of increasing dilution on the stable operating range across the different engine loads. The right-hand boundary of each stable region is constrained by the knock limit, while the left-hand boundary is constrained by the combustion stability threshold. At high load, an exception to this pattern is observed at stoichiometric,

$\lambda = 1.1$ and 5% EGR operation where further spark retardation is limited by excessive exhaust gas temperatures and not by the stability limit.

As visible, increasing dilution leads to reduced combustion stability while simultaneously extending the knock limit. The ability to further advance the spark timing at higher loads by the expanded knock margin, opens new potential for efficiency improvements. This is in accordance with experiments of Zhu et al. [9].

Both dilution strategies effectively reduce the in-cylinder temperature and suppress knock. Lean burn does this by the thermal capacity of excess air, while EGR reduces flame temperatures by increasing the heat capacity of the mixture with inert gases.

EGR allows for a greater shift of the knock limit due to replacement of oxygen with inert gases, suppressing the formation of reactive radicals such as O, H, and OH in the end-gas, reducing the tendency for auto-ignition [10]. In addition, EGR dilution behaves more stable, in contrast with lean mixtures where the lower fuel concentration increases the risk of partial burns and misfires at elevated dilution levels. As a result, the combined effect of improved knock tolerance and sustained stability makes EGR a more robust dilution strategy for extending the engine's efficient and reliable operating boundaries.

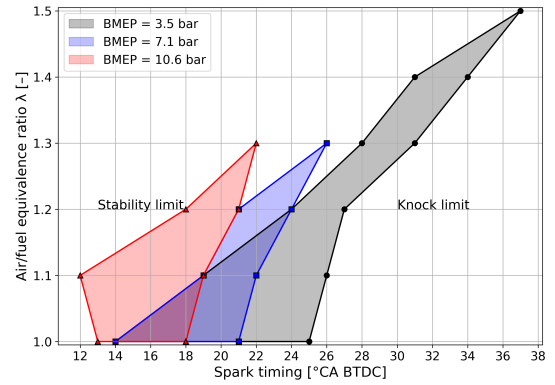


Figure 3: Stable operating range for lean burning at different loads.

A.2 Efficiency potential of lean burn and EGR dilution

Figure 5 presents the BTE without dilution as a function of the spark timing. Advancing the spark timing results in an increase in BTE as combustion occurs closer to top dead center (TDC), where the piston is optimally positioned to convert the combustion pressure into mechanical work.

This is confirmed by Figure 6 showing an increased peak in-cylinder pressure and narrowed heat release rate (HRR) trace at advanced spark timings illustrating a more favorable phasing of the combustion process. Optimizing combustion timing near TDC maxi-

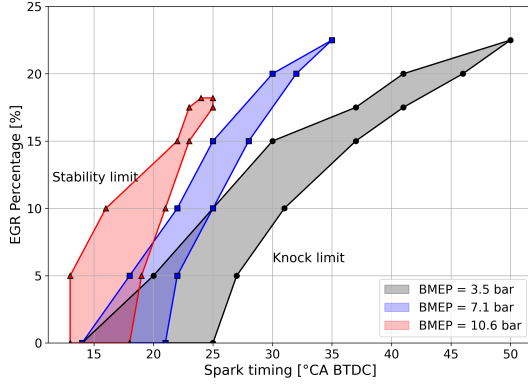


Figure 4: Stable operating range for EGR dilution at different loads.

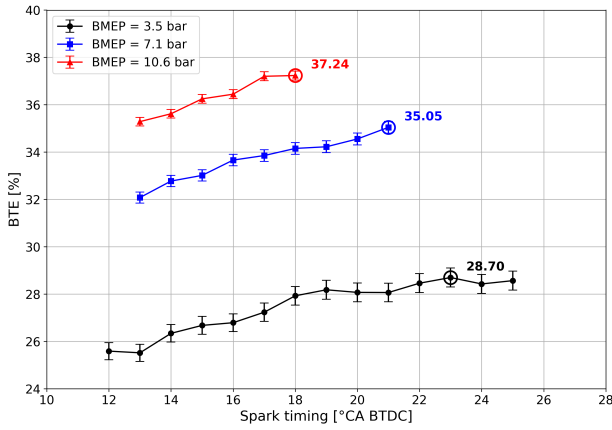


Figure 5: Brake thermal efficiency as a function of spark timing for different loads (BMEP) at stoichiometric operation.

mizes thermal efficiency by ensuring that a greater portion of released chemical energy is converted into mechanical work during the expansion stroke. However, advancing combustion timing excessively can raise in-cylinder temperatures, increasing heat losses to the cylinder walls and surrounding components. This creates a trade-off between maximizing energy conversion and minimizing thermal losses.

At medium and high load, the point of maximum BTE corresponds with the KLSA. This indicates that knock occurrence imposes a constraint on achieving the minimum spark advance for best torque (MBT) timing at these loads. At low load, a knock margin of 2° CA is observed.

At higher loads, thermal efficiency improves because relative heat and mechanical losses decrease, throttle valve opening reduces pumping losses, and the turbocharger boosts airflow, lowering the pumping mean effective pressure (PMEP).

This is confirmed by Figure 7 displaying the in-cylinder pressure as a function of the relative working volume for operating points corresponding to highest

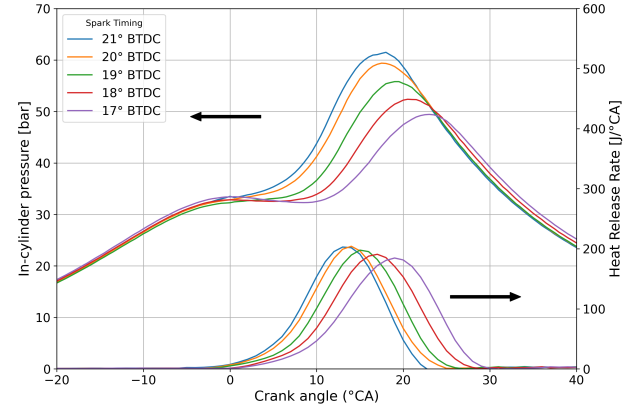


Figure 6: In-cylinder pressure and heat release rate as a function of degrees crank angle for different spark timings and a BMEP of 7.1 bar (1500 RPM, 400 Nm, $\lambda = 1$).

efficiencies. A detailed view of the negative work region, representing the pumping losses associated with throttling, is shown in the upper right corner of the plot. The plot reveals that as load increases, the negative work region representing pumping losses shrinks, indicating a reduction in throttling losses.

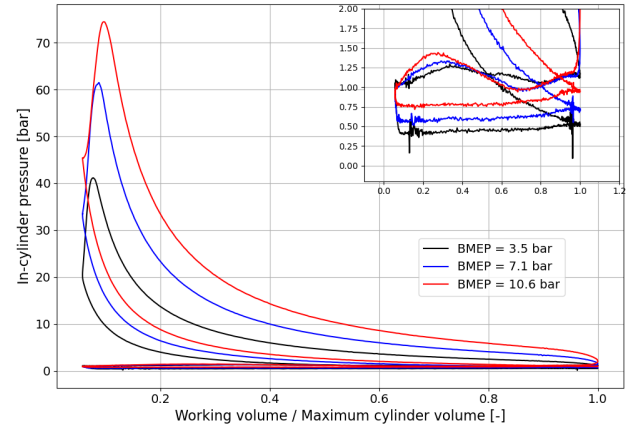


Figure 7: P-V diagram for different loads (BMEP), stoichiometric operation, and most optimal spark timing with a zoomed view of pumping losses.

Figure 8 and Figure 9 both show for each load an increase in efficiency at higher dilution rates. This can be attributed to several interrelated factors.

A primary contributor is the reduction in pumping losses as the throttle valve must be opened further resulting in a lower pressure drop across the intake system.

EGR reduces PMEP more effectively than lean operation because it displaces intake air with inert exhaust gases, requiring a larger throttle opening to maintain oxygen supply. This effect is further amplified by the higher temperature of recirculated gases, which lowers their density, requiring a greater intake volume and reducing throttling losses, as confirmed by

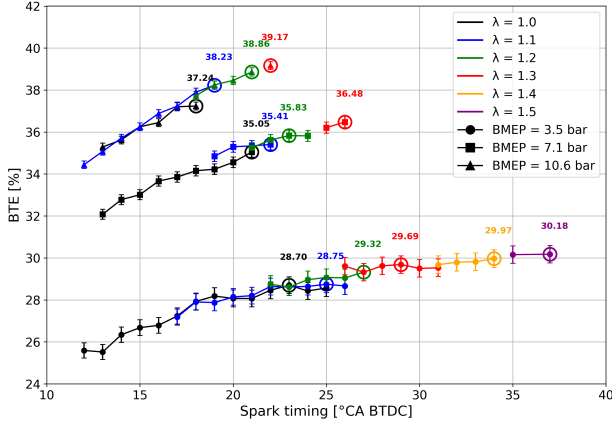


Figure 8: BTE as a function of spark timing for different loads and increasing lambda values.

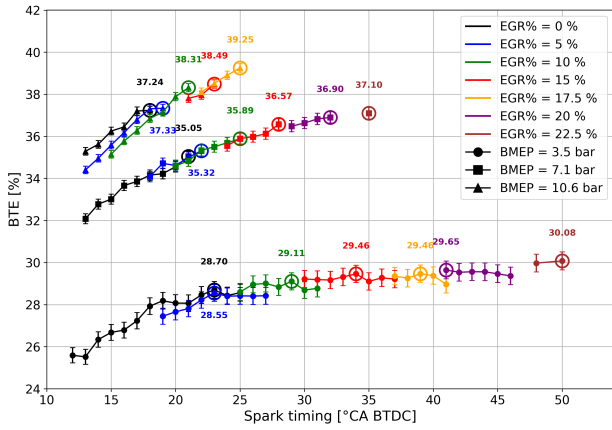


Figure 9: BTE as a function of spark timing for different loads and increasing EGR%.

Figure 10.

Alongside the reduction in pumping losses, charge dilution also significantly lowers peak in-cylinder temperatures. This temperature decrease impacts both heat transfer losses and endothermic dissociation losses within the combustion chamber [11]. The primary reason for this reduction is an increase in the total heat capacity of the in-cylinder mixture.

Lastly, dilution improves efficiency by shifting the KLSA to more advanced values. Reduced in-cylinder temperatures lower the engine's knock tendency. At low load, knock was entirely suppressed with sufficient dilution, especially with EGR, allowing spark timings as early as 50° BTDC. At medium and high loads, dilution allowed operation closer to optimal combustion phasing (MBT), but remained knock-limited. While further dilution could further suppress knock at higher loads, it would compromise combustion stability, highlighting a trade-off between knock suppression and cyclic variability.

One thing to highlight is the impact of the specific heat ratio (γ) on the BTE. If the combustion process is modeled as an ideal Otto cycle, the thermal efficiency

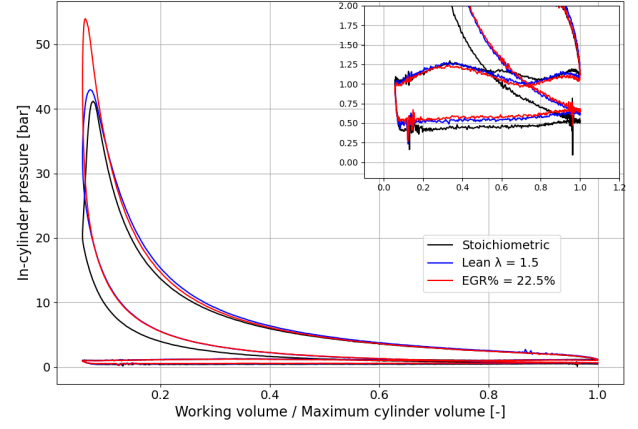


Figure 10: P-V diagram for the dilution strategies and most optimal spark timing with a zoomed view of pumping losses.

can be expressed as in Equation 3.

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{\gamma-1}} \quad (3)$$

with r being the CR and γ being the specific heat ratio.

Lean combustion, which increases the proportion of air in the mixture, raises γ , leading to a higher thermal efficiency, according to Equation 3. In contrast, EGR introduces CO_2 and H_2O , gases with lower γ , reducing the effective γ and thus limiting the theoretical efficiency. This is confirmed by Figure 11. Interestingly, at the minimum point of the γ curves, the EGR trace rises above the stoichiometric trace. This observation is attributed to the reduction in peak in-cylinder temperature caused by dilution, which significantly influences the specific heat ratio γ due to its strong temperature dependence. Although EGR allows for more advanced spark timing and knock suppression, it does not inherently boost thermal efficiency due to its lower γ effect. All of these findings are consistent with the findings reported by Zhu et al. [9].

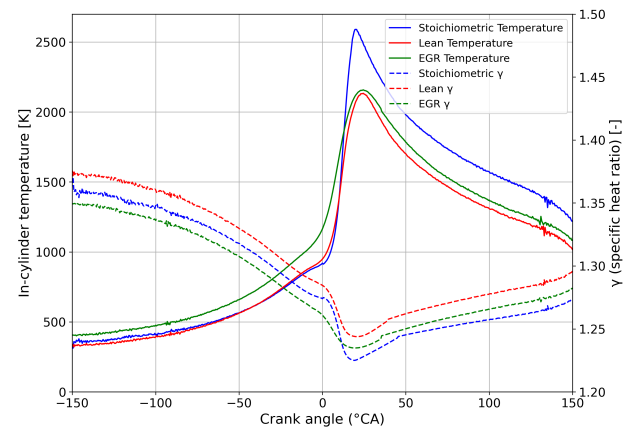


Figure 11: In-cylinder temperature and specific heat ratio for dilution at medium load.

A.3 Emissions

Methanol's inherent high latent heat of vaporization contributes to lower peak combustion temperatures, which inherently reduced nitric oxide (NO) emissions compared to conventional fuels. Furthermore, experimental results revealed that EGR suppresses NO emissions more compared to lean operation because of the lower in-cylinder temperatures at same dilution rates and the lower oxygen concentration.

Carbon monoxide (CO) emissions were primarily influenced by the air-fuel equivalence ratio. Lean combustion at 3.5 bar BMEP and $\lambda = 1.1$ reduced CO emissions by three to four times compared to stoichiometric operation. At low loads, emissions were elevated due to methanol's high latent heat of vaporization, which limited CO oxidation. EGR initially suppressed CO emissions by lowering combustion temperatures, thereby reducing CO_2 dissociation. However, beyond a critical EGR rate (17.5% at 3.5 bar BMEP), CO emissions increased again due to incomplete oxidation of unburned methanol in the crevices, consistent with the findings of Zhu et al. [9].

The aftertreatment systems of both dilution strategies also differ markedly. Lean operation fundamentally inhibits the use of a three-way catalyst (TWC) for NO_x reduction because the excess oxygen in lean exhaust prevents the simultaneous reduction of NO_x and oxidation of CO that a TWC requires.

In contrast, EGR dilution maintains a stoichiometric exhaust composition, making it fully compatible with TWC operation. This is critically important because EGR provides a similar diluting effect as lean combustion, improving efficiency and reducing emissions, but without exceeding the narrow air-fuel ratio range necessary for effective TWC performance.

A.4 WOT exploration

Wide Open Throttle (WOT) operation was explored to evaluate its potential for efficiency gains under lower load conditions using EGR-based dilution control. At 1500 RPM and 7.1 bar BMEP, WOT with EGR increased BTE by approximately 0.7% compared to conventional EGR control, primarily because of a more advanced spark timing. However, this approach also introduced significant combustion instability. The findings demonstrate that while WOT with EGR can enhance efficiency, it also requires stability improvements for reliable operation.

B Unlocking the potential of a passive prechamber

An investigation was made into a passive prechamber ignition system for a methanol-fueled SI engine, aiming to enhance combustion stability and efficiency under high dilution conditions. Although experimental validation was not completed, the prechamber is expected to extend the dilution limits enhancing BTE,

while improving combustion stability. A design of the passive prechamber was carefully made and manufactured, as shown in Figure 12.



Figure 12: Manufactured passive prechamber.

CONCLUSION

The key findings in this dissertation are:

- Both dilution strategies (Lean operation and EGR) result in higher BTEs, approaching diesel-like efficiencies as visible in Figure 13, but narrow the stable operating range.
- Both dilution strategies reduce emissions compared to stoichiometric operation, with EGR being the favorable option because of the stoichiometric exhaust and TWC compatibility.
- A passive prechamber ignition system would improve stability, thereby offering the possibility of reaching higher dilution rates and improving BTE.

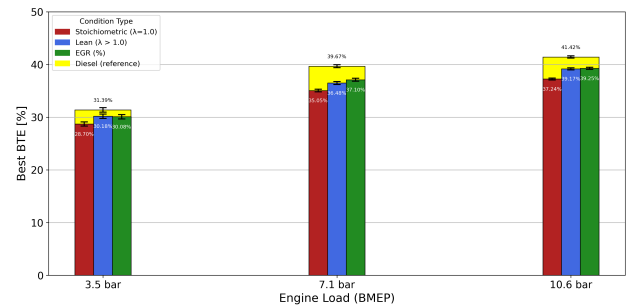


Figure 13: BTE at 1500 RPM for various load points and operating strategies.

REFERENCES

- [1] U. N. C. change, “The paris agreement,” Oct 2020.
- [2] K. Whiting, “How 6 heavy-emitting industries are working to decarbonize,” Sept 2024.
- [3] M. Wernke, “Diesel exhaust,” in *Encyclopedia of Toxicology (Third Edition)* (P. Wexler, ed.), pp. 111–114, Oxford: Academic Press, third edition ed., 2014.
- [4] R. Pearson and J. Turner, “2 - the role of alternative and renewable liquid fuels in environmentally sustainable transport,” in *Alternative Fuels and Advanced Vehicle Technologies for Improved Environmental Performance* (R. Folkson, ed.), pp. 19–51, Woodhead Publishing, 2014.
- [5] Z. Xie, Q. Jin, G. Su, and W. Lu, “A review of hydrogen storage and transportation: Progresses and challenges,” *Energies*, vol. 17, no. 16, 2024.
- [6] S. Verhelst, J. W. Turner, L. Sileghem, and J. Vancoillie, “Methanol as a fuel for internal combustion engines,” *Progress in Energy and Combustion Science*, vol. 70, pp. 43–88, 2019.
- [7] J. Vancoillie, J. Demuynck, L. Sileghem, M. Van De Ginste, S. Verhelst, L. Brabant, and L. Van Hoorebeke, “The potential of methanol as a fuel for flex-fuel and dedicated spark-ignition engines,” *Applied Energy*, vol. 102, pp. 140–149, 2013. Special Issue on Advances in sustainable biofuel production and use - XIX International Symposium on Alcohol Fuels - ISAF.
- [8] T. V. Sheldon Demiddeleer, “Experimental evaluation of the performance of methanol in a marine diesel engine retrofitted to si operation,” Master’s thesis, Ghent University, 2024.
- [9] Z. Zhu, Z. Mu, Y. Wei, R. Du, and S. Liu, “Experimental evaluation of performance of heavy-duty si pure methanol engine with egr,” *Fuel*, vol. 325, p. 124948, 2022.
- [10] Z. Zhang and Z. Zheng, “Numerical simulation of the effects of the egr ratio and ignition timing on a supercharged and high compression ratio hybrid gasoline engine,” *Fuel*, vol. 341, p. 127695, 2023.
- [11] Vancoillie, Jeroen and Verhelst, Sebastian and Demuynck, Joachim and Galle, Jonas and Sileghem, Louis and Van De Ginste, Maarten, “Experimental evaluation of lean-burn and EGR as load control strategies for methanol engines,” in *SAE TECHNICAL PAPER SERIES*, p. 18, Society of Automotive Engineers, 2012.

Contents

I	Literature Review	1
1	Introduction	2
1.1	De-fossilizing the heavy-duty industry	2
1.2	Fueling the Future	3
1.2.1	Battery-Electric	4
1.2.2	Gaseous Fuels	4
1.2.3	Liquid Fuels	5
1.2.4	A Compelling Choice for Heavy-Duty Applications	6
2	Retrofitting Diesel CI To Methanol SI	7
2.1	Methanol properties relevant to ICEs	7
2.1.1	Key Chemical and Physical Properties	7
2.1.2	Comparison with Diesel and Gasoline	9
2.2	From CI to SI	11
2.2.1	Why use retrofitting?	11
2.2.2	Retrofitting Diesel Engine for Methanol Spark Ignition	11
3	Engine Optimization	13
3.1	Performance parameters	13
3.1.1	Efficiency	13
3.1.2	Maximal load	14
3.1.3	Operational limits	15
3.2	Limiting factors	17
3.2.1	Knock	17
3.2.2	Stability	22
3.2.3	Cold start	23

3.2.4	Emissions	24
3.3	Performance enhancing techniques	31
3.3.1	EGR	31
3.3.2	Lean combustion	37
3.3.3	Passive Prechamber Technology	40
3.4	Problem & Goal statement	44
3.4.1	Problem Statement	44
3.4.2	Goal Statement	44
II	Experimental Research	46
4	Experimental Setup	47
4.1	Baseline marine diesel engine	47
4.2	Retrofitting	48
4.2.1	Shifting from diesel to DMDF	48
4.2.2	Transitioning from DMDF to Methanol SI	50
4.3	Additional engine systems	52
4.3.1	EGR loop	52
4.4	Data collection	53
4.4.1	Measurement equipment	53
4.4.2	Software	53
4.4.3	Engine performance parameters	55
4.5	Setup modifications	57
4.5.1	Throttle body	57
4.5.2	Water circuit	59
4.5.3	Maintenance	59
5	Unveiling the Impact of Dilution on the Engine Performance	60
5.1	Measurement Methodology	60
5.2	Operational Limits	62
5.2.1	Stoichiometric operation	63
5.2.2	Dilution tolerance	66
5.3	Uncovering the Efficiency Potential of Lean Burn and EGR Dilution	76

5.3.1	Stoichiometric operation	76
5.3.2	Effect of dilution	80
5.4	Emissions	89
5.4.1	NO emissions	89
5.4.2	CO-emissions	93
5.5	Exploration of Wide Open Throttle (WOT) Operation	96
5.6	Conclusion: Lean Versus EGR Dilution	98
5.6.1	Knock Suppression and Combustion Phasing	98
5.6.2	Combustion Stability	99
5.6.3	Thermal Efficiency and Heat Release	99
5.6.4	Emissions and Aftertreatment Compatibility	99
5.6.5	Practical Implementation Considerations	99
5.6.6	Extending EGR Limits with Stability-Enhancing Technologies . . .	99
5.6.7	Overall Assessment	100
6	Unlocking the Potential of a Passive Prechamber to boost Heavy-Duty Methanol SI Engine Performance	101
6.1	Introduction	101
6.2	Prechamber Working Principle	101
6.3	Design Constraints and Assumptions	102
6.4	Geometric Design Parameters	102
6.4.1	Prechamber Volume	102
6.4.2	Nozzle Area and A/V Ratio	102
6.4.3	Spray Angle and Jet Targeting	103
6.5	Final Assembly and Installation Methodology	103
6.6	Summary and Future Work	103
7	Conclusion	106
7.1	Literature Review	106
7.2	Experimental research	107
8	Future Outlook	110
8.1	Future Research Directions	110
8.2	Future work on the Experimental Setup	111

III	Appendices	114
A	Volvo Penta operating procedures	115
B	Last Minute Startup Analysis - SI	119
C	Uncertainty analysis of experimental data	121
D	Transition from MATLAB to Python for Data Processing	126
E	LabVIEW Changes	130
F	Encountered Technical Issues and Solutions	132
G	Determining EGR rate	135
H	Throttle Valve Replacement	137
I	Procedures for Diluted Operation	141
J	Base Engine Data Sheet	143
K	Emission Results	146
	Bibliography	156

List of Figures

1.1	Overview of global greenhouse gas emissions [2].	2
1.2	Volumetric energy density as a function of gravimetric energy density for different energy carriers [7].	3
1.3	Net volumetric and gravimetric energy density of different fuels [8].	5
2.1	Effect of methanol properties on engine performance [27].	10
2.2	PFI versus GDI Injection Systems [32].	12
3.1	Minimum Advance for Maximum Brake Torque (MBT) with influence of ignition timing.	15
3.2	Performance map, showing BTE [39].	15
3.3	Stable operating range at different loads, with knock limiting the spark advancing and combustion stability limiting spark retarding [21].	16
3.4	Graphical representation of pre-ignition and conventional knock [42].	17
3.5	Engine damage by conventional knock [41].	18
3.6	Cylinder pressure traces of typical pre-ignition and normal cycles [41].	18
3.7	Knock overview [41].	19
3.8	In-cylinder pressure and amplitude of pressure oscillations for normal and knock engine behavior [46].	20
3.9	High tumble intake port [41].	21
3.10	COV of IMEP for Conventional, CO ₂ -only and N ₂ -only EGR at 3000 RPM / 16.0 bar nIMEP [51].	22
3.11	Spark Advance and Combustion Response with Conventional, CO ₂ -only, and N ₂ -only EGR at 3000 RPM / 16 bar nIMEP [51].	23
3.12	The formation of emissions of soot (in ppm by mass, in ‘island’ at top of plots, i.e. at the high equivalence ratios) and NO _x (in ppm by volume, bottom right of plots, i.e. at high temperatures and lower equivalence ratios) as a function of equivalence ratio and temperature, for methanol and diesel [58].	27

3.13	Performance map for the heavy-duty methanol SI engine showing the NO_x emissions [39].	27
3.14	Performance map for the heavy-duty methanol SI engine showing the CO emissions [39].	29
3.15	Performance map for the heavy-duty methanol SI engine showing the HC emissions [39].	31
3.16	Schematic showing the principle of EGR in an ICE [66].	33
3.17	(a) LP versus (b) HP EGR [65].	34
3.18	(a) clean versus (b) dirty EGR [65].	34
3.19	Variations in the BTE with respect to the EGR rate at different speeds [39].	35
3.20	Effects of the EGR rate on CA0-10, CA50 and CA10-90 under different loads [39].	35
3.21	Effects of the EGR rate on the BTE under different loads [39].	36
3.22	NO_x emissions and bsfc versus EGR rate [70].	36
3.23	BTE at optimal spark timing for different loads and varying lambda values at 1500 RPM [21].	37
3.24	Pumping mean effective pressure (PMEP) as a function of the air/fuel equivalence ratio for different loads at 1500 RPM [21].	38
3.25	Emissions of HC, CO and NO_x of engine fueled with gasoline and methanol at different λ under idle conditions [71].	39
3.26	Coefficient of variation of indicated mean effective pressure (COV_{IMEP}) as a function of λ at different loads with constant engine speed of 1500 RPM [21].	40
3.27	Schematic of the jet ignition concept with an actively fueled prechamber. The star represents the spark and the arrows highlight the locations where fuel injections [77].	41
3.28	Schematic of the prechamber configurations, passive on the (left) and active on the (right) [76].	42
3.29	Spray angle of prechamber indicated with symbol α [80].	43
4.1	Volvo Penta D7C-B TA, the baseline marine diesel engine used in this dissertation.	47
4.2	Methanol supply system, adapted from [83].	50
4.3	Retrofit from diesel injectors to spark plugs [22].	51
4.4	Ignition system.	51
4.5	Schematic representation of the Engine setup with additional engine equipment, adapted from [21].	52

4.6	Exhaust Gas Recirculation (EGR) loop. Blue path: cooled EGR ("cold EGR"); red path: bypass over EGR cooler ("hot EGR"), Adapted from [22].	53
5.1	Effect of spark timing on COV_{IMEP} for different loads (BMEP) under stoichiometric conditions, with the spark timing corresponding to peak BTE highlighted.	63
5.2	Visualization of combustion duration parameters plotted alongside the pressure trace and cumulative heat release (CHR) curve.	64
5.3	Effects of spark timing on CA_{0-10} , CA_{50} , and CA_{10-90} across different loads (BMEP) under stoichiometric conditions, with the spark timing for maximum BTE indicated.	65
5.4	Stable operating range for lean burning at different loads, with knock constraining further spark advance and combustion stability setting the limit for spark retardation.	68
5.5	Stable operating range for EGR burning at different loads, with knock constraining further spark advance and combustion stability setting the limit for spark retardation.	68
5.6	Effect of spark timing on COV_{IMEP} for lean burning at BMEP = 3.5 bar with the spark timing corresponding to peak BTE highlighted.	69
5.7	Effect of spark timing on COV_{IMEP} for EGR operation at BMEP = 3.5 bar with the spark timing corresponding to peak BTE highlighted.	70
5.8	Effect of dilution on the in-cylinder temperature at the same spark timing (7.1 bar BMEP at 25°BTDC).	71
5.9	Effect of dilution ($\lambda = 1.2$ and EGR% = 15%) on the in-cylinder temperature.	72
5.10	Effect of spark timing on COV_{IMEP} for lean burning at BMEP = 10.6 bar with the spark timing corresponding to peak BTE highlighted.	72
5.11	Effect of spark timing on COV_{IMEP} for lean burning at BMEP = 10.6 bar with the spark timing corresponding to peak BTE highlighted.	73
5.12	Effect of lean burn on combustion durations for different loads, with averaged durations over all stable measurements.	74
5.13	Effect of EGR on combustion durations for different loads, with averaged durations over all stable measurements.	75
5.14	Brake thermal efficiency as a function of spark timing for different loads (BMEP) at stoichiometric operation.	76
5.15	Average exhaust temperature as a function of spark timing for all loads (BMEP) at $\lambda = 1$	78
5.16	In-cylinder pressure and heat release rate as a function of degrees crank angle for different spark timings and a BMEP of 7.1 bar (1500 RPM, 400 Nm, $\lambda = 1$).	78

5.17	In-cylinder pressure and heat release rate as a function of degrees crank angle for different spark timings and a BMEP of 3.5 bar (1500 RPM, 200 Nm, $\lambda = 1$).	79
5.18	P-V diagram for different loads (BMEP), stoichiometric operation, and most optimal spark timing with a zoomed view of pumping losses.	80
5.19	BTE as a function of spark timing for different loads and increasing lambda values.	80
5.20	BTE as a function of spark timing for different loads and increasing EGR%.	81
5.21	P-V diagram for the dilution strategies and most optimal spark timing with a zoomed view of pumping losses.	82
5.22	PMEP as function of amount of dilution.	82
5.23	Effect of dilution on the HRR and pressure trace for different loads.	83
5.24	Effect of dilution on the in-cylinder temperature at medium load (BMEP = 7.1 bar).	84
5.25	Effect of dilution on the HRR and pressure trace for different dilution strategies.	84
5.26	In-cylinder pressure and specific heat ratio for dilution at medium load.	86
5.27	In-cylinder temperature and specific heat ratio for dilution at medium load.	87
5.28	Best brake thermal efficiency (BTE) for each dilution strategy and engine load, under stable operating conditions ($\text{COV} \leq 3\%$).	88
5.29	Influence of spark timing on NO-emissions across different loads at stoichiometric operation.	89
5.30	Influence of spark timing on NO-emissions across different loads at Lean operation ($\lambda = 1.2$).	90
5.31	Peak in-cylinder temperature as a function of the spark timing for stoichiometric operation at different loads.	91
5.32	Influence of dilution strategies on NO-emissions at 3.5 bar BMEP.	92
5.33	Influence of dilution strategies on peak in-cylinder temperatures at 3.5 bar BMEP.	92
5.34	NO_x emission limits as a function of the engine speed, as defined by MARPOL Annex VI [92].	93
5.35	Influence of spark timing on CO-emissions across different loads.	94
5.36	Influence of dilution strategies on CO-emissions at 3.5 bar BMEP.	95
5.37	Minimum exhaust temperature across different cylinders and spark timings as a function of EGR rate.	96
5.38	COV_{IMEP} for different loads at WOT.	97

5.39	Comparison of BTE at 400 Nm with conventional EGR control ($\text{COV}_{\text{IMEP}} \leq 3\%$) versus WOT operation with extended dilution.	98
6.1	Final design of the passive prechamber assembly.	104
6.2	Manufactured prechamber.	104
6.3	Technical drawing of final prechamber design.	105
7.1	Brake thermal efficiency (BTE) at a constant engine speed of 1500 RPM for various load points and operating strategies.	108
7.2	Calculated in-cylinder temperatures for different loads in diesel and methanol operation.	109
B.1	10 last checks before starting the Volvo Penta engine.	120
D.1	In-cylinder temperature at 1500 rpm and 3.5 bar BMEP for Matlab processing file and Python processing file.	129
D.2	Heat capacity ratio at 1500 rpm and 3.5 bar BMEP for Matlab processing file and Python processing file.	129
E.1	Added wait-statements in EGR control loops reducing CPU usage.	131
F.1	Fluctuating exhaust temperature of cylinder 1 indicating a malfunctioning methanol injector for that cylinder.	132
F.2	Dropping of exhaust temperature of cylinder 6 indicating a faulty spark plug.	133
F.3	Corroded EGR Valve (left) and new EGR Valve (right).	133
H.1	VDO 408-238-327-003Z Throttle Valve.	137
H.2	Connector and pinout for throttle body connector.	138
H.3	Pin layout of the MoTeC M800 module using the MoTeC DBW user manual.	139
K.1	Influence of dilution strategies on NO emissions at 7.1 bar BMEP	146
K.2	Influence of dilution strategies on NO emissions at 10.6 bar BMEP	147
K.3	Influence of dilution strategies on CO emissions at 7.1 bar BMEP	147
K.4	Influence of dilution strategies on CO emissions at 10.6 bar BMEP	148

List of Tables

2.1	Comparison of key properties of Methanol, Diesel, and Gasoline relevant to ICEs [21], [22], [25], [26], [27].	9
3.1	Overview of mechanisms of methanol on NO _x emissions [57].	26
3.2	Overview of mechanisms of methanol on CO emissions [57].	28
3.3	Overview of mechanisms of methanol on HC emissions [57].	30
3.4	A comparison of the characteristics between cooled EGR and hot EGR [65].	33
4.1	Technical specifications of the baseline Volvo Penta D7C-B TA marine diesel engine.	48
4.2	Retrofit components DMDF, adapted from [22].	49
4.3	Retrofit components SI, adapted from [21].	51
4.4	Overview of engine systems.	52
4.5	Details of measurement equipment, adapted from [22].	54
4.6	Overview of calculated engine performance parameters, adapted from [21].	58
5.1	Overview of measurement sets at different load points.	61
5.2	Stable spark timing range for each load condition under stoichiometric operation.	66
5.3	Air Dilution Percentage for various λ -values.	67
5.4	KLSA for low, medium and high load operating points at stoichiometric combustion.	77
C.1	Range, accuracy and resolution of measured quantities, adapted from [21].	122
G.1	Molar Fractions and Molecular Weights of EGR Components.	136
H.1	Pin configuration for connecting the SI throttle valve to the MoTeC M800 module, following the guidelines provided in the MoTeC DBW user manual.	138

H.2	Configuration settings for the VDO Valve (06F 133 062 A) with M800 ECU setup.	140
-----	--	-----

Nomenclature

Abbreviations

A/V	Area to Volume
AFR	Air/Fuel Ratio
AI	Artificial Intelligence
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake-Specific Fuel Consumption
BTDC	Before Top Dead Center
BTE	Brake Thermal Efficiency
CA	Crank Angle
CAD	Computer-Aided Design
CC	Cubic Centimeter
CCV	Cycle-to-Cycle Variability
CFD	Computational Fluid Dynamics
CH ₃ OH	Methanol
CH ₄	Methane
CHR	Cumulative Heat Release
CI	Compression Ignition
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
COV	Coefficient of Variation
CPU	Central Processing Unit
CR	Compression Ratio
DBW	Drive-By-Wire

DI Direct Injection
 DISI Direct Injection Spark Ignition
 DMDF Diesel-Methanol Dual-Fuel
 ECU Engine Control Unit
 EGR Exhaust Gas Recirculation
 EOC End Of Combustion
 EOI End Of Injection
 ETB Electronic Throttle Body
 FAME Fatty Acid Methyl Ester
 GDI Gasoline Direct Injection
 GHG Greenhouse Gas
 GWP Global Warming Potential
 HC Hydrocarbons
 HD Heavy-Duty
 HP High-Pressure
 HRR Heat Release Rate
 HVO Hydrotreated Vegetable Oil
 ICE Internal Combustion Engine
 IJPU Injector Opening Duration
 IMEP Indicated Mean Effective Pressure
 IMO International Maritime Organization
 IT Ignition Timing
 ITE Indicated Thermal Efficiency
 KLSA Knock Limited Spark Advance
 LBV Laminar Burning Velocity
 LNG Liquefied Natural Gas
 LP Low Pressure
 LPG Liquefied Petroleum Gas
 LSPI Low-Speed Pre-Ignition
 MAPO Maximum Amplitude of Pressure Oscillation
 MBT Minimum Spark Advance for Maximum Brake Torque
 MC Main Chamber
 MON Motor Octane Number

MPI	Multi-Point Injection
MW	Molecular Weight
N ₂ O	Nitrous Oxide
NO	Nitric Oxide
NO ₂	Nitrogen Dioxide
NO _x	Nitrogen Oxides
NRMM	Non-Road Mobile Machinery
O ₂	Oxygen
P-V	Pressure-Volume
PC	Prechamber
PFI	Port Fuel Injection
PID	Proportional-Integral-Derivative
PLN	Pump-Line-Nozzle
PM	Particulate Matter
PMEP	Pumping Mean Effective Pressure
PPM	Parts Per Million
RON	Research Octane Number
RPM	Rotations Per Minute
SAE	Society of Automotive Engineers
SI	Spark Ignition
SO _x	Sulfur Oxides
SOI	Start Of Injection
ST	Spark Timing
TDC	Top Dead Center
TP	Throttle Position
TWC	Three-Way Catalyst
VPM	Volume Parts per Million
WOT	Wide Open Throttle

Greek Symbols

γ	Specific Heat Ratio
λ	Air/Fuel Equivalence Ratio
ϕ	Fuel/Air Equivalence Ratio

Part I

Literature Review

Chapter 1

Introduction

1.1 De-fossilizing the heavy-duty industry

To meet the climate goals stated by the Paris agreement [1], large carbon dioxide (CO_2) emitting industries will need to drastically reduce their carbon footprint. The heavy-duty (HD) transport industry is responsible for 9% of global greenhouse gas (GHG) emissions [2]. This includes aviation (3%), shipping (2%) and trucking (4%) as can be seen in Figure 1.1.

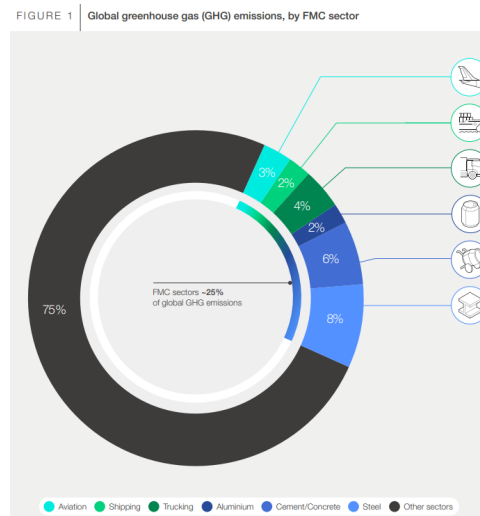


Figure 1.1: Overview of global greenhouse gas emissions [2].

Furthermore, by 2050, the demand for sea freight is predicted to triple and that for trucking to double in the same time frame [2]. This emphasizes the pressure on the heavy-duty fleet to reach the obliged carbon-neutral goals.

Most of the current HD fleet is powered by internal combustion engines (ICEs), running on diesel. This is due to their high efficiency, high energy density and reliable performance under harsh conditions. Nevertheless, diesel exhaust contains a complex mix of greenhouse gases and pollutants. The GHGs typically include carbon dioxide, nitrous oxide (N_2O), and smaller quantities of methane (CH_4), which contribute to climate change. Pollutants

in diesel exhaust, on the other hand, are more directly harmful to human health and the environment. These include particulate matter (PM), nitrogen oxides (NO_x), carbon monoxide (CO), hydrocarbons (HC), sulfur oxides (SO_x), and black carbon. Black carbon, in particular, is a potent short-lived climate pollutant, with a warming potential thousands of times greater than CO_2 in the near term [3] [4]. These emissions are the result of incomplete combustion and high pressure conditions within diesel engines [4].

Modern after-treatment systems can eliminate most NO_x , HC, CO, and PM from exhaust gases [5]. This is a solution for pollutant emission, but does not pose a solution to the climate impact of the fuel. It is obvious that an alternative approach to driving heavy-duty vehicles is needed to achieve a carbon neutral future.

1.2 Fueling the Future

To get a comprehensive overview of possible alternatives for Diesel use in HD applications, each following section will dive into the feasibility of emerging green technologies. They will be evaluated on their ability to comply with the requirements of a HD energy carrier [6]:

- **High energy density:** The alternative energy carrier should have a high energy density (both volumetric and gravimetric) to ensure sufficient power and range for heavy-duty applications without excessively large or heavy fuel storage systems. Figure 1.2 shows a graph of the volumetric and gravimetric energy density of multiple energy carriers. The blue arrows indicate the effect of specific storage systems needed on the energy density, like cryogenic or high pressure tanks for hydrogen storage for example.

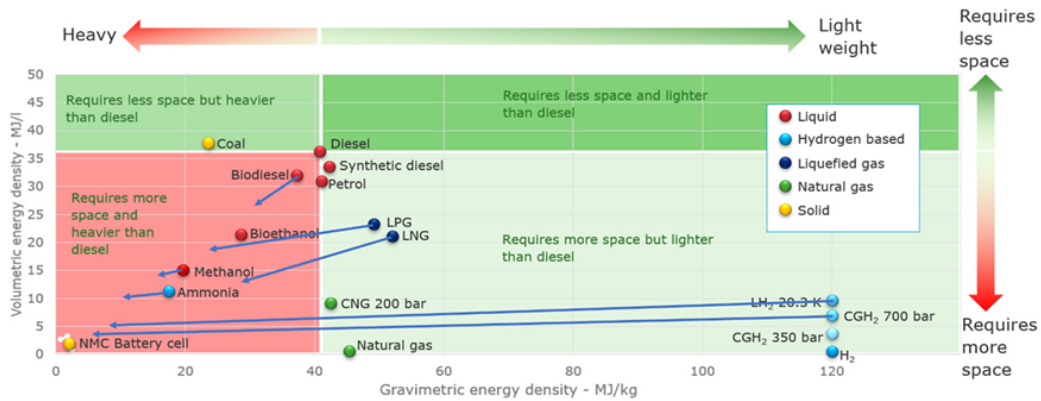


Figure 1.2: Volumetric energy density as a function of gravimetric energy density for different energy carriers [7].

- **Power density:** The alternative energy converter e.g. the engine, motor or fuel cell, should support power densities that allow machinery to maintain performance levels comparable to diesel engines. It should also integrate with existing engine technologies or be adaptable for use with ICEs, electric motors, or fuel cells.

- **Infrastructure availability and scalability:** The fuel or technology must be supported by existing or rapidly scalable infrastructure for production, distribution, and refueling.
- **Environmental sustainability:** The alternative should offer significant reductions in GHG and pollutant emissions over its life-cycle, including production, distribution, and end-use.
- **Economic viability:** The alternative should be cost-competitive with diesel in terms of production, operation, and maintenance. This includes potential retrofitting costs for existing engines or the price of a new powertrain.
- **Reliability and safety:** The replacement should be reliable in various operating conditions, ensuring durability and consistent performance. Safety in handling, storage, and use is also crucial.

1.2.1 Battery-Electric

A battery package driving an electric motor is the most efficient way of green energy storage, compared to hydrogen or e-fuel conversion. No tailpipe emissions are produced and an electric setup can co-exist in hybrid solutions. On the contrary, batteries have a low energy density [8], resulting in large, heavy battery packs for heavy-duty machinery [9]. Additionally, this limits the range for applications operating long hours or high energy loads. Although locally no emissions are produced, there are still a lot of question marks around the sustainability of battery packs and the use of rare earth materials [10]. New battery technologies, like solid-state batteries [9], are being developed improving recyclability of batteries. Unfortunately it is estimated that only 5% of the lithium-ion batteries are being recycled at the moment [11]. Thus concerns about resources and scalability are another disadvantage of this technology.

1.2.2 Gaseous Fuels

Hydrogen

Hydrogen can be easily produced from green electricity by means of electrolysis. It is very versatile as it can be used as a fuel in a combustion engine or fuel cell and as a foundational component for other carbon-neutral synthetic fuels [12]. The main issue with hydrogen as an energy carrier is again the low energy density. Hydrogen has a gross gravimetric energy density of 120 MJ/kg, which is approximately three times higher than that of diesel. The volumetric energy density on the other hand is 0.01079 MJ/L at atmospheric conditions [12]. This is due to its nature as a lightweight gas, with a density of just 0.08 kg/m³ at atmospheric conditions. This requires high-pressure or cryogenic storage, increasing the density to 25 kg/m³ (at 350 bar) or 70.85 kg/m³, respectively. Both of these methods are energy-intensive and require bulky, heavy tanks. Another problem is the limited infrastructure for hydrogen production, distribution and refueling [12].

Methane

Methane, the primary component of natural gas, can be generated from biological material or produced using renewable energy, resulting in bio-methane or e-methane, respectively. It can be used in compressed (CNG) or liquefied (LNG) form [13]. In this way it can reach half the volumetric energy density of diesel, storage tanks not included [6]. Furthermore, the existing gas infrastructure can be repurposed for the transportation and storage of methane. The main downside of methane is its global warming potential (GWP). The GWP is defined to quantify the global warming effect of a gas. Each GHG is evaluated based on its ability to trap heat over a specific time period compared to CO₂. For example, methane has a GWP that is 28 times greater than CO₂ over a 100-year period and even 86 times over a 20-year period [14]. Leakage during production and distribution can offset the benefits.

Ammonia

Ammonia offers advantages over hydrogen in terms of storage, as it can be stored in liquid form at ambient temperature under a pressure of only 8 bar. Additionally, as the world's second most produced chemical, ammonia benefits from well-established production and transportation infrastructure [6]. Ammonia does not contain carbon, resulting in a CO and CO₂ free combustion. On the other hand, nitrogen oxides, including N₂O which has a GWP 300 times higher than CO₂, are being produced. Additionally, the lower flame speed, as well as its high ignition energy and temperature requirements, limits the performance of ammonia in ICEs [15].

1.2.3 Liquid Fuels

Liquid fuels have notable benefits over gaseous fuels, primarily due to their higher energy density, as can be seen in Figure 1.3. This makes onboard storage easier for mobile applications and supports long-distance transport. They are also easier to handle and can leverage existing infrastructure for storage and use. Additionally, liquid fuels are typically safer, posing fewer risks than compressed or cryogenic gas storage, which requires specialized safety protocols to manage potential hazards [8] and decrease the potential energy density of the system, see Figure 1.2.

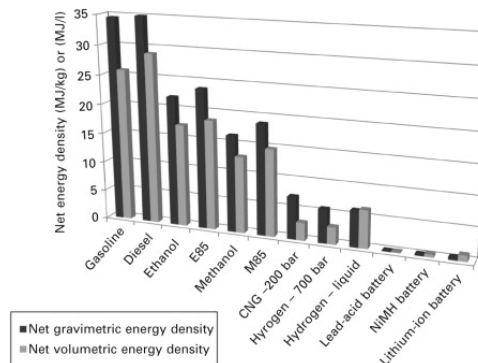


Figure 1.3: Net volumetric and gravimetric energy density of different fuels [8].

Methanol

Methanol is a simple alcohol that remains liquid at ambient temperature and pressure. It can be synthesized from second generation biomass or through the use of electricity, or a combination of both, resulting in bio-methanol and e-methanol. Methanol, being one of the top five chemicals used globally, is already abundantly available worldwide, and its use and safe handling are well-established [6]. However, it is important to note that the majority of the currently available methanol, known as grey methanol, is derived mainly from natural gas in America and Europe, and from coal in Asia [16]. New engines capable of running on methanol using the dual-fuel principle are already available, highlighting the progress in methanol adoption. NO_x is still being produced during combustion of methanol and it has a volumetric energy density of less than half that of diesel [8].

Bio-diesel or HVO

Fatty acid methyl esters (FAME), known as biodiesel, are typically mixed with fossil diesel to produce a blend called B7, containing up to 7% biodiesel. Higher concentrations of biodiesel require modifications to existing engines [6]. Another bio-based diesel alternative is hydrotreated vegetable oils (HVO). Unlike FAME, HVO can be used directly as a drop-in fuel without needing to be mixed with fossil diesel or requiring engine modifications, as it meets diesel fuel standards, having a relatively equal volumetric energy density as fossil diesel [17]. The problems with these two diesel replacements are that these primarily rely on first-generation biological material, which may compete with food production [6]. Bio-diesel also entails higher production costs than fossil diesel.

1.2.4 A Compelling Choice for Heavy-Duty Applications

For the heavy-duty industry, the driving requirement is high power and volumetric energy density for sufficient autonomy and mobility. Here, liquid fuels have the edge on gaseous fuels and battery-electric solutions. Liquid fuels also have an advantage with respect to storability and safety. Unlike Bio-diesel or HVO, green methanol production does not rely on bio-based feedstock but can be made with captured CO_2 and hydrogen, ensuring a circular carbon life cycle. Moreover, methanol is a promising renewable fuel for internal combustion engines, offering clean-burning properties and lower emissions. Its molecular structure, free of carbon-carbon bonds, reduces soot formation, while its low-temperature combustion minimizes NO_x emissions. Additionally, methanol's high octane rating allows for improved engine efficiency, making it a viable alternative for cleaner and more efficient heavy-duty applications. This is why methanol will be fueling our research on diesel replacement in the heavy-duty industry. Existing dual-fuel (methanol and diesel) technology is already in practice on large vessels [18] [19] [20]. The problem here is that it still relies on diesel (for some load points, almost no methanol is used). To combat this, dedicated engines need to be designed, covering the entire power range for renewable fuel options.

A more in depth discussion about the current production, availability, distribution and costs of methanol and green methanol can be found in previous research [21] [22], but is not further covered here.

Chapter 2

Retrofitting Diesel CI To Methanol SI

In the maritime industry, the shift toward sustainable energy sources is important to target stringent emission reductions [2]. Methanol has emerged as a promising alternative fuel because of its potential for reducing greenhouse gas emissions and its compatibility with existing infrastructure [23]. Retrofitting existing diesel engines to operate on methanol in a spark ignition (SI) mode offers a practical pathway to achieve these environmental goals without the immediate need for entirely new engine designs.

Methanol's high octane rating (cf. Table 2.1) and clean combustion characteristics make it suitable for SI engines. However, transitioning from compression ignition (CI) diesel engines to methanol-fueled SI engines includes several technical challenges. These encompass modifications to the fuel injection system, ignition system and engine control strategies to accommodate methanol's different combustion properties. Studies have demonstrated that with appropriate modifications, diesel engines can be effectively converted to operate on methanol, achieving reasonable performance while significantly reducing emissions [22] [24].

This chapter explores the technical considerations of retrofitting marine diesel engines to methanol SI operation. It begins by examining methanol's unique properties and their relevance to ICEs compared to diesel and gasoline. The discussion is then followed by the usefulness of retrofitting and some differences between CI and SI engines.

2.1 Methanol properties relevant to ICEs

2.1.1 Key Chemical and Physical Properties

- **Flame Speed:** Methanol's high laminar flame speed, approximately 42 cm/s at stoichiometric conditions, surpasses that of gasoline, which is around 37 cm/s [25]. This characteristic facilitates faster and more efficient combustion cycles in SI engines, enhancing thermal efficiency and reducing combustion duration. The rapid flame propagation allows for more complete combustion, leading to higher power

output and improved fuel economy.

- **Latent Heat of Vaporization:** One of methanol's standout features is its high latent heat of vaporization (approximately 1100 kJ/kg), which is considerably higher than that of diesel or gasoline, see Table 2.1. This cooling effect is beneficial as it lowers in-cylinder temperatures, further helping to reduce knock and NO_x emissions [26]. However, as methanol needs more heat to vaporize fully, this can lead to challenges with cold starts.
- **Auto-Ignition Temperature:** Methanol has a relatively high auto-ignition temperature (around 738 K), which means it is less prone to auto-ignition at typical engine operating temperatures. This is ideal for SI engines that rely on spark ignition rather than compression for fuel ignition. The high auto-ignition temperature of methanol reduces the likelihood of pre-ignition and knocking, particularly in high-compression engines being retrofitted from diesel. This property allows methanol SI engines to operate safely under demanding conditions, such as high-load scenarios in marine applications. Methanol's higher auto-ignition temperature and flash point compared to gasoline makes it also safer for transportation and storage [25].
- **Energy Density:** Methanol's volumetric energy density is approximately 15.9 MJ/l, which is significantly lower than that of gasoline (about 31.7 MJ/l) and diesel (around 35.8 MJ/l) [27]. This lower energy content means that, for a given volume, methanol provides less energy, necessitating larger fuel tanks or more frequent refueling to achieve the same operational range as diesel or gasoline engines. In marine applications, this can impact vessel design and payload capacity, as additional space must be allocated for fuel storage. Moreover, the increased fuel consumption associated with methanol may affect overall operational efficiency and logistics planning. Therefore, when retrofitting diesel engines to methanol, careful consideration of fuel storage solutions and consumption rates is essential to maintain performance and range. The combination of methanol's lower energy density (requiring approximately twice the fuel volume) and its high latent heat of vaporization (roughly four times that of diesel) results in a substantially greater cooling effect. This strong cooling can reduce in-cylinder temperatures, helping to suppress knock and increase volumetric efficiency in SI engines. However, it may also complicate cold start performance and fuel vaporization under low-temperature conditions.
- **Octane Rating:** Methanol's high Research Octane Number (RON), often around 109, makes it well-suited for SI engines [27]. This high octane level indicates excellent resistance to knocking, which is a primary advantage in SI applications, particularly when high compression ratios are used in heavy-duty engines. Methanol's Motor Octane Number (MON) of around 92, though slightly lower than the RON, also contributes to its knock-resistant properties [27]. Together, these values imply that methanol can handle the pressures and temperatures typical in SI combustion without premature ignition.

Table 2.1 compares the most relevant key properties of methanol, diesel and gasoline, while Figure 2.1 highlights the effect of these properties to the engine performance.

Table 2.1: Comparison of key properties of Methanol, Diesel, and Gasoline relevant to ICEs [21], [22], [25], [26], [27].

Property	Methanol	Diesel	Gasoline
Chemical Formula	CH ₃ OH	Various	Various
Density (STP) (kg/m ³)	790	830	740
Energy Density (MJ/l)	15.9	35.8	31.7
Flame Speed (cm/s)	42	30	37
Latent Heat of Vaporization (kJ/kg)	1100	250	180-350
Lower Heating Value (MJ/kg)	20.09	42.74	42.90
Auto-Ignition Temperature (K)	738	530	465-743
Research Octane Number (RON)	109	-	91-98
Motor Octane Number (MON)	92	-	81-86
Cetane Number	3	40-55	0-14
Flash point (°C)	11	55	-45
Adiabatic flame temperature (K)	2143	-	2275
Flammability limits in air (vol%)	6.70-36.00	-	1.30-7.60
Flammability limits in air [λ]	0.23-1.81	-	0.26-1.60
Stoichiometric Air-Fuel Ratio	6.5:1	14.6:1	14.7:1
Soot Emissions	Negligible	High	Moderate

2.1.2 Comparison with Diesel and Gasoline

- **Efficiency in SI:** Methanol’s high octane rating and flame speed (cf. Table 2.1) make it superior to gasoline in terms of efficiency and knock resistance in SI engines, particularly under heavy-duty demands. When compared to diesel, methanol’s potential to achieve diesel-like efficiencies in SI operation is promising [21]. Diesel engines benefit from the thermodynamic advantages of the Diesel cycle, such as higher compression ratios and lean combustion, contributing to Brake Thermal Efficiency (BTE) values exceeding 40 percent. However, studies indicate that retrofitted methanol SI engines can approach similar efficiency levels under optimized conditions. For instance, methanol’s high latent heat of vaporization contributes to charge cooling, improving volumetric efficiency, compensating for some of the efficiency gap [28].

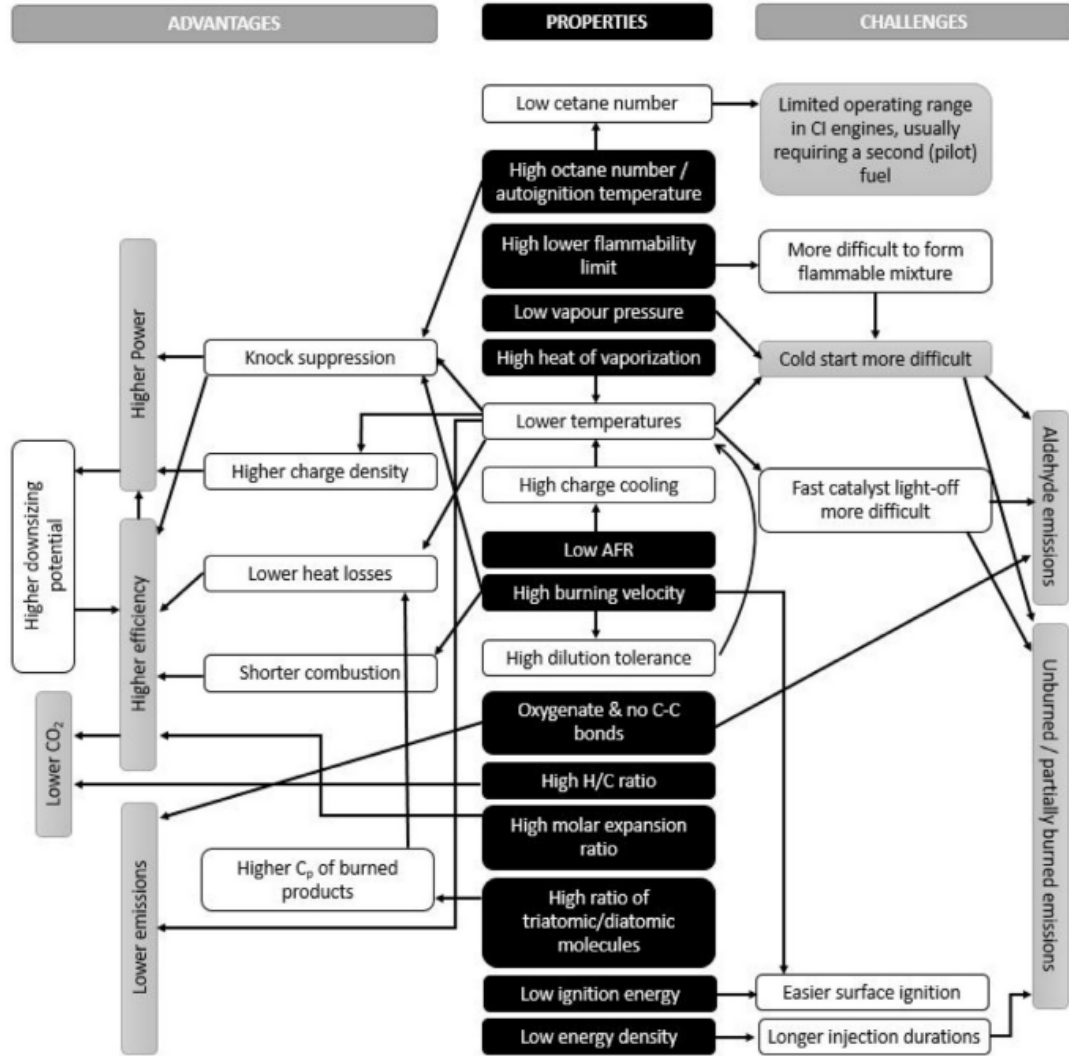


Figure 2.1: Effect of methanol properties on engine performance [27].

- Emission Benefits:** Methanol's chemical structure, which includes a high oxygen content and no carbon-carbon bonds, contributes to cleaner combustion with almost no soot. Compared to gasoline, NO_x production is reduced, owing to its lower combustion temperatures and more complete burning process. On the other hand, however, methanol can produce higher levels of unburned hydrocarbons and formaldehyde [25]. The emissions produced by burning methanol are further discussed in subsection 3.2.4.
- Knock Resistance and Heavy-Duty Application:** Methanol's high knock resistance makes it an appealing choice for SI engines in heavy-duty at high compression ratios. The fuel's cooling effect helps mitigate knock even further, allowing higher thermal efficiencies. A further discussion of knock resistance is given in chapter 3.

2.2 From CI to SI

2.2.1 Why use retrofitting?

Retrofitting existing marine engines to run on methanol offers a practical solution to bridge the gap between current diesel engines and the new, purpose-designed methanol engines expected in the future. Given that the average lifespan of a cargo ship ranges from 20 to 30 years, with some vessels exceeding 30 years [29], relying solely on the gradual replacement of the fleet to achieve carbon neutrality by 2050 is impractical. As methanol-ready engines are not yet widely available, retrofitting allows us to begin reducing emissions in the existing fleet without waiting for the complete transition to new technologies. This approach not only extends the life of current engines in a more sustainable way but also supports a gradual shift in fuel infrastructure, helping to build the necessary support systems for methanol use while addressing immediate environmental concerns in marine and heavy-duty transport [30].

2.2.2 Retrofitting Diesel Engine for Methanol Spark Ignition

Conceptual differences between CI and SI

Compression ignition and spark ignition operate on fundamentally different principles which influence their design, performance and suitability for specific applications. SI engines use an ignition spark from the spark plug to initiate combustion in a pre-mixed air-fuel mixture, where CI engines rely on the auto-ignition of the fuel due to elevated pressure in the cylinder.

CI engines are optimized for fuels with high energy density and low auto-ignition temperatures, such as diesel. Diesel's high cetane number ensures reliable ignition under the high pressures typical in CI operation. SI engines, on the other hand, require fuels with high octane ratings, such as gasoline or methanol to resist knocking (cf. Table 2.1).

The compression ratio, a critical parameter for engine performance, also varies significantly between CI and SI engines. CI engines typically operate with compression ratios in the range of 14:1 to 22:1. This contributes to their higher thermal efficiency. In SI engines, the compression ratio is generally lower, around 8:1 to 12:1, to prevent knocking and pre-ignition. However, the high knock resistance of methanol allows for higher compression ratios in retrofitted SI engines, which is also the case in this thesis with a compression ratio of 17.6:1.

Another major difference lies in the fuel injection system. CI engines employ high-pressure direct injection systems, often exceeding 2000 bar, to ensure proper atomization of fuel for efficient combustion. In contrast, SI engines commonly use port fuel injection (PFI) or direct injection (DI) systems at significantly lower pressures.

Injection systems

Retrofitting a CI engine to operate as an SI engine on methanol often involves replacing the diesel injection system with a methanol-compatible injection system and integrating spark ignition hardware.

Both PFI and DI are options in SI engines, visually represented in Figure 2.2. Each option has its advantages and drawbacks. PFI introduces fuel into the intake manifold where it mixes with air before entering the cylinder. This method is mechanically simpler, less costly and provides excellent mixing at low and medium loads. However, due to mixing outside the cylinder, particularly under high load conditions, PFI can lead to worse volumetric efficiency compared to DI. Furthermore, PFI systems are more prone to wall wetting and cold start issues due to methanol's high latent heat of vaporization and low energy density [31] [25].

DI, on the other hand, involves injecting fuel directly into the combustion chamber enabling greater control over injection timing and fuel stratification. This approach improves combustion efficiency and power output, particularly under high load conditions. The downside of DI systems lies in their increased cost, mechanical complexity and the challenges of optimizing injector design to handle methanol's lower lubricity and potential for corrosion [27] .

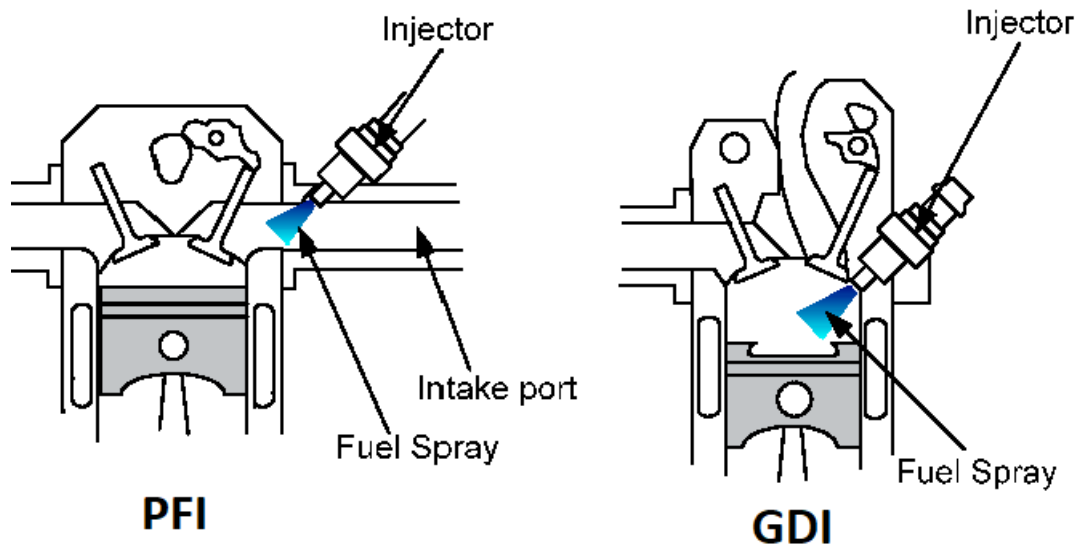


Figure 2.2: PFI versus GDI Injection Systems [32].

Chapter 3

Engine Optimization

This dissertation aims to evaluate the feasibility of methanol as a viable alternative to the fossil fuels currently powering the heavy-duty industry. For methanol to serve as an effective substitute, it must ideally meet or exceed the power output and performance benchmarks of modern diesel engines while achieving substantial emission reductions. This chapter will cover the specific targets and discuss the limits with modern internal combustion engines. Furthermore, it includes an overview of recent research in the field, laying the groundwork for the experimental measurements and findings detailed in Part II.

3.1 Performance parameters

To reach optimal engine performance for heavy-duty applications, three important parameters can be distinguished. Namely, the efficiency, the maximal load and the operational limits will be discussed in the following sections.

3.1.1 Efficiency

Efficiency is represented by Brake Thermal Efficiency (BTE). BTE is a measure of an engine's efficiency in converting the energy in the fuel into mechanical work at the output shaft. It is crucial for an engine to be able to operate at high efficiencies. This will reduce fuel consumption, extending the operating range of the engine. Higher efficiency will also result in less emissions, because less fuel is needed for the same power output. Modern diesel engines can reach BTEs up to 45% [33]. This superior efficiency compared to SI engines is achieved due to several key factors. These include higher compression ratios, which enhance thermal efficiency, direct fuel injection enabling lean combustion, the inherently high energy density of diesel fuel, no throttling losses and a more complete combustion [33]. In contrast, SI engines typically achieve efficiencies of up to 35% [33], falling short of the efficiency levels attainable by CI engines.

In 2.1.1, properties of methanol are mentioned that have the ability to enhance the efficiency of SI engines relative to gasoline. Methanol has a high Research Octane Number (RON) and a high Motor Octane Number (MON). These properties give a distinct view

of the knock-resistivity of a fuel. In a RON test, only the air is initially heated before introducing the fuel. In contrast, in a MON test, the air and fuel mixture is heated simultaneously. This means that in a RON test, methanol can cool down the mixture due to its high latent heat of vaporization, increasing knock-resistivity. This is why the MON number is typically a bit lower. Additionally, the fast flame speed of methanol leads to a faster flame propagation, leaving less time available for end-gas heating.

The combination of the charge cooling effect and the high flame speed extend the knock-limit of the engine. This implies higher compression ratios are possible, enhancing the thermal efficiency of the engine.

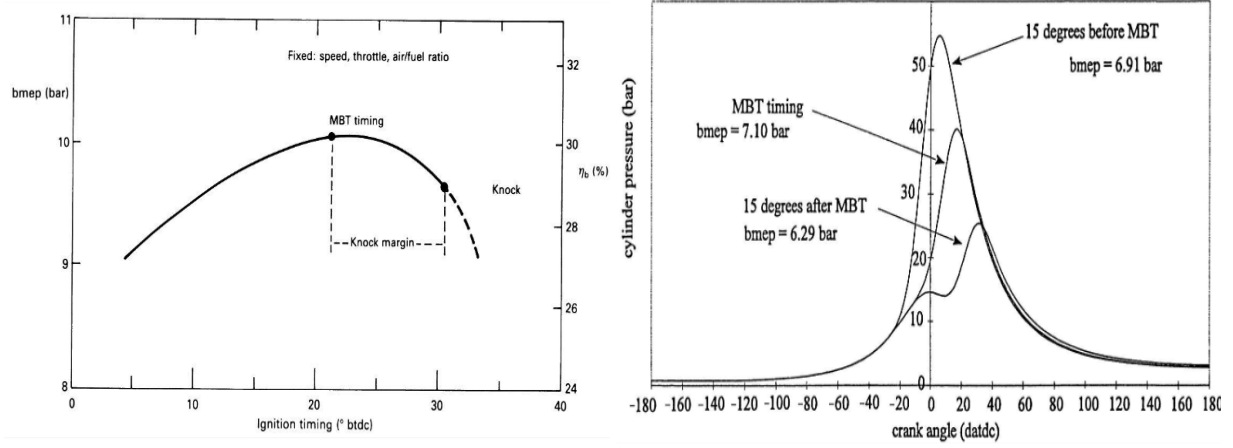
Furthermore, methanol can support leaner air-fuel mixtures than gasoline, meaning it can operate with more air relative to fuel [34]. This lean combustion improves thermal efficiency as it reduces the peak combustion temperature and energy losses, enhancing fuel economy and reducing emissions.

The potential for increased engine efficiency by replacing gasoline with methanol has been extensively validated experimentally. Li et al. [35] report indicated efficiencies of over 50% on a heavy-duty directly injected SI engine, with a bore of 130 *mm* using a compression ratio of 16:1 and stratification. These experiments were conducted at 1600 RPM at full load. An emphasis is placed on the direct injection of methanol with stratification. This way of injecting the fuel will enhance methanol's efficiency increase with respect to the port injection strategy used in our experimental research. Brusstar et al. conducted experiments on a 1.9 liter turbocharged diesel engine with a compression ratio of 19.5:1 that was converted for SI operation on methanol [36]. The high compression ratio enabled peak brake thermal efficiencies higher than the baseline diesel engine (40%) for operation on methanol (42%). A side-note here is the size of the engine cylinders compared to the 7.15 liter Volvo Penta, used in the experimental results of this dissertation.

3.1.2 Maximal load

In HD industry, operating under heavy loads is the norm. Therefore the maximal load should be extended as much as possible to withstand these heavy loads. CI engines reach maximum load at relatively low RPM's. For SI engines, this is not possible due to knock occurring for high loads at low RPM. Therefore, again the knock limit will be a constraint to reach maximal load, comparable to that of an CI engine [37].

The knock limit at high load can be extended by implementing spark retardation. This spark retardation is shifting the combustion away from top dead center. In this way, end-gas pressure and temperature are diminished leading to a prolonged ignition delay as can be seen on Figure 3.1b, lowering the chance of knock occurring [38]. On the other side, retarding the spark timing has an adverse effect on the efficiency. At each operating point, a Minimum Advance for Maximum Brake Torque (MBT) can be defined. It refers to the optimal ignition timing at which the engine produces its maximal power output without knocking, see Figure 3.1a.



(a) BMEP in function of ignition timing.

(b) Cylinder pressure in function of crank angle.

Figure 3.1: Minimum Advance for Maximum Brake Torque (MBT) with influence of ignition timing.

Deviating from this point will lower the brake mean effective pressure (BMEP) causing the BTE to deteriorate. If MBT cannot be reached before knock occurrence, the term Knock Limited Spark Advance (KLSA) is used to indicate the point of maximum efficiency limited by knock.

3.1.3 Operational limits

Every engine has a safe operating range in which normal engine behavior is assured. This operating range can be demarcated by a performance mapping. Zhu et al. [39] performed such a mapping in their research on a heavy-duty SI-engine on pure methanol. Figure 3.2 plots the brake mean effective pressure in function of the engine speed, indicating the brake thermal efficiency at each load point. The black lines show the load limit for low and high engine speed, marking the engines operating range.

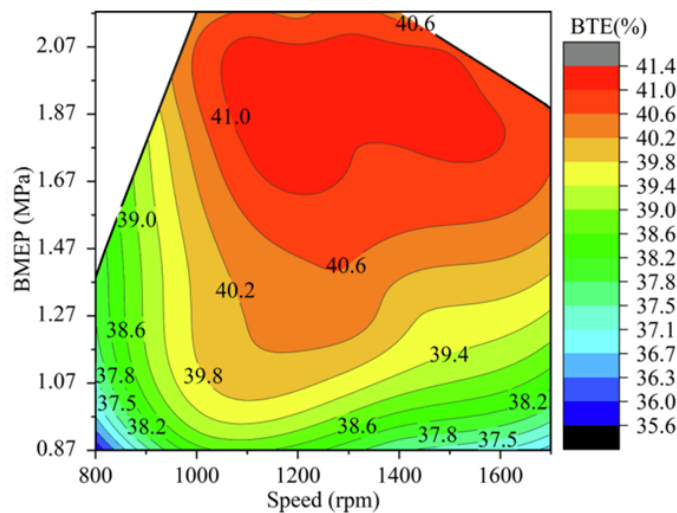


Figure 3.2: Performance map, showing BTE [39].

Demiddeleer and Vanderbeken [21] performed measurements, seeking the operating range of the Volvo Penta by varying the ignition timing for multiple loads and lambda values at 1500 RPM, see Figure 3.3. In this way, three recurring limits were established.

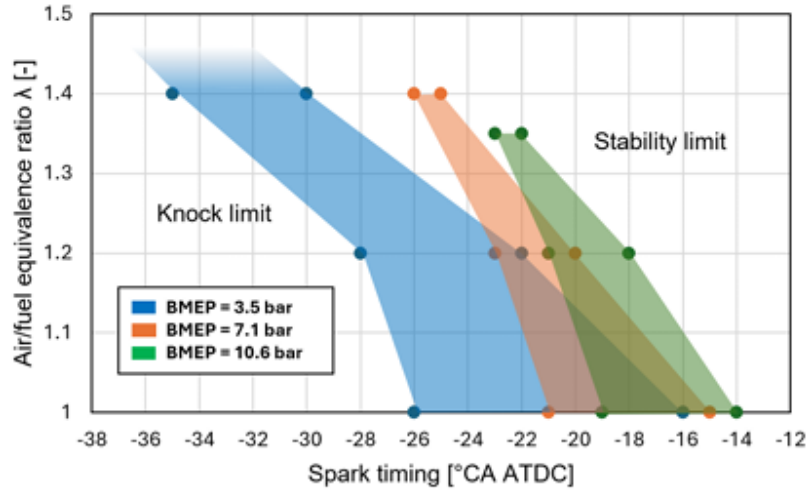


Figure 3.3: Stable operating range at different loads, with knock limiting the spark advancing and combustion stability limiting spark retarding [21].

Knock limit

As discussed in subsection 3.1.2, varying the spark timing has an influence on the achievable efficiency and can mitigate knock. To achieve optimal efficiency, the spark timing needs to be advanced until MBT is reached. Advancing the spark timing increases the probability of knock. In this way, at certain operating points, MBT can't be reached due to knock occurring. This is called Knock Limited Spark Advance (KLSA) and is the left limit on Figure 3.3. This limit can prevent the engine from running at optimal efficiency if MBT is not reached.

Stability limit

As discussed above, when increasing the load, knock is the limiting factor. Therefore, spark retarding is used as an anti-knock measure to be able to reach higher loads. However, not only the efficiency, but also the engine stability is affected by this measure. Engine stability becomes a problem when the Cycle-to-Cycle Variability (CCV) becomes too high. This is the consequence of inconsistent combustion in successive power cycles. To get a measure for the stability, a Coefficient Of Variance (COV) of the Indicated Mean Effective Pressure (IMEP) is calculated [40]. When the COV-value exceeds 3%-5%, the engine is running unstable. On Figure 3.3 the instability limit is at the right side and can be seen as the maximum possible spark retardation.

Lean limit

As mentioned, there is a third factor limiting the operating range of the engine. The lean limit is a special case of the two other mentioned above. When running at higher lambda values, the stability limit is affected to a higher extent than the knock limit, causing them

to converge with each other. The point where spark advance leads to knock and spark retarding leads to instability, is referred to as the lean limit. At this point, the engine can't run smoothly with more excess of air. On Figure 3.3, it can be seen that the lean limit decreases for higher loads.

3.2 Limiting factors

3.2.1 Knock

A significant challenge for SI engines is the risk of end-gas auto-ignition as power output increases, whether through higher demands on the same engine or by enlarging the engine size. This phenomenon, known as knock, should not be interchanged with pre-ignition.

Pre-ignition vs Normal knock

Pre-ignition occurs when the air-fuel mixture in one or more engine cylinders ignites spontaneously before the spark plug fires. This happens due to high temperatures reached during the compression process or the presence of hot spots, such as an overheated spark plug [41], presented in Figure 3.4a. In contrast, knock occurs after the piston has passed top dead center. It happens when the advancing flame front raises the temperature of the remaining unburned air-fuel mixture (the end-gas) to the point of auto-ignition before the flame can consume it [41]. This results in the sudden release of a substantial portion of the mixture's energy ahead of the main turbulent flame, as represented in Figure 3.4b.

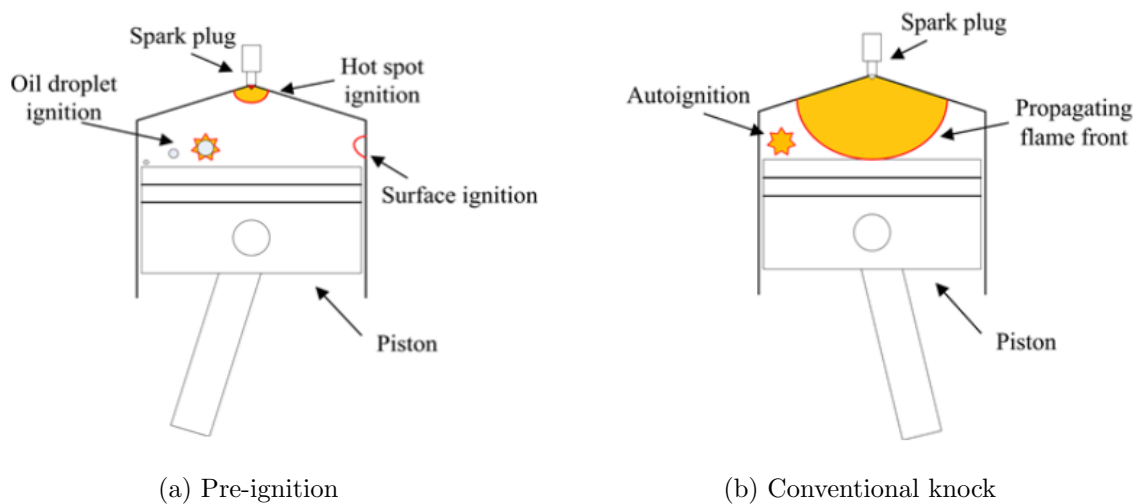


Figure 3.4: Graphical representation of pre-ignition and conventional knock [42].

This rapid energy release generates pressure waves that resonate within the cylinder, producing the distinct knocking or pinging sound and often leading to increased engine vibrations. Knock represents abnormal combustion, which destabilizes engine operation and can cause severe damage to both the engine and its after-treatment systems if sustained, see Figure 3.5.

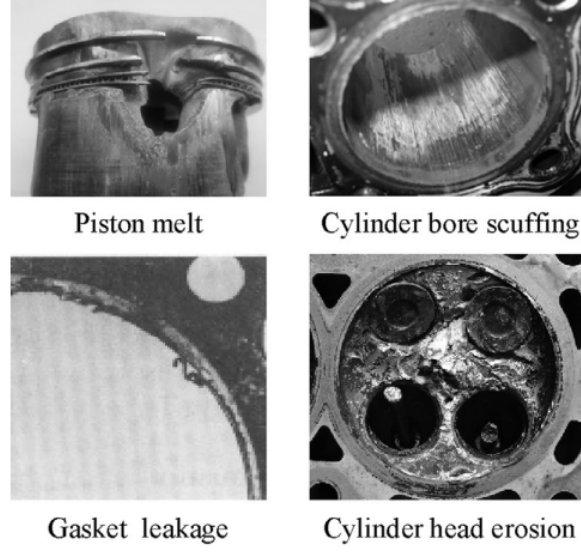


Figure 3.5: Engine damage by conventional knock [41].

Super-knock

Even though pre-ignition is distinctly different from conventional knock, it can trigger severe engine knock. This phenomena is called Super-knock [41]. This intense combustion produces a powerful pressure wave, with the reaction front propagating outward from the ignition source at supersonic speeds. This leads to pronounced pressure wave activity, causing extreme in-cylinder pressures, as illustrated in Figure 3.6, which can result in instant engine damage. Furthermore, this form of abnormal combustion appears randomly with little direct relationship to engine control parameters, such as ignition timing, equivalence ratio, intake temperature, etc [41].

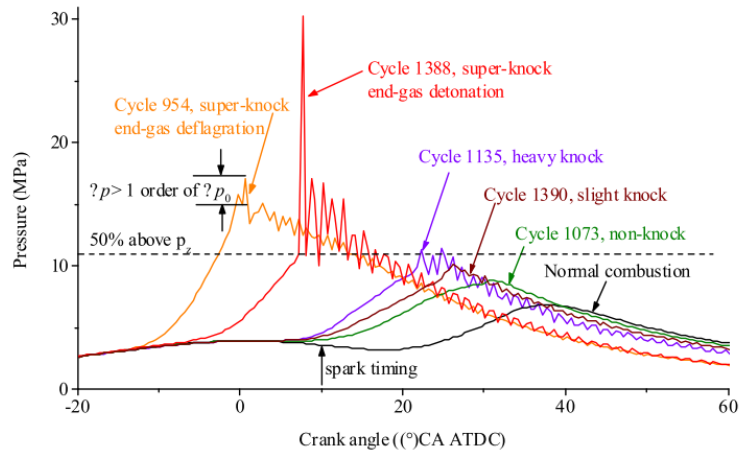


Figure 3.6: Cylinder pressure traces of typical pre-ignition and normal cycles [41].

Super-knock is a very specific type of knock that usually occurs at low engine speeds and high loads, typical for boosted engines [41]. In this dissertation, super-knock is expected to occur only if the engine experiences a drop in speed at full load. The relationship

between pre-ignition, conventional knock, and super-knock in a boosted gasoline engine is summarized in Figure 3.7. The figure shows that generally, SI combustion can be categorized into normal combustion and engine knock. In cases of normal combustion, the flame front progresses smoothly via turbulent flame propagation, resulting in stable operation without knock. However, under certain conditions - especially at high load - abnormal combustion events can occur. These are classified as engine knock, which can be further divided into conventional knock and super knock. Conventional knock typically arises from auto-ignition of the end gas ahead of the main flame front. The auto-ignition mode is then end-gas deflagration: a slower, subsonic combustion of the end-gas. Super knock, a more severe form can also be associated with end-gas detonation: a supersonic combustion mode with extremely high pressure rates. At the root of these abnormal combustion processes is pre-ignition. Depending on the conditions, pre-ignition can thus lead to end-gas detonation (super knock), end-gas deflagration (heavy or mild knock) or turbulent flame propagation (non-knock).

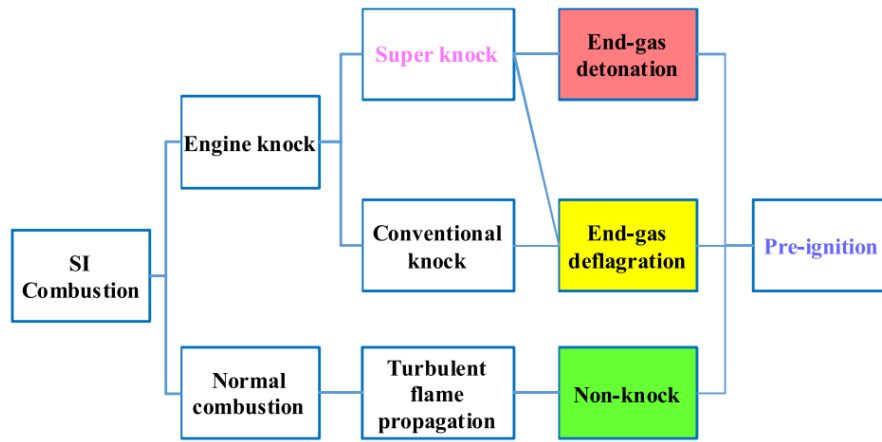


Figure 3.7: Knock overview [41].

Knock detection & quantification

For heavy-duty SI engines, knock poses a significant limitation in achieving high thermal efficiencies and engine loads. The large bore size of these engines results in a longer flame travel distance, providing more opportunity for knock to occur. Furthermore, in this study, the HD SI engine was retrofitted from an existing diesel engine with a high compression ratio (17.6:1). Although the higher compression ratio will lead to a theoretically more efficient cycle, there will be a trade off because of the elevated temperatures and pressures, increasing the risk of knock and imposing stricter constraints on efficiency.

Therefore, it is crucial to quantify knock in an accurate way. In-cylinder pressures can be used as a representative measure to detect knock [43]. A straightforward method for measuring knock intensity is the Maximum Amplitude of Pressure Oscillation (MAPO) method [44]. This metric simply looks at the maximum value of the high-frequency pressure oscillations during a specific time window after ignition. The high-frequency pressure oscillations are obtained by applying a band-pass filter into the in-cylinder pressure signal. The knock intensity is then given by Equation 3.1.

$$\text{MAPO} = \max \left(|p_{\text{filt}}(\theta)|_{\theta_0}^{\theta_0+w} \right) \quad [44] \quad (3.1)$$

where p_{filt} is the band-pass-filtered in-cylinder pressure, θ_0 is the crank angle corresponding to the start of the calculations, and w is the knock window length.

The resulting index is then compared to a predefined knock threshold. This threshold must be sufficiently high to exclude pressure oscillations from normal combustion, yet low enough to identify mild knocking cycles [43]. Typically, the threshold is determined through statistical analysis or experimental testing. However, because knock behavior varies with engine operating conditions, a fixed threshold may lack precision. To combat this, multiple approaches can be taken, Galloni et al. [45] proposed a method that dynamically calculates the knock threshold for example.

Alternatively, a knock sensor could be implemented that measures vibrations close to the cylinder. For this research, the use of a knock sensor could be a potential upgrade regarding precise knock measurement, although its calibration is quite complex. Until then, knock will be evaluated audibly and through pressure graphs, as shown in Figure 3.8.

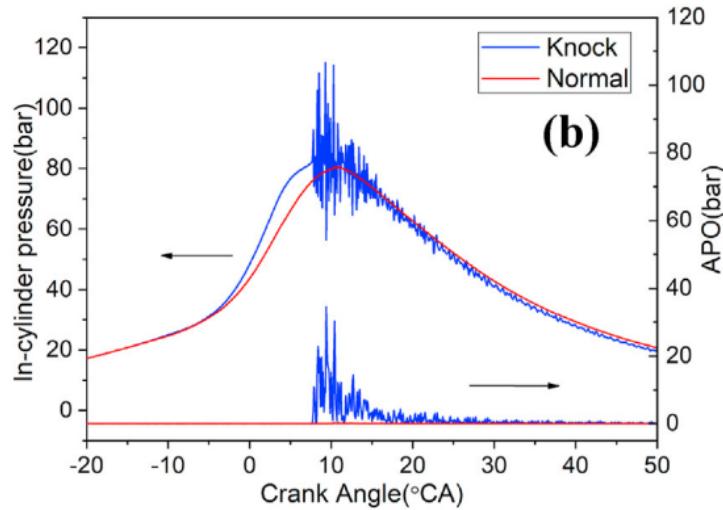


Figure 3.8: In-cylinder pressure and amplitude of pressure oscillations for normal and knock engine behavior [46].

An important side-note, the engine used in this research is equipped with only one in-cylinder pressure sensor. Consequently, the remaining five cylinders do not record in-cylinder pressures, making any pressure-based method unreliable, as knock can occur in any of the cylinders.

Knock limiters

In the following discussion, a short overview will be given of possible knock suppression strategies. Other enhancements, that will be the main focus in this dissertation, will be discussed later on.

In subsection 3.1.2, the effect of spark retarding on knock was already discussed in detail. An other way to suppress knock is by enriching the fuel mixture. Injecting excessive fuel could increase the effect of charge cooling and decrease the temperature of the mixture, which could prolong the time to end-gas auto-ignition and finally suppress knock. However, under the condition of fuel enrichment, HC and CO emissions increase, and since the Three-Way Catalyst (TWC) can only work under stoichiometric conditions, this causes problems of fuel economy and emissions [41].

Similarly, use of stratified mixture is a flexible approach to suppress knock and is usually achieved using direct injection. Stratification ensures a non-uniform air-fuel ratio within the cylinder. A richer mixture is formed close to the spark plug, for ignition purposes, while elsewhere in the cylinder the mixture is rather lean. This creates an overall lean mixture, with less fuel far away from the spark plug, leaving less opportunity for auto-ignition of the end gas, restricting knock. Additionally, fuel directly injected in the cylinder can lower the combustion temperatures due to the cooling associated with fuel vaporization. Therefore, the cooling can lower knock sensitivity and the maximum achievable load at a certain compression ratio can be increased [41].

Increasing turbulence intensity is also a viable option to suppress engine knock. Once combustion has started, increased turbulence leads to fast combustion and decreases the tendency to knock [47] [48]. However, at the start of combustion, increased turbulence also increases heat transfer from the spark electrode. To solve this problem, modern engines adopt high energy ignition (more than 100 *mJ*) to support strong turbulence. For example, through optimizing the geometry of the intake port as shown in Figure 3.9, the tumble ratio of a dedicated engine for the Toyota Prius hybrid increased from 0.8 in the previous 2ZR-FXE model to 2.8 in the ESTEC 2ZR-FXE. Introduction of high tumble-induced-high-turbulence is accompanied by low tendency to knock [41].

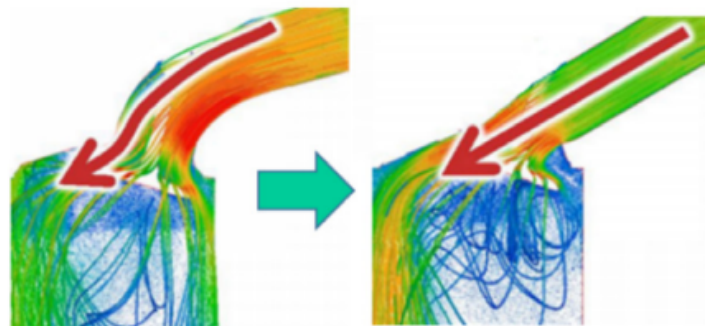


Figure 3.9: High tumble intake port [41].

3.2.2 Stability

In subsection 3.1.3, a brief overview was given on stability, limiting the operating range of the engine. The most common causes of combustion instability originate from partial burn and misfires [49]. A misfire in an engine occurs when one or more cylinders fail to complete the combustion cycle properly. This means that the air-fuel mixture in the cylinder does not ignite as expected, leading to a loss of power and efficiency. The engine's performance becomes erratic, and you may hear a popping or sputtering noise, indicating an incomplete combustion event.

Bahri et al. [50] define three types of misfires: Fuel cut-off (I), high dilution (II) and low temperatures (III). For type-I, misfires and partial burning cycles can be caused by fluctuations in the engines air-fuel ratio (AFR). Where partial burn cycles are defined as cycles where the heat release is reduced by 10% or more compared to a well-burning cycle [50]. With increasing lambda value (λ) the amount of heat released from the air-fuel mixture decreases and misfires type-II are generated. By reducing the intake temperature, the reactivity of air-fuel mixture significantly decreases. Misfires will occur once the intake temperature is not sufficient to support normal/partial combustion.

Randolph et al. [51] confirm that excessive Exhaust Gas Recirculation (EGR) has a disadvantageous effect on the engine stability. The Coefficient Of Variance (COV) of the IMEP was used as a measure for combustion stability. Dilution sweeps were performed to see the effect on this parameter, shown in Figure 3.10. Beyond a diluent mass fraction of approximately 20%, a noticeable increase in COV is observed, indicating the onset of combustion instability.

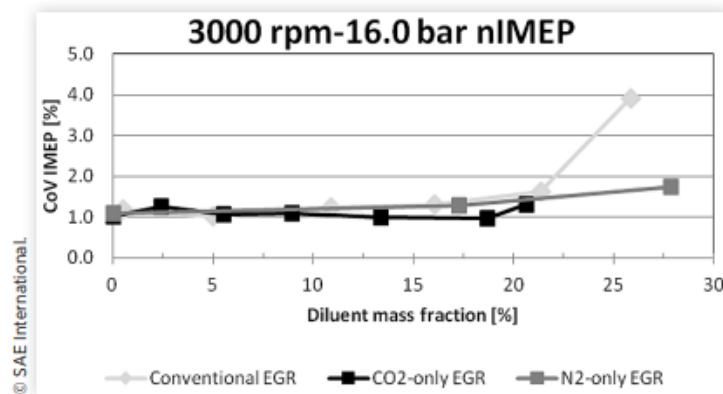


Figure 3.10: COV of IMEP for Conventional, CO₂-only and N₂-only EGR at 3000 RPM / 16.0 bar nIMEP [51].

As dilution increases, the temperature in the cylinder at end of compression is lower. This leads to slower reaction rates during the early phase of combustion when the flame is still laminar, thus a slower Laminar Burning Velocity (LBV), and requires spark to be advanced, see Figure 3.11. As dilution increases, the spark advance become so early that the laminar flame either extinguishes before transitioning to a fully developed turbulent flame (complete misfire), or it transitions too late to achieve nominal combustion (partial mis-fire). Either event leads to high COV of IMEP.

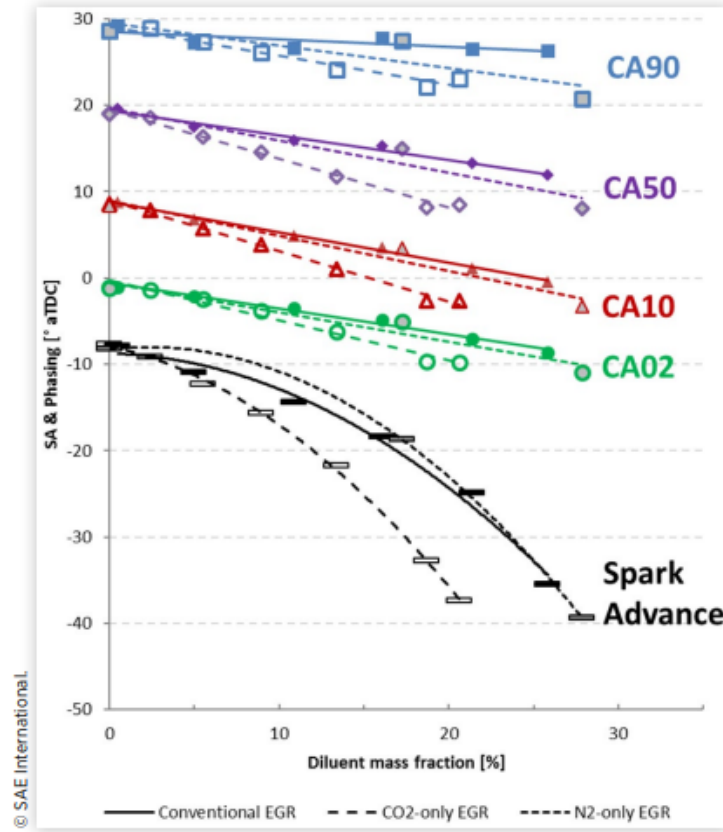


Figure 3.11: Spark Advance and Combustion Response with Conventional, CO₂-only, and N₂-only EGR at 3000 RPM / 16 bar nIMEP [51].

3.2.3 Cold start

In this dissertation, the retrofitted SI engine operates using methanol as its fuel. As highlighted in subsection 2.1.1, methanol possesses a high latent heat of vaporization, approximately four times greater than that of diesel and gasoline Table 2.1. Combined with the need for twice the amount of fuel due to methanol's energy density being half that of gasoline and diesel Table 2.1, this leads to a cooling power that is eight times greater compared to gasoline and diesel. This high cooling power, combined with methanol's low vapor pressure, can result in cold start challenges for the engine under low ambient temperature conditions [52]. At lower ambient temperatures, the intake port and combustion chamber temperatures decrease, necessitating a richer air-fuel mixture for a successful startup [53]. However, increasing the injected fuel mass leads to the formation of thicker fuel films on the intake port and valve surfaces, which slows down evaporation [53]. As a result, there is an optimal injected methanol mass required for a given engine to achieve efficient combustion. Once a cycle successfully fires, the backflow of hot residual gases into the intake port enhances fuel evaporation in subsequent cycles [53]. Therefore, cold start challenges are typically limited to the initial cycles after the engine starts.

Gong et al. [52] conducted cold start tests on a single-cylinder, four-stroke SI engine with a 52.4 mm bore and a 10.55:1 compression ratio, fueled by methanol. They found that at ambient temperatures below 16 °C, reliable engine starts were not possible without

auxiliary starting aids, even with significant methanol injection per cycle. Using the same engine, Li et al. [53] investigated various starting aids, including air preheating, methanol preheating, resistance heating of the intake manifold, glow plug preheating, and additional LPG injection into the intake manifold. While air and methanol preheating proved ineffective, resistance wire and glow plug preheating successfully extended the critical firing boundary, enabling starts at minimum ambient temperatures of 5 °C and 0 °C, respectively. Furthermore, LPG injection into the intake manifold significantly expanded the critical firing boundary, permitting cold starts at temperatures as low as -13 °C. The latter solution seems to be the most practical and effective in terms of extending cold start temperatures. Important to mention is that our predecessors [21] did not encounter cold start during any of their experiments. This could be explained by the high compression ratio and the fact that the engine was not used under very cold conditions. So, implementation of one of these solutions will probably not be a necessity.

3.2.4 Emissions

The primary aim of this dissertation is to establish a bridge between conventional diesel heavy-duty engines and purpose-built heavy-duty engines powered by renewable fuels, contributing to a greener heavy-duty fleet. This implies that emissions must be reduced for the transition to be meaningful. In addition, the European Commission has introduced the European Stage V standards [54], among the strictest emission regulations globally for Non-Road Mobile Machinery (NRMM). As a result, the transition to more carbon-neutral alternatives has become unavoidable. In the following section, a detailed analysis of the emissions produced by a SI engine running on methanol will be presented, highlighting its environmental impact and potential benefits compared to traditional fuels.

As was done in the introduction of this literature review section 1.1, a distinction can be made between green house gases and pollutant emissions.

GHG contribute to the greenhouse effect, where heat is trapped in the Earth's atmosphere, maintaining temperatures suitable for life but also driving climate change when concentrations are elevated.

Pollutants on the other hand, are substances or agents introduced into the environment that have harmful or toxic effects on ecosystems and human health.

Green House Gases

CO₂

Methanol (CH₃OH) is a carbon rich fuel, so it is inevitable to produce CO₂ during combustion of the fuel. Although CO₂ emissions occur during combustion, methanol can serve as a renewable fuel if its carbon cycle is closed. This involves capturing the CO₂ released from the tailpipe, either by biomass photosynthesis or by direct air capture, and utilizing it as a raw material for synthesizing renewable methanol. By recycling the CO₂ in this manner, no additional greenhouse gases are introduced into the atmosphere.

In addition to the fact that methanol's CO₂ emissions can be made circular through carbon recycling, the fuel itself possesses intrinsic properties that result in a lower net

CO₂ output compared to gasoline and diesel. Vancoillie et al. [26] found that methanol operation reduces the specific CO₂ emissions by more than 10% compared to gasoline. This is partly due to the lower CO₂ formation per unit of energy when burning methanol (68.4 g/MJ compared to 74 g/MJ for gasoline). Another reason is the higher brake thermal efficiency when running on methanol. Turner et al. [55] and Wang et al. [56] reached a similar conclusion, reporting an 8.8% reduction in CO₂ emissions when the engine operated on methanol compared to gasoline in the latter research.

Additionally, methanol’s reduced carbon-to-hydrogen ratio minimizes the potential for CO₂ formation [27]. The oxygen content in methanol further enhances combustion efficiency, reducing specific CO₂ emissions [27]. These combined factors position methanol as a cleaner alternative to gasoline and diesel, offering significant potential for reducing overall carbon emissions.

Pollutants

NO_x

Atmospheric air consists of approximately 78% nitrogen. During the combustion of the air-fuel mixture, nitrogen oxides (NO_x), consisting primarily of nitric oxide (NO) and nitrogen dioxide (NO₂), are produced. The amount of NO_x produced, depends on several parameters, with the most influential being: Combustion temperature and in-cylinder temperature, oxygen concentration, and residence time of the flue gases in the combustion chamber [57].

When compared to gasoline, methanol reduces the in-cylinder temperatures because of its lower adiabatic flame temperature, high burned gas heat capacity and evaporation cooling. Consequently, NO_x can be expected to be lower if the engine is operated at the same equivalence ratio [27]. Song Ho et al. [57] provide a detailed review of the mechanisms through which methanol influences NO_x emissions, as summarized in Table 3.1. The study distinguishes between two types of effects: direct and indirect. Direct effects are mechanisms associated with the elemental composition of the methanol molecule, such as the presence of oxygen, which tends to increase NO_x emissions. In contrast, indirect effects are mechanisms that affect the working conditions of the engine by physicochemical properties of methanol, such as its high heat of vaporization, which influences factors like in-cylinder temperatures. These indirect effects generally reduce NO_x emissions, although certain conditions, like a faster flame speed, can contribute to increased NO_x formation, as shown in Table 3.1.

The authors also note that their analysis includes studies involving both pure methanol and methanol-gasoline blends. Despite these variations, the findings offer valuable insights into understanding and evaluating methanol’s impact on engine performance and emissions.

Table 3.1: Overview of mechanisms of methanol on NO_x emissions [57].

Impact	Mechanism	Class
Methanol reduces NO_x emissions	Methanol-blended fuel generates more triatomic products, which decrease the peak combustion temperature, subsequently reducing NO_x emissions.	Indirect
	The higher heat of vaporization of methanol-blended fuel results in lower temperatures of the combustible mixture and subsequently lowers peak combustion temperatures in the cylinder, reducing NO_x emissions.	Indirect
	Methanol-blended fuel's lower heating value and faster flame speed lead to lower spark advance, thereby decreasing combustion temperature and reducing NO_x emissions.	Indirect
	The lower adiabatic flame temperature of methanol contributes to the reduction in NO_x emissions when used in methanol-blended fuel.	Indirect
Methanol increases NO_x emission	Oxygen content in methanol may supply additional oxygen for NO_x production.	Direct
	The in-cylinder air-fuel mixture may become slightly lean when using methanol-blended fuel, resulting in increased NO_x production.	Indirect
	The faster flame propagation speed associated with methanol-blended fuel results in the increase of combustion temperature, potentially resulting in increased NO_x production.	Indirect

Svensson et al. [58] conducted stochastic reactor simulations incorporating detailed chemical kinetics and a soot model to compare the emission formation pathways of methanol and diesel. They developed emission maps for various species as functions of equivalence ratio ($\phi=1/\lambda$) and temperature (T), examining typical trajectories in (ϕ ,T)-coordinates expected during engine operation. Figure 3.12 highlights the regions where NO_x (bottom) and soot (top) are produced, along with the adiabatic flame temperature.

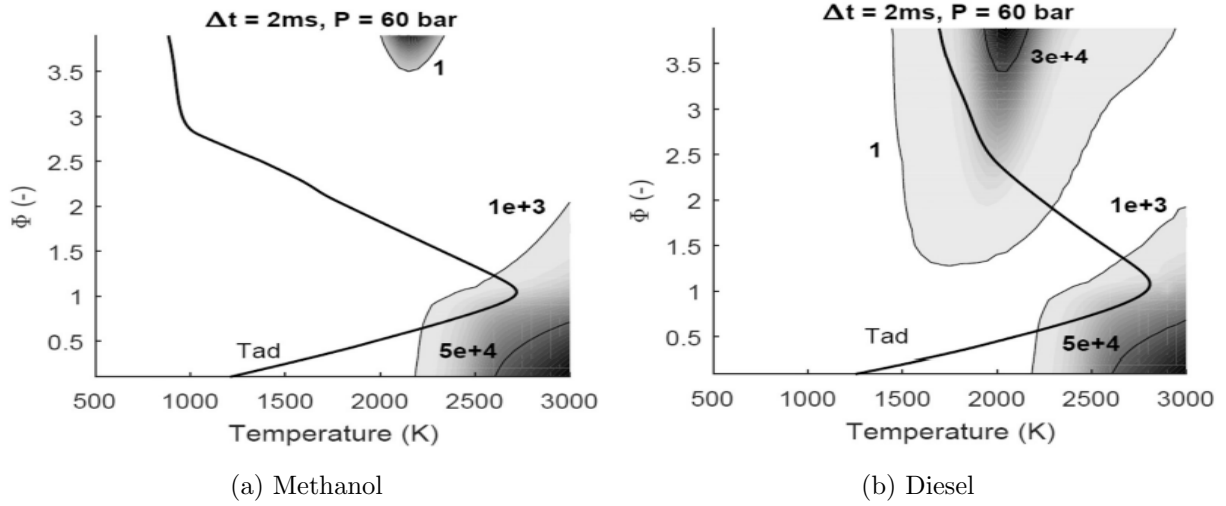


Figure 3.12: The formation of emissions of soot (in ppm by mass, in ‘island’ at top of plots, i.e. at the high equivalence ratios) and NO_x (in ppm by volume, bottom right of plots, i.e. at high temperatures and lower equivalence ratios) as a function of equivalence ratio and temperature, for methanol and diesel [58].

The results indicate that the NO_x formation regions are similar for methanol and diesel. However, due to the lower in-cylinder temperatures achieved with methanol injection, NO_x emissions are expected to be lower for methanol at the same equivalence ratio.

Zhu et al. [39] provide a mapping of the NO_x emissions for the entire operating range. Figure 3.13 shows that NO_x emissions are highest at lower engine speeds or higher engine loads owing to the lower EGR rate.

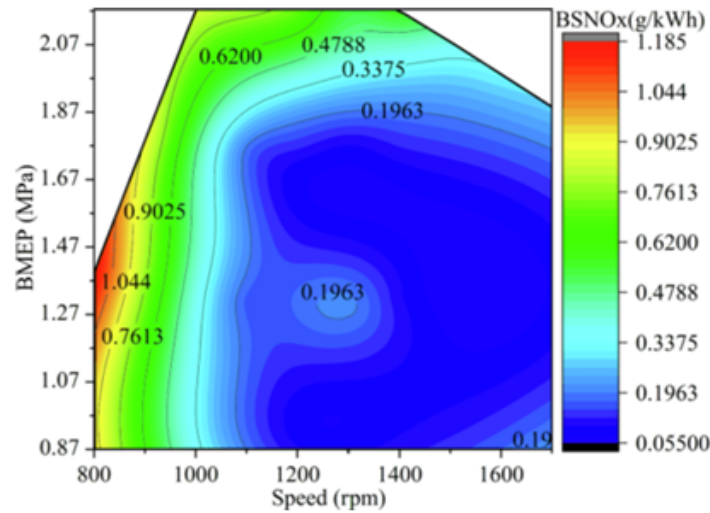


Figure 3.13: Performance map for the heavy-duty methanol SI engine showing the NO_x emissions [39].

CO

Carbon monoxide (CO) is a byproduct of incomplete fuel combustion in the engine combustion chamber and represents the loss of chemical energy that is not fully utilized in the engine. The amount of CO produced depend heavily on combustion conditions like: air-fuel ratio, combustion temperature and interruption of combustion cycle time [57].

Compared to gasoline, methanol can ensure a more complete combustion of the fuel. This is due to methanol's elemental composition characteristics (e.g., lower carbon content, presence of oxygen, lower C/H ratio, etc.), structural characteristics (e.g., non-presence of C-C bond), and physicochemical characteristics (e.g., lower boiling point, etc.). Therefore, it is considered that a decrease in CO emission is natural when methanol is used as an alternative to gasoline [57].

In Table 3.2, a summary of the mechanisms of methanol on the increase and decrease of CO emissions is given, compared to gasoline.

Table 3.2: Overview of mechanisms of methanol on CO emissions [57].

Impact	Mechanism	Class
Methanol reduces CO emissions	Methanol has a lower carbon content than gasoline. The carbon in the fuel is directly converted into CO during the combustion process, so the use of methanol-blended fuel reduces the formation and emission of CO.	Direct
	Methanol has a lower C/H ratio than gasoline.	Direct
	The oxygen enrichment from methanol leads to a “pre-mixed oxygen effect” that promotes complete combustion.	Direct
	The lower stoichiometric AFR of methanol leads to the leaning effect of the methanol-blended fuel, and this promotes complete combustion.	Direct
	Methanol has no C-C bond in its structure, which could help complete the combustion of the methanol-blended fuel.	Direct
	The lower boiling point of methanol makes the methanol-blended fuel completely vaporize, allowing for complete combustion.	Indirect
	The higher heat of vaporization of methanol-blended fuel leads to lower intake manifold temperatures, and more air access occurs during fuel combustion.	Indirect
Methanol increases CO emission	Methanol-blended fuel produces more triatomic products, which lowers the combustion temperature and slows down CO oxidation.	Indirect
	The shorter combustion process of methanol-blended fuel might result in insufficient oxygenation of CO.	Indirect

Methanol also possesses properties that could potentially increase CO production. Especially at low loads, it can be found that the higher latent heat of evaporation of methanol limits the oxidation of CO, increasing CO content. Zhu et al. [39] performed emission measurements on a SI engine running on methanol and confirmed this observation, as can be seen in Figure 3.14.

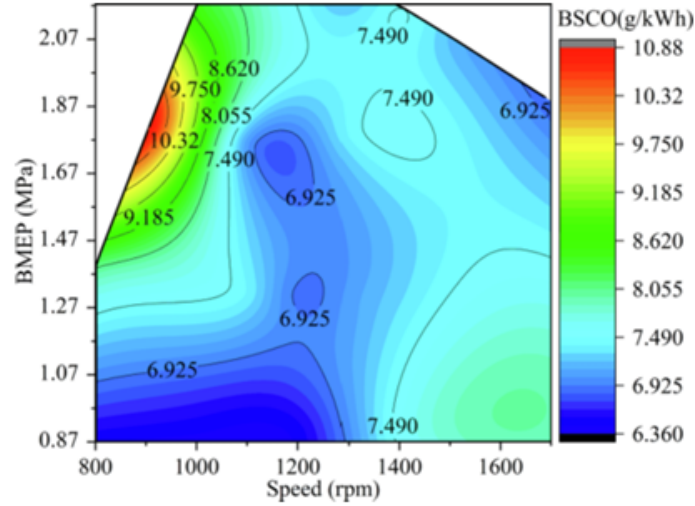


Figure 3.14: Performance map for the heavy-duty methanol SI engine showing the CO emissions [39].

SO_x & Soot

Methanol inherently has no soot formation due to the absence of carbon-carbon bonds and aromatic component in its molecular structure [57]. Furthermore, the carbon atom in methanol is bonded to an oxygen atom, providing chemically premixed oxygen within the fuel. Consequently, the "soot island," illustrated in Figure 3.12, is almost non-existent. For this study, which focuses on the combustion of pure methanol, soot emissions are expected to be too low to measure with the available equipment. This property offers a distinct advantage in reducing NO_x emissions, as it eliminates the typical NO_x-soot trade-off commonly encountered in diesel engines.

In contrast with diesel, sulfur oxides (SO_x) are not present in methanol combustion due to the lack of sulfur content in the fuel.

Unburned Hydrocarbons

Hydrocarbon (HC) emission is an unburned or incomplete combustion product of fuel in the combustion chamber. Like CO, it represents the loss of chemical energy that is not fully utilized in the engine [57]. HC is emitted in such conditions as misfires, regulated valve leakage, liquid fuel effects (especially during cold start and warm-up), fuel or fuel/air mixture protected from the combustion process in crevices, oil films and deposits, lack of complete air supply, low cylinder compression, lean mixture and lower in-cylinder temperature [57].

An overview of the effect of methanol on HC emissions is given in Table 3.3.

Table 3.3: Overview of mechanisms of methanol on HC emissions [57].

Impact	Mechanism	Class
Methanol reduces HC emissions	Methanol has no heavy HC.	Direct
	The oxygen in methanol provides sufficient oxygen, and this improves combustion efficiency.	Direct
	Methanol has a lower carbon content than gasoline, which makes the combustion reaction more complete.	Direct
	Methanol has a lower C/H ratio than gasoline	Direct
	The lower stoichiometric AFR of methanol leads to the leaning effect of the methanol-blended fuel, and this promotes complete combustion.	Direct
	The higher heat of vaporization of methanol-blended fuel leads to lower intake manifold temperatures, and more air access occurs during fuel combustion.	Indirect
	The lower boiling point of methanol makes the methanol-blended fuel completely vaporize, allowing for complete combustion.	Indirect
Methanol increases HC emission	The lower AFR of methanol leads to more fuel injection, which results in more fuel entering the crevice volumes or being absorbed in the oil layers.	Direct
	The combustion process of methanol-blended fuel is advanced, decreasing the exhaust temperature, which weakens the oxidation of HC.	Indirect
	The higher heat of vaporization of methanol-blended fuel causes lower in-cylinder temperature, which may lead to misfire and partial combustion.	Indirect

Additionally, Güdden et al. [59] found that HC emissions increased heavily with λ . As shown in Figure 3.15, the HC emissions were higher at lower engine speeds and loads because of the lower post-oxidation rate caused by the lower exhaust temperature.

Formaldehyde

Next to all the other well-known emission species, methanol combustion also produces the toxic compound formaldehyde which is a human carcinogen. Formaldehyde (CH_2O) emission is the post-oxidation production of unburned methanol. The formation occurs due to the partial oxidation of methanol in the combustion process, especially under specific conditions such as low combustion temperatures or incomplete oxidation.

Zhang et al. [60] conducted several experiments to identify the effects on formaldehyde formation. Injection timing, load, speed and EGR rate were varied during this study. It

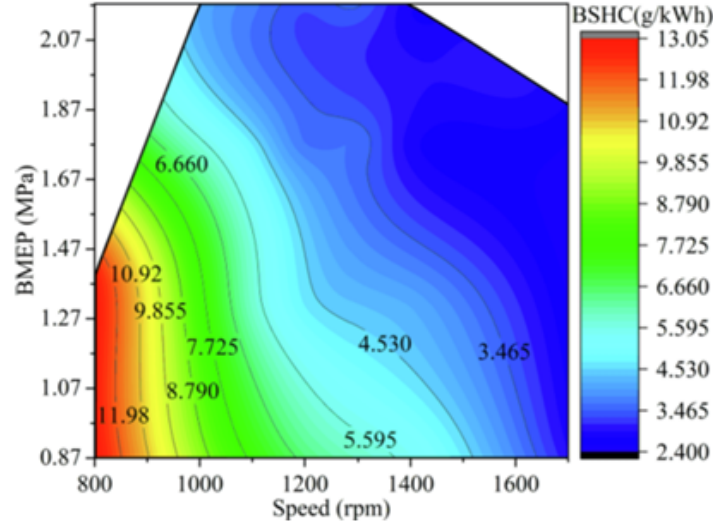


Figure 3.15: Performance map for the heavy-duty methanol SI engine showing the HC emissions [39].

was found that a retardation of the injection timing decreased formaldehyde content in the exhaust gases, while an increase is observed with the retardation of ignition timing. The temperature of unburned methanol post-oxidation is affected by engine speed, load, and EGR ratio, affecting formaldehyde emissions. With increasing exhaust temperature, unburned methanol emission decreases while formaldehyde emission first increases and then decreases.

When looking at Figure 3.13, Figure 3.14 and Figure 3.15, it is expected that emissions will be highest when operating at lower loads and low engine speeds. This will be examined in the experimental research of this dissertation, focusing on CO and NO_x emissions.

3.3 Performance enhancing techniques

In the following section, three possible performance enhancing techniques will be explored: Exhaust Gas Recirculation, Lean Combustion and Passive Prechamber Technology. Their working principle, performance parameters and limiting factors will be discussed. These techniques will also be implemented and evaluated in the experimental research conducted in this dissertation.

3.3.1 EGR

Exhaust Gas Recirculation (EGR) is a widely applied technique in internal combustion engines to improve efficiency, suppress knock, and reduce NO_x emissions [61]. By reintroducing a portion of the exhaust gases back into the intake system, see Figure 3.16, EGR lowers the combustion temperature within the cylinder, which in turn has multiple benefits: knock suppression, enhancing thermal efficiency and reducing NO_x emissions for methanol-fueled SI engines, especially in high-load marine applications [62]. The EGR rate (EGR%) is a quantity that expresses the amount of exhaust gases recirculated back

into the engine inlet, which can be defined in multiple ways [62]. Typically, it is defined as the ratio of the mass of the recirculated exhaust gas to the total mass of the intake mixture, including exhaust gases, air and fuel as shown in Equation 3.2 [63]. In Equation 3.3, the recirculated exhaust gas is only compared to the combined mass flow rate of fuel and air. Additionally, the EGR% can also be defined using the volumetric ratio of different intake gases, though this method is less representative due to differences in temperature and density between the recycled exhaust gases and intake air, see Equation 3.4 [63].

$$\text{EGR}(\%) = \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{EGR}} + \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}} \cdot 100 \quad [63] \quad (3.2)$$

$$\text{EGR}(\%) = \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}} \cdot 100 \quad [63] \quad (3.3)$$

$$\text{EGR}(\%) = \frac{V_{\text{EGR}}}{V_{\text{air}} + V_{\text{fuel}} + V_{\text{EGR}}} \cdot 100 \quad [63] \quad (3.4)$$

However, due to the hard environment created by the exhaust gases, the EGR mass rate is never directly measured [64]. An approach to counteract this is based on focusing on the concentration of CO_2 in the intake and exhaust gases, providing a measure of the EGR rate. This approach provides a direct calculation of EGR percentage using Equation 3.5. Similarly, this is also possible using the O_2 concentration as written down in Equation 3.6

$$\text{EGR}(\%) = \frac{[\text{CO}_{2,\text{amb}}] - [\text{CO}_{2,\text{mix}}]}{[\text{CO}_{2,\text{amb}}] - [\text{CO}_{2,\text{EGR}}]} \cdot 100 \quad [65] \quad (3.5)$$

$$\text{EGR}(\%) = \frac{[\text{O}_{2,\text{amb}}] - [\text{O}_{2,\text{mix}}]}{[\text{O}_{2,\text{amb}}] - [\text{O}_{2,\text{EGR}}]} \cdot 100 \quad [63] \quad (3.6)$$

EGR systems are typically classified based on two key distinctions: hot versus cold EGR and high-pressure (HP) versus low-pressure (LP) EGR. Hot EGR refers to exhaust gases being recirculated directly back into the intake without cooling, while cold EGR incorporates a cooler to reduce the temperature of the recirculated gases [65]. Cooled EGR increases intake density and thereby also increases the volumetric efficiency. The decreased temperature also reduces NO_x emissions. On the other hand, the HC-emissions and cycle-to-cycle variations are increased compared to that of hot EGR [65]. Hot EGR uses the high temperature exhaust to heat the intake, promote combustion and improve the thermal efficiency. These characteristics of both types of EGR are given in Table 3.4 [65].

Another distinction in EGR is based on the pressure loop. HP EGR systems extract exhaust gases upstream of the turbine or close to the exhaust manifold and reintroduce them into the intake manifold downstream of the compressor, see Figure 3.17 [65]. This configuration allows for rapid response due to the shorter flow path, making it effective during transient conditions [67].

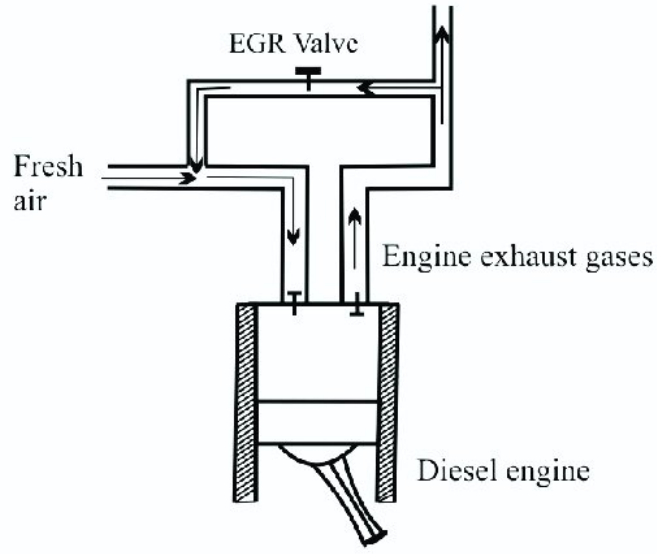


Figure 3.16: Schematic showing the principle of EGR in an ICE [66].

Cooled EGR	Hot EGR
• Lower NO_x values • Better knock suppression • Complex structure	• Lower combustion duration • Lower HC values • Simple structure

Table 3.4: A comparison of the characteristics between cooled EGR and hot EGR [65].

LP EGR systems, on the other hand, extract exhaust gas downstream of the turbocharger turbine and reintroduce it upstream of the compressor, as can be seen in Figure 3.17. This setup enables more homogeneous mixing of exhaust gas with fresh air via the compressor and improved EGR cooling. Despite the slower response compared to HP EGR, it is effective at higher engine loads and speeds because of the longer flowpath of the recirculated exhaust gases and thus the better cooling [67].

LP EGR is further divided into clean and dirty EGR, as shown in Figure 3.18 [65]. Despite some sticky hydrocarbons in dirty EGR, there are also some benefits such as improved fuel economy, reduced pumping losses and improved EGR cooling [65] [68].

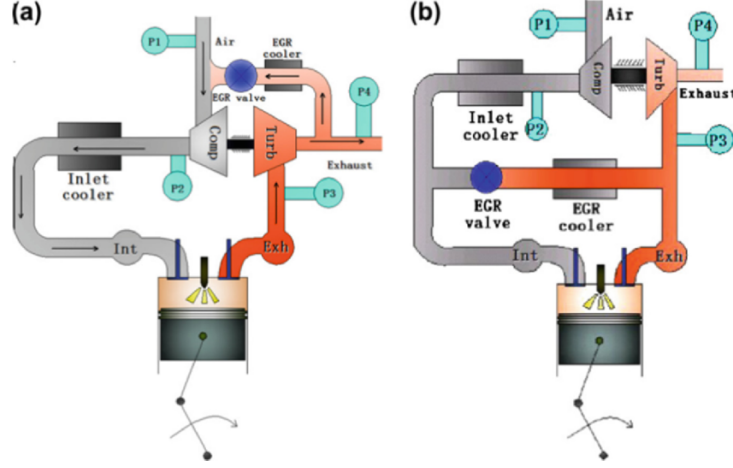


Figure 3.17: (a) LP versus (b) HP EGR [65].

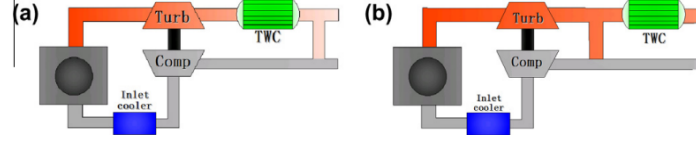


Figure 3.18: (a) clean versus (b) dirty EGR [65].

Knock Mitigation

Knock, as explained in subsection 3.2.1, is a significant limitation in methanol SI engines, particularly at high loads. EGR mitigates this issue by introducing recirculated exhaust gases into the intake charge, which absorb heat during combustion and lower the peak temperatures inside the cylinder. This reduces the likelihood of auto-ignition of the end gas (the unburned portion of the air-fuel mixture ahead of the flame front). Research indicates that with optimal EGR rates, engines can operate with more advanced spark timing, which improves efficiency without the risk of knock [69] [61]. Zhu et al. [39] investigated the impact of EGR on the performance of a pure methanol-fueled heavy-duty SI engine. Figure 3.19 shows the BTE as a function of the EGR rate for different engine speeds. The solid circles indicate that the MBT is equal to the KLSA at the EGR rate [39]. It can be seen from this figure that when increasing the EGR rate above 20%, knock is no longer the factor limiting the thermal efficiency. Figure 3.20 also shows the resulting knock mitigation. It can be seen that the combustion phase was advanced with increasing EGR rate because of this knock mitigation. At a BMEP of 0.87 MPa, knock was no longer a factor hindering the improvement of the BTE when the EGR rate was bigger than 25%. However, the EGR rate of 25% was insufficient to mitigate knock at BMEPs of 1.31 and 1.74 MPa [39]. EGR tolerance is inherently dependent on both engine-specific factors and engine load. At lower load, the engine operates at cooler temperatures and lower pressures, resulting in reduced knock tendency and allowing for more extensive dilution without compromising combustion stability. In contrast, at higher BMEP, while EGR continues to suppress knock, the combustion process becomes more sensitive to dilution effects such as slower flame propagation and increased cycle-to-cycle variability.

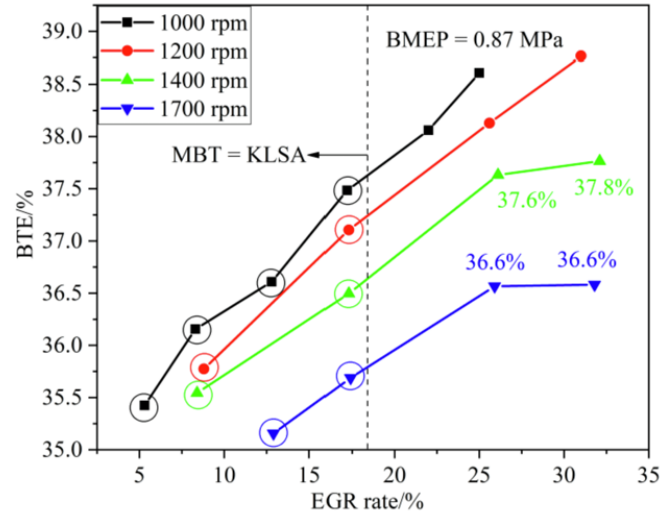


Figure 3.19: Variations in the BTE with respect to the EGR rate at different speeds [39].

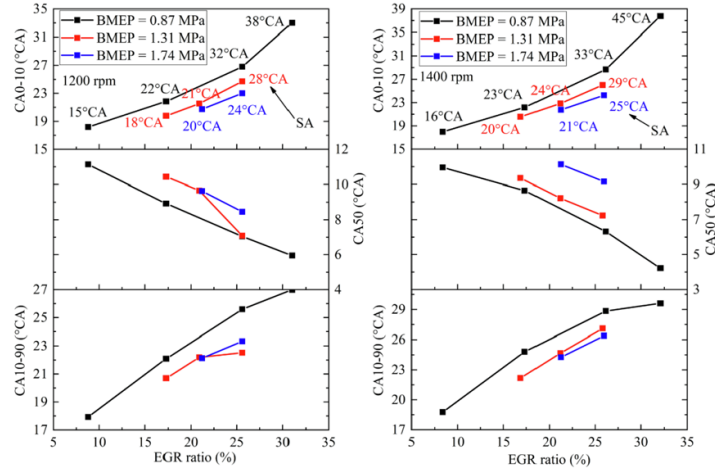


Figure 3.20: Effects of the EGR rate on CA0-10, CA50 and CA10-90 under different loads [39].

Efficiency

The lower combustion temperatures facilitated by EGR not only suppress knock but also allow the engine to achieve higher thermal efficiency. Studies, including those on methanol SI engines, have shown that the cooling effect of EGR enables further spark advance, leading to more complete combustion near top dead center (TDC) and improved BTE [4]. Figure 3.21 shows the BTE as a function of the EGR rate at BMEPs in the range of 0.87-1.74 MPa. As can be seen, the BTE increases with increasing EGR rate by affecting both the combustion process and the gas exchange process [39].

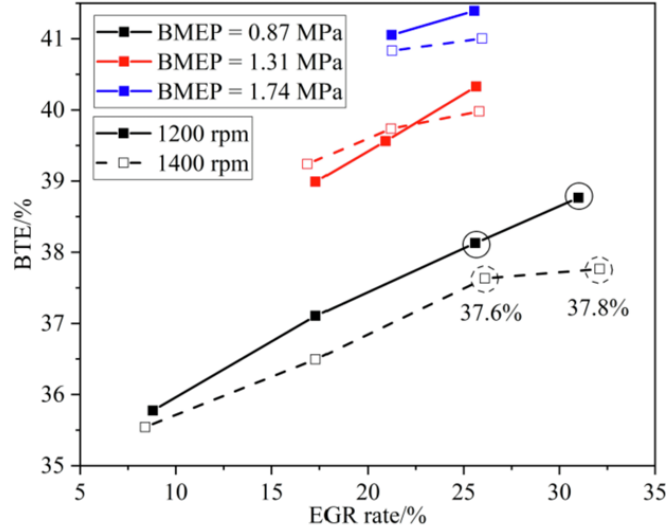


Figure 3.21: Effects of the EGR rate on the BTE under different loads [39].

NO_x Emissions Reduction

As NO_x formation is highly temperature-dependent, lower combustion temperatures have a direct impact on NO_x emissions. The reintroduced exhaust gases lower the available oxygen and reduce combustion temperature, making EGR an effective strategy for reducing NO_x, especially in applications targeting stringent emissions regulations [25]. Li et al. [70] investigated the effect of EGR on NO_x emissions for a natural gas engine and found that these emissions effectively decrease with increasing EGR rate, as can be seen in Figure 3.22 [70]. This figure depicts the NO_x emissions as a function of the EGR rate. A downward trend is observed.

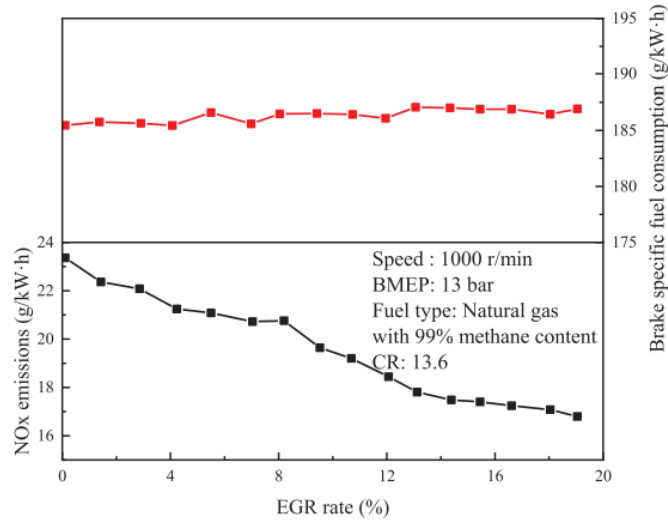


Figure 3.22: NO_x emissions and bsfc versus EGR rate [70].

Challenges and Considerations

While EGR provides clear advantages, its application in methanol SI engines involves a careful balance to avoid adverse effects. Excessive EGR rates can lead to poor combustion stability, resulting in incomplete combustion and increased hydrocarbon emissions [39]. Integrating EGR systems also adds complexity to engine control, as the balance of recirculated gases must be carefully managed to prevent efficiency loss. Optimizing EGR for knock suppression while maintaining stable and efficient combustion necessitates precise calibration, which can be challenging, especially in large marine engines.

3.3.2 Lean combustion

Lean combustion is a widely adopted strategy to increase the BTE. By increasing the air-to-fuel equivalence ratio beyond the stoichiometric value ($\lambda = 1$), lean combustion promotes complete combustion and reduces emissions [59].

Thermal efficiency

The experiments conducted by Demiddeleer and Vanderbeken [21] on a retrofitted methanol SI marine engine (CR = 17.6, Bore = 108 mm) demonstrated significant efficiency improvements under lean conditions. Figure 3.23 denotes the BTE as a function of varying lambda values for different loads at 1500 RPM operation. It can be seen from this figure that BTE increases with lean operation compared to stoichiometric operation, e.g. at a BMEP of 10.6 bar and a $\lambda = 1.4$, the BTE reached 39.3% compared to 38.5% at stoichiometric operation.

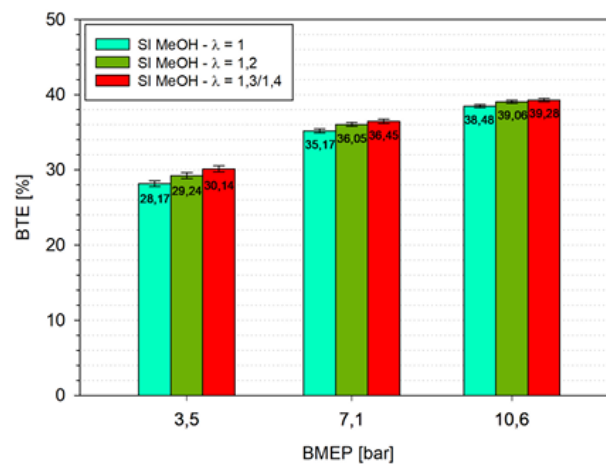


Figure 3.23: BTE at optimal spark timing for different loads and varying lambda values at 1500 RPM [21].

This enhanced efficiency is attributed to several factors, encompassing reduced pumping losses (cf. Figure 3.24) due to wider throttle valve openings and improved combustion completeness at leaner operation[21]. Secondly, the excess of air promotes more complete combustion and such reduces the amount of unburned methanol in the exhaust which contributes to more efficient combustion.

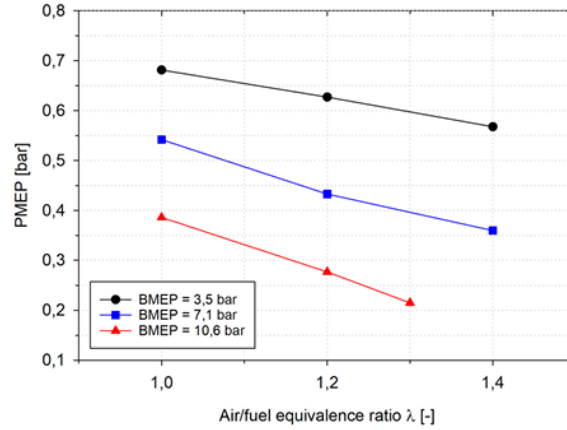


Figure 3.24: Pumping mean effective pressure (PMEP) as a function of the air/fuel equivalence ratio for different loads at 1500 RPM [21].

Methanol's high latent heat of vaporization also contributes to charge cooling, allowing for advanced spark timing and further improving thermal efficiency [59] [21]. This advanced spark timing leads to a combustion cycle that more closely aligns with the ideal Otto cycle, further boosting efficiency.

Emissions

NO_x emissions are also significantly reduced by lean combustion by the lower peak combustion temperature, a phenomenon observed in the studies by Güdden et al. [59]. However, unburned methanol and formaldehyde emissions increased at higher λ values, indicating the need for an oxidation catalyst to mitigate these emissions [59].

Wu et al. [71] investigated the lean burn characteristics of gasoline and methanol under idle conditions (at an engine speed of 800 RPM). They found the emissions from Figure 3.25 where a significant decrease in NO_x emissions was found. These graphs depict different emissions as a function of the air/fuel ratio for methanol compared to gasoline under idle conditions. Figure 3.25c shows decreasing NO_x emissions with increasing λ .

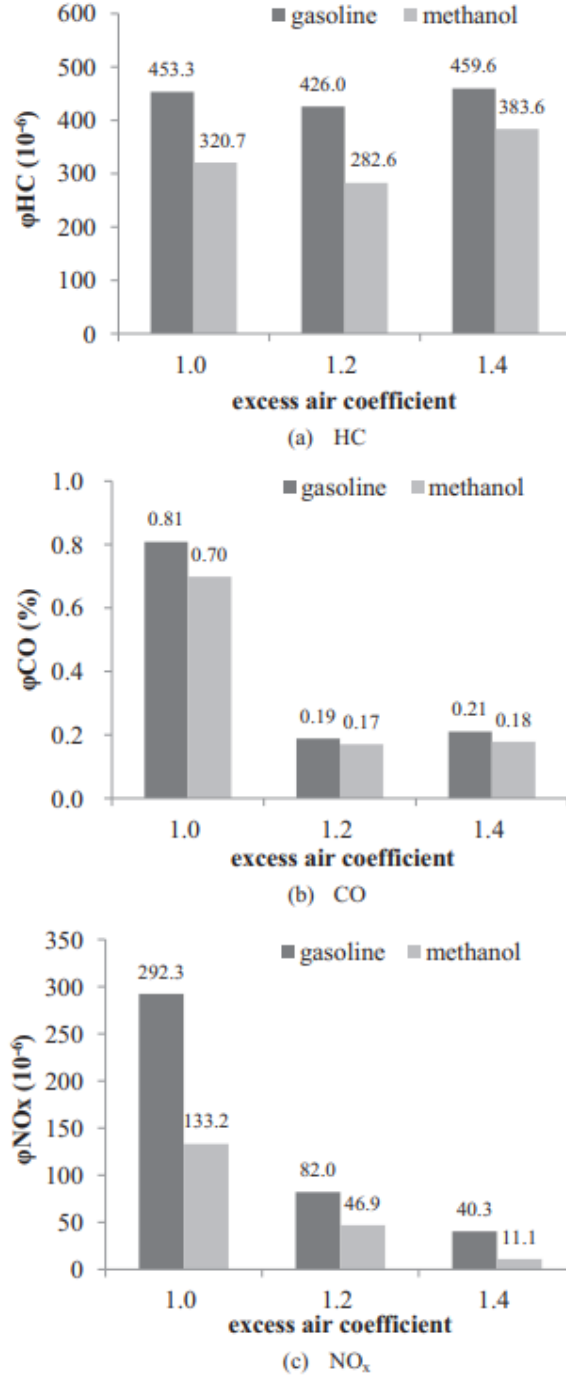


Figure 3.25: Emissions of HC, CO and NO_x of engine fueled with gasoline and methanol at different λ under idle conditions [71].

Combustion stability

The slower combustion process associated with leaner mixtures leads to higher cycle-to-cycle variations in the combustion process. Since instable combustion increases fuel consumption and pollutant emissions, this is an important factor to consider [71]. Figure 3.26 shows the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}) as a function of the air/fuel equivalence ratio. Out of this graph, it can be concluded that the

cycle-to-cycle variations increase with increasing air/fuel equivalence ratio. This effect is explained by the slower combustion at higher λ values [71].

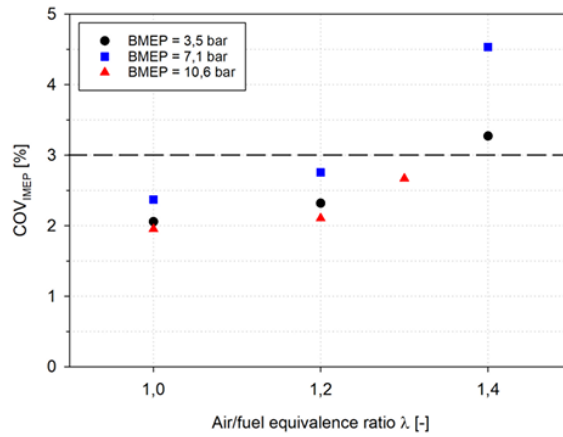


Figure 3.26: Coefficient of variation of indicated mean effective pressure (COV_{IMEP}) as a function of λ at different loads with constant engine speed of 1500 RPM [21].

Challenges and future work

While lean combustion offers numerous advantages: enhanced thermal efficiency, NO_x reduction and exhaust gas temperature reduction, it is not without challenges [59]. The narrowing of stable operating ranges at higher air/fuel equivalence ratio coupled with increased HC emissions, necessitate advanced combustion control strategies. Next section gives an example of such an advanced combustion control strategy.

3.3.3 Passive Prechamber Technology

Prechamber technology has emerged as an effective method to enhance combustion in large-bore high-compression natural gas engines, such as those used in marine applications [72] [73] [74]. This technology enhances thermal efficiency and emissions performance by addressing the limitations of traditional SI engines. Methanol's high flame speed (subsection 2.1.1) and lean burn compatibility make it well-suited for integration with prechamber systems, ensuring robust performance under various operating conditions [72].

Mechanism

A prechamber is a small, auxiliary chamber connected to the main combustion chamber through small precisely calibrated orifices (nozzles). During the compression stroke, the air-fuel mixture from the main chamber (MC) is forced into the prechamber (PC), where it is ignited by a spark plug. This ignition causes combustion within the PC, generating a rapid pressure rise. The resulting pressure forces the burning mixture through small orifices into the MC. As the flames pass through these narrow openings, they are quenched, forming highly turbulent and reactive jets instead of a continuous flame front. These jets, consisting of partially combusted reactants and high-energy chemical species, mix with the main air-fuel charge and serve as distributed ignition sources. This process

is represented in Figure 3.27. This results in a substantial increase in the combustion velocity, establishing a more concentrated heat release closer to the optimum, leading to highly efficient combustion of the MC fuel while at the same time suppressing knock [75][76].

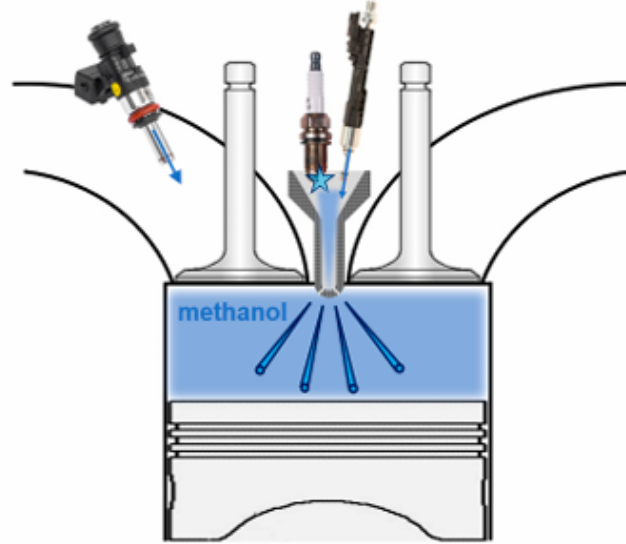


Figure 3.27: Schematic of the jet ignition concept with an actively fueled prechamber. The star represents the spark and the arrows highlight the locations where fuel injections [77].

When examining prechambers used in premixed combustion, two distinct layouts can typically be identified. The key difference lies in whether an auxiliary fuel system is employed, resulting in either an active or passive PC.

In an active PC, a secondary fuel injection system enriches the mixture near the spark plug, creating a locally richer fuel-air composition. In contrast, a passive PC, often designed as a cap with orifices around the spark plug, relies on scavenged gases that enter the PC during the compression stroke as its fuel source.

A schematic comparison of these two systems is shown in Figure 3.28. While an additional injection system in the PC has been shown to offer advantages, it also adds complexity and increases system costs [77]. As a result, a passive prechamber system was chosen for our experimental setup. Moving forward, only passive prechambers will be discussed.

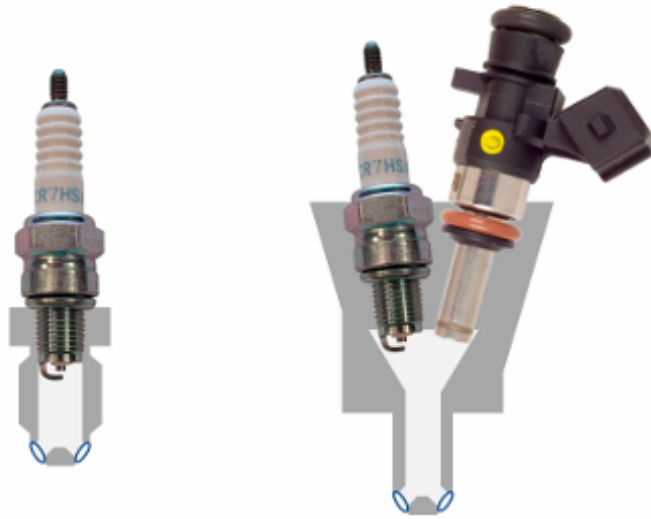


Figure 3.28: Schematic of the prechamber configurations, passive on the (left) and active on the (right) [76].

Influence of Prechamber Design

The performance of the engine system is heavily influenced by the geometry and configuration of the prechamber. Gussak et al. [78] performed research on engines equipped with prechambers in the 1970s. They concluded that a PC with a size of 2% to 3% of the clearance volume combined with an orifice total area of 0.03 to 0.04 cm^2 per cm^3 of PC volume is optimal for combustion stability, efficiency, and power. More recent work on the matter by Shah et al. [79], which studies the effect of PC volume and nozzle diameter in a single-cylinder engine, is in line with Gussak's findings on geometrical parameters.

In addition to the prechamber's volume and orifice design, the spray angle also affects the system's effectiveness. The spray angle is defined as the largest angle between two orifices of the prechamber, as seen in Figure 3.29.

The spray angle not only determines the angle at which the jets are ejected into the MC, but also determines the scavenging and swirl of the air-fuel mixture in the PC itself. Li et al. [80] concluded that there exists an optimal spray angle that maximizes scavenging and swirl and minimizes the amount of dissipation. This optimum can be computed with advanced CFD models and experimental testing.

Efficiency gains

The use of prechambers has shown substantial improvements in thermal efficiency, combustion stability and emission characteristics for methanol engines. In a study by Liu et al. (2023) [72], computational fluid dynamics (CFD) simulations and optical engine experiments revealed that prechamber ignition reduced combustion duration compared to conventional spark ignition. Li et al. [80] measured a 30% to 50% reduction in combustion duration after PC implementation. Especially the early combustion stage (mass fraction burned 10 50%) is significantly shortened, but the later combustion stage (mass fraction burned 50 90%) presents no noticeable improvement. The shorter combustion en-

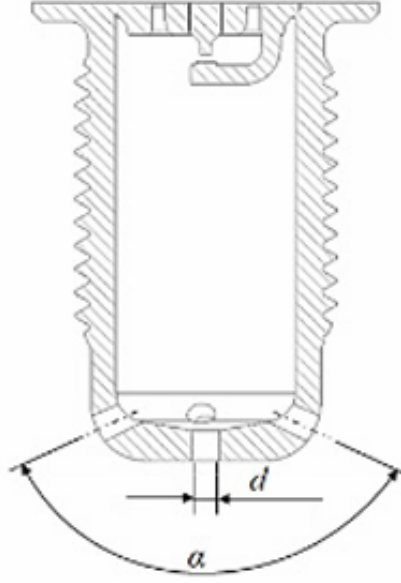


Figure 3.29: Spray angle of prechamber indicated with symbol α [80].

hances the heat release rate of near-stoichiometric MC charges, directly translating into an increased BTE. Especially under high-load conditions, where methanol prechamber combustion achieved indicated thermal efficiencies (ITE) exceeding 42% [72].

Emission Reductions

Locally distributed ignition sites with PC operation allow for increased levels of dilution (lean burn/EGR) compared to conventional spark ignition combustion [81]. This significantly reduces NO_x emissions by lowering of the peak combustion temperatures. For example, Novella et al. (2024) [74] demonstrated that the combination of prechamber ignition and EGR reduced NO_x emissions by 50% compared to baseline SI systems. Methanol prechamber systems also showed reduced carbon monoxide (CO) emissions because the distributed ignition sites facilitate more complete combustion. However, hydrocarbon emissions are still a problem under ultra-lean conditions, which requires an efficient after-treatment system [73] [74].

Challenges

Optimizing a prechamber requires extensive CFD work and experimental testing. In addition, the implementation of prechambers increases heat loss from an increase in wall area. The cooling effect of methanol compensates for this, but increased wall heat losses remain particularly evident at certain operating points [77]. It is worth mentioning that these results are specific to a light-duty engine, where heat losses are more pronounced than in larger engines. However, there is a lack of available measurements on heavy-duty passive PC engines running on methanol to provide a comparative analysis. Measuring the effect of wall heat losses in terms of ITE will be part of our experimental research to fill the literature gap on heavy-duty passive PC engines running on methanol. In passive prechambers, achieving highly diluted mixtures presents a significant challenge due to the

absence of externally supplied fuel within the prechamber. As a result, the mixture must ignite solely through the prevailing in-cylinder conditions, making the combustion process more demanding. The difficulty in maintaining stable ignition restricts the extent to which dilution can be used, thereby constraining the previously mentioned NO_x reduction potential of passive prechamber systems [77].

3.4 Problem & Goal statement

3.4.1 Problem Statement

Marine engines traditionally rely on diesel as a result of its high energy density and favorable combustion characteristics under compression ignition. However, diesel engines contribute significantly to greenhouse gas emissions, NO_x and particulate matter, creating an urgent need for cleaner alternatives. Alternative fuels, like methanol, offer a lower carbon footprint and the potential for clean combustion.

Methanol's high octane number and fast flame speed make it well-suited for SI applications. Its high resistance to auto-ignition reduces the risk of pre-ignition; however, the high CR and the elevated in-cylinder temperatures and pressures associated with marine operation can still lead to knock, particularly under high-load conditions.

Retrofitting marine diesel CI engines to methanol SI operation introduces several technical challenges, requiring innovative solutions to achieve diesel-like performance and efficiency. Achieving high brake thermal efficiency is a primary challenge, which could be addressed by advanced strategies such as lean combustion, exhaust gas recirculation, and prechamber ignition systems. Knock control remains critical, especially under high-load conditions, to ensure reliable and efficient operation.

Despite the favorable properties of methanol and the potential of advanced strategies, the behavior of methanol in heavy-duty applications under high CR and load conditions remains not entirely understood. Further investigation is needed to optimize engine efficiency and emissions, ensuring methanol's successful integration into marine applications.

3.4.2 Goal Statement

Overall goal

This research aims to optimize the performance of a heavy-duty, marine engine retrofitted from diesel CI to methanol SI operation, targeting improvements in efficiency, emissions and combustion stability. This study investigates three key strategies: lean combustion, EGR and passive prechamber combustion. By implementing these methods, the goal is to expand engine mapping limits while maintaining high efficiency and robust performance across varying operating conditions.

Methodology

A key objective is to experimentally measure critical performance metrics, including BTE, NO_x emissions and combustion stability parameters (coefficient of variation of indicated mean effective pressure). These measurements will be conducted across a range of air/fuel

equivalence ratios, EGR rates with and without prechamber configuration to evaluate their individual and combined effects.

Expected results

Based on the literature, it is hypothesized that moderate EGR rates (20-30%) will enable higher load operation while reducing NO_x emissions. Prechamber technology should facilitate stable, fast and distributed combustion, extending the lean burn limit. By combining these strategies, this research seeks to enable reliable operation at leaner conditions and across all loads, thereby advancing the potential of methanol as a marine fuel.

Towards cleaner and more efficient heavy-duty engines

The insights and data generated from this study aim to support the maritime industry's transition toward low-emission operations. By advancing sustainable fuel solutions with methanol, this work contributes to global efforts to minimize the environmental impact of shipping.

Part II

Experimental Research

Chapter 4

Experimental Setup

This chapter covers the experimental setup used in this dissertation. Expanding on previous research, this summary outlines the evolution of the engine setup, covering the baseline engine, retrofitting processes, additional engine systems, and data collection elements. This chapter concludes with a detailed description of the modifications implemented this year to address challenges associated with obtaining accurate measurement sets.

4.1 Baseline marine diesel engine

The baseline diesel engine utilized in this dissertation is the Volvo Penta D7C-B TA, a six-cylinder in-line four-stroke engine with "TA" indicating turbo aftercooler. This heavy-duty (HD) high-speed marine diesel engine model is commonly employed in smaller vessels, typically seen in coastal regions, inland waterways, or as auxiliary engine on large sea-going vessels. The baseline engine is depicted in Figure 4.1, with main technical specifications represented in Table 4.1. The engine data sheet can be found in Appendix J.



Figure 4.1: Volvo Penta D7C-B TA, the baseline marine diesel engine used in this dissertation.

The marine diesel engine is fitted with a cooling system consisting of a seawater pump.

Table 4.1: Technical specifications of the baseline Volvo Penta D7C-B TA marine diesel engine.

Description	Technical specification
Model	Volvo Penta D7C-B TA
Aspiration	Turbocharged with air intercooler
Engine layout	4-stroke, 6 cylinders in-line
Compression ratio	17.6:1
Bore x stroke	108 mm x 130 mm
Displacement volume	7.15 l
Diesel injection system	Cam-driven single injection pumps
Diesel injection pressure	1200 bar
Maximum torque / speed / bmep	904 Nm / 1500 RPM / 15.9 bar
Rated power / speed / bmep	195 kW / 2300 RPM / 14.2 bar
Inlet Valve Closes (IVC)	225 °CA
Exhaust Valve Opens (EVO)	480 °CA
Exhaust Valve Closes (EVC)	12 °CA
Emission compliance	IMO NO _x Tier 2
Maximum allowable p_{cyl} / speed	140 bar / 1500 RPM

In the experimental setup, rainwater (instead of seawater) is pumped to the intercooler in order to cool the air exiting the turbo-compressor. Subsequently, the rainwater cools down the internal engine cooling liquid via a shell-and-tube water-to-water heat exchanger, before flowing back to the rainwater reservoir. The internal engine cooling liquid takes up heat from all the other engine components such as the crankcase, exhaust manifold and turbo-compressor. Finally, rainwater is also used to power the Froude G-type hydraulic dynamometer, which functions as a water brake, providing load to the engine.

Furthermore, The engine is originally equipped with cam-driven single injection pumps, commonly referred to as a pump-line-nozzle (PLN) system, which supplies diesel to the injectors. With this system, adjustments to injection timing, duration, and pressure cannot be made while the engine is running. Injection timing can only be modified when the engine is off by altering the shim thickness, which controls the distance between the injection pump foot and the roller tappet that transfers camshaft motion to the pump.

4.2 Retrofitting

This section presents an overview of the engine modifications implemented over the years. It begins with a discussion of the retrofit for diesel-methanol dual-fuel (DMDF) operation, followed by the adaptations required for 100% methanol fueling.

4.2.1 Shifting from diesel to DMDF

DMDF type of combustion uses methanol as the main fuel and diesel as a pilot to ignite the methanol-air mixture. In this way, the engine can still operate according to the com-

pression ignition (CI) principle. During the European Horizon 2020 LeanShips project, short for "Low Energy and Near-to-Zero Emission Ships", preceding his doctoral research, Dierickx [82] retrofitted the baseline diesel engine as described in section 4.1. The retrofit primarily consisted of adding a methanol supply system which could be regulated independently from the original diesel supply system.

A port fuel injection (PFI) system, specifically a multi-point injection (MPI) system, was installed on the engine. In this configuration, each cylinder is equipped with its own fuel injector located in the intake manifold. Fuel is injected separately into each intake port, allowing for more precise control of the air-fuel mixture. These six injectors are connected to a shared methanol fuel rail. A 1,000 liter reservoir is used to store the methanol, which is located adjacent to the setup. To prevent any potential methanol vapor from escaping to the surroundings, the reservoir is housed within a safety cabinet maintained at sub-atmospheric pressure. Figure 4.2 illustrates the complete fuel supply system, including all components and an additional return line to facilitate the flow of excess methanol back to the reservoir.

To control the supply of methanol, a MoTeC M800 engine control unit (ECU) was added alongside the existing mechanically driven diesel injection system. To ensure a homogeneous methanol-air mixture in dual-fuel mode, methanol was injected at a pressure of 5.5 bar, with the end of injection (EOI) consistently set at 280 degrees crank angle before top dead center ($^{\circ}\text{CA BTDC}$). The start of injection (SOI) varied depending on the quantity of fuel required, which was specified in the MoTeC ECU Manager software as an IJPU percentage, representing the injector opening duration. A 100% IJPU corresponded to an injector opening duration of 15 ms.

For a more detailed description of the full retrofitting process, readers are referred to [82].

Table 4.2: Retrofit components DMDF, adapted from [22].

Description	Technical specification
Methanol tank	1000 l IBC reservoir
Twin valved coupling	QV-series quickconnect
Fuel filter (75 μm)	FueLab 818 xxx series
Fuel pump	MARCH MASTER GEAR series - MMG313 AISI316L
Pressure regulator	FueLab 51501
Mass flow meter	Bronkhorst mini CORI-FLOW
Ball valve	Earls UltraPro
Fuel filter (6, 40 & 75 μm)	FueLab 818 xxx series
MPI fuel rail	Stainless steel - made in house
Fuel injectors	Bosch EV14 CKxT (piezo-injector)
Engine control unit (ECU)	MoTeC M800
Crank angle encoder	Magneti Marelli SEN 8D inductive
Cam angle encoder	Volvo 22435058 variable reluctance

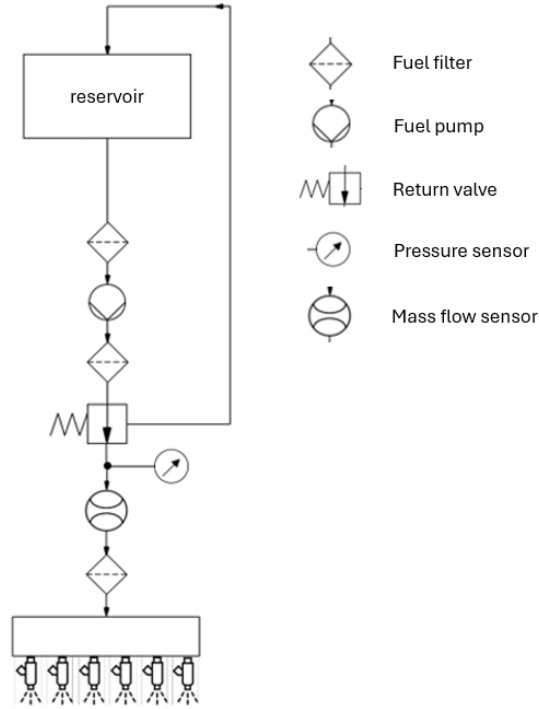


Figure 4.2: Methanol supply system, adapted from [83].

4.2.2 Transitioning from DMDF to Methanol SI

Previous research conducted by Bracke and Gyselinck [22], explored the potential of methanol combustion without the assistance of diesel on a HD CI engine. Therefore, the engine was retrofitted from DMDF operation to methanol spark ignition (SI) operation. As a methanol supply system was already in place, only an ignition system was needed to enable methanol SI operation.

The retrofit involved installing an ignition system comprising spark plugs, ignition coils, and ignition leads, which were carefully selected to match the existing engine configuration. Since the original diesel injectors were no longer needed, the spark plugs were installed in their position, as illustrated in Figure 4.3. NGK SILMAR8A9S spark plugs were selected based on their heat grade and compatibility with the previous diesel injector dimensions. Custom-designed ignition leads were created to connect high-performance ignition coils to the spark plugs. These ignition coils are driven by the MoTeC ECU manager software. The settings in the ECU software were calibrated to optimize ignition parameters, such as firing order and spark timing, ensuring reliable SI operation. Figure 4.4 illustrates the implementation of the ignition system.

In addition to the ignition system, a throttle valve is a crucial component for SI operation. Initially, the electronic throttle body ETB1 valve (VDO A2C59513666 [84]) from the heat exchanger setup (see section 4.3) was integrated, utilizing the MoTeC drive-by-wire (DBW) feature. However, challenges arose with this implementation, which will be discussed in detail in section 4.5.

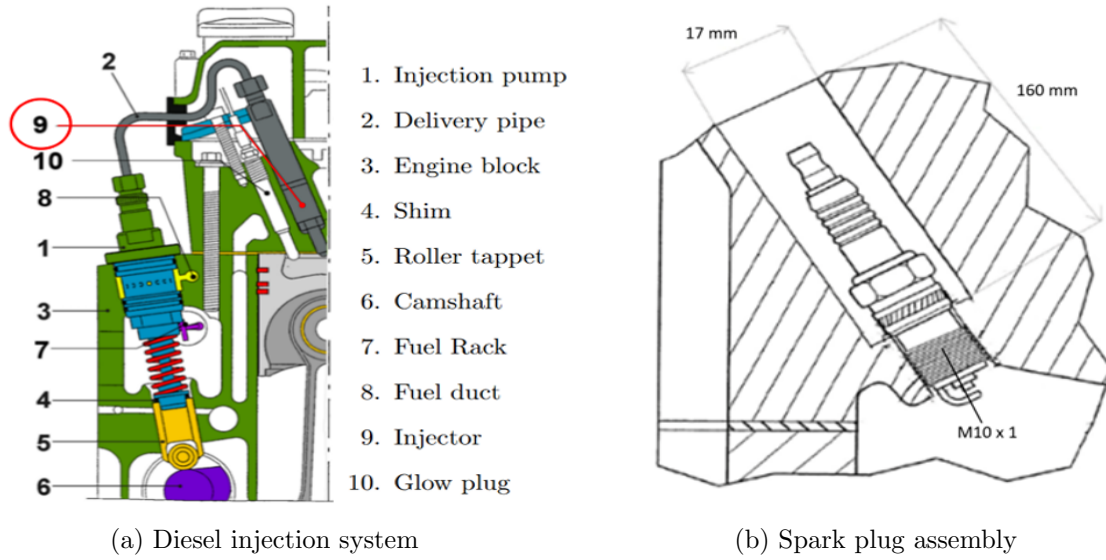


Figure 4.3: Retrofit from diesel injectors to spark plugs [22].



Figure 4.4: Ignition system.

Table 4.3: Retrofit components SI, adapted from [21].

Description	Technical specification
Spark plugs	NGK SILMAR8A9S
Ignition coils	SEM I2CM Inductive ignition coils for CNG, LPG & Biogas fueled engines
Ignition lead	Custom-made, designed and manufactured by Hatraco
Power supply	Output voltage of 24V, output current of 100A
Throttle body	VDO A2C59513666 electronic throttle body

4.3 Additional engine systems

Over time, additional engine systems were incorporated into the setup to meet specific experimental requirements. This section provides an overview of the additional systems that will be used in this year's experimental research, namely, the Exhaust Gas Recirculation (EGR) loop. For other modifications, readers are referred to the respective dissertations, listed in Table 4.4. The schematic of the DMDF setup, presented in Figure 4.5, shows the complete engine setup with additional components.

Table 4.4: Overview of engine systems.

Engine system	Year	Design by
Wastegate	2020-2021	Dejaegere [85]
Heat Exchanger	2020-2021	Dejaegere [85]
Three-way valve	2020-2021	Dejaegere [85]
EGR loop	2020-2021	Van Gijzeghem [86]

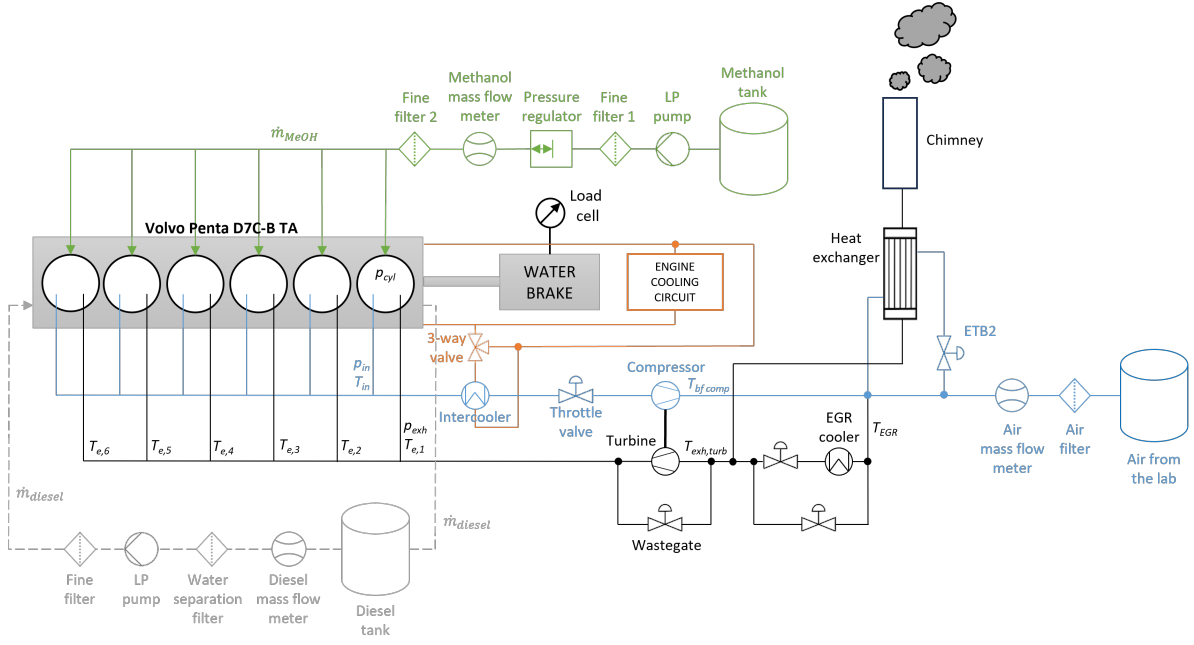


Figure 4.5: Schematic representation of the Engine setup with additional engine equipment, adapted from [21].

4.3.1 EGR loop

Van Gijzeghem [86] integrated an EGR loop into the engine setup to assess its effectiveness as a knock suppressor across the entire load range in DMDF operation. Similarly, in this dissertation, EGR will be used to expand the knock limit and increase efficiency in methanol SI operation.

As illustrated in Figure 4.6, this system allows for the introduction of both hot and cold exhaust gases into the intake air mixture. Cold exhaust gases are obtained through an

EGR cooler, which operates using engine cooling water, while hot exhaust gases bypass the cooler. For clarity, these configurations are referred to as "cold EGR" and "hot EGR", respectively.

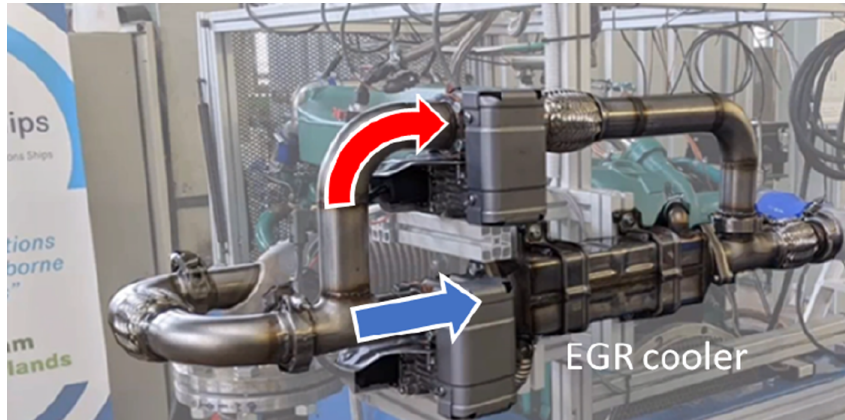


Figure 4.6: Exhaust Gas Recirculation (EGR) loop. Blue path: cooled EGR ("cold EGR"); red path: bypass over EGR cooler ("hot EGR"), Adapted from [22].

4.4 Data collection

To collect all valuable data during measurement campaigns, a series of sensors and processing software are used. This section will give a detailed overview of all measurement equipment and software used to acquire, process, and post-process data sets.

4.4.1 Measurement equipment

The setup is equipped with a variety of measurement instruments, including mass flow sensors, pressure sensors, and temperature sensors. It also features a load cell, exhaust emission analyzers, and crank angle encoders. A detailed overview of all measurement instruments is provided in Table 4.5. It should be noted that, due to malfunctioning of the NO_2 - NO converter, total NO_x emission measurements could not be performed during this dissertation.

4.4.2 Software

The data processing in this thesis utilizes a combination of LabVIEW and Python, each serving a distinct role in data collection, visualization, and analysis.

LabVIEW is used for real-time monitoring and data acquisition, providing an interface for collecting various engine parameters. It records pressure measurements [V], temperature readings [$^{\circ}\text{C}$], and mass flow rates [kg/h], along with engine operating parameters such as speed [RPM] and load [Nm]. All acquired data is logged into Excel files for further processing and visualization.

Emission data, including the concentrations of NO_x [ppm], CO_2 [vol%], CO [vol%], and O_2 [vol%], are retrieved from the emission cabinet. As this data is not automatically logged, it is manually entered into the corresponding Excel files.

Table 4.5: Details of measurement equipment, adapted from [22].

Description	Technical specification
Chard chassis	NI CompactDAQ 9178
Analog input module	NI 9215 (for pressure sensors)
Digital I/O module	NI 9401 (for crank angle encoder)
Thermocouple module	NI 9213 (for thermocouples)
Voltage input module	NI 9205 (for flow meters and load cell)
Voltage output module	NI 9263 (for throttle control)
In-cylinder pressure sensor	Kistler 6045b
Intake and outlet pressure sensor	Keller M5HB
Crank angle encoder	Kistler 2614CK
Charge amplifier	Kistler 5018A
Thermocouples exhaust, EGR loop and inlet air HEX	K-type thermocouples
Other thermocouples	T-type thermocouples
Air mass flow meter	Bronkhorst IN-FLOW
Diesel and methanol mass flow meters	Bronkhorst lini CORI-FLOW
Wideband lambda sensor (MoTeC)	Bosch Motorsport LSU 4.9
Load cell	Zemic H3
Emissions analysers	Siemens Ultramat 23 (for CO, NO, NO ₂ and CO ₂)

To analyze combustion dynamics, pressure signals from the intake manifold, exhaust manifold, and in-cylinder pressure, only in cylinder 1, are recorded. These signals are captured over a full engine cycle at a resolution of two values per crank angle degree and stored in .prn files for further analysis.

Once all measurement data is captured, Python is used for mathematical computations and data analysis. This Python script is an updated version of the MATLAB code previously utilized in earlier dissertations. The revision was made to improve ease of use and streamline the MATLAB code for greater efficiency and clarity. In the Python code, two main parts for analysis can be found: a *calculations* class and a *basefile*. The *calculations* class processes the data for each measurement point, using the corresponding .prn file and the information in the Excel file. Afterwards, the results of these calculations are aggregated in the *basefile*, which compiles data from all measurement points for a comprehensive analysis and interpretation of the experimental results. Ultimately, the overall results for each measurement point are exported to new Excel files. More information about the Matlab to Python transition can be found in Appendix D. Furthermore, a post-processing Python file is made to plot important engine performance parameters (see subsection 4.4.3). In this way, measurement sets can be immediately evaluated, which aids in detecting faulty equipment or inaccurate data.

4.4.3 Engine performance parameters

In the following, the key engine performance parameters are presented, along with details of their measurement and processing methods.

Pressures

The gathering of pressure information is the key to further investigation of engine performance. As stated in subsection 4.4.2, the intake manifold pressure, exhaust manifold pressure, and in-cylinder pressure are stored in a .prn file. Data stored in the .prn files contain samples of 200 consecutive combustion cycles, with a resolution of 2 samples per crank angle degree. These signals, in their raw, unfiltered form, are processed by the *calculations* class.

First, the absolute inlet and outlet pressures are determined. Since the in-cylinder pressure sensor provides a relative measurement, it must be referenced to the intake pressure to derive the absolute cylinder pressure. This process involves integrating the absolute inlet and outlet pressures over the crank angle range during which the respective valves are open. To ensure proper synchronization with the intake valve opening event, a verification step is performed, and if necessary, the pressure traces are adjusted by 360° CA. Finally, the pressure data is reorganized into a matrix format, where each column corresponds to an individual engine cycle, facilitating cycle-by-cycle analysis and comparison.

Subsequently, the absolute pressure traces for the intake, exhaust, and in-cylinder measurements are averaged across all recorded cycles. This averaging provides a clear representation of the pressure behavior over the entire measurement period. Following this, a best-fit pressure cycle is identified using mathematical optimization techniques to determine the most representative pressure profile. The cycle exhibiting the lowest mean square deviation from the average trace is selected as the reference cycle for further analysis.

This reference cycle is utilized in subsequent evaluations, including the calculation of in-cylinder temperature, combustion durations and more, further discussed in later sections.

Finally, pressure filtering techniques, such as a seven-point moving average, are applied to prepare the pressure data for heat release rate ($dQ/d\theta$) calculations. Filtering is essential for reducing signal noise, thereby improving the accuracy of combustion analysis.

Coefficient Of Variation

In engine performance analysis, the coefficient of variation (COV) serves as an indicator of the variability of a given parameter by relating its standard deviation to its mean value. Within the scope of this dissertation, two key COV metrics are evaluated: the COV of the indicated mean effective pressure (IMEP) and the exhaust gas temperature. These are denoted COV_{IMEP} and $COV_{T,exh}$, respectively, and are used to assess in-cylinder combustion stability and cylinder-to-cylinder variation. COV_{IMEP} , in particular, is instrumental in detecting combustion irregularities such as misfires and partial burns, providing valuable information on overall engine performance. While both COV metrics are calculated during post-processing, they are also computed in real time during engine operation and

visualized in LabVIEW, ensuring continuous monitoring and accurate representation of engine behavior.

Combustion Dynamics

In order to assess combustion efficiency and overall engine performance, several key parameters are computed and analyzed. The air-fuel ratio (AFR) is first determined, along with the stoichiometric AFR and the air-fuel equivalence ratio (λ). These metrics provide essential information on the combustion process, particularly on fuel utilization and mixture strength under various operating conditions.

Following this, a set of performance and efficiency indicators is derived. These include effective power output (P_e), brake-specific fuel consumption (BSFC), brake thermal efficiency (BTE), and brake mean effective pressure (BMEP). Together, these parameters characterize the engine's ability to convert chemical energy into mechanical work, offering a comprehensive understanding of energy utilization and system efficiency.

Exhaust gas temperatures are also evaluated to supplement the analysis, providing additional information about combustion quality and the thermal behavior of the exhaust system.

To gain a deeper understanding of combustion phasing and heat release characteristics, the heat release rate (HRR) is calculated using filtered in-cylinder pressure data. The HRR reflects the rate at which chemical energy is converted into thermal energy within the cylinder, and calculated based on the assumption of a constant specific heat ratio (γ) initially set at 1.35. This initial assumption is refined through an iterative process that adjusts γ dynamically, accounting for changes in in-cylinder temperature and gas properties throughout the combustion event.

Key combustion timing metrics are extracted from the HRR profile. The Ignition timing (IT) or spark timing (ST) refers to the crank angle before top dead center (TDC), at which the spark plug fires and the end of combustion (EOC) is identified based on the point where the heat release drops below 1.5 J/°CA. Although defining these points at zero HRR might seem intuitive, such an approach is often unreliable due to pressure signal noise, which can introduce significant timing inaccuracies. Therefore, the crank angles corresponding to 10%, 50%, and 90% of the cumulative heat release, designated as CA_{10} , CA_{50} , and CA_{90} , respectively, are used as standardized markers. The end of combustion is defined by CA_{90} , as contributions to HRR beyond this point are considered negligible. The overall combustion duration (CB_{dur}) is then quantified as the interval between CA_{10} and CA_{90} , offering a robust measure of the temporal duration of the combustion event.

In-cylinder Temperature

The in-cylinder temperature is estimated using the ideal gas law. For this calculation, the pressure corresponds to the filtered and cycle-averaged in-cylinder pressure trace, while the instantaneous cylinder volume is determined as a function of the crank angle. The mass of the trapped gas mixture is derived from the measured intake mass flow. The specific gas constant dynamically changes in response to the intake mixture and exhaust gas composition, with interpolation applied between them.

Emissions

A Siemens Ultramat 23 gas analyzer was used to obtain CO₂, CO, NO, and NO₂ engine-out emissions. The cabinet is equipped with a NO₂-NO converter. This allows the analyzer, which measures only NO, to determine the total NO_x emissions. Unfortunately, the NO₂-NO converter could not be used during the measurement campaign due to internal overheating. In this way, total NO_x emissions could not be measured, but trends in NO emissions were still observed. The analyzer has a full-scale measurement range of 0 - 25 vol% CO₂, 0 - 5 vol% CO and 0 - 5000 vpm NO. To express emission concentrations as specific emissions (in [g/kWh]), the measured values are converted according to the guidelines established by the IMO (International Maritime Organization) in Marpol/-CONF.3/34.

Table 4.6 summarizes the main engine performance parameters calculated during post-processing of the measurements and their respective formulas.

4.5 Setup modifications

This section outlines the modifications made to the setup in comparison to the previous year, ensuring that future work on the setup remains aligned with its current operating principles.

4.5.1 Throttle body

During the initial SI methanol measurements by [21], the ETB1 valve (VDO A2C59513666) was used as the throttle valve. Positioned upstream of the turbo-compressor, this arrangement caused issues during SI operation, although it functioned adequately during CI and DMDF operations. In SI operation, during startup, the throttle valve is almost completely closed to "choke" the engine, a common practice that helps facilitate ignition. By restricting the intake air, the throttle valve increases the air-fuel ratio, enriching the mixture and improving the likelihood of a successful start. This creates a vacuum over the turbo-compressor, which results in engine oil intended for lubrication being drawn from the turbocharger into the inlet airflow. As a result, oil enters the intercooler, leading to contamination. A significant portion of this oil was condensed and collected in the condensation trap, while the remainder was combusted, as confirmed by an oil level inspection.

To address this issue, the intercooler underwent industrial cleaning. To prevent the creation of a vacuum, the throttle valve was placed downstream of the turbo-compressor. At this location, the air pressure increases, requiring a throttle body with a smaller diameter compared to the original ETB1 valve.

Several solutions were tested last year, ultimately leading to the implementation of a PID-controlled throttle valve managed through the LabVIEW program. Although this approach resolved the issue of oil drainage from the turbocharger, it introduced significant lag in the LabVIEW control, resulting in considerable overshooting and undershooting when adjusting to measurement points. This caused delays in the measurement process

Table 4.6: Overview of calculated engine performance parameters, adapted from [21].

Air-fuel ratio	$AFR = \frac{\dot{m}_{air}}{\dot{m}_{MeOH}}$	[-]
Stoichiometric air-fuel ratio	$L_{s,MeOH} = 6.5$	[-]
Air-fuel equivalence ratio	$\lambda = \frac{AFR}{L_{s,MeOH}}$	[-]
Specific emission 'X'	$X_{specific} = \frac{\dot{m}_X}{P_e}$	[g/kWh]
Brake power	$P_e = T_e \cdot RPM_e \cdot \frac{2\pi}{60} \cdot 10^{-3}$	[kW]
Brake specific fuel consumption	$BSFC = \frac{\dot{m}_{MeOH}}{P_e} \cdot 10^{-3}$	[g/kWh]
Energy consumption	$E_{cons} = \frac{\dot{m}_{MeOH} \cdot LHV_{MeOH}}{P_e}$	[MJ/kWh]
Brake thermal efficiency	$BTE = \frac{3.6 \cdot P_e}{\dot{m}_{MeOH} \cdot LHV_{MeOH}}$	[-]
Brake mean effective pressure	$BMEP = \frac{P_e \cdot 10^3}{V_s \cdot \frac{1}{2} \cdot \frac{RPM_e}{60}} \cdot 10^{-5}$	[bar]
Indicated work per cycle	$W_i = \int_0^{720} p_{av} \cdot \frac{dV}{dCA} \cdot dCA$	[J]
Indicated mean effective pressure	$IMEP = \frac{W_i}{V_s} \cdot 10^{-5}$	[bar]
Indicated power	$P_i = W_i \cdot 6 \cdot \frac{RPM_e}{60} \cdot \frac{1}{2} \cdot 10^{-3}$	[kW]
Indicated efficiency	$\eta_i = \frac{3.6 \cdot P_i}{\dot{m}_{MeOH} \cdot LHV_{MeOH}}$	[-]
Indicated pumping work	$W_{i,p} = \int_{540}^{180} P_{av} \cdot \frac{dV}{dCA} \cdot dCA$	[J]
Pumping mean effective pressure	$PMEP = \frac{W_{i,p} \cdot 6}{V_s} \cdot 10^{-5}$	[bar]
Mechanical efficiency	$\eta_m = \frac{BMEP}{IMEP}$	[-]
Standard variation of IMEP	$\sigma_{IMEP} = \sqrt{\frac{1}{cycles} \sum (IMEP_i - IMEP_{av})^2}$	[-]
Coefficient of variation of IMEP	$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{av}}$	[-]
Volumetric efficiency	$\eta_{vol} = \frac{m_{air}}{V_{s,tot} \cdot \rho_{air,intake} \cdot RPM_e \cdot \frac{60}{2}}$	[-]
Gross heat release rate	$HRR = \frac{dQ}{dCA} = \frac{1}{\gamma-1} \cdot V \cdot \frac{dp_{av,f}}{dCA} + \frac{\gamma}{\gamma-1} \cdot p_{av,f} \cdot \frac{dV}{dCA}$	[-]
Combustion duration	$CB_{dur} = CA_{90} - CA_{10}$	[CA°]
In-cylinder temperature	$T = \frac{p_{av,f} \cdot V}{m_{mix} \cdot R}$	[K]

and led to instability in engine operation. Consequently, it was decided to revert to the DBW principle in MoTeC for throttle body control, as this method had proven reliable in previous setups. A suitable MoTeC compatible DBW throttle body (VDO 408-238-327-003Z) with the correct diameter (57 mm) and integrated cooling channels was located, installed, and successfully tested.

For a detailed discussion of the electrical installation process of the throttle body, refer to Appendix H.

4.5.2 Water circuit

As outlined in section 4.1, rainwater is utilized both to cool the engine and to provide load via the hydraulic dynamometer. The rainwater is pressurized by two centrifugal pumps arranged in series. However, during the measurement campaign, these pumps began to experience cavitation. Due to the series configuration, the pumps were unable to simultaneously support both engine cooling and brake power demands. As a temporary measure, engine cooling was switched back to city tap water. While this solution addressed the immediate issue, it resulted in unnecessary waste of clean water. To resolve this more sustainably and prevent further cavitation, it is recommended to reconfigure the pump system to a parallel arrangement. This modification is expected to effectively address the cavitation issue while maintaining efficient operation for both engine cooling and brake power provision.

4.5.3 Maintenance

Throughout the year, multiple components were replaced as a result of malfunctioning or degradation. Specifically, the inlet and outlet pressure sensors were replaced due to inaccuracies in the measurements. Additionally, two EGR valves were fully corroded and therefore replaced to ensure proper exhaust gas recirculation. In addition, six methanol injectors were replaced to maintain optimal fuel delivery and combustion efficiency. Lastly, six spark plugs were replaced to prevent misfires and ensure stable ignition. These replacements were essential to maintain engine performance and reliability during testing. Refer to Appendix F for a detailed description of encountered technical problems.

Chapter 5

Unveiling the Impact of Dilution on the Engine Performance

This chapter investigates the impact of charge dilution on the performance and operating range of a heavy-duty methanol-fueled spark-ignition engine. Two primary dilution strategies, lean operation and exhaust gas recirculation, are explored and compared relative to stoichiometric operation. The comparison focuses on key engine performance parameters such as brake thermal efficiency, combustion stability, emissions, and knock mitigation potential.

By analyzing experimental results, the chapter aims to provide a comprehensive understanding of how each dilution method influences engine behavior and to identify the benefits and trade-offs associated with each strategy in the context of heavy-duty methanol combustion.

5.1 Measurement Methodology

Before presenting the measurement results, this section outlines the experimental approach and test conditions used to compare lean operation and EGR dilution. All experiments were carried out at a fixed engine speed of 1500 RPM, which represents the peak torque point for the investigated engine under diesel operation.

To establish a stoichiometric baseline for comparison, tests were first performed at three distinct load points: 200 Nm (BMEP = 3.5 bar) representing low load, 400 Nm (BMEP = 7.1 bar) representing medium load, and 600 Nm (BMEP = 10.6 bar) representing high load. For reference, the base engine has a maximum load of 904 Nm (BMEP = 15.9 bar). At each of these load levels, an ignition timing sweep was carried out. The limits of the sweep were defined by the onset of audible knock on the advanced side and by a coefficient of variation in indicated mean effective pressure (COV_{IMEP}) exceeding 3% on the retarded side, which was used as a threshold for combustion instability.

Following the stoichiometric tests, lean operation measurements were conducted using ignition timing sweeps at the same three load points. At each load, the lambda value was

progressively increased, starting from $\lambda = 1.1$ in increments of 0.1. The process continued until further increases in lambda triggered knock or exceeded the 3% instability threshold. The maximum lambda values achieved while maintaining stable operation were $\lambda = 1.5$ at 3.5 bar BMEP, $\lambda = 1.3$ at 7.1 bar BMEP, and $\lambda = 1.3$ at 10.6 bar BMEP. The lambda value was monitored using a sensor connected to the MoTeC system, see Table 4.5.

The EGR measurements followed a similar approach. Again using the three predefined load points, EGR percentage was increased in 5% increments, starting from 5%, and in 2.5% increments near the EGR stability limit. The testing continued until knock or combustion instability emerged as limiting factors. The maximum EGR rates achieved under stable operating conditions were 22.5% at 3.5 bar BMEP, 22.5% at 7.1 bar BMEP, and 17.5% at 10.6 bar BMEP. All EGR measurements were conducted for $\lambda = 1$. The EGR percentage was not directly monitored but instead calculated based on CO₂ concentration measurements obtained from the emission cabinet, allowing an accurate determination of the fraction of recirculated exhaust gases, see Appendix G. The procedure for establishing lean and EGR operating conditions is detailed in Appendix I.

All measurements were conducted with controlled accuracy. The engine speed was maintained within ± 10 RPM, and the engine load was controlled within ± 5 Nm. The lambda value was kept within a ± 0.01 tolerance, while the EGR percentage was controlled to within $\pm 0.3\%$.

A summary of the measurement sets used in the experimental campaign is provided in Table 5.1.

Table 5.1: Overview of measurement sets at different load points.

BMEP [bar]	Lean Operation (max λ)	EGR (max %)
3.5	$\lambda = 1.5$	22.5%
7.1	$\lambda = 1.3$	22.5%
10.6	$\lambda = 1.3$	17.5%

Remark on Spark Timing Accuracy

It is important to note a potential source of uncertainty regarding the accuracy of the spark timing values reported throughout the experimental analysis. The ignition timings referenced in this work are those defined within the MoTeC engine control software. However, discrepancies may exist between the programmed spark timing and the actual ignition event occurring within the engine. This deviation is likely attributable to a small delay between the electronic trigger signal and the physical discharge of the spark plug.

This observation was prompted by the consistently long ignition delays recorded during testing, which raised concerns about the fidelity of the reported spark timings. As a result, it is plausible that the actual spark events occurred slightly later, by several degrees of crank angle, than indicated by the MoTeC system.

Despite this potential offset, the relative trends observed across the various measurements

remain valid. Since all test points were subject to the same timing reference framework, the comparative analyses and resulting conclusions regarding combustion behavior and performance characteristics are not compromised.

Use of AI Assistance

Artificial Intelligence (AI) tools were utilized throughout this thesis project as a means of improving workflow efficiency and enhancing productivity. Specifically, AI was employed for tasks such as summarizing lengthy academic papers, refining writing structure, and supporting coding-related activities including data post-processing and the generation of plotting scripts.

It is important to emphasize that AI was used solely as a supportive tool to accelerate routine or repetitive tasks. No AI-generated content was directly copied into this thesis without thorough review, adaptation, and verification by the author. The core technical work, including experimental setup, data analysis, interpretation of results, and the development of conclusions, was carried out independently.

5.2 Operational Limits

This section examines the stable operating boundaries across different combustion strategies: stoichiometric, lean, and EGR-diluted combustion. As described in section 5.1, all experiments were conducted within two primary operational constraints: the knock limit and the combustion stability limit. Additionally, under significantly retarded spark timings, an upper limit for exhaust gas temperature of 620°C was reached, imposing an additional boundary on the operating range.

The knock limit in this dissertation is defined as the point at which audible knocking is detected during engine operation. While the authors acknowledge that this method does not represent a scientifically rigorous approach, as it lacks the precision of dedicated knock detection sensors, it was deemed sufficient given the experimental setup available. As outlined in subsection 3.2.1, sustained knock can cause accelerated engine wear, mechanical damage, and in severe cases, complete engine failure.

The combustion stability limit is quantified using the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}), calculated as the standard deviation of IMEP (σ_{IMEP}) divided by its mean value (μ_{IMEP}). For each operating condition, 200 consecutive in-cylinder pressure cycles were recorded to ensure statistical relevance.

Elevated COV_{IMEP} values are indicative of increased cycle-to-cycle variability in combustion, which may manifest as partial burns or misfires. Such instabilities negatively affect thermal efficiency, as the fuel's chemical energy is not fully converted into mechanical work. Furthermore, large fluctuations in in-cylinder pressure can result in perceptible engine vibrations, rendering the engine unsuitable for practical applications [87].

Literature reports a range of COV_{IMEP} thresholds to define acceptable combustion stability, typically between 3% and 10%, depending on the specific engine architecture and its intended application [87]. In this study, a threshold value of 3% was adopted, following

the methodology described by Demiddeleer and Vanderbeken [21]. Combustion conditions exceeding this threshold were classified as unstable.

5.2.1 Stoichiometric operation

Figure 5.1 illustrates the variation in combustion stability, quantified by COV_{IMEP} , as a function of spark timing across different engine loads under stoichiometric conditions. A general trend is evident: advancing the spark timing reduces cycle-to-cycle variability, indicating improved combustion stability. However, combustion stability will decrease with overly advanced ignition timing. Conversely, retarding the spark timing results in a rapid increase in COV_{IMEP} , signaling deteriorating stability.

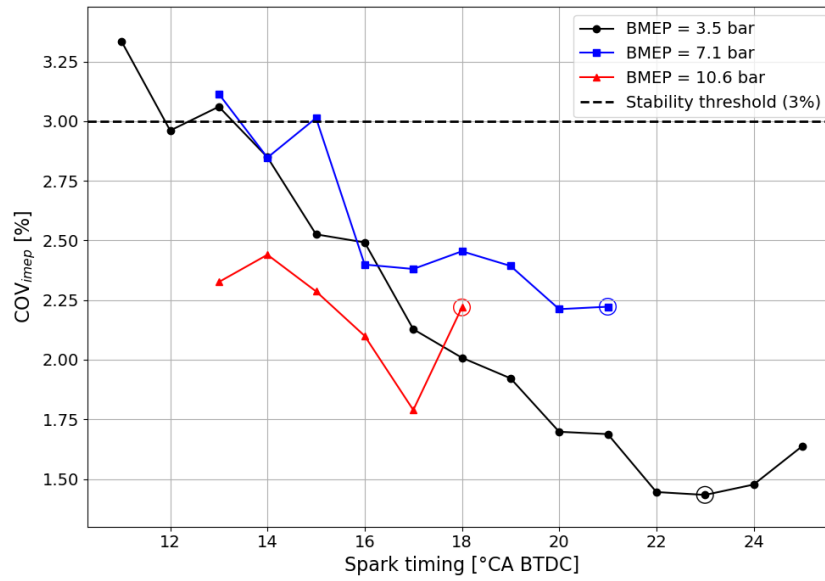


Figure 5.1: Effect of spark timing on COV_{IMEP} for different loads (BMEP) under stoichiometric conditions, with the spark timing corresponding to peak BTE highlighted.

To explain this behavior, it is essential to investigate the combustion durations, which serve as critical indicators of how rapidly the air-methanol mixture is burned and provide insight into the phasing of the combustion process. Several key parameters are of particular interest: CA_{10} , CA_{50} , and CA_{90} , representing the crank angles at which 10%, 50%, and 90% of the cumulative heat release have occurred, respectively. Ignition timing or spark timing refers to the crank angle before top dead center, at which the spark plug fires and EOC is defined at the crank angle where the HRR crosses a threshold of 1.5 J/°CA. These combustion phases are schematically represented in Figure 5.2.

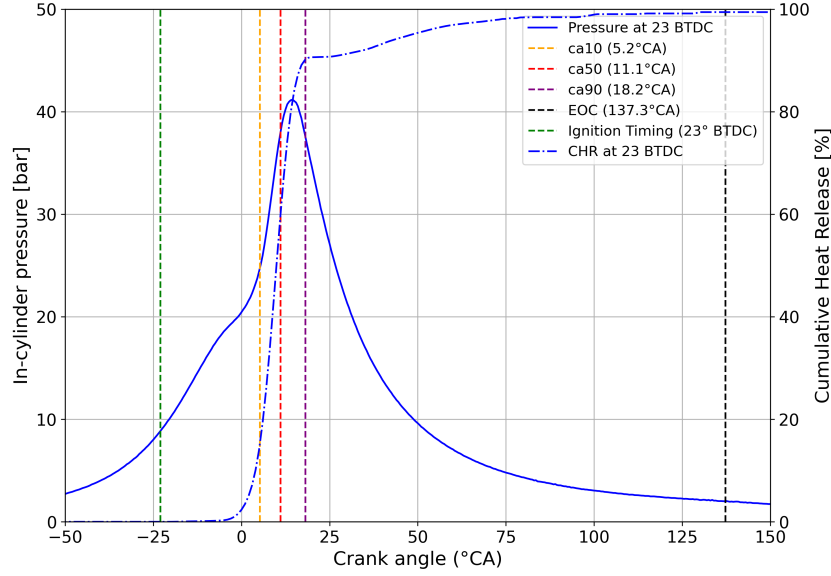


Figure 5.2: Visualization of combustion duration parameters plotted alongside the pressure trace and cumulative heat release (CHR) curve.

For comparative analysis, two composite parameters are defined:

- CA_{0-10} : The crank angle duration from the IT to the point where 10% of the total heat release has occurred (CA_{10}). This interval is commonly referred to as the **ignition delay** and characterizes the initial combustion development phase, still impacted by the laminar phase of flame kernel growth.
- CA_{10-90} : The crank angle interval between CA_{10} and CA_{90} , referred to as the **main combustion duration**, which describes the bulk burning phase, and is governed by fully developed turbulent flame propagation.

These definitions provide a consistent framework for comparing combustion behavior under varying engine conditions.

Figure 5.3 presents the influence of spark timing on the combustion duration metrics across multiple load conditions. As expected, advancing the spark timing shifts the center of combustion, CA_{50} , closer to TDC. The trends of CA_{0-10} and CA_{10-90} can be attributed to the thermodynamic conditions that dominate at ignition.

Advancing the spark timing causes ignition to occur earlier in the compression stroke, which in turn leads to earlier flame kernel development. When the main combustion initiates closer to TDC, the mixture benefits from higher pressure and temperature, as well as increased in-cylinder turbulence. Turbulence intensity generally peaks near TDC and decays linearly with crank angle after TDC. These conditions promote faster chemical reaction rates and more efficient flame propagation, thereby reducing both the ignition delay (CA_{0-10}) and the main combustion duration (CA_{10-90}).

However, it is important to note that excessively advancing the spark timing can be counterproductive. If the ignition occurs too early in the compression stroke, the mixture may still be relatively cold and turbulence levels lower, leading to a slower and less stable

initial flame development. This defines the existence of an optimal spark timing, often referred to as the minimum spark advance for best torque (MBT), where the balance between thermodynamic and turbulence conditions is most favorable.

In the present study, however, engine operation remained significantly knock-limited under stoichiometric conditions. As a result, spark timing could not be advanced sufficiently to reach or exceed the MBT point at higher loads. Thus, all observed reductions in combustion duration occurred within the regime where advancing the spark consistently improved the ignition environment without encountering the diminishing returns typically associated with over-advanced timing.

In addition, higher loads inherently shorten the overall combustion duration due to elevated in-cylinder pressure and temperature, which accelerate flame propagation.

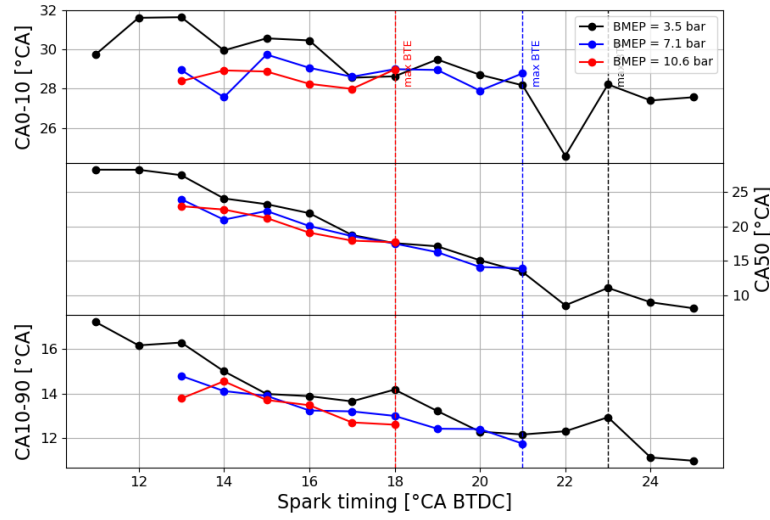


Figure 5.3: Effects of spark timing on CA_{0-10} , CA_{50} , and CA_{10-90} across different loads (BMEP) under stoichiometric conditions, with the spark timing for maximum BTE indicated.

It becomes clear that combustion duration and combustion stability are closely linked. Specifically, shorter ignition delays are associated with reduced COV_{IMEP} , indicating more stable operation. Conversely, prolonged combustion durations, typically observed with retarded spark timings, lead to elevated cycle-to-cycle variability and reduced stability. This observation is consistent with the findings of Heywood [88], who notes that late spark timing increases the total combustion duration, thereby amplifying combustion variability and raising the likelihood of unstable cycles. Notably, Figure 5.1 further confirms that at lower loads, excessively retarded spark timings can push the system beyond the 3% stability threshold, resulting in potentially rough engine operation.

All measurement points falling below the stability threshold line in Figure 5.1 are considered stable operating points. An error margin of ± 0.2 COV_{IMEP} is applied to account for measurement uncertainty. It has to be noted that this is a somewhat arbitrary choice. Repeating these experiments multiple times would allow for a more precise conclusion about whether or not it exceeds the 3% threshold. However, due to time constraints, this

could not be conducted here.

Based on the previously discussed relationship between load, combustion duration, and stability, it would be logical to expect that the stable operating range increases at higher loads. This expectation stems from the observation that combustion duration decreases with increasing load, which enhances stability by reducing cycle-to-cycle variability. However, the experimental data reveal the opposite trend: the number of stable operating points diminishes as the load increases.

This apparent contradiction can be explained by considering the second limiting factor, namely the knock limit. The most advanced spark timing in Figure 5.1 corresponds with the knock limited spark advance (KLSA) at that load. At higher loads, the knock limit becomes the dominant constraint, shifting more significantly than the stability limit and thereby reducing the range of stable spark timings. Elevated in-cylinder temperatures and pressures at high loads increase the propensity for knock, narrowing the permissible operating window.

Table 5.2 summarizes the stable spark timing ranges determined for each load condition. It is important to note that under high load, the stability limit was not technically reached. However, further spark retardation was prevented due to exhaust gas temperatures exceeding the tolerable threshold of 620°C, as shown in Figure 5.15.

Table 5.2: Stable spark timing range for each load condition under stoichiometric operation.

Load (BMEP)	Spark Timing Range (°CA BTDC)	Remarks
Low	12–25	Stability limit reached
Medium	13–21	Limited by exhaust temperature
High	13–18	Limited by exhaust temperature

5.2.2 Dilution tolerance

With the stoichiometric baseline established, the effect of dilution on the engine’s operating limits can now be systematically assessed. As outlined in the methodology section, this dissertation focuses on two specific dilution strategies: lean combustion (excess air) and EGR dilution.

Before analyzing the measurement results, it is essential to establish a consistent basis for comparing the level of dilution introduced by different strategies. EGR is typically quantified directly as a percentage by mass. The way of determining the EGR rate is established in Appendix G. In contrast, lean operation is usually represented using the excess air ratio, λ .

To enable a fair comparison between EGR and lean operation, the dilution effect of a lean mixture can be converted to an equivalent air dilution percentage. This air dilution percentage represents the relative amount of excess air in the cylinder compared to stoichiometric operation. It is calculated using the following formula:

$$\% \text{Air Dilution} = \frac{\lambda - 1}{\lambda + \frac{1}{\text{AFR}_{\text{stoich}}}} \times 100 \quad (5.1)$$

where:

- λ is the air-fuel equivalence ratio (excess air ratio),
- $\text{AFR}_{\text{stoich}}$ is the stoichiometric air-to-fuel ratio for methanol. For methanol, this value is 6.5.

Table 5.3 shows the calculated air dilution percentages corresponding to several λ -values.

Table 5.3: Air Dilution Percentage for various λ -values.

λ	Air Dilution [%]
1.1	$\frac{1.1-1}{1.1+\frac{1}{6.5}} \times 100 = 7.98$
1.2	$\frac{1.2-1}{1.2+\frac{1}{6.5}} \times 100 = 14.77$
1.3	$\frac{1.3-1}{1.3+\frac{1}{6.5}} \times 100 = 20.63$
1.4	$\frac{1.4-1}{1.4+\frac{1}{6.5}} \times 100 = 25.74$
1.5	$\frac{1.5-1}{1.5+\frac{1}{6.5}} \times 100 = 30.23$

Figure 5.4 and Figure 5.5 illustrate the effect of increasing dilution on the stable operating range across various engine loads, with the data plotted as a function of spark timing. In these figures, the right-hand boundary of each stable region is constrained by the knock limit, beyond which further spark advance is no longer possible without inducing audible knock. Conversely, the left-hand boundary is defined by the combustion stability threshold, set at a COV_{IMEP} of 3%.

An important exception to this pattern is observed under stoichiometric conditions as well as under mild dilution levels, specifically at $\lambda = 1.1$ and 5% EGR, at high load. In these cases, further spark retardation is not constrained by the stability limit, but rather by excessive exhaust gas temperatures, which exceeded the maximum allowable threshold of 620°C.

At first glance, both dilution strategies exhibit a similar influence on the operating limits: increasing dilution leads to a reduction in combustion stability, reflected by a rise in cycle-to-cycle variability, while simultaneously extending the knock limit.

This combination of effects yields a noteworthy advantage. Although the stable operating range becomes narrower due to stability constraints, the ability to further advance the spark timing at higher loads, made possible by the expanded knock margin, opens new potential for efficiency improvements. Under stoichiometric conditions, spark advance is typically constrained by the early onset of knock. By contrast, dilution, whether through excess air or EGR, suppresses knock, enabling operation at more optimal combustion

phasing. The specific efficiency benefits associated with this, will be discussed in detail in section 5.3.

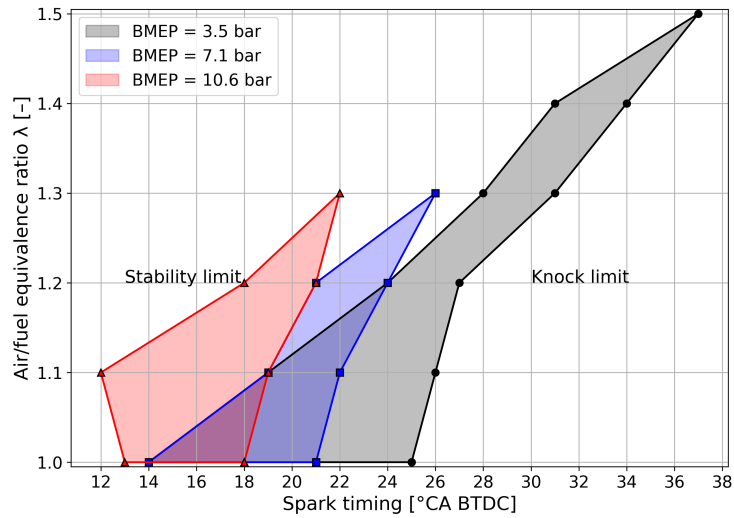


Figure 5.4: Stable operating range for lean burning at different loads, with knock constraining further spark advance and combustion stability setting the limit for spark retardation.

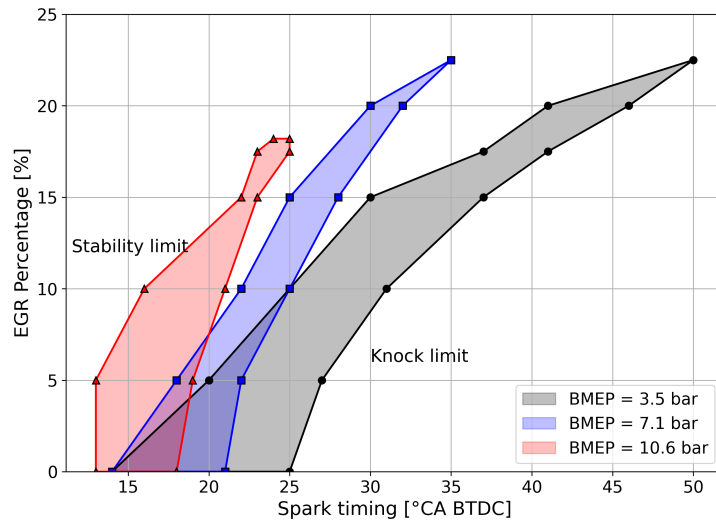


Figure 5.5: Stable operating range for EGR burning at different loads, with knock constraining further spark advance and combustion stability setting the limit for spark retardation.

For both dilution strategies, the dilution tolerance of the engine decreases as the load increases. At first sight, this trend may appear counterintuitive. Typically, at higher loads, the combustion process becomes inherently more stable due to the elevated pressure and temperature conditions, which enhance flame propagation and robustness.

However, when revisiting the experimental mappings presented in Figure 5.4 and Figure 5.5, this phenomenon can be easily explained. These figures clearly demonstrate that at higher loads, combustion indeed remains stable even at more retarded spark timings, confirming the expected improvement in combustion stability with increasing load.

The primary factor limiting the achievable dilution level at higher loads is not combustion stability, but rather the onset of knock. As engine load increases, the knock limit shifts to occur at earlier spark timings, thereby restricting the permissible operating range. In other words, although the stability limit improves with load, the knock limit tightens more significantly, ultimately constraining the maximum level of dilution that can be applied.

Figure 5.6 and Figure 5.7 illustrate the effect of spark timing on combustion stability for varying dilution levels under low load conditions. For both lean and EGR-diluted combustion, a consistent trend is observed: advancing the spark timing improves combustion stability, indicated by a reduction in COV_{IMEP} . This trend aligns with observations made under stoichiometric conditions and reflects the benefit of enhanced thermodynamic conditions at earlier ignition phasing.

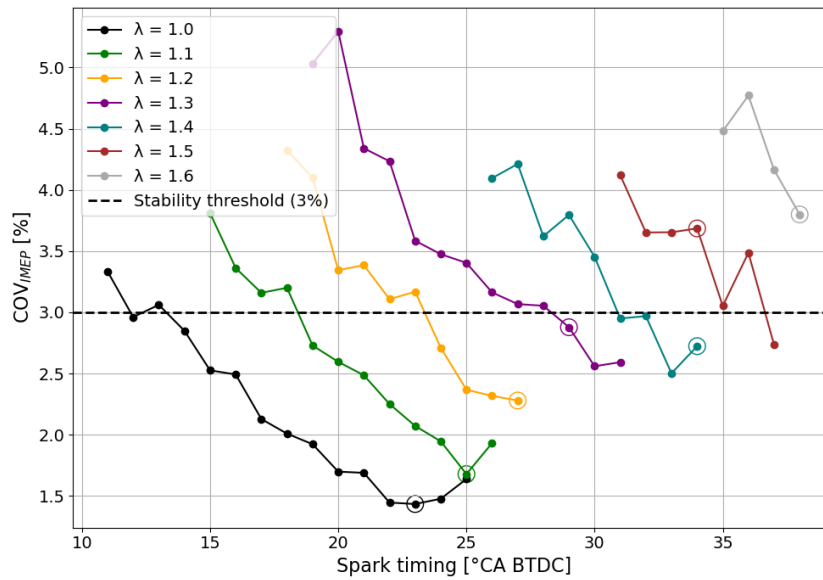


Figure 5.6: Effect of spark timing on COV_{IMEP} for lean burning at $BMEP = 3.5$ bar with the spark timing corresponding to peak BTE highlighted.

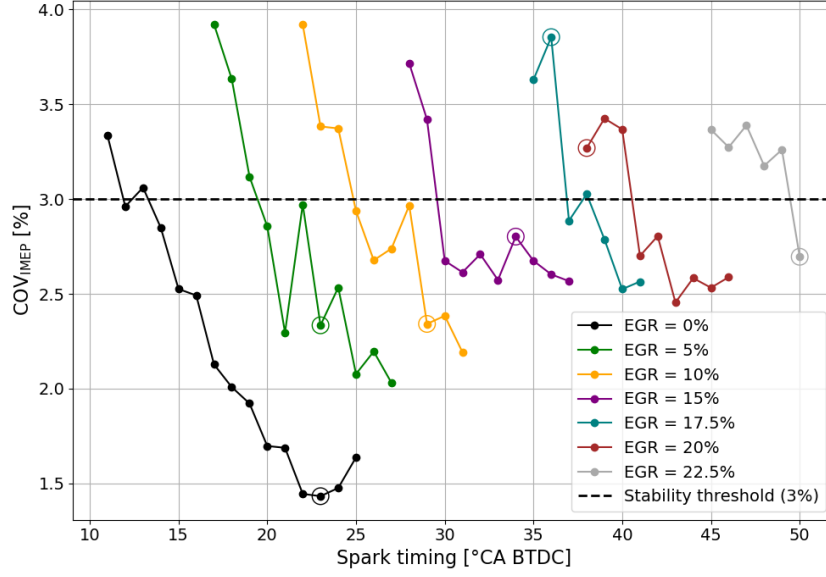


Figure 5.7: Effect of spark timing on COV_{IMEP} for EGR operation at $\text{BMEP} = 3.5$ bar with the spark timing corresponding to peak BTE highlighted.

Both figures confirm that increasing the dilution rate, either via excess air or EGR, generally leads to decreased combustion stability. While lean burn and EGR dilution produce qualitatively similar effects on COV_{IMEP} , the underlying mechanisms differ significantly, resulting in distinct operational characteristics and limitations.

An important distinction between the two strategies lies in the ability to advance the spark timing. As observed in both figures above and the broader operating range mapping results, EGR allows for a broader spark timing range than lean burn under equivalent conditions. This characteristic can be attributed to two factors favoring EGR dilution.

Higher knock mitigation capacity

Both dilution strategies effectively reduce the in-cylinder temperature and therefore suppress knock. However, this is done in two completely different ways. Lean burn reduces peak combustion temperatures through the thermal capacity of the excess air, while EGR reduces flame temperatures by displacing oxygen and increasing the heat capacity of the mixture with inert gases. Interestingly, Vancoillie et al. [87] reported that EGR has a higher potential to reduce peak in-cylinder temperature than lean combustion due to the inherently higher specific heat capacity of exhaust gases compared to air. At first sight, this claim is contradicted by the present experimental results. Figure 5.8 shows that at identical spark timing (with matching CA_{50}), the peak in-cylinder temperature is lower for lean combustion than for EGR dilution.

However, the comparison presented in Figure 5.8 does not reflect equivalent dilution levels between the two strategies. Specifically, the EGR case represents a dilution rate of 15%, whereas the lean case with $\lambda = 1.3$ corresponds to an air dilution of approximately 20.63% (as shown in Table 5.3). This discrepancy implies that the total mass, and thus the total heat capacity, is greater in the lean case, naturally leading to lower peak tem-

peratures. Therefore, a meaningful assessment of the relative thermal effects of EGR and lean combustion must be made at comparable dilution rates and identical combustion phasing (i.e., similar spark timing or CA_{50}).

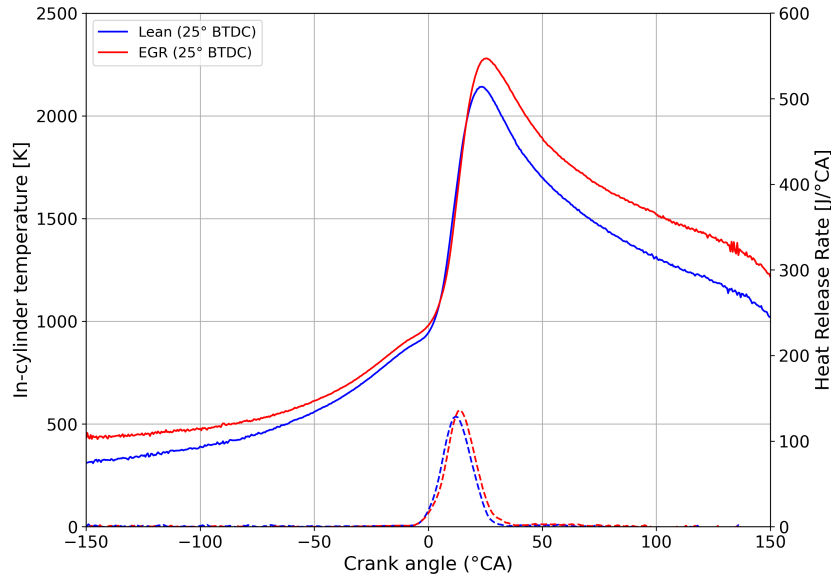


Figure 5.8: Effect of dilution on the in-cylinder temperature at the same spark timing (7.1 bar BMEP at 25°BTDC).

When this condition is satisfied, as illustrated in Figure 5.9 for medium and high load conditions, EGR demonstrates a greater capacity to suppress in-cylinder temperature than lean combustion. This finding aligns with the conclusions of Vancoillie et al., confirming that the superior specific heat capacity of exhaust gases plays a dominant role when dilution levels are properly matched.

Although in-cylinder temperature is decreased more effectively with EGR for a similar dilution level, it is observed that at lower EGR rates the knock limit is shifted further compared to lean, suggesting that an additional knock-mitigating mechanism must be considered. In EGR-diluted operation, inert exhaust gases replace part of the intake air volume, allowing the engine to operate at or near stoichiometric conditions. This reduces the availability of oxygen in the end-gas region and suppresses the formation of reactive radicals such as O, H, and OH [89]. As a result, the tendency for auto-ignition is significantly lowered providing superior knock resistance compared to lean burn.

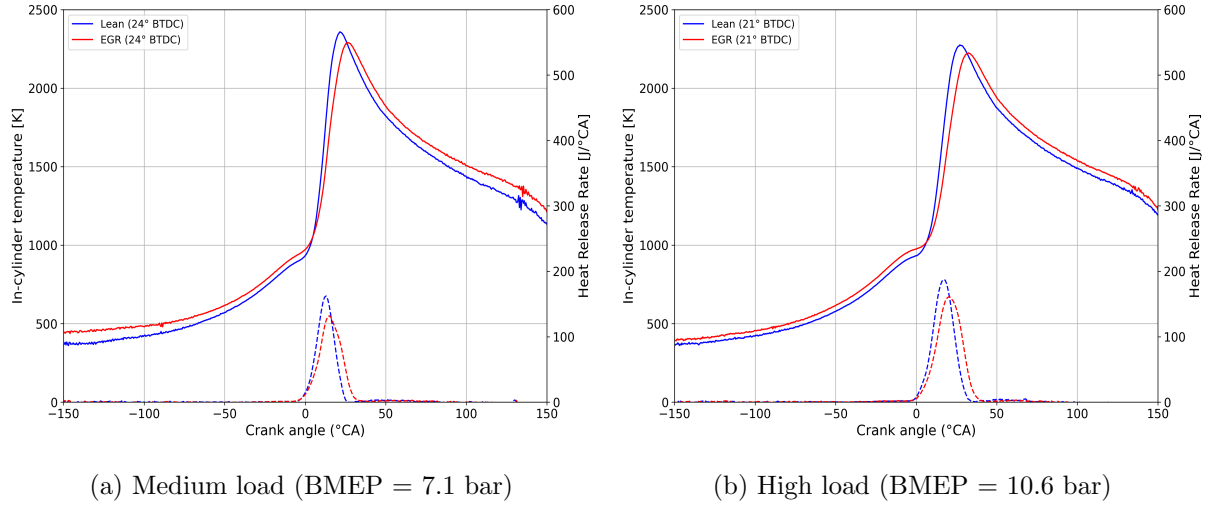


Figure 5.9: Effect of dilution ($\lambda = 1.2$ and EGR% = 15%) on the in-cylinder temperature.

Stable combustion at higher dilution rates

A comparison of Figure 5.6 and Figure 5.7 reveals that under EGR-diluted operation, a greater number of operating points remain below the COV_{IMEP} stability threshold at higher dilution rates. This trend is not limited to low load conditions. Figure 5.10 and Figure 5.11 show that this behavior persists across higher loads as well.

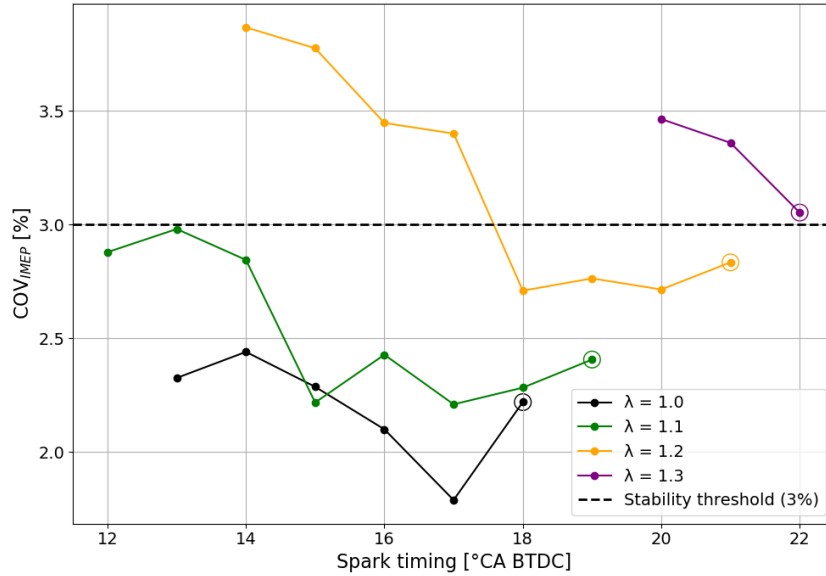


Figure 5.10: Effect of spark timing on COV_{IMEP} for lean burning at BMEP = 10.6 bar with the spark timing corresponding to peak BTE highlighted.

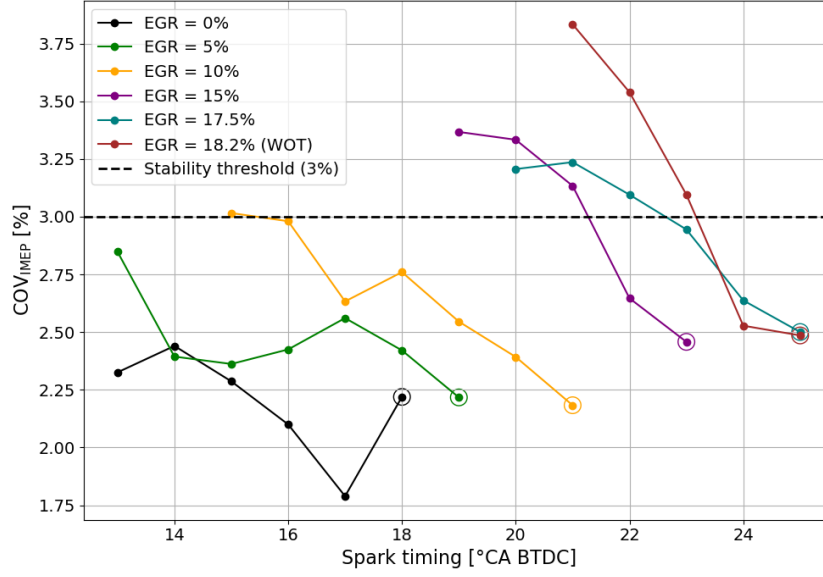


Figure 5.11: Effect of spark timing on COV_{IMEP} for lean burning at $BMEP = 10.6$ bar with the spark timing corresponding to peak BTE highlighted.

Zhu et al. [39] reported that increasing the EGR rate generally leads to decreased engine stability, which is consistent with the findings of this study. However, in their experiments, stable combustion was achieved even at EGR rates exceeding 30%, with COV_{IMEP} values remaining below 2%. This is in contrast to the present results, where instability was observed at significantly lower EGR levels.

A plausible explanation for the observed discrepancy in combustion stability between the present study and the results reported by Zhu et al. lies in the fundamental differences in engine architecture. The engine employed by Zhu et al. was originally designed for SI operation with natural gas, whereas the engine used in this thesis is a converted CI diesel engine that retains its original piston geometry and cylinder head configuration.

SI engines, particularly those optimized for gaseous fuels, are specifically designed to generate strong tumble motion during the intake and compression strokes. This is achieved through tailored intake port designs and combustion chamber geometries that promote turbulence near TDC, thereby enhancing flame propagation and combustion stability under diluted conditions [90]. In contrast, diesel engines are engineered to induce mostly swirl motion to support fuel-air mixing and control spray-wall interaction during direct injection [91]. While swirl is effective for stratified auto-ignition, it is suboptimal for stabilizing flame propagation in homogeneous SI combustion.

Consequently, the turbulence characteristics in the diesel-based configuration used in this work are not well-suited for spark-ignited combustion, particularly under high EGR conditions. This likely contributes to the increased cycle-to-cycle variation observed in this study and helps explain the superior dilution tolerance achieved in the SI-optimized engine used by Zhu et al.

To understand the underlying mechanisms responsible for the observed difference in stability between lean combustion and EGR-dilution, the combustion durations are examined, as combustion duration is known to correlate strongly with cycle-to-cycle variability. Figure 5.12 and Figure 5.13 illustrate the effect of increasing dilution, via excess air and EGR respectively, on the ignition delay and total combustion duration across different load conditions. The metrics evaluated include CA_{0-10} (ignition delay), CA_{50} (center of combustion), and CA_{10-90} (main combustion duration).

In both cases, the ignition delay and total combustion duration increase with greater dilution levels, see Table 5.3 for ease of comparison. These trends are consistent with the previously discussed decline in stability. The CA_{50} moves closer to TDC as dilution increases, reflecting the more advanced spark timings made possible by reduced knock tendency. Interestingly, the total combustion durations (CA_{10-90}) are found to be quite similar for both lean and EGR dilution strategies. This similarity suggests that further insights can be gained by isolating the ignition delay, which is particularly critical in determining combustion stability.

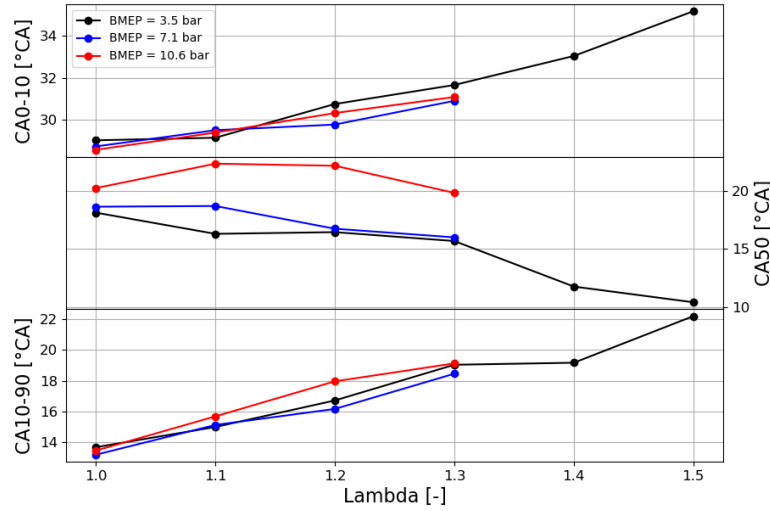


Figure 5.12: Effect of lean burn on combustion durations for different loads, with averaged durations over all stable measurements.

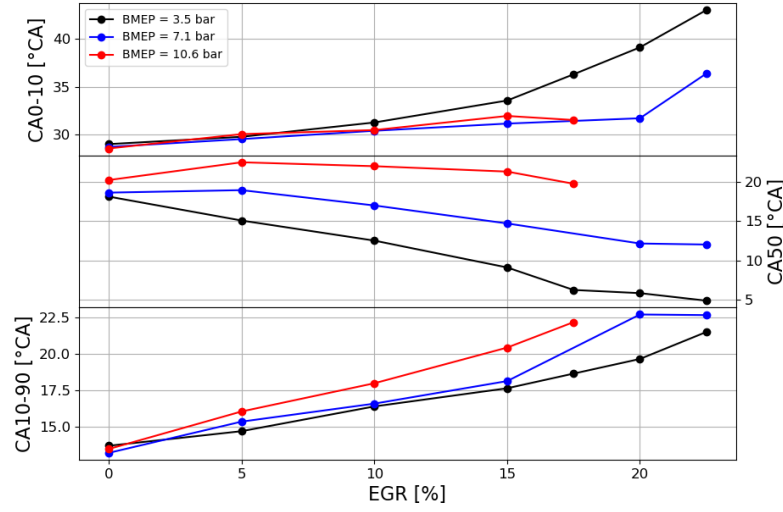


Figure 5.13: Effect of EGR on combustion durations for different loads, with averaged durations over all stable measurements.

Zhu et al. [39] argue that cycle-to-cycle variability is most sensitive to the early stages of combustion. In this context, the ignition delay (CA_{0-10}) becomes a key parameter. Comparing the two dilution methods, it is observed that EGR operation exhibits a steeper increase in ignition delay with increasing dilution. Vancoillie et al. [87] similarly reported that lean mixtures tend to burn more rapidly than EGR-diluted mixtures. However, this finding appears to contradict the earlier observation that EGR yields a more stable combustion process.

This contradiction can be resolved by considering additional contributing factors. First, EGR dilution allows for more advanced spark timings due to its stronger knock suppression effect, as discussed previously. As established in subsection 5.2.1, operating at advanced spark timings moves the main combustion closer to TDC, where the higher turbulence and higher temperature provide better stability. Second, EGR dilution enables the engine to operate at $\lambda = 1$, maintaining better flame propagation characteristics. In contrast, lean mixtures at high dilution levels experience increased susceptibility to misfires and partial burns, which severely impact stability. These phenomena result in an exponential increase in COV_{IMEP} under lean conditions.

Conclusion

In summary, both dilution strategies effectively reduce the in-cylinder temperature and suppress knock. Lean burn does this by the thermal capacity of excess air, while EGR reduces flame temperatures by increasing the heat capacity of the mixture with inert gases. EGR allows for a greater shift of the knock limit due to replacement of oxygen with inert gases, suppressing the formation of reactive radicals in the end-gas. In addition, EGR dilution behaves more stable, in contrast with lean mixtures where the lower fuel concentration increases the risk of partial burns and misfires at elevated dilution levels. As

a result, the combined effect of improved knock tolerance and sustained stability makes EGR a more robust dilution strategy for extending the engine's efficient and reliable operating boundaries. The efficiency implications of these findings are further analyzed in the following section, section 5.3.

5.3 Uncovering the Efficiency Potential of Lean Burn and EGR Dilution

As previously introduced in section 3.3, the potential efficiency benefits of in-cylinder charge dilution were outlined. In this section, the experimental results are presented and compared against assumptions and trends reported in the literature. Furthermore, a detailed comparison between the two dilution strategies, lean operation and EGR, is conducted, highlighting their respective impacts on engine performance.

5.3.1 Stoichiometric operation

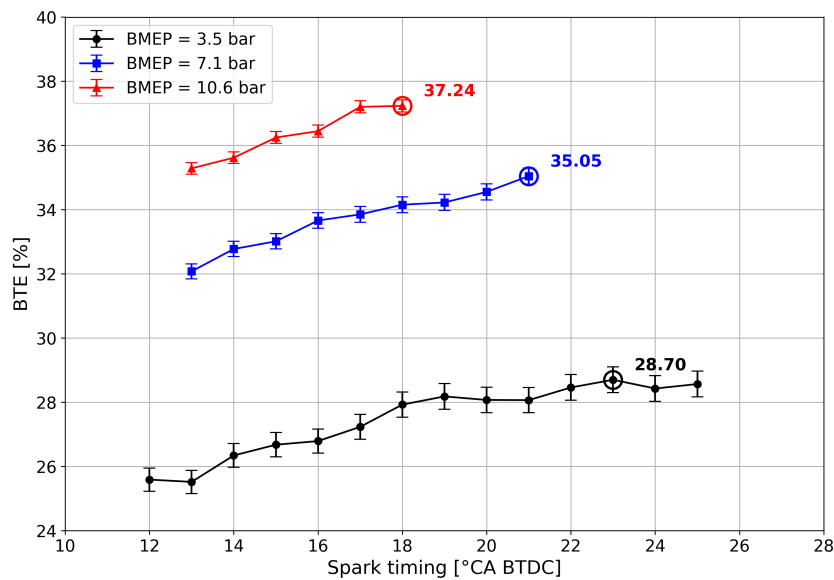


Figure 5.14: Brake thermal efficiency as a function of spark timing for different loads (BMEP) at stoichiometric operation.

Figure 5.14 presents the BTE under stoichiometric conditions as a function of spark timing, expressed in crank angle degrees before top dead center, across low, medium, and high load conditions. In the figure, the points of maximum BTE are encircled. Establishing this baseline is essential to facilitate a meaningful comparison with the subsequent dilution strategies discussed in this chapter.

Clear trends can be observed in Figure 5.14. For all load conditions, advancing the spark timing results in an increase in BTE. This behavior is expected, as advancing the ignition allows combustion to occur closer to TDC, where the piston is optimally positioned to convert the combustion pressure into mechanical work.

The knock-limited spark advance can also be identified on the graph. In this dissertation, KLSA is defined as the spark timing at which the onset of audible knock is detected, retarded by 1° crank angle to ensure safe operation. It can be observed that, at medium and high load conditions, the KLSA corresponds to the point of maximum BTE. This indicates that knock occurrence imposes a constraint on achieving the MBT timing at these loads, see Figure 3.1a. At low load, a knock margin of 2° CA is observed.

Table 5.4 provides an overview of the KLSA and spark timing (ST) for maximum BTE for different loads.

Table 5.4: KLSA for low, medium and high load operating points at stoichiometric combustion.

Operating point	BMEP [bar]	KLSA [$^\circ$ CA BTDC]	ST for BTE _{max} [$^\circ$ CA BTDC]
1500 RPM, 200 Nm	3.5	25	23
1500 RPM, 400 Nm	7.1	21	21
1500 RPM, 600 Nm	10.6	18	18

To gain a more detailed understanding of the in-cylinder phenomena associated with varying spark timing, Figure 5.16 is presented with both the in-cylinder pressure and the HRR as functions of the crank angle, for different spark timings at medium load conditions. In this figure, 0° CA corresponds to the TDC position. It is observed that advancing the spark timing leads to an increase in peak in-cylinder pressure, as well as a narrowing of the pressure trace. Furthermore, the peak HRR increases with advanced spark timing, but the combustion duration decreases, narrowing the HRR curve. The observed shift of the HRR peak toward TDC signifies a more favorable phasing of the combustion process. When combustion occurs closer to TDC, a larger fraction of the released chemical energy is transformed into useful mechanical work during the expansion stroke, thereby improving thermal efficiency. This optimal positioning minimizes the portion of energy lost to the cylinder walls and other heat transfer paths. However, excessively advancing the combustion timing can result in elevated in-cylinder temperatures prior to TDC. These higher temperatures increase the rate of heat transfer to the surrounding components, potentially leading to greater thermal losses. Consequently, there exists a trade-off between maximizing the conversion of combustion energy into mechanical output and minimizing heat losses. Similarly, the amount of heat leaving the exhaust pipe is also an indication of how much energy potential is wasted to the environment. In Figure 5.15, it is observed that the exhaust gas temperature increases for spark retardation. This confirms the ascending BTE seen in Figure 5.14.

Advancing the spark timing leads to higher efficiencies, however, the peak pressure observed at the KLSA remains significantly delayed, occurring at 17° after top dead center (ATDC). This indicates that the engine is still operating away from the optimal combustion phasing, or MBT timing. In comparison, under low load conditions (3.5 bar BMEP), the peak pressure at MBT timing (23° BTDC) is achieved earlier, at 14° ATDC, as illustrated in Figure 5.17. This comparison again confirms that at higher loads, the efficiency of the engine under stoichiometric combustion is constrained by knock, preventing operation at the true MBT point.

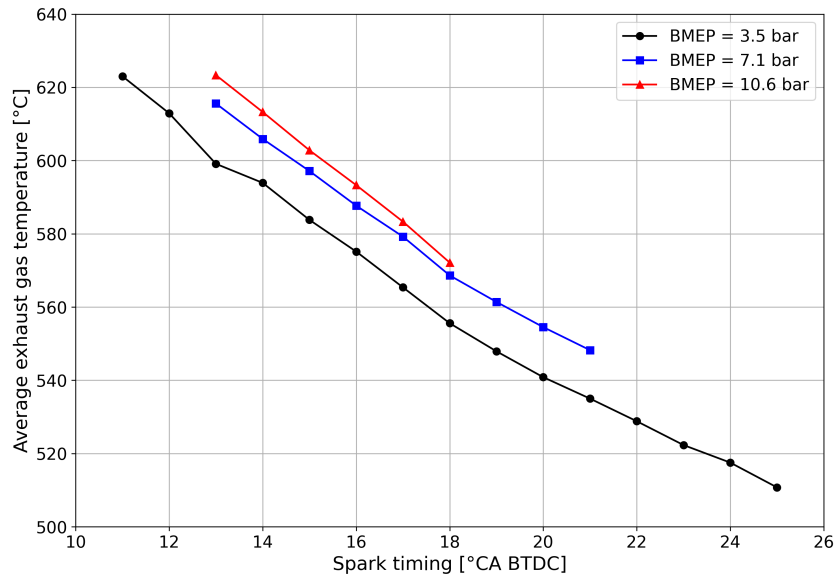


Figure 5.15: Average exhaust temperature as a function of spark timing for all loads (BMEP) at $\lambda = 1$.

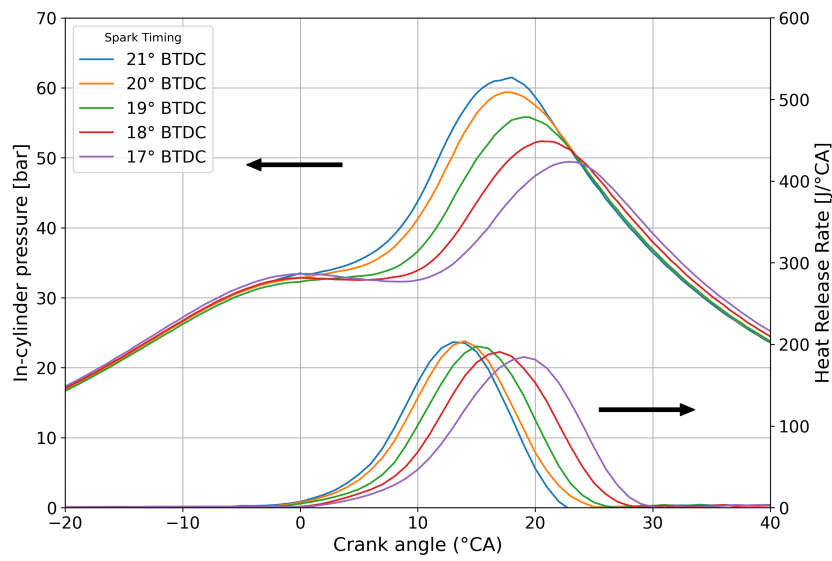


Figure 5.16: In-cylinder pressure and heat release rate as a function of degrees crank angle for different spark timings and a BMEP of 7.1 bar (1500 RPM, 400 Nm, $\lambda = 1$).

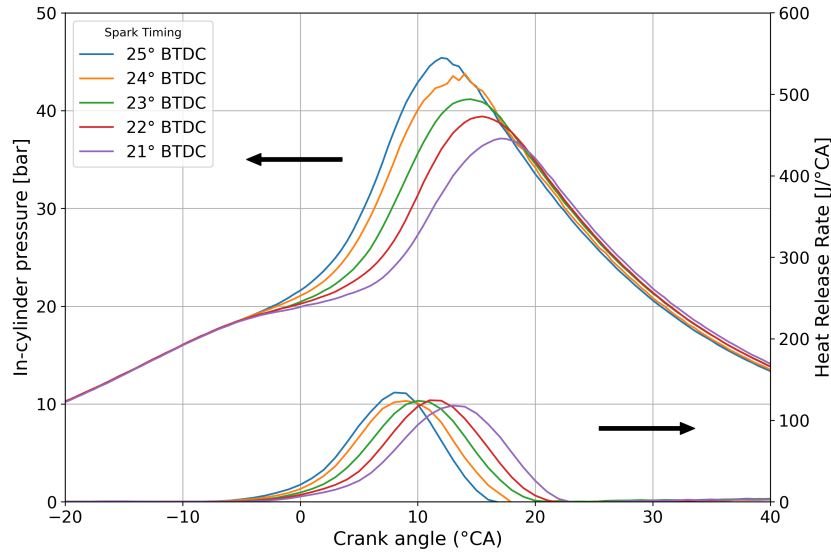


Figure 5.17: In-cylinder pressure and heat release rate as a function of degrees crank angle for different spark timings and a BMEP of 3.5 bar (1500 RPM, 200 Nm, $\lambda = 1$).

An increase in efficiency for higher loads is also observed, and can be allocated to several factors. First, when the load increases, the heat and mechanical losses relative to the power output become smaller. Additionally, at higher loads, the throttle valve opens more, reducing pumping losses. Because of the larger airflow in the engine at higher load, the turbo compressor also starts to work. The compressor then increases the mass of air in the engine cylinders, which further decreases the pumping mean effective pressure (PMEP).

The effects discussed above are also reflected in the Pressure-Volume (P-V) traces. Figure 5.18 displays the in-cylinder pressure as a function of the ratio of the working volume to the maximum cylinder volume for the operating points corresponding to the highest efficiencies, highlighted by the encircled points in Figure 5.14, at low, medium, and high load conditions. A detailed view of the negative work region, representing the pumping losses associated with throttling, is shown in the upper right corner of the plot. As load increases, the extent of this negative work area diminishes, indicating a reduction in pumping losses. This reduction contributes directly to the observed improvements in engine efficiency at higher loads. Furthermore, the combination of increased boost pressure from the turbocharger and a more open throttle valve leads to a rise in inlet manifold pressure, as also illustrated in Figure 5.18.

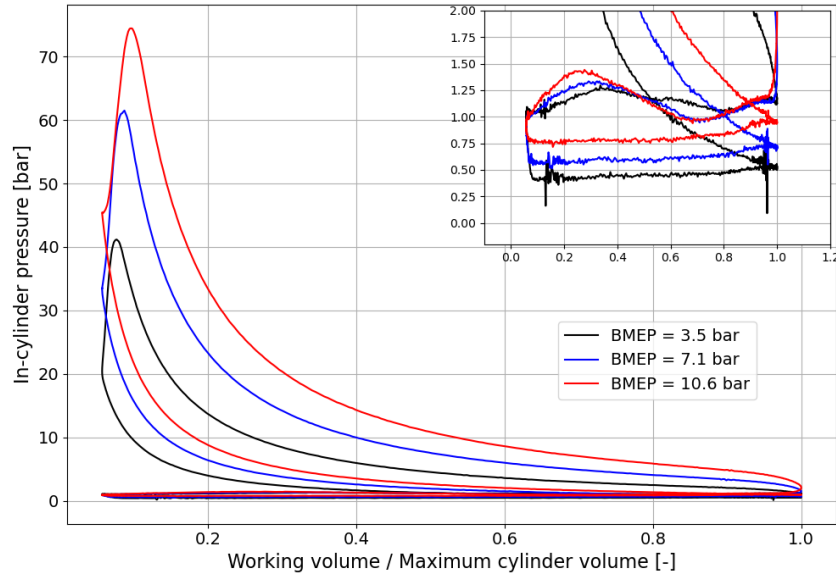


Figure 5.18: P-V diagram for different loads (BMEP), stoichiometric operation, and most optimal spark timing with a zoomed view of pumping losses.

5.3.2 Effect of dilution

Having established a clear stoichiometric baseline and examined the influence of spark advance and load variations, the next step is to investigate the impact of mixture dilution on the various engine efficiency parameters.

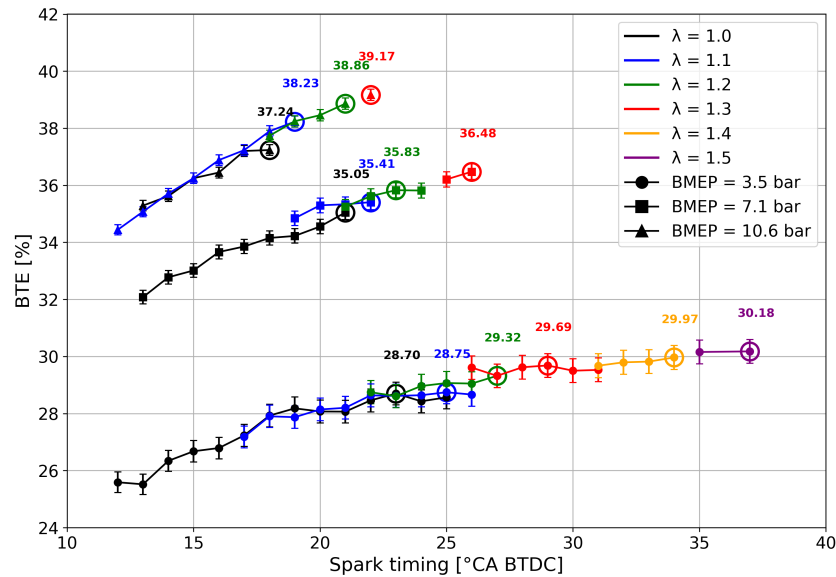


Figure 5.19: BTE as a function of spark timing for different loads and increasing lambda values.

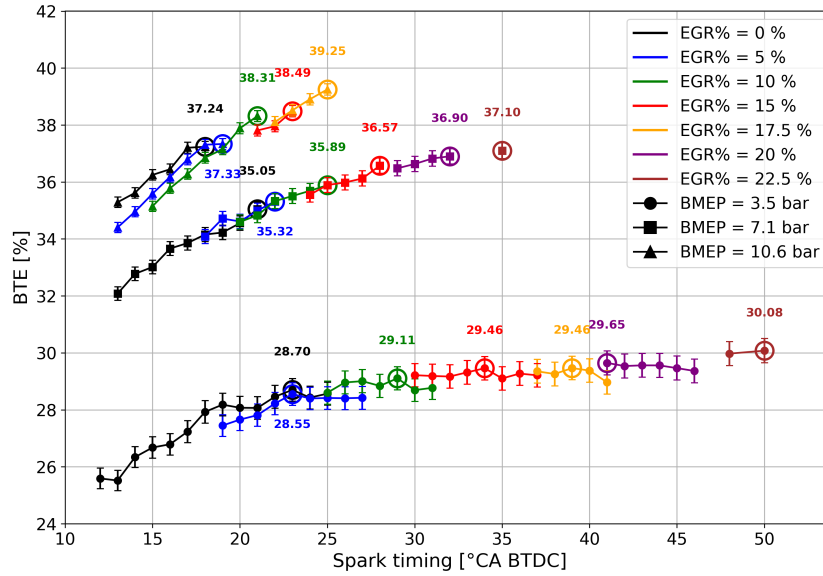


Figure 5.20: BTE as a function of spark timing for different loads and increasing EGR%.

Figure 5.19 and Figure 5.20 show the effect of increasing dilution on the BTE for multiple loads, in function of the spark timing. Similar effects for both dilution strategies can be observed: an increase in efficiency at higher dilution rates, at each load. The observed increase in engine efficiency with mixture dilution can be attributed to several interrelated factors, which are explained below.

Pumping losses

A primary contributor is the reduction in pumping losses. When operating with leaner mixtures or at higher levels of EGR, the throttle valve must be opened further to maintain the required mass of air entering the cylinder. This results in lower pressure drops across the intake system and thus reduced pumping work.

This trend is clearly demonstrated in Figure 5.21, where the area representing pumping losses on the P-V diagram is noticeably smaller for the diluted cases compared to the stoichiometric baseline. Furthermore, this observation is quantitatively supported by the analysis of the PMEP as a function of the dilution level, see Table 5.3 for ease of comparison. As shown in Figure 5.22, PMEP decreases progressively with increasing dilution, confirming the reduction in pumping losses.

The reduction in PMEP is most pronounced with EGR compared to lean operation. This indicates that, for the same load, EGR requires a larger throttle opening, thereby reducing throttling losses more effectively. The underlying reason is that EGR displaces part of the intake air with inert exhaust gases. Since the engine's oxygen demand remains unchanged, the intake system must compensate by increasing the intake manifold pressure to maintain the necessary air mass flow. Even if the EGR were at the same temperature as the intake air, additional throttle opening would still be needed to maintain oxygen supply. The effect is further amplified by the higher temperature of the recirculated gases, which lowers their density and thus requires a greater intake volume to deliver

the same oxygen content. In short, the combined influence of pressure and temperature on gas density means that EGR displaces air, but not the engine's oxygen requirement, necessitating a higher intake pressure and reducing PMEP.

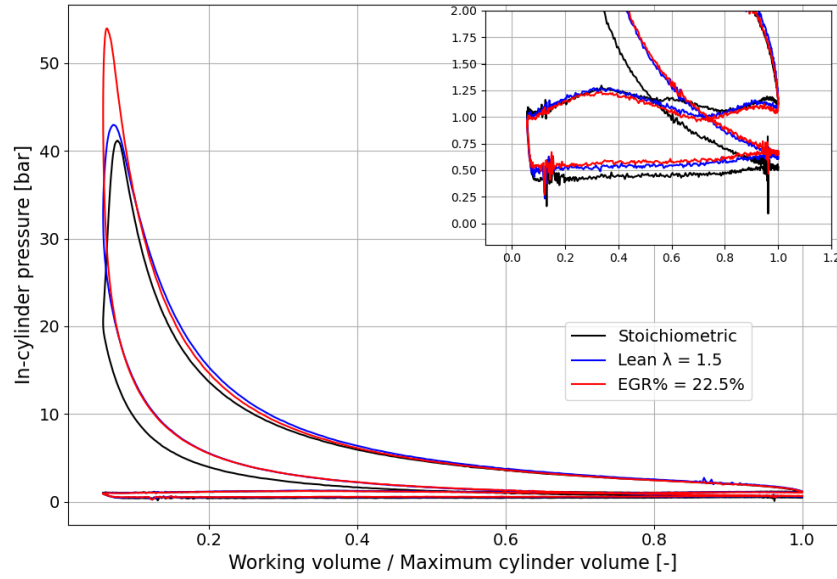
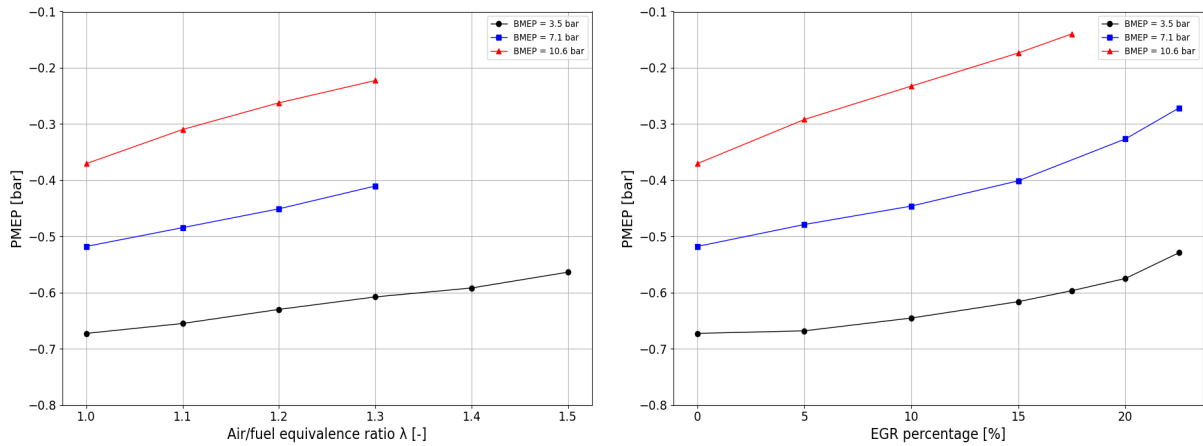


Figure 5.21: P-V diagram for the dilution strategies and most optimal spark timing with a zoomed view of pumping losses.



(a) PMEP as function of air/fuel equivalence ratio.

(b) PMEP as function of EGR percentage.

Figure 5.22: PMEP as function of amount of dilution.

Heat Release Rate

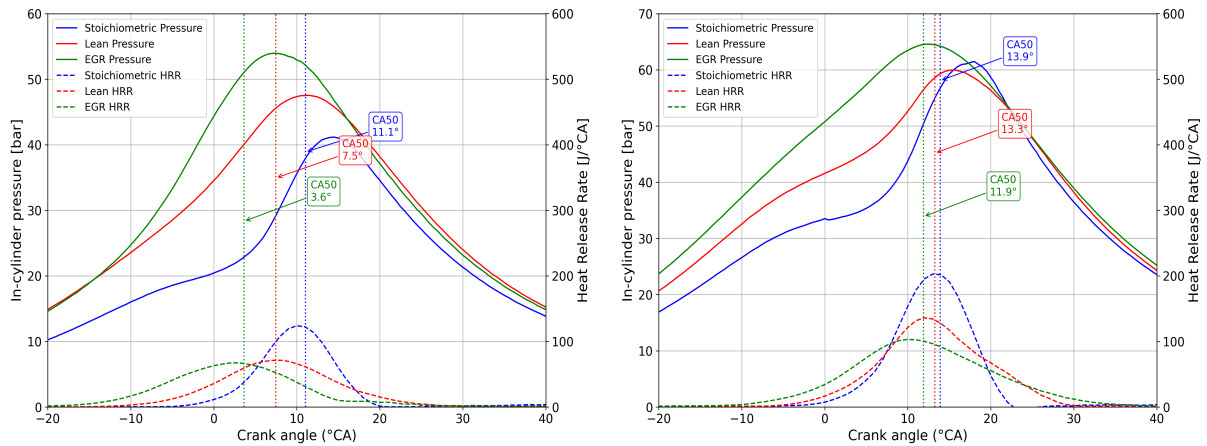
In addition to the reduction in pumping losses, dilution of the charge also leads to a noticeable decrease in peak in-cylinder temperatures. This temperature reduction has important implications for both heat transfer losses and endothermic dissociation losses within the combustion chamber [87]. The primary mechanism behind the lower temperatures is the increase in the total heat capacity of the in-cylinder mixture.

Both lean combustion, characterized by excess air, and EGR, characterized by the rein-

roduction of combustion products, contribute to this increase in total heat capacity. As a result, the charge can absorb a greater amount of heat during combustion without experiencing a significant rise in temperature. Notably, EGR has a particularly strong effect in this regard, as it introduces CO_2 and H_2O into the mixture, two components with excellent heat-absorbing properties, which further attenuate the temperature rise during combustion events.

This thermal buffering effect manifests clearly in the heat release profiles. As shown in Figure 5.23, the peak HRR is noticeably lower under both lean and EGR conditions when compared to stoichiometric operation. Among the two dilution strategies, EGR demonstrates a greater capacity for knock suppression, enabling more advanced spark timing. Consequently, the combustion phasing, defined by CA_{50} , shifts closer to TDC. This advancement results in a larger fraction of the heat release occurring before TDC, thereby increasing peak pressures and improving the effective expansion work.

Despite the overall reduction in HRR, the earlier combustion phasing enabled by EGR leads to higher peak in-cylinder temperatures compared to lean operation. This phenomenon is illustrated in Figure 5.23 and further supported by the temperature evolution shown in Figure 5.24. Although EGR generally exhibits a greater ability to reduce in-cylinder temperatures than lean combustion at equivalent dilution levels, as discussed in subsection 5.2.2, the increased spark advance permissible under EGR shifts the combustion closer to TDC. As a result, the associated rise in pressure and temperature outweighs the thermal buffering advantage, ultimately producing higher peak temperatures than those observed under lean conditions. Nonetheless, this observation confirms the effectiveness of EGR in managing in-cylinder thermal conditions and mitigating temperature-related losses.



(a) Pressure and HRR at low load (BMEP = 3.5 bar). (b) Pressure and HRR at medium load (BMEP = 7.1 bar).

Figure 5.23: Effect of dilution on the HRR and pressure trace for different loads.

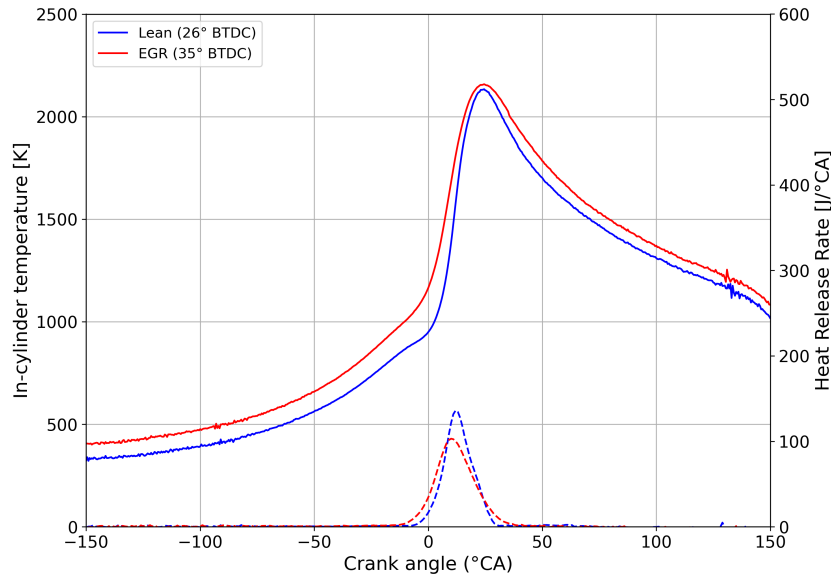
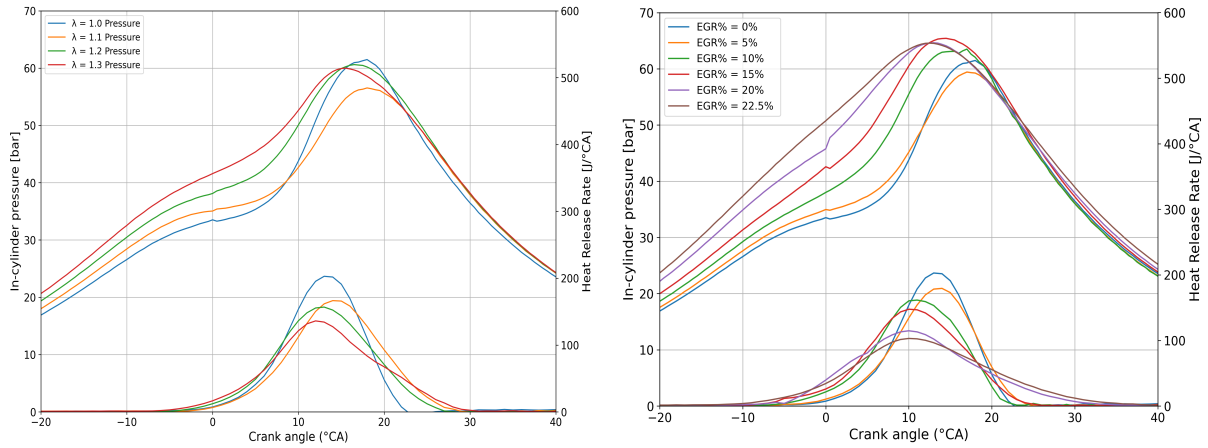


Figure 5.24: Effect of dilution on the in-cylinder temperature at medium load (BMEP = 7.1 bar).

The progressive decline in peak in-cylinder temperature as the level of dilution increases is clearly illustrated in Figure 5.25. This figure effectively demonstrates the relationship between the degree of dilution, whether through excess air or EGR, and the corresponding reduction in HRR, highlighting the thermal-moderating effect of both strategies.



(a) Pressure and HRR at medium load (BMEP = 7.1 bar) for lean burn. (b) Pressure and HRR at medium load (BMEP = 7.1 bar) for EGR.

Figure 5.25: Effect of dilution on the HRR and pressure trace for different dilution strategies.

Knock limit

A third major factor contributing to the observed increase in efficiency with dilution is the shift in the KLSA. The combined effects of reduced in-cylinder temperatures and more complete combustion work to lower the knock tendency of the engine.

Experimental results demonstrate that at low load conditions, knock can be fully suppressed through the application of sufficient dilution, particularly with the use of EGR, where spark timings as early as 50° BTDC were reached. At medium and high load conditions, the shift in KLSA remains especially beneficial, as it enables operation closer to the optimal combustion phasing, illustrated in Figure 5.23. As previously established in subsection 5.3.1, under stoichiometric conditions the engine was knock-limited from achieving MBT timing.

The effect of advancing the spark without inducing knock is also evident in the pressure traces shown in the figures above. These traces illustrate a clear shift of the peak pressure toward TDC, accompanied by smoother pressure profiles, characteristic of more favorable combustion phasing. However, an analysis of Figure 5.19 and Figure 5.20 reveals that at medium and high loads, the KLSA still coincides with the highest BTE point. This indicates that, despite the benefits of dilution, engine operation remains knock-limited even under elevated dilution levels.

All measurements were conducted up to the point at which further spark advance would have exceeded the defined stability threshold of 3% coefficient of variation. These findings suggest that knock could, in principle, also be eliminated at higher loads by applying additional dilution. However, this would come at the expense of combustion stability, highlighting the inherent trade-off between knock suppression and cyclic variability.

Impact of Specific Heat Ratio on Thermal Efficiency

subsection 5.2.2 concluded that EGR-dilution has a greater ability to shift the knock limit. Although spark timing can be advanced further under EGR conditions, the peak brake thermal efficiencies achieved with both EGR and lean operation are quite similar. Interestingly, for identical CA_{50} , lean combustion consistently yields a higher BTE compared to EGR. This finding aligns well with theoretical expectations.

If the combustion process is modeled as an ideal Otto cycle, the thermal efficiency can be expressed as:

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{\gamma-1}} \quad (5.2)$$

where:

- r = compression ratio
- $\gamma = \frac{C_p}{C_v}$ = specific heat ratio

From this relationship, it is evident that a higher γ leads to a higher theoretical efficiency.

Pure methanol vapor exhibits a relatively low specific heat ratio of approximately $\gamma \approx 1.26$, while air at atmospheric conditions has a significantly higher value of around $\gamma \approx 1.4$. When methanol is mixed with air to form an air-fuel mixture, the effective specific heat ratio of the mixture reaches approximately $\gamma \approx 1.35$. During combustion, the mixture produces typical exhaust gases such as CO_2 and H_2O , which have even lower specific heat

ratios, typically around $\gamma \approx 1.3$ or less.

As the air-to-fuel ratio increases under lean combustion conditions, the proportion of air in the mixture rises, resulting in an overall increase in the effective γ of the in-cylinder gas mixture. This increase in γ directly contributes to improved thermal efficiency under lean operating conditions.

In contrast, when EGR is employed, the introduction of CO_2 and H_2O into the fresh charge lowers the effective γ of the in-cylinder mixture. As a result, the potential efficiency gains from a higher γ are diminished. This explains why EGR, despite offering clear advantages such as knock suppression and an extended stable operating range, does not inherently improve thermal efficiency through the specific heat ratio effect.

The variation of γ across stoichiometric, lean, and EGR operation, as well as the influence of temperature, is illustrated in Figure 5.26. This figure provides clear experimental confirmation of the theoretical reasoning described above. Specifically, under lean conditions, γ increases compared to stoichiometric operation, whereas under EGR conditions, γ decreases. Furthermore, the figure demonstrates the temperature dependence of γ : as temperature increases, additional molecular degrees of freedom are activated, particularly rotational and vibrational modes. This increases the specific heat at constant volume (C_v), thereby reducing the specific heat ratio γ .

Interestingly, at the minimum point of the γ curves, the EGR trace rises above the stoichiometric trace. This observation can be attributed to the reduction in peak temperature associated with dilution effects. Due to the lower combustion temperatures under EGR operation, the effective γ appears higher, however, this must be interpreted in the context of the corresponding temperature evolution shown in Figure 5.27.

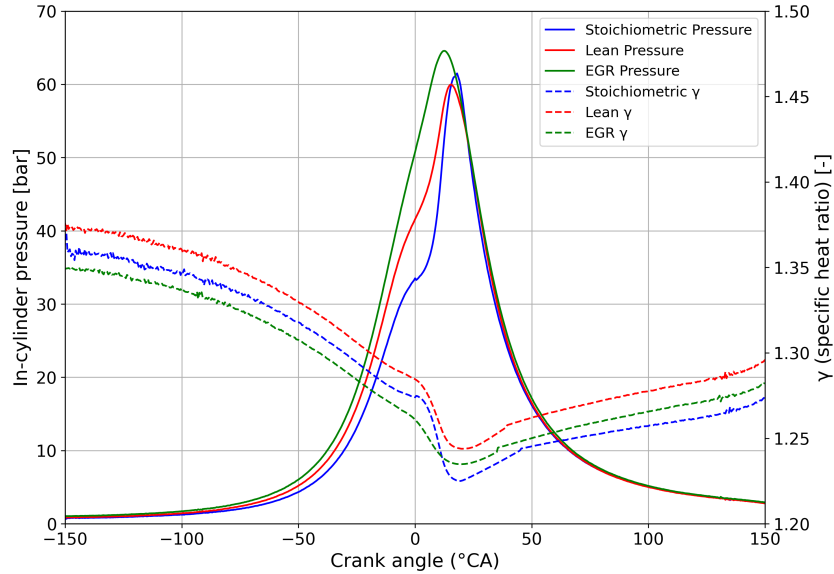


Figure 5.26: In-cylinder pressure and specific heat ratio for dilution at medium load.

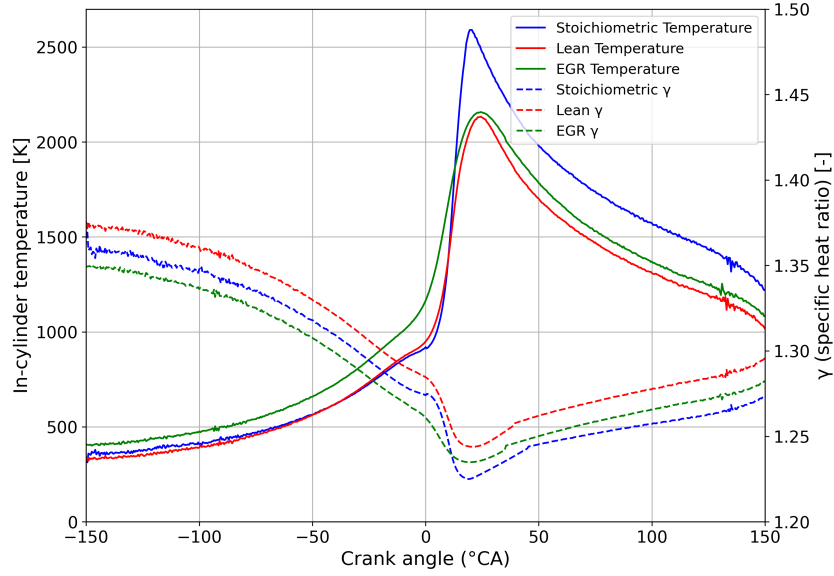


Figure 5.27: In-cylinder temperature and specific heat ratio for dilution at medium load.

Literature validation

The efficiency benefits of exhaust gas recirculation observed in the present study are consistent with the findings reported by Zhu et al. [39], who conducted an extensive evaluation of a heavy-duty spark-ignition methanol engine under varying EGR rates.

Their results confirm that increasing EGR leads to a reduction in pumping losses, as reflected by a drop in PMEP at low to medium loads. For example, at a BMEP of 0.87 MPa and 1200 RPM, PMEP decreased progressively as the EGR rate increased up to 31%. This corresponds to reduced throttling effort due to the greater intake charge mass flow at wider throttle openings.

Additionally, Zhu et al. report a significant decline in peak heat release rate with rising EGR levels, attributing it to slower flame propagation and increased specific heat capacity of the in-cylinder mixture. Despite this reduction in combustion intensity, thermal efficiency improved due to reduced heat transfer losses. Specifically, BTE increased from 35.8% at 8.8% EGR to 38.7% at 31% EGR at 1200 RPM, an absolute improvement of 2.9 percentage points. Efficiency improvements for this dissertation are summarized in Figure 5.28.

Finally, the knock suppression effect of EGR is clearly quantified in their study: advancing the spark timing from 33° to 41° CA BTDC was possible as the EGR rate increased from 25.9% to 31.8% at 1700 RPM, enabling operation closer to optimal combustion phasing and further enhancing efficiency. These observations provide strong experimental confirmation of the mechanisms underlying the efficiency improvements described in the present work.

Conclusion

The results of this investigation clearly demonstrate the beneficial impact of mixture dilution, through both lean combustion and EGR, on engine efficiency across a range of loads. The primary mechanisms driving this improvement include reduced pumping losses, attenuated peak heat release rates, and an extended knock-limited spark advance. These factors collectively enable operation closer to optimal combustion phasing while mitigating thermal and part-load losses.

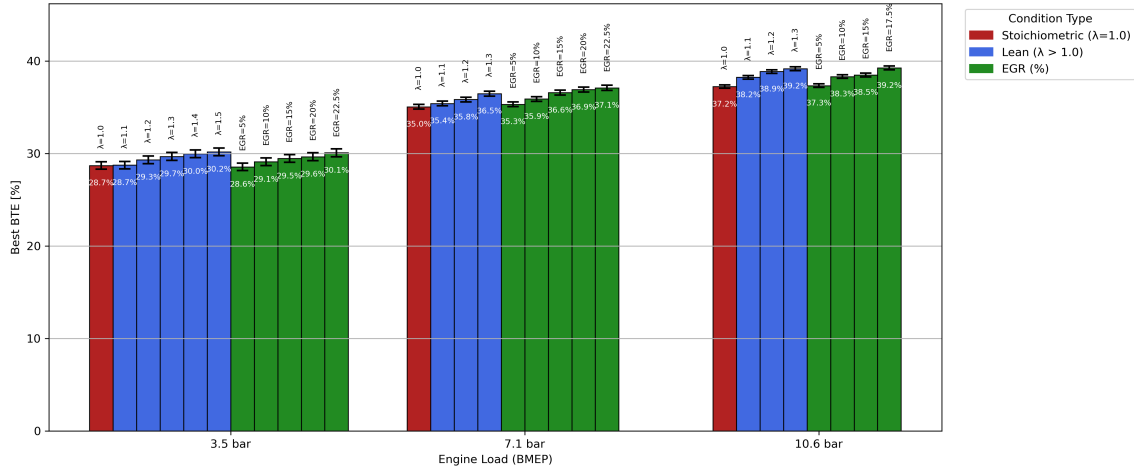


Figure 5.28: Best brake thermal efficiency (BTE) for each dilution strategy and engine load, under stable operating conditions ($\text{COV} \leq 3\%$).

Figure 5.28 summarizes the best BTE achieved under each operating condition and dilution strategy. It shows a consistent increase in BTE with higher levels of dilution at all load points. For example, at 7.1 bar BMEP, BTE increases from 35.0% (stoichiometric) to 37.1% at 22.5% EGR, while at 10.6 bar BMEP, the BTE improves from 37.2% to 39.2% at 17.5% EGR. Similar upward trends are observed under lean conditions, particularly at medium and high loads.

These findings confirm that both lean operation and EGR are effective strategies to improve efficiency in methanol-fueled SI engines. However, each strategy operates through distinct thermodynamic mechanisms and presents different trade-offs in terms of combustion stability, specific heat ratio, and knock mitigation.

5.4 Emissions

During all measurement campaigns, the exhaust gas composition was continuously monitored to evaluate the environmental impact and combustion characteristics of methanol in SI operation. Specifically, CO-, CO₂-, O₂- and NO-concentrations were measured using a Siemens Ultramat 23 gas analyzer. Due to internal overheating of the NO₂-NO converter of the emission cabinet, it was not possible to measure NO₂ or total NO_x emissions this year. Only NO-concentrations could be measured. More information about the emission cabinet can be found in appendix C of [21].

This section analyzes how the CO- and NO-emissions are affected by changes in spark timing, engine load, air/fuel equivalence ratio (λ) and the application of EGR. For each emission, trends are presented across the different combustion regimes - stoichiometric, lean and EGR - along with a discussion of the underlying mechanism influencing their formation.

5.4.1 NO emissions

Nitric oxide (NO) is the dominant component of NO_x emissions in SI engines and forms primarily through the thermal Zeldovich mechanism. As already discussed in subsection 3.2.4, the main parameters influencing NO_x-formation are in-cylinder temperature, oxygen concentration and high temperature residence time.

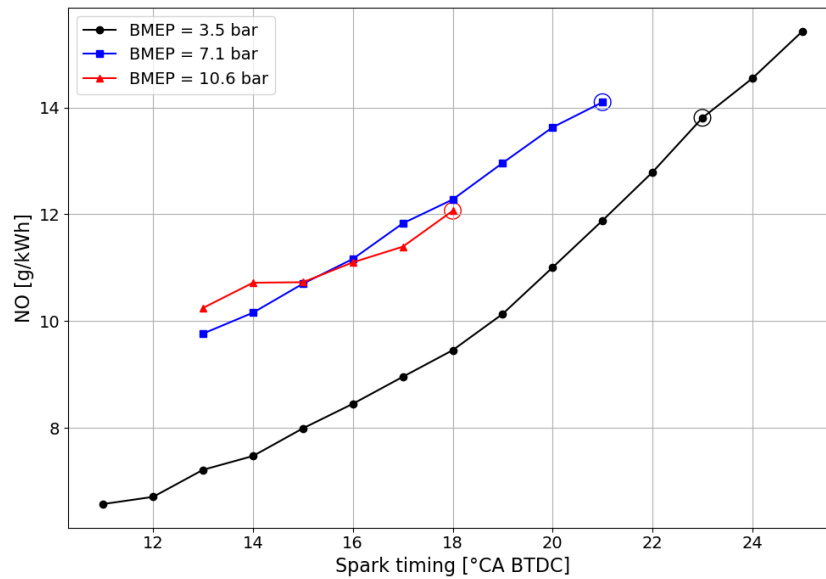


Figure 5.29: Influence of spark timing on NO-emissions across different loads at stoichiometric operation.

Figure 5.29 shows the brake specific NO emissions across different loads as a function of the spark timing. Advancing the spark timing results in combustion shifting closer to TDC and higher peak pressures and temperatures. As a result of higher peak in-cylinder temperatures, the NO emissions rise with advanced spark timing. This is verified by Figure 5.31 which shows the effect of the spark timing on the maximum in-cylinder tempera-

ture. At retarded spark timings, combustion occurs later and peak temperatures decrease, resulting in lower NO emissions. However, for the KLSA at low load, the NO emissions are higher than at medium and high load, despite the peak in-cylinder temperatures being lower. This is because earlier spark timing extends the high-temperature residence time, enabling increased NO formation. For the same spark timing, it is generally expected that higher NO emissions will be observed at higher loads due to increased peak in-cylinder pressures and temperatures. However, this trend is not evident in Figure 5.29 when comparing medium load with high load. On the other hand, when examining NO emissions across different loads at various dilution rates, a clear increase in NO emissions with increasing load is observed. This trend is exemplified in Figure 5.30, which shows a rise in NO emissions with increasing load under lean operation at $\lambda = 1.2$. The discrepancy with Figure 5.29 can possibly be the result of saturation temperatures for NO formation being reached. This is not the case when comparing loads at a certain dilution level, as in-cylinder temperatures are much lower there.

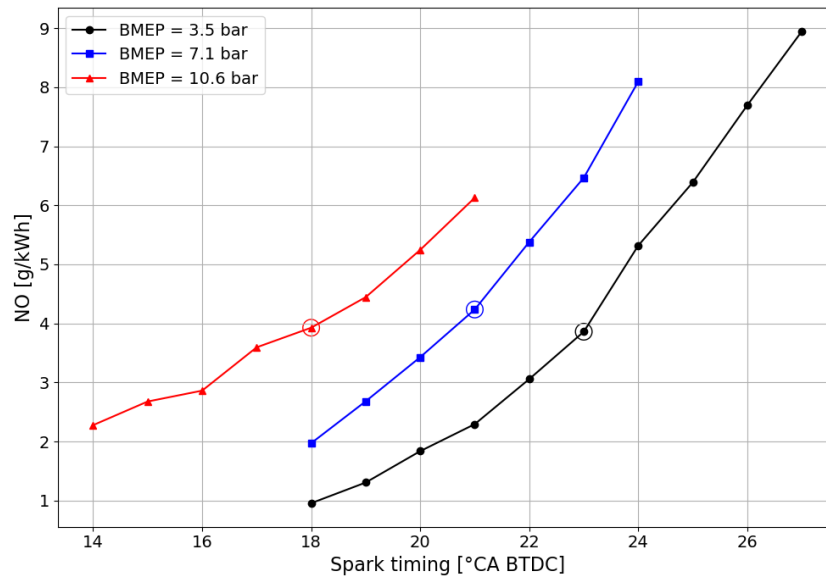


Figure 5.30: Influence of spark timing on NO-emissions across different loads at Lean operation ($\lambda = 1.2$).

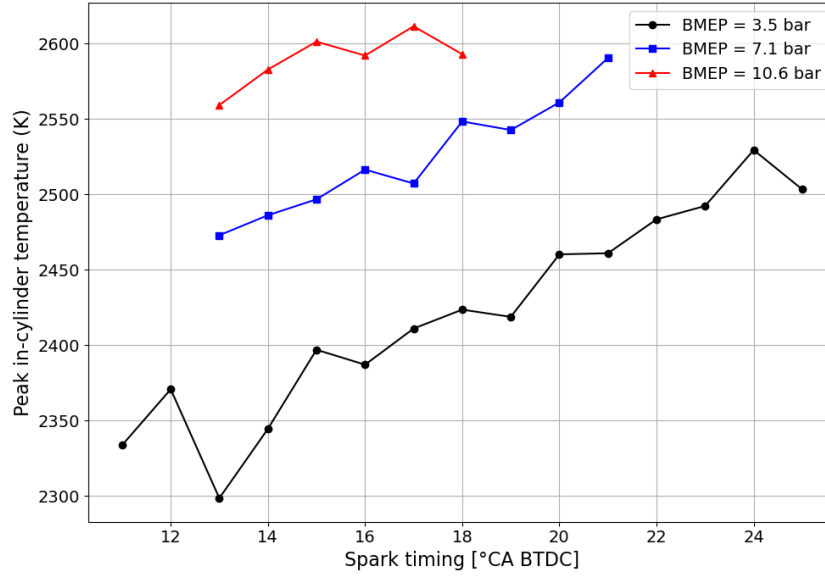
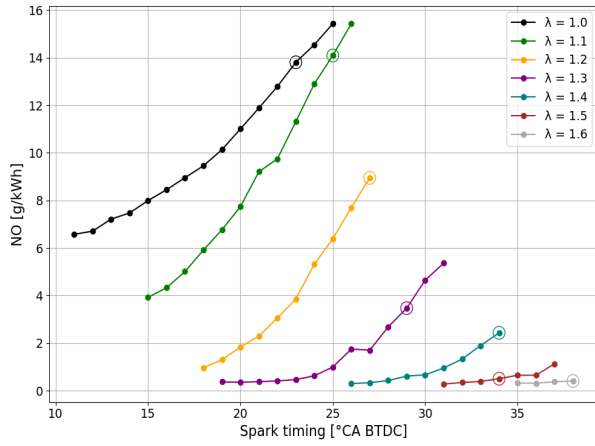
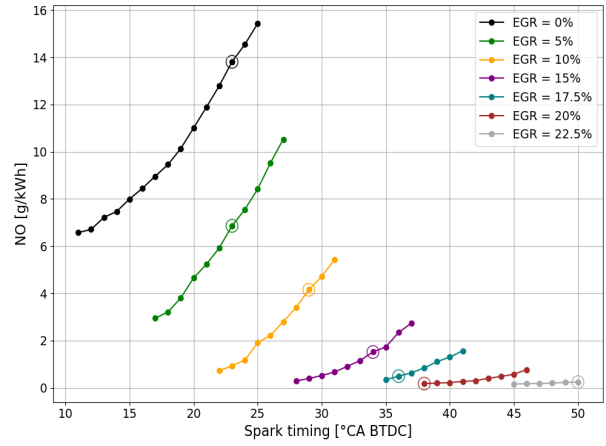


Figure 5.31: Peak in-cylinder temperature as a function of the spark timing for stoichiometric operation at different loads.

The effect of diluting the mixture on the NO emissions is visually represented in Figure 5.32. Both lean burn and EGR significantly suppress NO emissions. The excess air in lean operation lowers the overall in-cylinder temperature, visually represented in Figure 5.33a, resulting in lower NO emissions. The prime effect of EGR operation is the presence of inert gases with high specific heat capacity. They take up part of the combustion heat, resulting in decreasing peak in-cylinder temperatures and decreasing NO emissions. This is verified by Figure 5.33b showing the peak in-cylinder temperature as a function of the spark timing for different EGR percentages. At same dilution rates (see Table 5.3), EGR is more effective due to its additional impact on the in-cylinder oxygen concentration. While lean combustion increases the air-to-fuel ratio, thereby reducing flame temperature to some extent, it still results in a relatively high oxygen availability, which supports NO formation. EGR operation, on the other hand, has lower oxygen concentration available for NO formation compared to lean operation. In addition, EGR has a higher effect of reducing in-cylinder temperatures at equal dilution rates. The influence of dilution strategies on NO emissions at 7.1 bar BMEP and 10.6 bar BMEP is given in Appendix K. The discussed trends are the same.

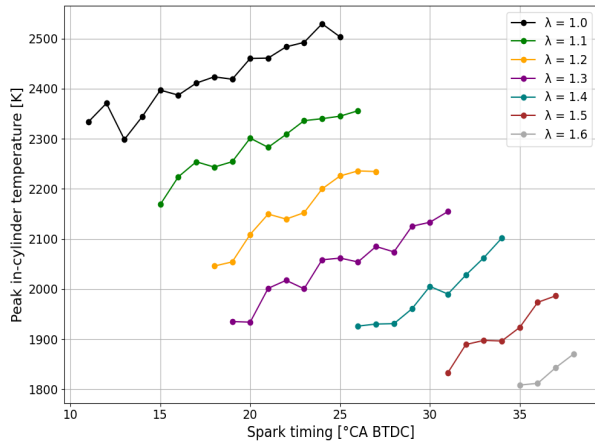


(a) Lean operation

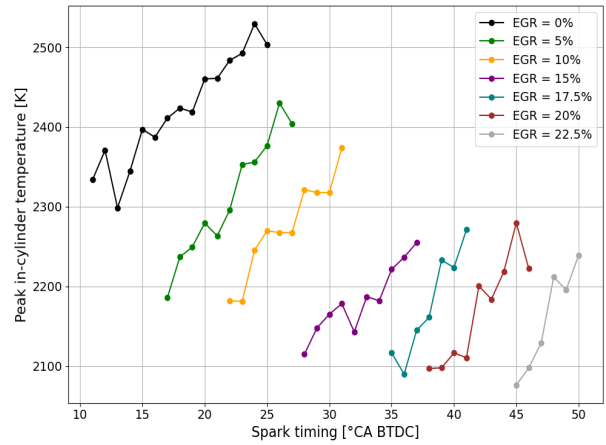


(b) EGR operation

Figure 5.32: Influence of dilution strategies on NO-emissions at 3.5 bar BMEP.



(a) Lean operation



(b) EGR operation

Figure 5.33: Influence of dilution strategies on peak in-cylinder temperatures at 3.5 bar BMEP.

Marine engines, these days, are required to meet Tier III NO_x emissions standards. Figure 5.34 presents MARPOL Annex VI NO_x emission limits [92], which are applicable to marine diesel engines under various Tier categories. While these limits are not directly applicable here as also total NO_x emissions could not be measured this year, they are used as a reference benchmark to assess the relative NO_x performance of the proposed engine concept. It can be seen that points corresponding with max BTE at stoichiometric combustion, without aftertreatment system, don't meet the Tier III. However, with increasing dilution, as can be seen in Figure 5.32, the criteria will probably be met as NO is the dominant component in NO_x . The results from Appendix K especially show that at medium and high load, EGR is the more favorable solution to reduce NO emissions and meet the Tier III NO_x emissions standards.

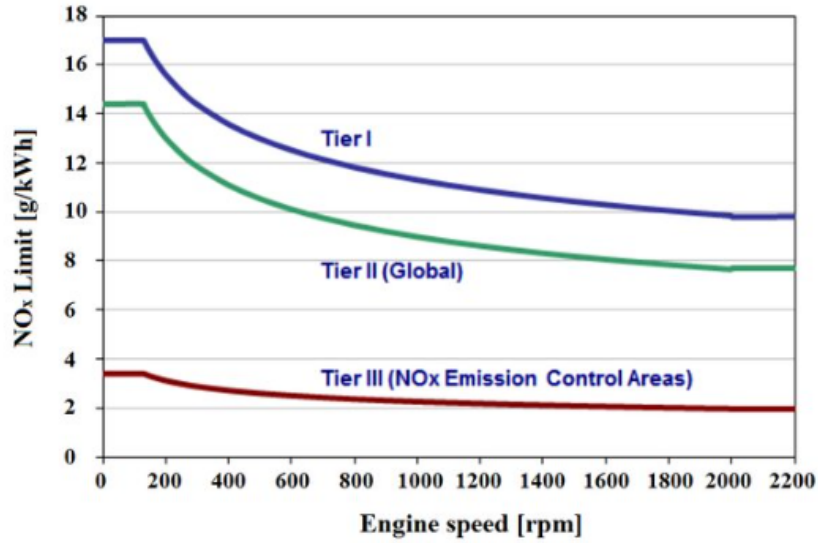


Figure 5.34: NO_x emission limits as a function of the engine speed, as defined by MARPOL Annex VI [92].

5.4.2 CO-emissions

Carbon monoxide (CO) emissions are the result of incomplete combustion in ICEs. It is formed when the fuel only partially oxidizes to CO_2 , which typically occurs if there is insufficient oxygen or inadequate time or temperature to complete the CO-to- CO_2 oxidation process. This subsection discusses different parameters (spark timing, load and dilution strategies) influencing the CO emissions measured during the experiments.

Figure 5.35 shows the effect of spark timing and load on the brake specific CO emissions. At low load (3.5 bar BMEP), an erratic pattern is observed, indicating the absence of a clear specific trend of spark timing on the CO-emissions. The fluctuating behavior observed at low load may be attributed to the formation of local fuel-rich regions at different spark timings, which have a more significant impact on the effective CO emissions than the direct effect of spark timing itself. At medium and high load, a small trend is visible with decreasing CO emissions for advanced spark timing. This can be the result of a more complete combustion process and more time to complete the CO-to- CO_2 oxidation process.

The results of Figure 5.35 also confirm the effect of load on the CO emissions. As already stated in section 3.2.4, the CO content is increased at low loads as a result of the higher latent heat of evaporation of methanol limiting the oxidation process of CO. This was also confirmed by the experiments of Zhu et al. [39].

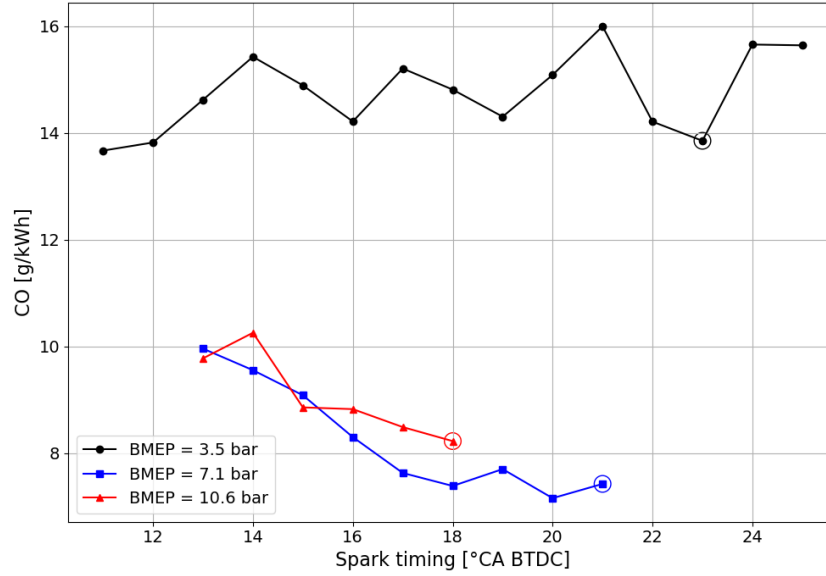


Figure 5.35: Influence of spark timing on CO-emissions across different loads.

The influence of the two different dilution strategies (lean and EGR) on CO-emissions is represented in Figure 5.36. It can be observed that both strategies significantly suppress CO-emissions. For lean operation at an air-fuel equivalence ratio of 1.1, the CO-emissions are 3 to 4 times less compared to stoichiometric. It can be noticed that spark timing, in this case, does not have a big influence on the CO emissions. As already stated above, the effect of the air-fuel equivalence ratio (λ) has a more pronounced effect compared to ignition timing.

With an increasing EGR rate, CO emissions tend to decrease due to the reduction in the level of dissociation of CO_2 because the combustion temperature decreases [93], as can be verified in Figure 5.33b. This is also verified by the results of Zhu et al. [39]. However, from a certain EGR rate on, the CO emissions do not decrease anymore and even tend to increase. This is a result of peak combustion temperatures not significantly decreasing anymore and the partial oxidation of unburned methanol from the crevices [39]. This critical EGR rate, at 3.5 bar BMEP, is taken here as 17.5%. For low EGR rates (i.e. smaller than 5%), the CO emissions as a function of the spark timing are still very volatile as a result of stoichiometric combustion being not perfectly stoichiometric. At higher EGR rates, advancing the ignition timing leads to increased CO emissions due to enhanced dissociation of CO_2 at elevated combustion temperatures. Additionally, the lower exhaust temperature promotes increased post-oxidation of the unburned methanol [39]. The graphs showing the influence of dilution strategies on CO emissions at 7.1 bar BMEP and 10.6 bar BMEP are again given in Appendix K.

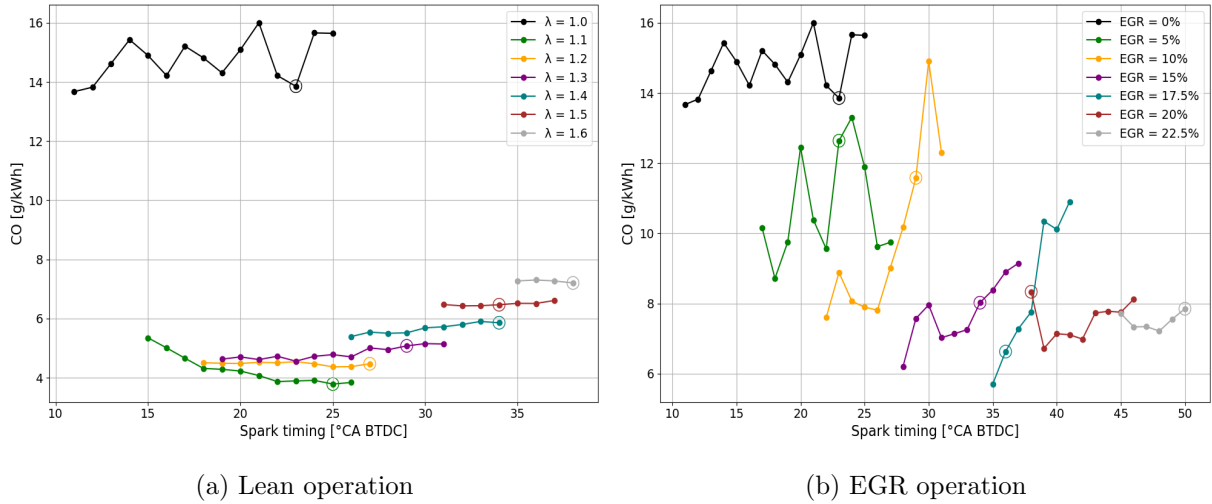


Figure 5.36: Influence of dilution strategies on CO-emissions at 3.5 bar BMEP.

Conclusion

Dilution strategies, both lean burn and EGR, effectively suppress NO emissions by lowering peak temperatures, with EGR achieving this through heat absorption by inert gases. EGR suppresses NO emissions more compared to lean combustion because of the lower in-cylinder temperatures at same dilution rates and the lower oxygen concentration. CO emissions are influenced by incomplete combustion, with higher values at low loads due to the high latent heat of evaporation of methanol. Both dilution methods also reduce CO emissions, though EGR eventually reaches a threshold where CO begins to rise again due to partial oxidation of unburned methanol.

The aftertreatment systems of both dilution strategies also differ markedly. Lean operation fundamentally inhibits the use of a three-way catalyst (TWC) for NO_x reduction because the excess oxygen in lean exhaust prevents the simultaneous reduction of NO_x and oxidation of CO that a TWC requires. Under lean burning, conventional TWCs will preferentially oxidize CO instead of reducing NO_x, leaving most NO_x largely untreated. As a result, meeting stringent NO_x standards with lean combustion necessitates additional NO_x aftertreatment devices which add extra cost and complexity to the engine.

In contrast, EGR dilution preserves a stoichiometric exhaust, which is fully compatible with a TWC. The significance of this cannot be overstated. Recent heavy-duty engine developments favor this approach: stoichiometric combustion with EGR and a TWC has been shown to attain ultra-low emissions, avoiding the need for extremely complex NO_x aftertreatment systems [94].

The effectiveness of TWCs is highly dependent on temperature, with a typical light-off temperature ranging from 200°C to 300°C, below which their catalytic activity is significantly limited [95]. Figure 5.37 demonstrates the minimum exhaust temperatures of different cylinders as a function of EGR rate across various operating conditions. For each EGR rate and engine load, the spark timing resulting in minimum exhaust temperature was chosen. Notably, all recorded exhaust temperatures exceed the TWC light-off threshold, indicating that the catalyst should maintain effective conversion of NO_x, CO,

and unburned hydrocarbons.

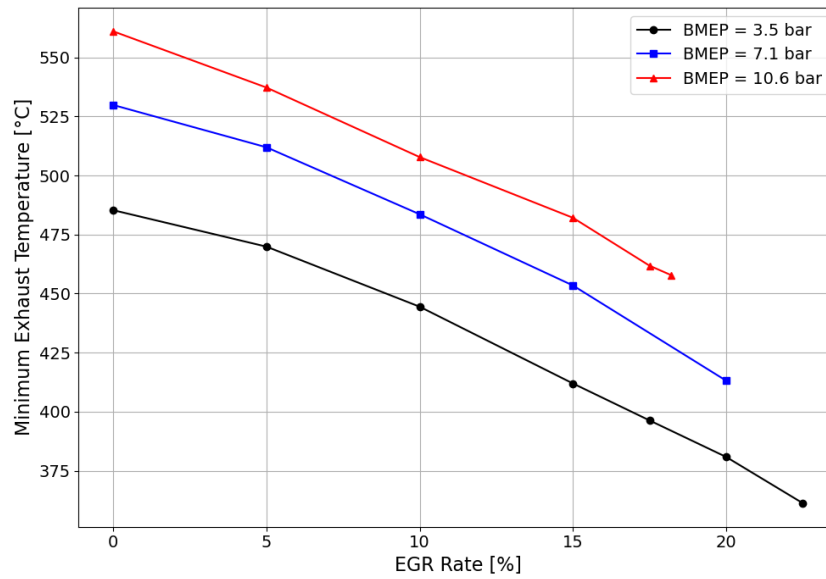


Figure 5.37: Minimum exhaust temperature across different cylinders and spark timings as a function of EGR rate.

5.5 Exploration of Wide Open Throttle (WOT) Operation

In addition to the primary measurement campaigns described earlier, an exploratory measurement series was conducted to evaluate the feasibility and implications of operating the engine at wide open throttle (WOT) under lower load conditions. This investigation was motivated by observations made during the high-load EGR experiments, where it was noted that the throttle valve approached a nearly fully open position at the dilution limit. This observation suggested the potential benefit of extending WOT operation into the part-load regime, potentially improving efficiency by further reducing throttling losses.

To examine this concept, a dedicated test series was carried out at a constant engine speed of 1500 RPM, with the throttle held fully open throughout. As the engine load was progressively reduced, the quantity of injected fuel was correspondingly decreased. To maintain stoichiometric conditions ($\lambda = 1$) despite the reduction in fuel flow, the EGR rate was increased proportionally. This method enabled WOT operation at lower brake torques by using EGR as a control variable to balance the air–fuel mixture.

It was recognized in advance that this approach would result in increased cycle-to-cycle variability due to the high levels of dilution required. Therefore, the conventional stability threshold of a coefficient of variation (COV_{IMEP}) below 3% was intentionally relaxed for this exploratory study. The objective was to observe how rapidly combustion instability would rise under extreme dilution conditions.

Measurements were performed at brake torques of 600 Nm (BMEP = 10.6 bar), 550 Nm (BMEP = 9.7 bar), 500 Nm (BMEP = 8.8 bar), 450 Nm (BMEP = 7.9 bar), and 400 Nm

(BMEP = 7.1 bar). As anticipated, the COV_{IMEP} increased progressively with decreasing load. At 400 Nm, the COV exceeded 10%, indicating a level of instability that could pose a risk to engine durability. Consequently, the test series was terminated at this point. The increase in instability is illustrated in Figure 5.38.

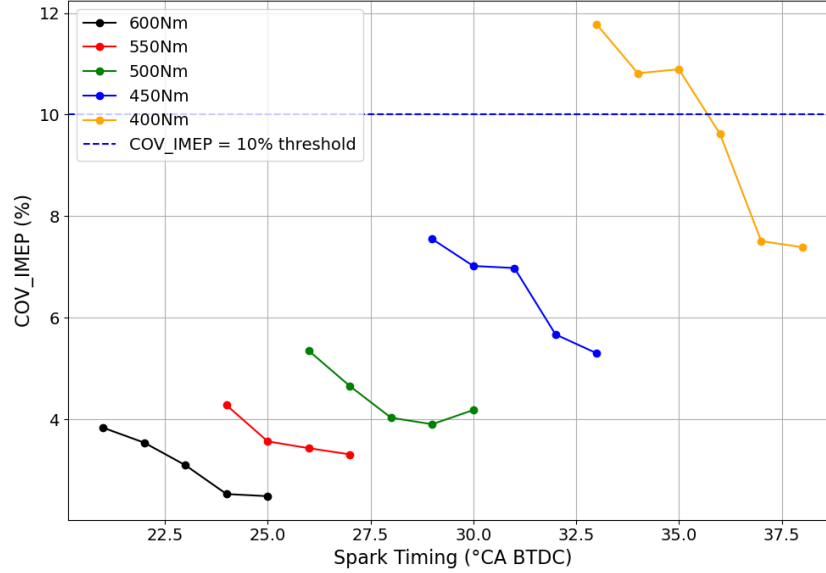


Figure 5.38: COV_{IMEP} for different loads at WOT.

The objective of this measurement series was to investigate the potential efficiency gains at part-load conditions by implementing WOT operation combined with EGR-based dilution control. The only valid point of reference for this analysis is the earlier EGR measurement at 400 Nm, which adhered to a combustion stability limit of $CoV_{IMEP} \leq 3\%$.

A comparison between the conventional and WOT-based EGR strategies at 400 Nm reveals a modest but measurable improvement in brake thermal efficiency. As shown in Figure 5.39, the WOT approach yielded an increase of approximately 0.6% in BTE. This improvement is primarily attributed to the ability to advance spark timing further due to the higher EGR rate employed (25% under WOT, compared to 22.5% under stability-constrained conditions). The elevated dilution level effectively extended the knock limit, allowing combustion phasing closer to the optimal MBT point.

While the knock suppression effect appears to be the dominant mechanism behind the observed efficiency gain, it should also be noted that reduced throttling losses likely contribute more significantly at lower engine loads. As discussed previously and shown in Figure 5.22, the relative importance of pumping losses increases as load decreases, indicating that the benefits of WOT operation could be more pronounced at even lower torque levels.

Although the current measurements did not extend into very low load operation, the results validate key observations from earlier sections. In particular, they confirm that higher dilution rates can further shift the knock limit, enabling more favorable combustion phasing. However, this benefit comes at the cost of increased combustion instability. The rise in COV_{IMEP} above acceptable levels at higher EGR rates highlights the need for a

stability enhancement mechanism.

Future improvements in combustion stability, such as through the implementation of passive prechamber ignition, could enable the use of even higher dilution rates under WOT conditions. This, in turn, would allow further efficiency improvements at part load and bring methanol-fueled spark-ignition engines closer to the thermal efficiency levels typically associated with diesel combustion.

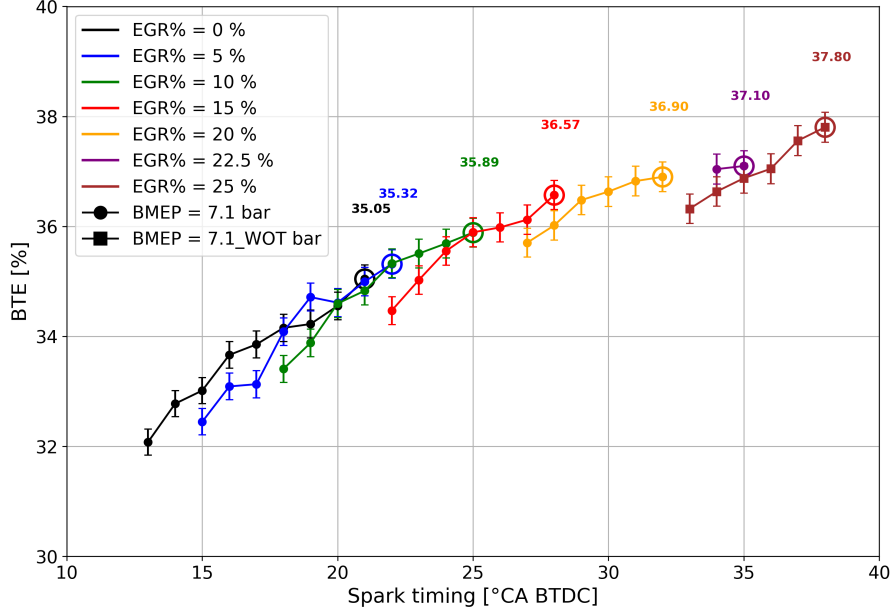


Figure 5.39: Comparison of BTE at 400 Nm with conventional EGR control ($\text{COV}_{\text{IMEP}} \leq 3\%$) versus WOT operation with extended dilution.

5.6 Conclusion: Lean Versus EGR Dilution

This section synthesizes the key findings from chapter 5 and offers a comparative evaluation of the two investigated charge dilution strategies: lean combustion and EGR. Both approaches were studied across a range of load conditions to understand their influence on engine performance, efficiency, and combustion behavior. The objective is to determine which strategy presents the most promising path forward for high-efficiency methanol-fueled spark-ignition engines.

5.6.1 Knock Suppression and Combustion Phasing

EGR provides a significantly stronger knock mitigation effect compared to lean operation. This is primarily attributed to the oxygen displacement and radical quenching characteristics of recirculated exhaust gases. Consequently, spark timing can be advanced further with EGR, bringing CA_{50} closer to TDC and increasing peak pressure and power output.

Lean combustion also delays knock onset due to charge cooling from excess air. However, at higher dilution levels, the elevated oxygen concentration in the end-gas increases its

reactivity, which can accelerate knock onset compared to EGR-diluted mixtures.

5.6.2 Combustion Stability

In terms of combustion stability, EGR again shows superiority. Across all load levels, EGR allows stable operation at higher dilution rates compared to lean combustion. Although EGR increases ignition delay, it avoids the lean limit instability that arises from insufficient fuel concentration, which can lead to misfires and excessive cycle-to-cycle variability.

Lean combustion, on the other hand, sees a steep rise in COV_{imep} beyond a certain λ threshold, particularly at high loads, where stability and knock limits converge. This severely limits the practical dilution ceiling for lean operation.

5.6.3 Thermal Efficiency and Heat Release

Lean combustion consistently demonstrated a slightly higher BTE at identical combustion phasing (CA_{50}), as predicted by the theoretical increase in the specific heat ratio (γ) when excess air is added to the charge. In contrast, EGR reduces γ due to the introduction of high heat-capacity components (CO_2 , H_2O), resulting in slightly lower theoretical efficiencies.

Despite this, EGR enables more advanced spark timing due to its superior knock suppression capabilities. As a result, EGR can better align the combustion process with TDC, improving the effective expansion ratio and reaching similar efficiencies as lean operation.

5.6.4 Emissions and Aftertreatment Compatibility

Both strategies significantly reduce NO_x emissions due to lower peak in-cylinder temperatures. However, EGR holds a decisive advantage in exhaust aftertreatment compatibility. Unlike lean combustion, which produces an oxygen-rich exhaust stream incompatible with TWCs, EGR maintains near-stoichiometric exhaust conditions. This enables the use of TWCs for simultaneous NO_x and CO reduction, avoiding the complexity and cost of alternative NO_x aftertreatment systems required in lean-burn configurations.

5.6.5 Practical Implementation Considerations

While EGR adds complexity to engine hardware and calibration, requiring precise control over recirculation rates and intake conditions, it provides a more robust and flexible approach. EGR preserves the benefits of dilution without compromising the stoichiometric operation required for simple and effective emissions control. Moreover, it supports stable combustion across a broader range of operating points, including those constrained by thermal or knock limitations.

5.6.6 Extending EGR Limits with Stability-Enhancing Technologies

A notable advantage of EGR-diluted combustion is its capability to maintain stable operation even at WOT conditions. As a result, engine load can be modulated by varying

the EGR rate rather than relying solely on throttle control, thereby significantly reducing pumping losses associated with throttling. However, the primary limitation in further advancing EGR dilution at WOT is not knock but combustion stability. As dilution increases, the system becomes increasingly susceptible to cycle-to-cycle variability, which restricts the extent to which spark timing can be advanced and dilution can be applied.

To overcome this stability barrier, further technological development is essential. One promising approach is the integration of passive prechamber ignition systems. These systems can significantly enhance ignition stability and flame propagation under highly diluted conditions, offering a path forward to fully exploit the efficiency and knock mitigation benefits of EGR. Thus, passive prechamber technology holds strong potential for expanding the stable operating range of spark-ignition engines utilizing EGR dilution.

5.6.7 Overall Assessment

In conclusion, although lean combustion offers theoretical efficiency benefits due to a higher specific heat ratio, its practical implementation is hampered by stability issues, lean limit constraints, and aftertreatment incompatibilities. EGR, on the other hand, offers a more balanced and practical solution by:

- providing superior knock suppression,
- enabling operation closer to optimal combustion phasing,
- retaining combustion stability at high dilution rates,
- maintaining compatibility with established emission control systems,
- and supporting knock-free operation up to WOT conditions, with further improvement possible through passive prechamber technology.

Therefore, EGR emerges as the more favorable dilution strategy for enhancing engine performance and efficiency in methanol-fueled heavy-duty spark-ignition engines.

Chapter 6

Unlocking the Potential of a Passive Prechamber to boost Heavy-Duty Methanol SI Engine Performance

6.1 Introduction

This chapter was originally intended to investigate the influence of a passive prechamber on the combustion behavior of the methanol-fueled spark-ignition engine. As discussed in chapter 5, the implementation of a passive prechamber ignition system is a promising approach to further increase dilution tolerance and unlock the full efficiency potential of spark-ignition methanol engines is. Passive prechambers have shown the ability to extend the lean and EGR dilution limits by promoting faster and more stable combustion through multiple turbulent ignition jets [96].

However, due to time constraints encountered during the course of this research project, the experimental validation of a prechamber system could not be completed within the time-frame of this dissertation. Despite this limitation, a prechamber design was developed and manufactured based on a detailed review of existing literature and engineering principles.

The remainder of this section presents the rationale behind the proposed design, supported by theoretical considerations and relevant findings from prior studies. This lays the groundwork for potential future work aimed at experimentally validating the impact of passive prechamber technology on combustion stability and efficiency under diluted operating conditions.

6.2 Prechamber Working Principle

The passive prechamber is a small volume chamber integrated with a spark plug and several small nozzles directed toward the main combustion chamber. During the compression stroke, a small portion of the homogeneous air-fuel mixture is pushed into the

prechamber. Upon ignition, high-pressure combustion gases and active radicals are expelled through the nozzles, initiating combustion at multiple sites in the main chamber. This promotes rapid flame propagation and improves combustion stability, particularly under high dilution conditions [97, 96].

6.3 Design Constraints and Assumptions

The design had to comply with several constraints dictated by the geometry of the Volvo Penta base engine:

- The prechamber had to fit in the original spark plug location, which is oriented at a 20° angle.
- The retrofit had to be reversible to allow comparison between standard and prechamber operation.
- Standard PFI fueling was used, implying the prechamber would be passively filled during the compression stroke.

6.4 Geometric Design Parameters

6.4.1 Prechamber Volume

The volume of a passive prechamber plays a critical role in balancing ignition reliability and system efficiency. Literature suggests an optimal range of 2–5% of the clearance volume at TDC [96, 98, 97]. In this study, a volume of 1531.13 mm³ was selected, corresponding to 2.13% of the clearance volume, positioned deliberately at the lower end of this range.

Smaller prechamber volumes have been shown to reduce crevice volume and surface area, which in turn minimizes heat transfer losses and unburned hydrocarbon emissions [97]. This is especially important in passive systems without auxiliary fueling, where ignition energy must be used efficiently to generate high-velocity jets that ignite the main chamber mixture.

Moreover, compact volumes limit the mass and energy extracted from the main chamber, avoiding the drawbacks of excessive jet strength and prechamber dominance that can arise with larger volumes [98]. They also help preserve charge homogeneity, which is beneficial for stable ignition under lean or EGR-diluted conditions [96].

Thus, the selected 2.13% volume strikes a careful balance between generating sufficiently strong ignition jets and minimizing thermal and energetic losses, aligning well with established best practices for passive prechamber applications.

6.4.2 Nozzle Area and A/V Ratio

The nozzle configuration significantly influences jet structure and prechamber emptying. A derived parameter known as the A/V ratio, the ratio of total nozzle cross-sectional area

to prechamber volume, was used to guide the design. Literature suggests values between 0.03 and 0.04 cm²/cm³ [96]. With five nozzles (four angled and one centrally downward), each with a diameter of 1.2 mm, the design achieved an A/V ratio of 0.0369 cm²/cm³, in line with best practices.

6.4.3 Spray Angle and Jet Targeting

The selection of the prechamber nozzle spray angle is critical for ensuring that the ignition jets are effectively directed into the piston bowl. Ideally, a wide spray angle of around 160° would maximize flame kernel dispersion and promote rapid combustion throughout the main chamber. However, in this design, the prechamber is installed with a 20° tilt relative to the cylinder axis. This tilt introduces a geometric limitation: if a 160° spray angle were used, some jets would strike the cylinder wall rather than entering the bowl.

To avoid this, the spray angle was limited to 120°, which was determined to be the widest feasible option that still ensures all jets are directed into the 63 mm diameter bowl, even at a minimum clearance height of 2 mm. While this narrower angle may slightly reduce jet dispersion, it was a necessary compromise to maintain effective flame propagation without redesigning the prechamber orientation or cylinder head geometry.

6.5 Final Assembly and Installation Methodology

The prechamber design was completed in CAD, see Figure 6.3, featuring an asymmetric head to facilitate precise nozzle orientation, and a hexagonal external profile that enables separate tightening of the prechamber body and the spark plug. A 5 mm copper sealing ring is employed to ensure reliable sealing. To achieve accurate alignment of the nozzle array, a marked reference is used in combination with an iterative installation procedure. The final nozzle orientation is adjusted by modifying the seal thickness. This method leverages the fine M10x1 thread pitch, where each full rotation corresponds to a 1 mm vertical displacement, allowing for a precise angular correction of 1° per 1/360 mm change in height.

6.6 Summary and Future Work

The design presented fulfills key criteria from literature and adapts them to the unique implementation constraints of the Volvo Penta engine. Although not experimentally tested, the selected parameters fall within recommended values:

- Prechamber volume: 2.13% of clearance volume
- Nozzle diameter: 1.2 mm (5 nozzles)
- A/V ratio: 0.0369 cm²/cm³
- Spray angle: 120°

These values reflect a design optimized for part-load dilution enhancement and knock mitigation through improved ignition stability. Future work should include combustion

chamber CFD to refine nozzle orientation and placement, and experimental validation to quantify the expected gains in efficiency, stability, and dilution tolerance.

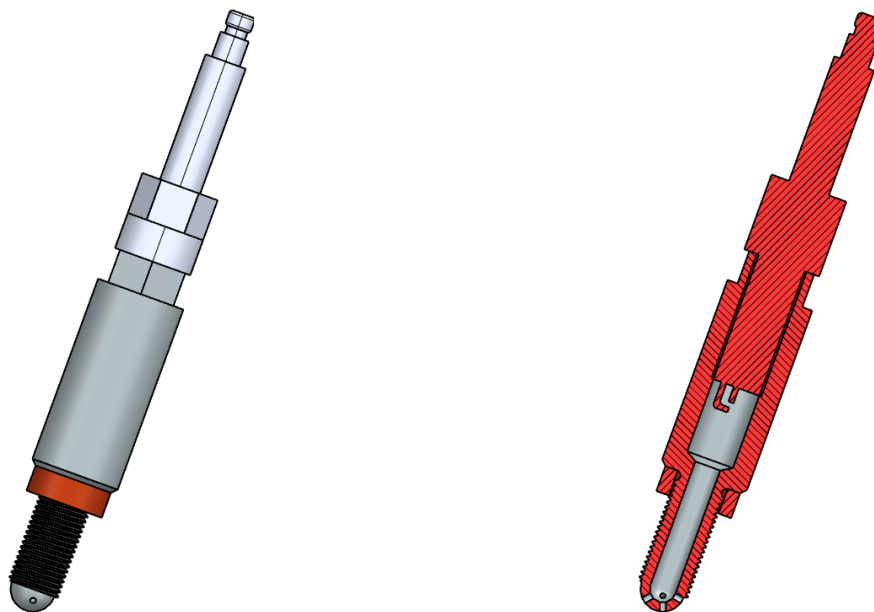


Figure 6.1: Final design of the passive prechamber assembly.



Figure 6.2: Manufactured prechamber.

Chapter 7

Conclusion

The transition to sustainable fuels in the heavy-duty transport sector requires not only cleaner alternatives but also technological pathways that maintain or improve current performance standards. Methanol is a promising candidate due to its carbon-neutral potential and positive combustion characteristics. However, its application in spark-ignition (SI) engines faces key limitations, particularly regarding knock resistance, combustion stability, and overall efficiency under high-load operation.

This study explores how advanced dilution strategies and ignition enhancements can optimize methanol-fueled SI engines, with the ultimate aim of delivering diesel-comparable or superior performance. By systematically analyzing the effects of lean operation, EGR, and passive prechamber ignition, the study provides a detailed assessment of how methanol SI combustion can be improved for future heavy-duty applications.

7.1 Literature Review

Heavy-duty engines remain a cornerstone of global freight and marine transport but are under increasing pressure to decarbonize. While current diesel compression-ignition (CI) engines offer high efficiency, they also produce significant NO_x and soot emissions. Methanol presents a clean alternative due to its oxygenated nature, high knock resistance, and potential for carbon-neutral production through bio- or e-methanol pathways. This positions methanol as a viable fuel for future HD spark-ignition (SI) applications.

Retrofitting existing diesel engines to operate in methanol SI mode has gained attention as a transitional pathway. However, such adaptations face key thermodynamic and combustion challenges. The high compression ratios typical of CI engine platforms heighten knock sensitivity in SI operation. Furthermore, the large bore geometry of heavy-duty engines increases the required flame travel distance, which elevates the risk of combustion instability, particularly under highly diluted or retarded ignition conditions.

Literature offers several knock mitigation strategies to address these issues. Among them, charge dilution via excess air (lean operation) or EGR has shown promise in delaying knock onset and improving efficiency. Additionally, passive prechamber ignition systems are emerging as a means to enhance flame initiation and combustion stability in highly

diluted environments. Collectively, these approaches form the foundation for further optimizing methanol SI combustion to make it competitive with diesel in terms of performance, reliability, and environmental impact.

7.2 Experimental research

The experimental investigation confirmed the strong potential of methanol-fueled SI operation in retrofitted HD engines. Through systematic testing under both lean and EGR-diluted conditions, substantial improvements in thermal efficiency were achieved compared to stoichiometric baseline operation. These gains were primarily attributed to reduced heat transfer and pumping losses, knock mitigation, and improved combustion phasing, as detailed in section 5.6.

In parallel, the development and integration of a passive prechamber system demonstrated promising capabilities to extend dilution tolerance and stabilize combustion under conditions that were previously limited by cycle-to-cycle variability. Although full validation of the prechamber is ongoing, the design shows clear potential to further enhance the performance of methanol SI combustion, especially under highly diluted and high-load conditions.

While previous comparisons in this work have primarily focused on improvements relative to stoichiometric methanol operation, the overarching objective of this thesis is to evaluate whether methanol-fueled SI combustion can approach or even match the performance of conventional diesel engines. This final comparison represents the missing benchmark in assessing methanol's viability as a true diesel alternative. By directly comparing the experimental results to the original diesel performance data, the practical potential of methanol SI combustion in retrofitted HD platforms can be more clearly assessed.

During the experimental phase of this study, no direct measurements were conducted under diesel operation. To facilitate a meaningful comparison, reference diesel performance data are drawn from the most recent measurements by Demiddeleer and Vanderbeken [21], who previously operated the same engine platform under conventional CI conditions.

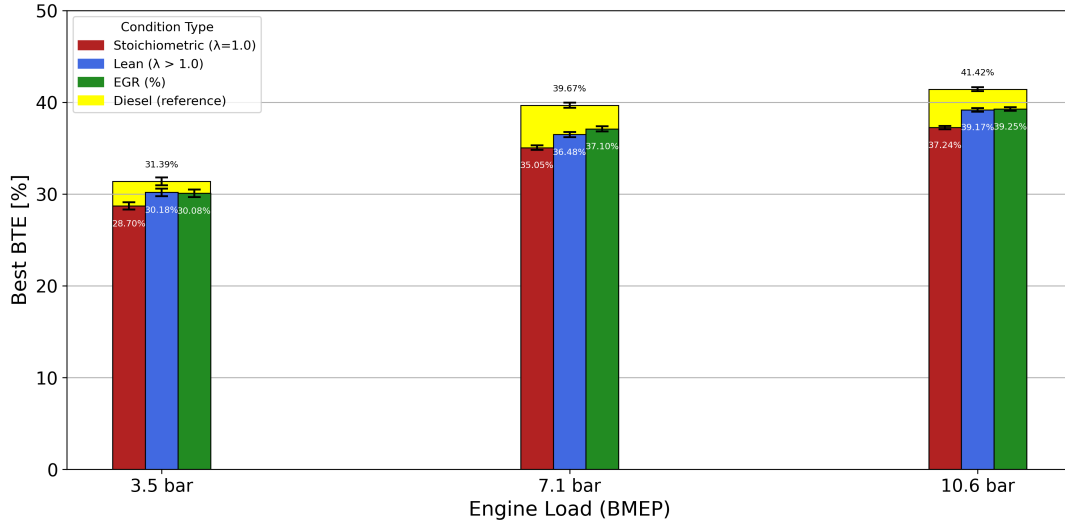


Figure 7.1: Brake thermal efficiency (BTE) at a constant engine speed of 1500 RPM for various load points and operating strategies.

Figure 7.1 illustrates the best brake thermal efficiencies (BTE) achieved under stoichiometric methanol operation, lean and EGR-diluted strategies, and conventional diesel CI operation across various load points. As anticipated, diesel combustion consistently delivers the highest thermal efficiency. Nonetheless, the application of charge dilution strategies, particularly EGR, enables methanol-fueled SI operation to significantly close the efficiency gap.

At low load (3.5 bar BMEP), the BTE difference between diesel and EGR-diluted methanol operation is reduced to only 1.2 percentage points. At medium and high loads (7.1 and 10.6 bar BMEP), this difference remains below 3%. Although seemingly small, such efficiency deficits represent a substantial amount of lost energy in absolute terms. In heavy-duty applications, where engines operate under high loads for extended durations, even a single percentage point improvement in thermal efficiency yields considerable reductions in fuel consumption and operating costs.

The superior efficiency of diesel combustion, even on the same engine platform, can be attributed to several intrinsic combustion advantages. Diesel engines operate under ultra-lean conditions due to direct injection into highly compressed air, resulting in high air-fuel equivalence ratios. This leads to inherently lower combustion temperatures, substantially below those observed in even diluted SI operation, see Figure 7.2, which minimizes heat transfer and dissociation losses. Additionally, diesel engines avoid throttling to control load, thereby eliminating the pumping losses that are characteristic of throttled SI operation, particularly at part loads.

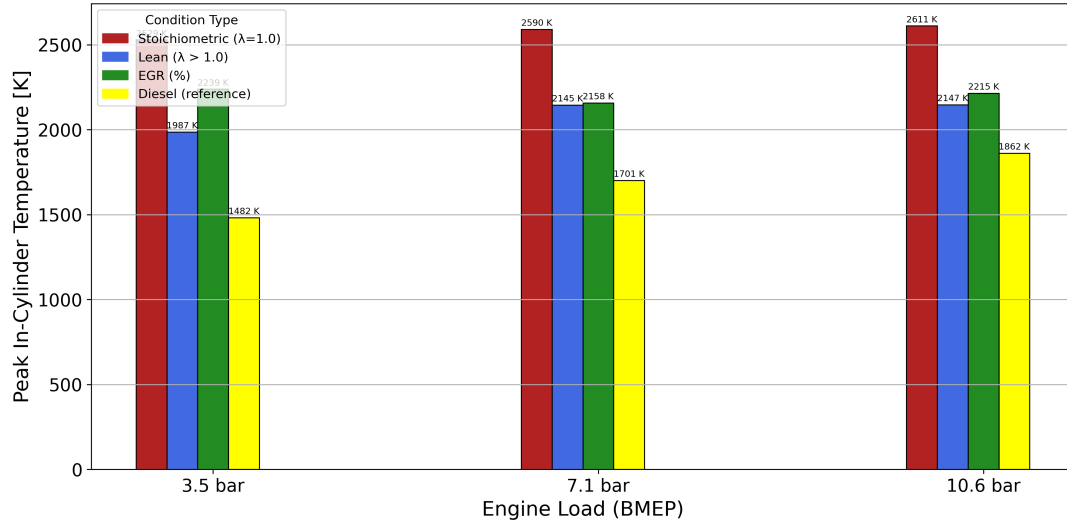


Figure 7.2: Calculated in-cylinder temperatures for different loads in diesel and methanol operation.

Furthermore, diesel combustion is initiated via auto-ignition, allowing for optimal phasing close to top dead center (TDC), with minimal sensitivity to knock. In contrast, SI engines must ignite the mixture earlier to avoid knock, sacrificing ideal combustion timing. Diesel's capability to operate with stratified mixtures, featuring localized fuel-rich zones within a globally lean environment, enhances combustion efficiency and stability under varying conditions.

Collectively, these combustion features explain why diesel continues to outperform methanol SI combustion in terms of BTE, even when both are applied to the same mechanical engine platform. Nevertheless, the close performance observed with EGR-diluted methanol operation highlights the substantial progress achieved in this study. Continued development of methanol SI strategies, such as refined dilution control and prechamber-assisted ignition, remains crucial to fully bridging the efficiency gap and establishing methanol as a viable, high-performance alternative to diesel in heavy-duty applications.

Chapter 8

Future Outlook

Throughout the experimental research and analysis conducted in this dissertation, several promising research directions have emerged that warrant further exploration. These potential avenues, outlined in section 8.1, could significantly enhance the understanding and optimization of engine performance for a marine engine retrofitted to methanol SI operation.

In addition, several issues and challenges with the experimental setup were encountered during the measurement campaign. These difficulties impacted the overall efficiency of the measurement process. section 8.2 outlines potential improvements aimed at streamlining data collection and improving the overall usability of the test setup.

8.1 Future Research Directions

- **Exploring complete engine map:**

While the current study focused on investigating dilution strategies (lean combustion and EGR) at 1500 RPM across three specific low, medium and high load points (3.5, 7.1, and 10.6 bar BMEP), further research should aim to extend this analysis across a broader range of operating conditions. This would include exploring even higher loads and varying engine speeds, allowing for a comprehensive understanding of the engine's behavior under diverse conditions. Such an expanded analysis could confirm the trends observed in this dissertation and provide insights into the lean limit and EGR limit at different operating conditions.

Particularly valuable would be conducting load sweeps to determine the maximum achievable load and efficiency under various dilution strategies. This was not feasible in the current study due to unsatisfactory performance of the pump supplying the water brake system. Addressing this limitation in future research would allow for the creation of a complete engine performance map, similar to the one developed by Zhu et al. [39], as illustrated in Figure 3.2. Such a map should include not only performance metrics like BTE, but also emissions behavior (CO , NO_x) across all conditions. This comprehensive engine map would provide a robust foundation for identifying the most efficient and environmentally friendly operating strategies.

- **Passive prechamber implementation:**

The results obtained from EGR operation demonstrated a clear benefit in extending the knock limit, effectively allowing for more advanced spark timing and increased efficiencies without the onset of knock. At higher EGR levels, a trade-off emerges between improved brake thermal efficiency and reduced combustion stability. The increased dilution can lead to higher cycle-to-cycle variability, eventually making it unsuitable for practical applications.

To mitigate these stability challenges while maintaining the benefits of EGR, advanced ignition enhancement technologies, such as a passive prechamber, offer a promising solution. An investigation and design of the passive prechamber has already been carried out, as detailed in chapter 6. Unfortunately, due to unforeseen circumstances and staff shortages, it was not possible to fully implement and test these prechambers during the current research period. Nonetheless, this remains a promising area for future work, with the potential to hold high BTE while significantly enhancing combustion stability under high EGR operation. This approach could also potentially extend the usable wide open throttle (WOT) range, enabling the use of EGR as a primary load control strategy across the entire engine operating map.

- **Effect of Motor Oil on Knock Sensitivity**

While the engine was originally designed for diesel operation, the same motor oil was retained during its adaptation to spark-ignition methanol combustion. However, literature indicates that lubricating oil composition can influence the knock sensitivity of spark-ignition (SI) engines, particularly under high load or boosted conditions [99].

One key mechanism is the potential for low-speed pre-ignition (LSPI), where small oil droplets or deposits form localized hot spots on the cylinder walls. These hot spots can prematurely ignite the air-fuel mixture ahead of the spark event, thus mimicking or exacerbating knock phenomena. Specialized LSPI-resistant motor oils have been developed to mitigate this issue. These formulations reduce the presence of catalytic additives and lower the likelihood of oil-fuel interactions that contribute to auto-ignition.

Given these insights, it would be valuable to experimentally assess the influence of LSPI-resistant oils compared to conventional diesel engine oils in the context of knock sensitivity. Such a comparison could help quantify the potential gains in knock suppression and combustion stability offered by optimized lubricant selection.

8.2 Future work on the Experimental Setup

- **Refinement of LabVIEW program:**

Measures have already been implemented to optimize the LabVIEW program, reducing CPU usage and improving speed, see Appendix E. However, as the program has evolved from diesel to DMDF and now to methanol SI operation, it has become increasingly complex. A thorough clean-up of the LabVIEW code is necessary to

improve measurement efficiency and facilitate easier debugging in the future.

Additionally, an issue has been identified where LabVIEW becomes unresponsive during data storage, which is particularly hazardous when measuring near the knock limit. Addressing this problem will require optimizing the data storage routine to maintain in control during critical measurements.

- **Water brake system and engine cooling circuit:**

During the research conducted this year, it was not feasible to perform load sweeps due to persistent issues with the pumps supplying rainwater to the water brake system and the engine cooling circuit. Prior to the start of this research, two centrifugal pumps were installed in series to supply water to both the water brake and cooling systems. However, this configuration proved inadequate, as it was unable to simultaneously support both engine cooling and brake power demands, leading to unstable and fluctuating loads during measurements.

To temporarily address this issue, the engine cooling circuit was switched back to city tap water, allowing for more stable operation under moderate conditions. Nevertheless, during measurements, particularly at higher loads, the pumps continued to exhibit unsatisfactory performance. In certain instances, they experienced cavitation, causing sudden pressure drops and further destabilizing the system.

To ensure reliable and consistent operation in future measurements, it is recommended to reconfigure the water supply system by installing the two pumps in parallel, one dedicated to the water brake system and the other to the engine cooling circuit. This setup would ensure independent and stable water supply to both systems, minimizing pressure fluctuations and enhancing measurement reliability.

- **Acquire a formaldehyde measurement device (adapted from [21]):**

In addition to the well-known emission species such as carbon monoxide and nitrogen oxides, it would be valuable to investigate the formation of formaldehyde. Despite its importance as an emission species, it cannot be measured with the current emission cabinet used in this study, which is primarily designed for conventional exhaust gases.

It is strongly recommended to acquire a dedicated formaldehyde measurement device in future research. Such an instrument would enable the precise quantification of formaldehyde concentrations in the exhaust gases, providing critical insights into the impact of operating conditions (e.g. lean combustion, EGR) on its formation.

- **Install a knock sensor (adapted from [21]):**

Currently, knock detection during methanol SI operation is performed using an auditory method, where the occurrence of knock is identified by the characteristic knocking sound. While this approach provides a basic indication of knock occurrence, it is inherently subjective and lacks the precision required for consistent and reliable measurements.

To ensure accurate knock detection and to facilitate more precise calibration of the engine, it is recommended to implement a dedicated knock sensor system mounted directly on the engine block in future research. However, it must also be noted that knock sensors alone may not provide a fully accurate indication of knock severity, especially under the unique combustion characteristics of methanol. Therefore, it is

further recommended to implement a direct knock quantification method based on in-cylinder pressure measurements, such as the Maximum Amplitude of Pressure Oscillations (MAPO). This pressure-based indicator enables objective, cycle-resolved knock assessment and is better suited for establishing knock intensity thresholds across a wide range of operating conditions. The combination of a knock sensor and a calculated indicator like MAPO would ensure both robust detection and quantitative assessment, forming a comprehensive knock evaluation framework.

Part III

Appendices

Appendix A

Volvo Penta operating procedures

This appendix presents an updated version of previously documented operating procedures for methanol SI operation on the Volvo Penta D7C-B TA [22] [21]. These procedures reflect all modifications made to the engine setup and measurement equipment over the years, as well as best practices established through ongoing experimentation. The focus here is exclusively on methanol SI operation, as this has been the area of investigation this year and will continue to be the focus in the coming years. A detailed operation procedure of diesel-only and dual-fuel (methanol-diesel) operation is incorporated in the thesis of Demiddeleer and Vanderbeken [21], as well as those of their predecessors.

Start-up

1. Turn on the NO₂-NO converter of the **Siemens Ultramat 23 emission analyser**. (It takes approximately half an hour to heat up to the operating temperature of 380 °C. Maximum accuracy of the Ultramat is obtained after 2 hours.) Check and remove displayed error messages. (Never shut off the power supply to the emissions cabinet for longer than a few minutes, as this would necessitate the recalibration of the Ultramat emission analyser.) A detailed analysis of the emission analyzer is available in last year's dissertation [21], as well as in the dissertations of their predecessors. However, this academic year (2024-2025) the NO₂-NO converter broke down due to internal overheating, making it impossible to activate and measure NO₂ emissions
2. Select the lowest expected measurement range for CO₂, CO and NO to get maximum accuracy.
3. Write down the **atmospheric conditions** (measuring apparatus is located on the back of the pole to the right of the test setup). Do this multiple times during a measurement campaign. Take the average afterwards and include it in the stored data.
4. Turn on the **exhaust gas ventilation system** by putting the switch to setting 2 (can be found at the pole near the Volvo T3 setup). Make sure the green light is turned on.

5. (Optional) Open the garage door and turn on the top ventilators in the laboratory by the switches in the cage to the left of the test setup.
6. Check the oil level (can be found at the front side of the engine).
7. Plug in the **RSP-2400-24 power supply** of the **ignition coils** (can be found underneath the table with the diesel mass flow meter).
8. Turn on the **power supply in the electric control box** mounted onto the frame of the test setup.
9. Turn on the **Kistler charge amplifier** on the back, press the green "Meas" button on the Kistler and verify if the green light is on.
10. Turn on the **power supply of the engine, red switch** on the right side of the engine. Vertical is off, horizontal is on.
11. Open the blue valve of the cubitainer in the **methanol tank**, close the door of the tank.
12. Turn on the **methanol fuel pump** by switching on the 'Fuel on' switch on the control panel situated on the table and turning the 'Pump speed' potentiometer until the pressure regulator (mounted on the rail between the test setup and the tank) shows 5.5 bar (overturning the potentiometer is not detrimental to the pump).
13. Open the **valve** of the water brake and the one of the engine cooling circuit.
14. Turn on the **ECU** on the control panel on the table. Open the **MoTeC ECU Manager software** and press 'Open ECU'.
15. Run the latest version of the **LabVIEW** program (located in the folder DataStudenten/labview and named mainPanel_Penta_241105_SI_MeOH_official.vi).
16. An example of a set of **engine settings** that enabled **successful cold starting**.
 - Set the **engine brake** to 0 Nm.
 - Open **ETB1 throttle valve** by turning the throttle potentiometer on the control panel on the table to approximately 8 %. The position can be monitored in the MoTeC software.
 - Close **ETB2, both EGR valves and wastegate (=0%)** in the LabVIEW program.
 - Set the **injection duration** (100 % IJPU = 15 ms) in the injection table in the MoTeC software to approximately 62 % IJPU.
 - Set the **ignition timing** in the ignition timing table in the MoTeC software to approximately 20 ° BTDC.
 - Set the **injection timing** in the injection timing table in the MoTeC software to approximately 280 ° BTDC.
17. Turn **the key** onto the frame of the test setup in order to **start the engine**.
18. If the Coolant Level Warning Light is on, turn off the engine and refill the coolant tank (situated on the right side of the engine) with coolant. (Never do this after

running for a long time as hot steam would come out.)

19. Reset errors in the MoTeC software (if any).

Measurements

1. **Go to the desired operating point** (load and speed) by adjusting the engine brake and controlling both the throttle valve position with the potentiometer and the methanol injection duration in the injection table in the MoTeC software (if the temperature of the brake gets too high, open the valve at the back of the brake, be aware that this changes the operating point). The operational procedures used during experimental testing to achieve stable engine operation under lean and EGR conditions is given in Appendix I.
2. Turn on the external **pump of the Ultramat** (check if it is connected).
3. Give the engine some time to settle in a **stable operating point**. Then, enter the IJPU %, lambda value (displayed in the MoTeC software), speed and load in the LabVIEW program.
4. **Enter any peculiarities** (about the temperatures, pressures or audible phenomena) **in the remarks** in LabVIEW.
5. **Store the operating point** by clicking 'Store Data' in LabVIEW. While doing so, start the emission measurements:
 - Connect the **Ultramat** to the exhaust gas tap at the back of the engine. Wait at least 3 minutes for the analyser to settle. Write down the CO₂, CO and NO concentrations.
 - Turn on the NO₂-NO **converter**, wait 3 minutes and write down the NO_x concentrations.
 - In **EGR operation**:
 - Also connect the **Ultramat** to the inlet tap of the engine. Perform the measurements in the same way as described above. Write down the values in an additional Excel file.
 - Connect the **Eurolyzer STx** to both the inlet and exhaust to measure the O₂ concentration. (only open the gas tap for 2 minutes while measuring, otherwise excessive contamination will occur). Write down the values in the additional Excel file.

Shut-down

1. Gradually **decrease the throttle valve ETB1** opening until 8% using the throttle potentiometer on the control panel on the table, while simultaneously **decreasing the IJPU %** in the MoTeC software.
2. Bring the **engine to idling speed** by reducing the load to zero, while slowing down the engine by lowering the IJPU % even more. Let the engine run idle for some time.

3. Set the **IJPU %** to 0 and turn **the key** onto the frame of the test setup in order to **stop the engine**.
4. Turn off the **methanol fuel pump** by switching off the 'Fuel on' switch on the control panel situated on the table.
5. Close the **tap water valves** of the water brake and of the engine cooling circuit.
6. Switch off the **ECU** on the control panel on the table and close the **MoTeC software**.
7. Close the blue valve of the cubitainer in the **methanol tank**, close the door of the tank.
8. Turn off the **power supply of the engine, red switch** on the right side of the engine. Vertical is off, horizontal is on.
9. Save the **Excel file** and stop the **LabVIEW** program. Afterwards the appropriate emissions should be entered in the Excel file. More information about the data post-processing is provided in Section 4.4 of this dissertation.
10. Turn off the **Kistler charge amplifier** situated in the electric control box.
11. Turn off **the power supply in the electric control box** mounted onto the frame of the test setup.
12. Unplug the **RSP-2400-24 power supply** of the **ignition coils** (can be found underneath the table with the diesel mass flow meter).
13. (Optional) Close the garage door and turn off the top ventilators by the switches in the cage to the left of the test setup.
14. Turn off the **exhaust gas ventilation system** by putting the switch to setting 0 (can be found at the pole near the Volvo T3 setup). Make sure the green light is turned off.
15. Turn off the **emission measurement equipment**.

Appendix B

Last Minute Startup Analysis - SI

The engine startup process involves several steps, as detailed in Appendix A. This appendix provides a last-minute startup analysis to ensure all necessary measurements are properly conducted, based on [21]. Omitting any of these steps may result in inaccurate measurements or even a failed startup. To help researchers avoid oversights, the following enumeration and Figure B.1 present a clear last-minute startup checklist.

- ☐ Green extraction lamp ON?
- ☐ Red engine switch ON?
- ☐ Electricity ON in cabinet and Kistler measuring ON?
- ☐ Valve methanol barrel open?
- ☐ Valve water brake and valve engine cooling open?
- ☐ Valve engine cooling open?
- ☐ Water brake on 0 Nm?
- ☐ MoTeC ECU ON?
- ☐ MoTeC settings ready to go?
- ☐ Atmospheric conditions in Excel and LabVIEW document?
- ☐ LabVIEW running?
- ☐ Excel from LabVIEW saved in appropriate folder?
- ☐ Methanol fuel pump ON?
- ☐ Methanol pressure at 5.5 bar?
- ☐ Throttle valve behaves properly?
- ☐ EGR valves behave properly?

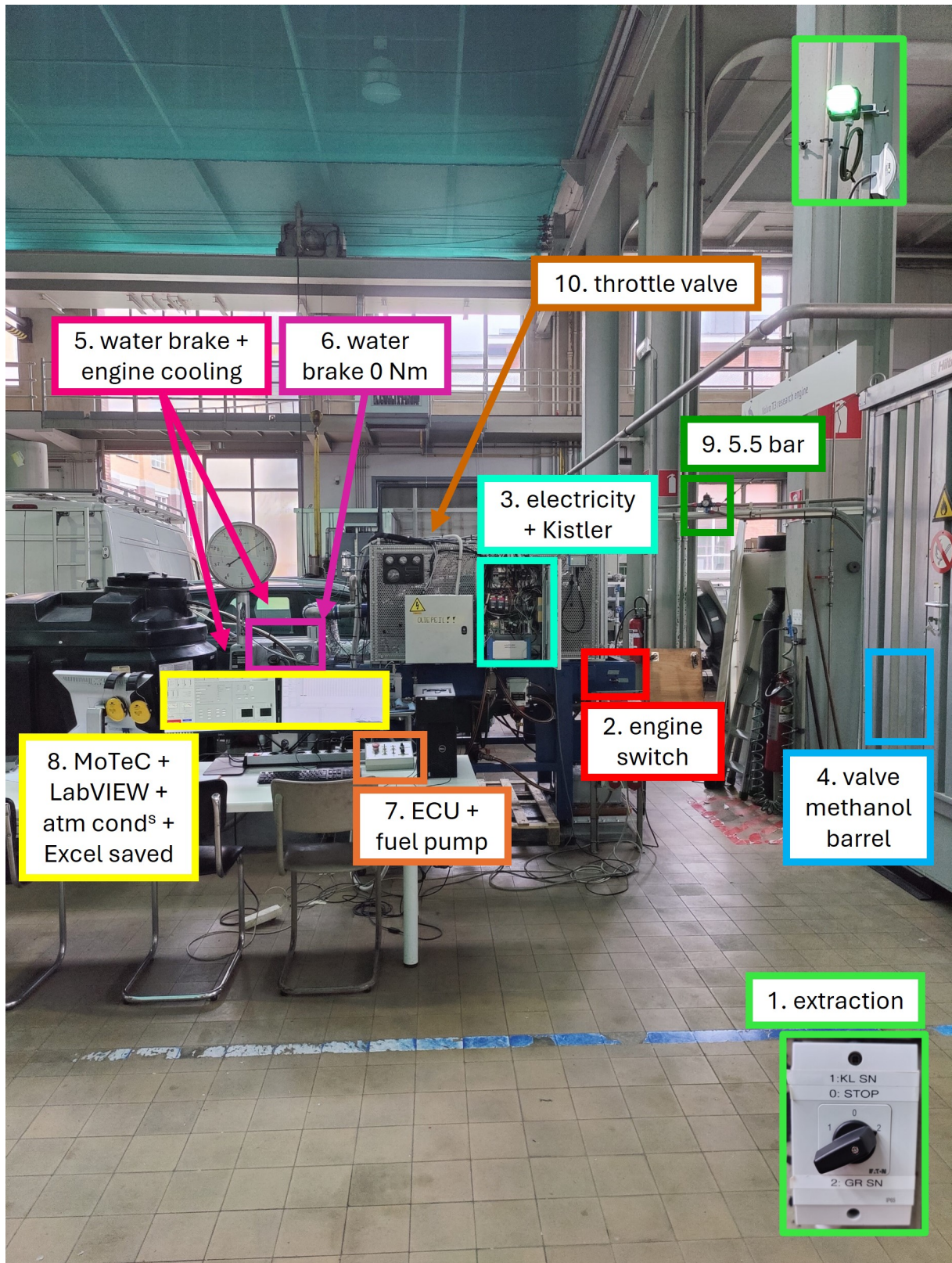


Figure B.1: 10 last checks before starting the Volvo Penta engine.

Appendix C

Uncertainty analysis of experimental data

In this appendix, the uncertainty analysis applied to the experimental data of Part II is comprehensively discussed. This analysis is based on the analysis of previous engine researchers [21] [22] [83] [84], but is specifically adapted for the case of pure methanol SI combustion.

The uncertainty analysis follows a systematic three-step methodology:

1. **Data Examination and Cleaning:** The initial stage involves carefully examining the raw experimental data to identify and remove any outliers or irrelevant data points. This was carried out directly during experimental researching and ensures that the analysis is based on clean and representative data.
2. **Measurement Uncertainty Evaluation:** In this stage, the accuracy, range, and resolution of the measurement equipment are analyzed, as these directly influence the uncertainty of the measured quantities. The specifications of the measurement equipment used are detailed in Table C.1. Furthermore, the uncertainty associated with these measurements is propagated to calculate the uncertainty of derived quantities.
3. **Statistical Evaluation of Uncertainty:** The final step involves a statistical analysis of the calculated uncertainties.

By following this structured approach, the uncertainty analysis ensures that comparative assessment of experimental data is reliable and that any observed difference can be attributed to actual physical phenomena rather than measurement errors.

Measured quantities

Table C.1 shows the range, accuracy and resolution of different measured quantities during testing, adapted from [21].

Table C.1: Range, accuracy and resolution of measured quantities, adapted from [21].

Mass flow meters			
Component	Range	Accuracy	
Methanol	1-100 kg/h	$\pm 0.2\% \cdot \text{RD}$ kg/h	
Air	25.86-1225 kg/h	$\pm (0.5\% \cdot \text{RD} + 1.225)$ kg/h	
Engine parameters			
Component	Range	Accuracy	Resolution
Lambda	0.65- ∞	± 0.05	
Torque		$\pm 9.64 - 0.002 \cdot \text{RD}$ Nm	
Speed	0-1200 RPM	± 5 RPM	
In-cylinder pressure	0-250 bar	$\pm 0.3\% \cdot \text{RD}$ bar	
Temperatures	K-type 0-1100°C	$\pm 1^\circ\text{C}$	0.1°C
	T-type -185-400°C	$\pm 1^\circ\text{C}$	0.1°C

RD = Reading

Derived quantities

The uncertainties on the derived quantities (q) are determined based on the uncertainties of their composite measured quantities ($x_1, x_2, \dots, x_n$). The general equation used for calculating the uncertainty of a derived quantity when the variables are random and independent follows from a Taylor series expansion and is given in Equation C.1.

$$\delta q(x_1, x_2, \dots, x_n) = \sqrt{\left(\frac{\partial q}{\partial x_1} \cdot \partial x_1\right)^2 + \left(\frac{\partial q}{\partial x_2} \cdot \partial x_2\right)^2 + \dots + \left(\frac{\partial q}{\partial x_n} \cdot \partial x_n\right)^2} \quad (\text{C.1})$$

This equation is only strictly accurate when the composite variables are independent and random. However, this ideal situation was not met during these experiments, as most variables exhibit correlations or dependencies, e.g. load and speed. Nevertheless, it is still acceptable to use this method as an approximation for the uncertainty estimation. An alternative and more conservative approach is to sum the absolute uncertainties, as given by Equation C.2.

$$\delta q \leq \left| \frac{\partial q}{\partial x_1} \right| \delta x_1 + \left| \frac{\partial q}{\partial x_2} \right| \delta x_2 + \dots + \left| \frac{\partial q}{\partial x_n} \right| \delta x_n \quad (\text{C.2})$$

This inequality provides an upper bound on the uncertainty when variables are not entirely independent or when their covariances are significant. In this dissertation, the derived uncertainties are calculated using the quadratic sum method, as it is more commonly applicable and provides a realistic estimate of combined uncertainties. In the remainder of this section, only the uncertainty of the quantities that are potted throughout this dissertation and its components are determined.

Brake power

The brake power is determined by taking the product of the engine load and speed using Equation C.3. The torque's unit is Nm, while the unit of engine speed is rounds per minute.

$$P_e = \frac{T \cdot 2 \cdot \pi \cdot n}{60} \cdot 10^{-3} [kW] \quad (C.3)$$

Consequently, the uncertainty on the brake power is calculated using Equation C.4.

$$\begin{aligned} \delta P_e &= \sqrt{\left(\frac{\partial P_e}{\partial T} \cdot \delta T\right)^2 + \left(\frac{\partial P_e}{\partial n} \cdot \delta n\right)^2} [kW] \\ &= \frac{2 \cdot \pi}{60} \sqrt{(n \cdot \delta T)^2 + (T \cdot \delta n)^2} \cdot 10^{-3} [kW] \end{aligned} \quad (C.4)$$

Brake thermal efficiency

The brake thermal efficiency, represented by Equation C.5, is denoted as the ratio of the brake power to the heat released by methanol combustion. The brake power is expressed in kilowatts, while the methanol mass flow rate is expressed in kg/h and the lower heating value of methanol in MJ/kg.

$$BTE = \frac{P_e \cdot 3600}{\dot{m}_{MeOH} \cdot LHV_{MeOH} \cdot 1000} [-] \quad (C.5)$$

Consequently, the uncertainty on the brake thermal efficiency is calculated using Equation C.6.

$$\begin{aligned} \delta BTE &= \sqrt{\left(\frac{\partial BTE}{\partial P_e} \cdot \delta P_e\right)^2 + \left(\frac{\partial BTE}{\partial \dot{m}_{MeOH}} \cdot \delta \dot{m}_{MeOH}\right)^2} \\ &= \frac{3600}{1000 \cdot LHV_{MeOH}} \sqrt{\left(\frac{1}{\dot{m}_{MeOH}} \cdot \delta P_e\right)^2 + \left(\frac{-P_e}{\dot{m}_{MeOH}^2} \cdot \delta \dot{m}_{MeOH}\right)^2} [-] \end{aligned} \quad (C.6)$$

Specific emissions

To calculate the specific emissions, the wet emissions must first be determined. The procedure for calculating wet emissions, starting from the dry emissions measured by the Siemens Ultramat gas analyzer in the emissions cabinet, is detailed in Appendix B of the master thesis by Verbiest and Janvier [84]. For further information on the calculation of specific emissions, readers are also directed to the same dissertation.

The mass flow rate of the CO and NO_x emissions are calculated according to Equation C.7 and Equation C.8, respectively.

$$\dot{m}_{CO} = \frac{CO_{wet}}{100} \cdot u_{CO} \cdot (\dot{m}_{air} + \dot{m}_{MeOH}) \left[\frac{g}{h} \right] \quad (C.7)$$

$$\dot{m}_{NO_x} = NO_{x,wet} \cdot u_{NO_x} \cdot (\dot{m}_{air} + \dot{m}_{MeOH}) \left[\frac{g}{h} \right] \quad (C.8)$$

The wet CO emissions are expressed in vol% and the wet NO emissions in ppm. The factors u_{CO} and u_{NO_x} are assigned values of 9.66 and 0.001587, respectively. The specific emissions are defined as the ratio of the mass flow rate of the respective emission to the brake power [21]:

$$X_{specific} = \frac{\dot{m}_X}{P_e} \left[\frac{g}{kWh} \right] \quad (C.9)$$

Where X denotes the respective emission. The uncertainties on the specific emissions are calculated according to Equation C.10. It is important to note that the factor $\frac{1}{100}$ only holds in the case of CO [21].

$$\begin{aligned} \delta X_{specific} &= \left\{ \left(\frac{\partial X_{specific}}{\partial X_{wet}} \cdot \delta X_{wet} \right)^2 + \left(\frac{\partial X_{specific}}{\partial P_e} \cdot \delta P_e \right)^2 \right. \\ &\quad + \left(\frac{\partial X_{specific}}{\partial \dot{m}_{air}} \cdot \delta \dot{m}_{air} \right)^2 \\ &\quad \left. + \left(\frac{\partial X_{specific}}{\partial \dot{m}_{MeOH}} \cdot \delta \dot{m}_{MeOH} \right)^2 \right\}^{\frac{1}{2}} \left[\frac{g}{kWh} \right] \\ \Rightarrow \delta X_{specific} &= \frac{1}{100} \cdot u_x \cdot \left\{ \left(\frac{(\dot{m}_{air} + \dot{m}_{MeOH})}{P_e} \cdot \delta X_{wet} \right)^2 \right. \\ &\quad + \left(\frac{-X_{wet} \cdot (\dot{m}_{air} + \dot{m}_{MeOH})}{P_e^2} \cdot \delta P_e \right)^2 \\ &\quad \left. + \left(\frac{X_{wet}}{P_e} \cdot \delta \dot{m}_{air} \right)^2 + \left(\frac{X_{wet}}{P_e} \cdot \delta \dot{m}_{MeOH} \right)^2 \right\}^{\frac{1}{2}} \left[\frac{g}{kWh} \right] \quad (C.10) \end{aligned}$$

Statistical analysis

The statistical analysis consist of determining the average uncertainty on the measurements $x_1, \dots, x_n$ in a certain operating point. The standard deviation σ_x is used to quantify this average uncertainty and is defined by Equation C.11 [21]. The standard deviation of the average result of the measurement set is then given by Equation C.12

$$\sigma_x = \sqrt{\frac{1}{n-1} \sum_{i=1}^n (x_i - \bar{x})^2} \quad (C.11)$$

$$\sigma_{\bar{x}} = \frac{\sigma_x}{\sqrt{n}} \quad (C.12)$$

The authors are aware that repeating these measurements multiple times would lead to smaller deviations and more powerful results. However, due to the extensive measurement set and time constraints, measurements could only be conducted once. As such σ_x equals $\sigma_{\bar{x}}$ and no smaller standard deviation is obtained.

Appendix D

Transition from MATLAB to Python for Data Processing

Accurate data processing is essential in engine research, especially when analyzing combustion characteristics and emissions. In this master’s dissertation, engine measurement data was processed using a Python-based data processing system developed by the authors as a direct replacement for a previously used MATLAB-based system [21]. This transition from MATLAB to Python was motivated by the need for greater flexibility, more efficient data handling, and the fact that Python is open-source and faster for this type of data processing. Python is also becoming increasingly popular among engineers, who are now commonly taught it as their primary programming language.

The original MATLAB scripts, while functional, contained several critical errors that affected the accuracy of the processed results. These errors were identified and systematically corrected in the Python implementation. The transition also introduced a more maintainable structure, significantly improving the clarity of the data processing.

This appendix provides an overview of the transition process. A detailed explanation of the Python structure is provided, alongside a thorough discussion of the critical corrections made, including the implementation and the effect of Exhaust Gas Recirculation.

Python Processing

The Python-based data processing system developed for this thesis is structured within a single Python file, but it is divided into two main sections: a main processing script and a dedicated calculation class *VolvoPentaD7Calculations*. This structure was designed to maintain a clear separation between data management and core calculations, adapted from the previously MATLAB-files. The *measurement_basefile*-MATLAB file corresponds with the main processing Python script, while the *measurement_calculations*-MATLAB file corresponds with the class *VolvoPentaD7Calculations* in Python. First, the main processing script is discussed, followed by a discussion delving deeper into the explicit calculations.

Main Processing Script

The main processing script is the control center of the Python-based data processing system and ensures each test case is processed systematically. This script begins by loading the Excel measurement file, which serves as the central database for all test conditions. This Excel file is generated directly from LabView, containing essential measurement data for each test such as the names of the corresponding .prn files, the engine operating conditions (engine speed, torque, air and fuel flow rates, various temperatures), ambient conditions and volumetric efficiencies. However, exhaust gas concentrations (CO , CO_2 , $\text{NO}_{(x)}$, O_2) are entered manually into the Excel file after the measurements are completed, as this is not automatically logged from the emission cabinet.

Each prn-file is a text file containing raw voltage signals for pressure measurements, specifically in-cylinder, intake, and exhaust. These voltage values are converted to actual pressure values using calibration constants, transforming the raw data into meaningful pressure information.

Once the measurement data is loaded and all fixed parameters are assigned, the main script initiates a control loop, processing each test measurement (.prn-file) one by one. For each test case, it initializes an instance of the `VolvoPentaD7Calculations` class, which is responsible for all core calculations. The .prn file for the test is read, and the voltage values are immediately converted to pressure values. The main script then calls the calculation methods of the class, performing all necessary analyses, including pressure processing, temperature processing, performance metrics, heat release rate, emission calculations, combustion durations and uncertainties.

The processed results are saved to six separate Excel files, each containing a specific aspect of the analysis. The first excel file contains all performance data for the different measurements taken. All other excel files contain pressures, temperatures, HRR, MFB ... as a function of the crank angle.

- **Performance Data:** Contains calculated performance metrics, including engine power, efficiency, fuel consumption, COV, combustion durations, emission values (CO , CO_2 , NO_x , O_2), exhaust temperatures, uncertainties etc.
- **Pressure Data:** Provides detailed pressure information, including in-cylinder, intake and exhaust pressure for each measurement.
- **Temperature Data:** Includes calculated temperature for the in-cylinder mixture (T_{mix}).
- **Heat Release Rate Data:** Contains HRR calculations, including specific heat and pressure derivatives.
- **Knock Detection Data:** Lists Maximum Amplitude of Pressure Oscillation (MAPO) and their frequencies, indicating knock intensity.
- **MFB Data:** Presents MFB values, showing combustion progress as a function of crank angle, both on a mass basis (x_{burned}) and molar basis (y_{burned}).

Calculations Class: VolvoPentaD7Calculations

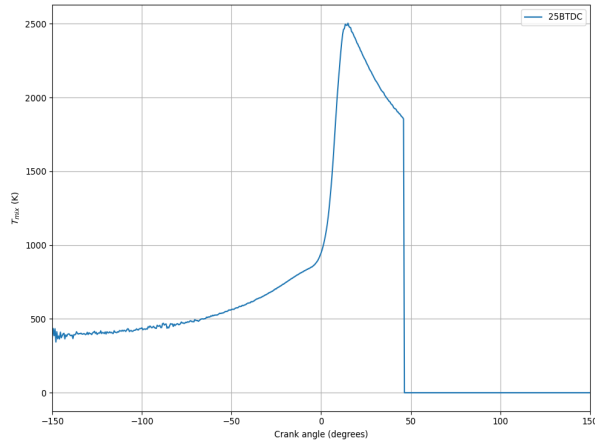
The VolvoPentaD7Calculations class is the core of the Python-based data processing system. It is designed to perform all key calculations in a modular and object-oriented manner, ensuring that each step of the data processing is clear and consistent. For an understanding of the real calculations, formulas and underlying principles of pressure processing, HRR ..., the reader is referred back to subsection 4.4.3.

Key Corrections Made in Python

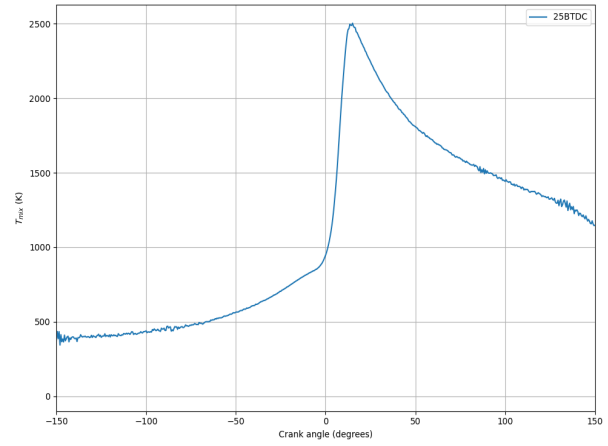
During the transition from MATLAB to Python, several calculation errors and methodological limitations that directly impacted the accuracy of the processed results were discovered.

Compression Ratio A first correction was the adjustment of the compression ratio, which was incorrectly set to 19 in the MATLAB code instead of the correct value of 17.6. This error influenced the calculated clearance volume, introducing inaccuracies in the temperature processing. As such small errors were introduced in calculating specific heat, HRR and combustion durations. However, adaptation of this compression ratio did not lead to significant differences compared to previously obtained results.

In-cylinder Temperature The calculation of the in-cylinder mixture temperature (T_{mix}) was another critical area of improvement. In the MATLAB script, T_{mix} was incorrectly calculated, with the temperature dropping to 0K beyond a certain crank angle due to an error in the mass calculation of the exhaust gas. This issue was caused by a unit mismatch, where the mass of the exhaust gas was calculated using an incorrect value, leading to a physically impossible temperature profile. In Python, this was corrected by ensuring that the mass of the exhaust gas was calculated using the correct variable, maintaining a physically realistic temperature profile for T_{mix} across the entire combustion period where the valves are closed. The effect is visible in Figure D.1. It has to be noted that as in-cylinder temperatures are calculated based on the ideal gas law, these temperatures are only valid when all valves are closed, i.e. between 135°CA BTDC and 120°CA ATDC. As a result of this corrected temperature, the evolution of the specific heat ratio as a function of the crank angle was more realistic, as can be seen in Figure D.2. This also influenced heat release calculations and the combustion duration metrics being shorter and more comparable to results from literature now [39].

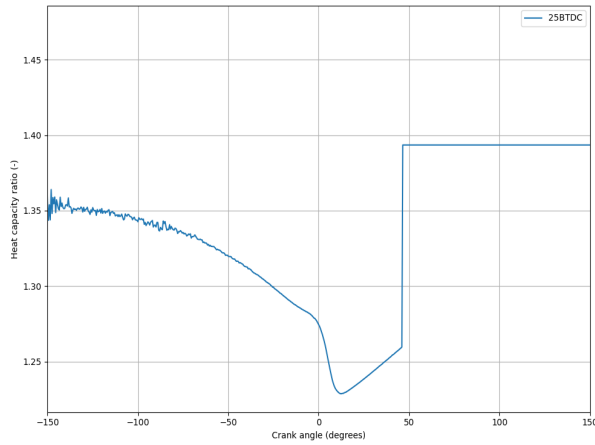


(a) Original Matlab processing file

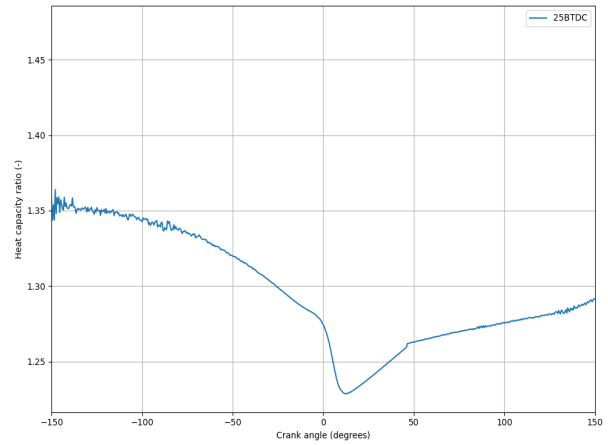


(b) New Python processing file

Figure D.1: In-cylinder temperature at 1500 rpm and 3.5 bar BMEP for Matlab processing file and Python processing file.



(a) Original Matlab processing file



(b) New Python processing file

Figure D.2: Heat capacity ratio at 1500 rpm and 3.5 bar BMEP for Matlab processing file and Python processing file.

EGR implementation The new python implementation consistently integrates the effect of recirculated exhaust gases, if present. The recirculated exhaust mass is calculated based on the specified EGR percentage, by rearranging Equation G.1. This inclusion is particularly significant in the determination of HRR and emissions, where the presence of inert components in the burned gas mixture can significantly influence thermodynamic and chemical characteristics.

Appendix E

LabVIEW Changes

The goal of this appendix is not to provide a comprehensive explanation of the LabVIEW program's functionality but rather to highlight some minor improvements made, resulting in a more reliable and stable program performance. These improvements were necessitated by recurring errors during measurement sets, which were traced back to the Central Processing Unit (CPU) of the control computer running at 100% capacity.

Initial diagnostics revealed that the LabVIEW program was consuming approximately 85% of the available CPU capacity. This high resource usage was problematic, especially during extended measurement campaigns, where it occasionally caused LabVIEW to crash or become unresponsive. Such failures were particularly concerning when operating near the knock limit, posing a safety risk. Moreover, there was a data loss whenever these errors occurred.

The VI Analyze tool, accessible via Tools > VI Analyzer in LabVIEW, proved very valuable during the troubleshooting process. This tool allowed for a systematic analysis of the program identifying areas where performance could be improved. A detailed review revealed that the while loops associated with the control of the EGR valves and associated with the stop procedure, were consuming an excessive amount of CPU resources. These loops were running without any form of delay, causing them to execute at maximum speed and monopolize CPU capacity.

To resolve this issue, wait functions were strategically introduced in the while loops controlling the EGR valves and the stop procedure. This ensured that the loops only performed their tasks at a controlled frequency, significantly reducing CPU usage from 85% to 40%. At this optimized level, the system maintained stable performance even during extended measurement periods. An example of the code modification, showing the addition of the 100 ms wait statements in the EGR control loops, is given in Figure E.1 as a visual reference. This modification effectively prevented further errors and ensured the reliability of measurement data.

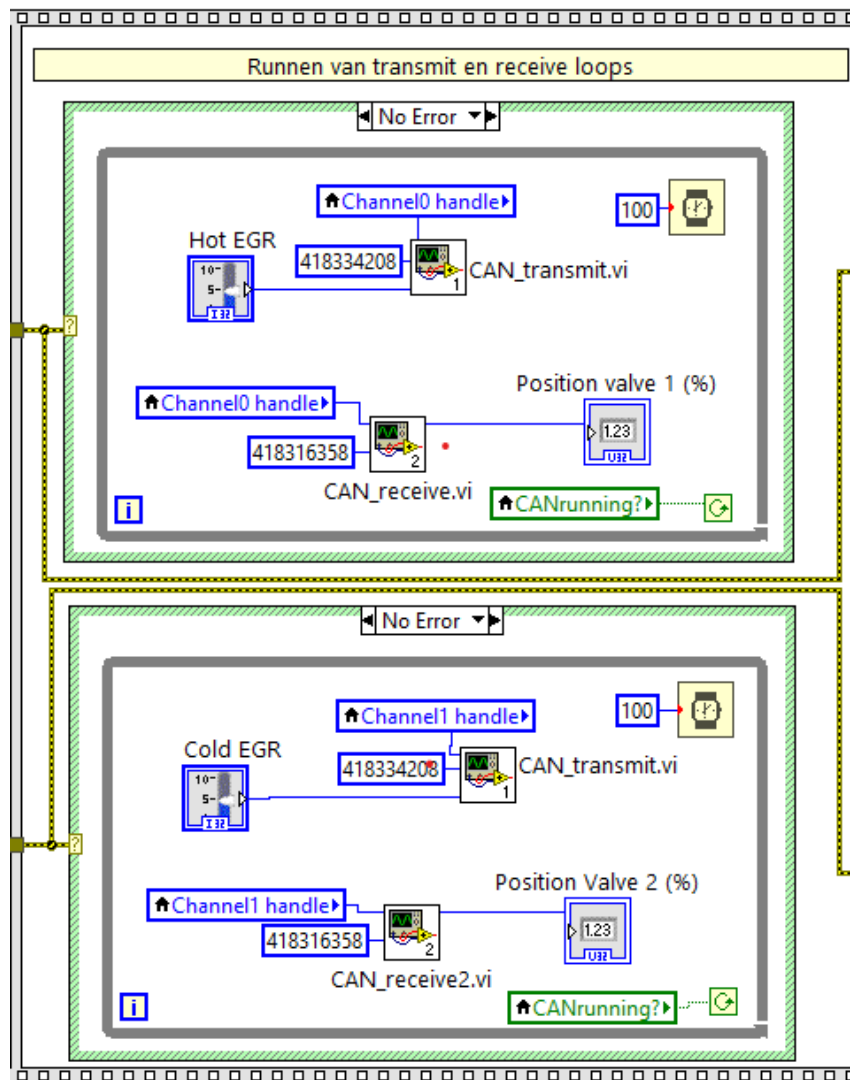


Figure E.1: Added wait-statements in EGR control loops reducing CPU usage.

Appendix F

Encountered Technical Issues and Solutions

This appendix provides an overview of a few technical issues encountered during operation of the experimental setup and the solutions that were developed to ensure reliable measurements.

Malfunctioning methanol injector

One issue observed was significant variation in the exhaust temperature of a certain cylinder. This fluctuation indicated inconsistent combustion in that cylinder, compromising measurement accuracy. A detailed investigation identified the cause as a malfunctioning methanol injector. Replacing this injector immediately stabilized the exhaust temperature, restoring consistent combustion. A visual representation of this issue is shown in Figure F.1.

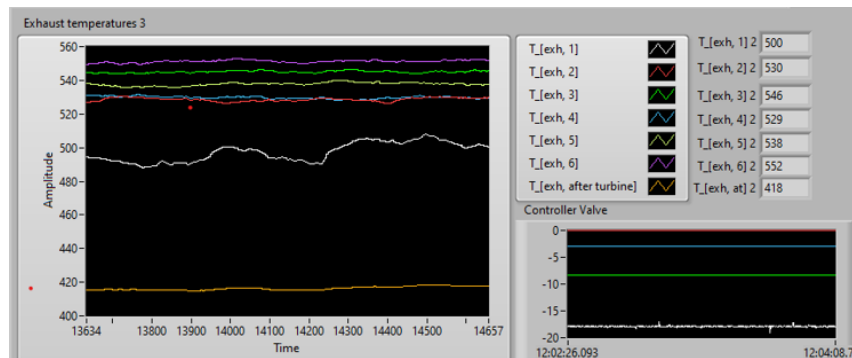


Figure F.1: Fluctuating exhaust temperature of cylinder 1 indicating a malfunctioning methanol injector for that cylinder.

Faulty spark plug

Another problem occurred when a cylinder stopped firing during operation. This issue was easily recognizable by a steep and rapid decline in the exhaust temperature of the affected cylinder. If this temperature drop continues to occur when restoring the operating point, it is highly likely that the spark plug is not reliable anymore. The solution is straightforward: replacing the spark plug of the affected cylinder immediately restored normal combustion. Figure F.2 visually demonstrates the encountered temperature drop.

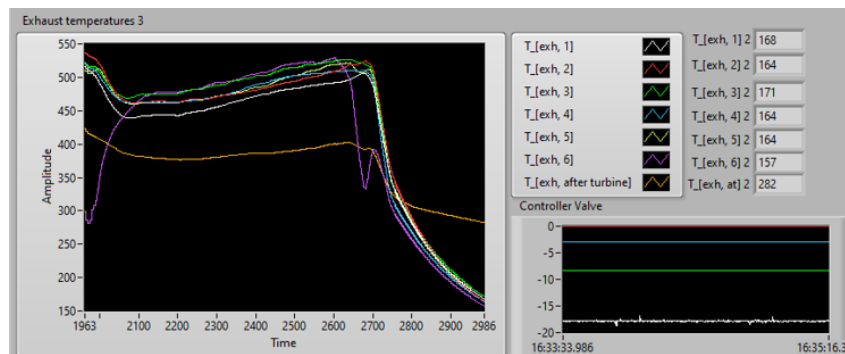


Figure F.2: Dropping of exhaust temperature of cylinder 6 indicating a faulty spark plug.

Corroded EGR Valve

A third issue involved an EGR valve failing to respond to slider adjustments in the control interface of LabVIEW. This problem was primarily due to internal corrosion of the valves, likely caused by methanol exposure. The most effective solutions are either performing a deep clean of the affected valve or replacing it entirely. The corroded valve and new EGR valve are both shown in Figure F.3



Figure F.3: Corroded EGR Valve (left) and new EGR Valve (right).

Pumps supplying water brake system and engine cooling circuit

As already stated in chapter 8, it was not feasible during this research to perform load sweeps due to persistent issues with the pumps supplying rainwater to the water brake system and the engine cooling circuit. A comprehensive overview of the problem and proposed solution was already represented in chapter 8.

Appendix G

Determining EGR rate

As already explained in subsection 3.3.1, there are different ways to quantify the used EGR rate [63] [64]. In this master's dissertation, the EGR rate was expressed in its most common way given by Equation G.1, by taking the ratio between the mass flow rate of EGR and the total mass flow rate consisting of recirculated exhaust gases, fresh air and methanol, following the approach of De Cock and Devos [83].

$$\text{EGR}(\%) = \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{EGR}} + \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}} \cdot 100 \quad (\text{G.1})$$

The mass flow rates of methanol and air are directly measured and displayed in LabVIEW. In contrast, the EGR mass flow rate is not directly measured but is instead calculated using other measured parameters. Specifically, it is determined as the product of the molecular weight of the recirculated exhaust gases and the molar EGR flow rate, illustrated by Equation G.2.

$$\dot{m}_{\text{EGR}} = MW_{\text{EGR}} \cdot \dot{n}_{\text{EGR}} \quad (\text{G.2})$$

Equation G.2 now consists of two terms to still determine: MW_{EGR} and $\dot{n}_{\text{EGR}}$.

Molecular weight

The molecular weight is determined based on the composition of the exhaust gas. Since not all components are measured, this calculation assumes complete stoichiometric combustion of the fuel. As a result, emissions of NO_x , CO and hydrocarbons are not considered in the calculation. However, their exclusion is expected to have only a minimal impact on the overall molecular weight of the exhaust gas. The molecular weight of the recirculated exhaust gas is calculated using Equation G.3

$$MW_{\text{EGR}} = y_{\text{CO}_2} \cdot MW_{\text{CO}_2} + y_{\text{H}_2\text{O}} \cdot MW_{\text{H}_2\text{O}} + y_{\text{N}_2} \cdot MW_{\text{N}_2} + y_{\text{O}_2} \cdot MW_{\text{O}_2} \quad (\text{G.3})$$

The stoichiometric combustion of methanol is written out in Equation G.4. The different molar fractions and molecular weights of the different exhaust products are represented

in Table G.1. This makes it possible to calculate the molar fractions of the exhaust gas of Equation G.3 resulting in the value of $MW_{EGR} = 27.55 \frac{kg}{kmol}$.

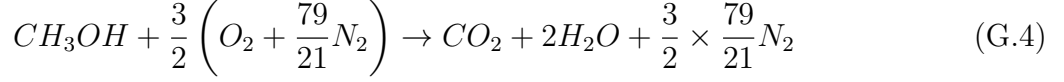


Table G.1: Molar Fractions and Molecular Weights of EGR Components.

Component	Molar Fraction	Molecular Weight (kg/kmol)
CO ₂	0.1157	44.01
H ₂ O	0.2314	18.02
N ₂	0.6529	28.01
O ₂	0	32.00

Molar EGR flow rate

The molar EGR flow rate $\dot{n}_{EGR}$ in Equation G.2 can be defined using Equation G.5 where $y_{EGR,mix}$ and $y_{air,mix}$ represent the mole fractions of EGR and air in the mixture, respectively. These fractions are determined based on the mole fractions of the components in ambient air, exhaust gas, and intake mixture. These concentrations are measured using a gas analyzer. The most common approach for determining these values is through a CO_2 molar balance, as shown in Equation G.6. Alternatively, balances based on O_2 , NO_x , or CO can also be used.

$$\dot{n}_{EGR} = \frac{y_{EGR,mix}}{y_{air,mix}} \cdot \dot{n}_{air} \quad (G.5)$$

$$\frac{y_{EGR,mix}}{y_{air,mix}} = \Delta y_{CO_2} = \frac{[CO_2]_{intake} - [CO_2]_{ambient}}{[CO_2]_{exhaust} - [CO_2]_{ambient}} \quad (G.6)$$

The molar air flow rate $\dot{n}_{air}$ from Equation G.5 is expressed by Equation G.7 with air assumed to be composed of 21% O_2 and 79% N_2 resulting in a molecular weight $MW_{air} = 28.84 \frac{kg}{kmol}$

$$\dot{n}_{air} = \frac{\dot{m}_{air}}{MW_{air}} \quad (G.7)$$

The mass flow rate of EGR can now be calculated, allowing for the determination of the applied EGR percentage during the experiment. The intake CO_2 percentage has the most significant influence on the calculated EGR rate. During the experiments, the EGR valve was adjusted until the intake CO_2 concentration matched the target EGR Rate (5%, 10%, 15%, 17.5%, 20% or 22.5%). Once this target concentration was reached, the emission cabinet was switched back to the exhaust, and the mass flows of fresh air and methanol were used in the formula to verify the actual EGR rate. If the calculated EGR rate did not match the target, an iterative adjustment process was performed. The EGR valve was further opened or closed until the desired operating point was achieved.

Appendix H

Throttle Valve Replacement

This year, the experimental setup's throttle control system underwent a significant upgrade to resolve persistent issues experienced with the previous configuration. Last year's researchers, Demiddeleer and Vanderbeken, reported difficulties in achieving consistent load control with the SI throttle valve, as documented in Appendix F of [21]. These issues included unreliable throttle response and calibration challenges, primarily due to the use of LabVIEW for throttle control. To address these problems, the existing electronic throttle body (ETB1) was replaced with a, 57 mm diameter and normally closed, VDO 408-238-327-003Z throttle valve (OE 06F 133 062), shown in Figure H.1. This change marked a complete transition from LabVIEW control to a MoTeC-based Drive-By-Wire (DBW) system.



Figure H.1: VDO 408-238-327-003Z Throttle Valve.

The new VDO throttle valve was installed following a precise procedure to ensure compatibility with the MoTeC software. This valve is passively cooled via two brass hose fittings connected to the engine's coolant system. This process involved acquiring a com-

patible connector, wiring the valve to the MoTeC ECU. A detailed overview of the pin configuration of this connector is provided in Figure H.2. The throttle position sensors (TP1 and TP2) were mapped to analog inputs (AV7 and AV8), while the motor control was assigned to AUX1 (Motor Control +) and AUX2 (Motor Control -).

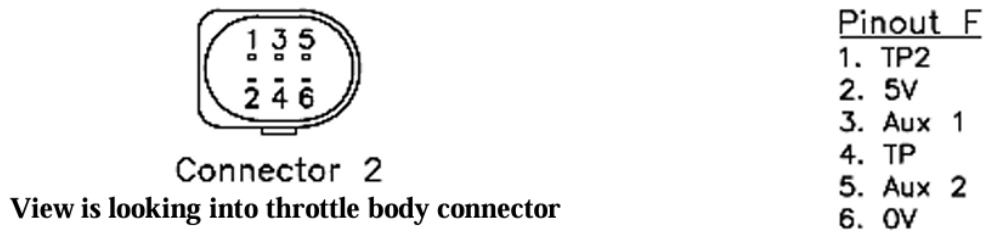


Figure H.2: Connector and pinout for throttle body connector.

Table H.1 illustrates the pins used to connect the throttle valve to the MoTeC M800 module, based on the MoTeC DBW user manual. Figure H.3 shows the pin locations on the MoTeC M800 module within the electrical cabinet of the engine setup.

Table H.1: Pin configuration for connecting the SI throttle valve to the MoTeC M800 module, following the guidelines provided in the MoTeC DBW user manual.

DBW pin function	Cable Color	ECU pin function	M800 pin no
TP Sensor 2 (Signal)	red	AV8 (Analog Input)	B22
5V reference (Power)	blue	5V Supply	A9
Motor Control (+) (Aux 1)	black	Aux 1 (Motor control +)	A1
TP sensor 1 (Signal)	white	AV7 (Analog Input)	B21
Motor Control (-) (Aux 2)	gray	Aux2 (Motor control -)	A18
0V (Ground)	brown	0V reference	B15

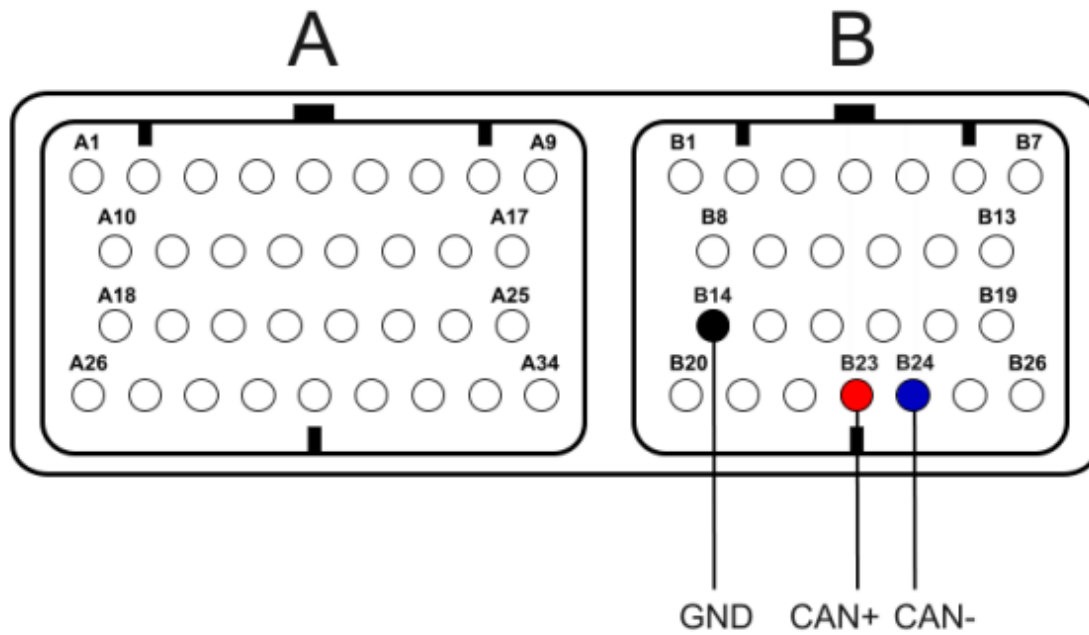


Figure H.3: Pin layout of the MoTeC M800 module using the MoTeC DBW user manual.

By connecting the cables to the pin layout of the MoTeC M800 module via Table H.1, the throttle body is now connected to MoTeC in a correct way.

Additionally, a series of new monitoring charts were added to the MoTeC setup, displaying key parameters including Throttle Position (TP), Throttle Position 2 (TP2), Throttle Driver Position (TP Driver), Throttle Driver Position 2 (TP Driver2), and Throttle Position Required (TP Required). These charts provide clear visual feedback, allowing for real-time monitoring. The throttle driver input is managed via a dedicated potentiometer. TP and TP2 show the throttle butterfly movement, while the TP required showed the butterfly position the ECU is aiming for.

Calibration of the throttle valve was performed using the MoTeC ECU Manager. The throttle's open and closed positions were carefully defined, ensuring consistent operation across the full range of movement. The DBW parameters including PID controller were installed according to Table H.2.

Compared to the previous setup, this upgrade eliminated the need for LabVIEW in throttle management, reducing system complexity and improving throttle response. Control is now achieved using a dedicated potentiometer for driver input, with the MoTeC ECU managing throttle response in real-time. This change not only enhanced reliability but also significantly improved the precision of throttle control.

Table H.2: Configuration settings for the VDO Valve (06F 133 062 A) with M800 ECU setup.

Motor Part Number	
VDO	06F 133 062 A
Pinout	F
Connector	2
M400/M600/M800/M880 ECU SETUP	
Proportional Gain	140
Integral Gain	80
Derivative Gain	165
Period	1
Dead Band	0.3
Feed Forward	0
Negative Integral Clamp	-20
Frequency	8000
Motor Volts	14
Feedback Source	0

Appendix I

Procedures for Diluted Operation

This appendix outlines the operational procedures used during experimental testing to achieve stable engine operation under lean and exhaust gas recirculation (EGR) conditions. While alternative approaches may be valid, the methodology described here was found to be effective and repeatable during the course of this research.

Lean Operation

To perform measurements under lean conditions, the engine must first be brought to the desired load point using a stoichiometric air–fuel ratio. This is achieved by increasing the load via the water brake while simultaneously opening the throttle valve. These adjustments allow the engine to maintain a stable operating speed, typically within the range of 1200 to 1700 RPM.

Once the target load is established, the mixture is progressively leaned by further opening the throttle and decreasing fuel injection, ensuring that engine speed remains constant. The air–fuel ratio is monitored in real time using the λ value provided by the MoTeC control system. It is essential to approach this adjustment gradually and with close monitoring, as high levels of dilution can compromise combustion stability.

As demonstrated in this study, spark timing becomes increasingly critical at higher λ values. Advancing the ignition timing is necessary to compensate for slower flame propagation and to maintain stable combustion. Care must be taken to optimize the timing appropriately at each lean condition to avoid misfire or excessive cycle-to-cycle variation.

EGR Operation

The procedure for reaching EGR-diluted measurement points begins similarly to the lean strategy. The engine is first brought to the desired load and speed under stoichiometric conditions. Since the EGR percentage is not available as a live parameter, an iterative approach is required to establish the desired dilution level.

To introduce EGR, the cold EGR valve is incrementally opened. This action displaces a portion of the intake air with recirculated exhaust gas, effectively enriching the mixture ($\lambda < 1$). To maintain stoichiometry and target engine speed, the throttle valve is opened further and the fuel injection is reduced.

Once the engine is stabilized at this new operating point, the intake CO_2 concentration is measured using a gas analyzer. Subsequently, the exhaust CO_2 concentration is recorded. These two values are entered into a pre-programmed Excel sheet to calculate the EGR percentage. If the measured value deviates from the target, further iteration is required by adjusting the EGR valve accordingly, either opening it further or closing it slightly depending on the direction of correction.

As with lean operation, spark timing must be carefully adjusted to preserve combustion stability under elevated EGR conditions. Due to the reduced in-cylinder reactivity and increased heat capacity of the charge, earlier ignition is generally required.

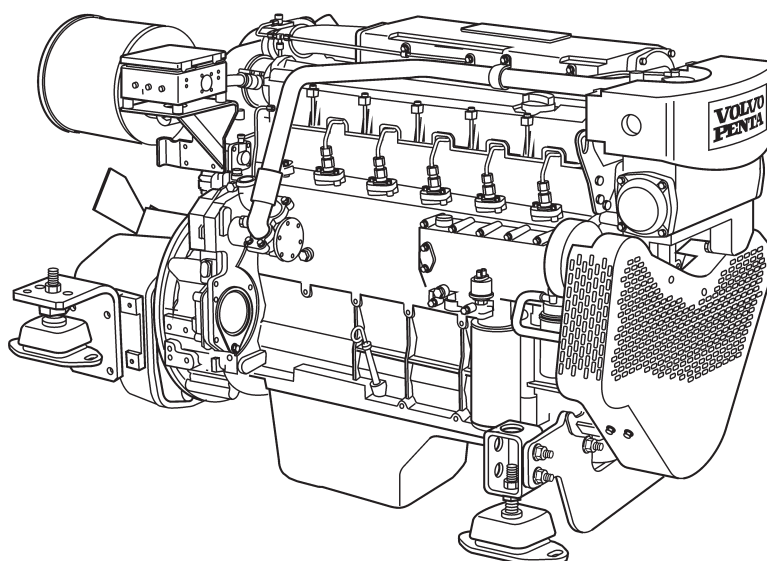
Shutdown and Transition Procedure

For both lean and EGR operation modes, it is strongly recommended to reverse the procedure when shutting down the engine or transitioning back to stoichiometric conditions. Gradually returning to lower dilution levels ensures that the engine does not encounter abrupt knock, misfire, or unstable operation, thereby protecting the engine hardware.

Appendix J

Base Engine Data Sheet

D7C TA

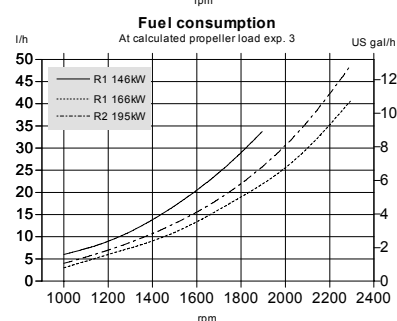
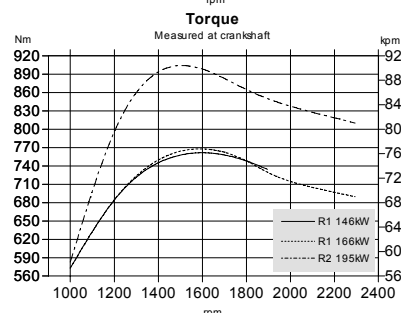
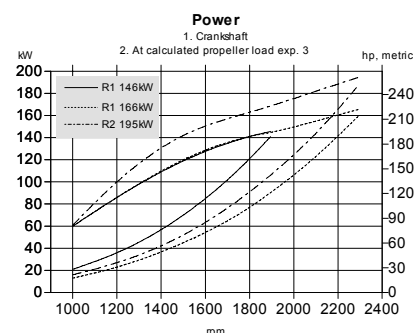


Technical Data

Engine designation	D7C TA		
No. of cylinders and configuration	in-line 6		
Method of operation	4-stroke, direct-injected, turbocharged diesel engine with aftercooler		
Bore/stroke, mm (in.)	108 (4.25)/130 (5.12)		
Displacement, l (in ³)	7.15 (436)		
Compression ratio	17.6:1		
Dry weight bobtail, kg (lb)	690 (1521)		
Dry weight with reverse gear ZF280, kg (lb)	760 (1676)		
	Rating 2 2300 rpm	Rating 1 2300 rpm	Rating 1 1900 rpm
Crankshaft power, kW (hp)	195 (265)	166 (226)	146 (199)
Max. torque, Nm (lbf.ft) @ 1500 rpm	904 (667)	764 (563)	758 (559)
Specific fuel consumption, g/kWh (lb/hph) @ 2300 rpm	228 (0.369)	230 (0.373)	
g/kWh (lb/hph) @ 1900 rpm			213 (0.344)
Recommended fuel to conform to	ASTM-D975 1-D & 2-D, EN 590 or JIS KK 2204		
Emission compliance	IMO NO _x , EU IWW, CCNR2		
Flywheel housing/SAE size	10"/11,5"/SAE2, 10"/11,5"/SAE3, 14"/SAE1		

Technical data according to ISO 3046 Fuel Stop Power and ISO 8665. Fuel with a lower calorific value of 42700 kJ/kg and density of 840 g/liter at 15 °C (60 °F). Merchant fuel may differ from this specification which will influence engine power output and fuel consumption.

Ratings R1 & R2, see explanation in Volvo Penta's Product Guide.



**VOLVO
PENTA**

D7C TA

Technical description:

Engine and block

- Cylinder block, head and flywheel housing made of cast iron
- Belt guard
- Combined 10" and 11.5" flywheel with SAE 2 housing
- Nitrocarburized transmission gears
- Press-hardened crankshaft
- Forged aluminum pistons
- Cylinder liners of wet type

Lubrication system

- Oil filler in valve cover
- Oil sump made of cast iron
- Manual oil drain pump
- Rotary lubrication oil pump
- Freshwater-cooled oil cooler
- Full flow oil filters of spin-on type
- Closed crankcase ventilation system

Fuel system

- Fuel feed pump
- Fine fuel filter of spin-on type
- Unit pumps
- Six-hole injectors
- Mechanical governor with smoke limiter
- Fuel stopping solenoid 24V

Cooling system

- 2-circuit keel cooling system with expansion tank

- Gear-driven seawater pump
- Belt-driven freshwater pump
- Freshwater-cooled turbocharger and exhaust manifold

Electrical system

- 2-pole 24V electrical system, 24V/55A alternator, 24V 4kW starter
- Senders and switches:
 - Tachometer
 - Lubrication oil pressure
 - Cooling-water temperature
 - Cooling-water level
- Rubber-suspended electrical terminal box with semi-automatic fuses and plug-in connection

Optional equipment

Engine

- Flexible suspension for engine and reverse gear
- Combined 10"/11,5" flywheel with SAE 3 housing

Lubrication system

- Shallow oil sump
- Twin oil filters, for remote mounting

Fuel system

- Hand pump
- Jacketed fuel pipes
- Single or twin fuel/water separating pre filter

- Twin fuel filters, for remote mounting

Exhaust system

- Exhaust elbow, wet
- Exhaust elbow, dry, with integrated flexible compensator
- Silencer, dry

Cooling system

- Engine-mounted tubular heat exchanger with integrated expansion tank
- Seawater filter

Electrical system

- 1-pole 12V electrical system, 12V/95A alternator, 12V/3.1kW starter
- 24V/140A alternator
- Engine heater 230V/820W
- Senders:
 - Charge air pressure
 - Gearbox oil pressure
- Cable harness in different lengths
- Various instrument panels

Power transmission

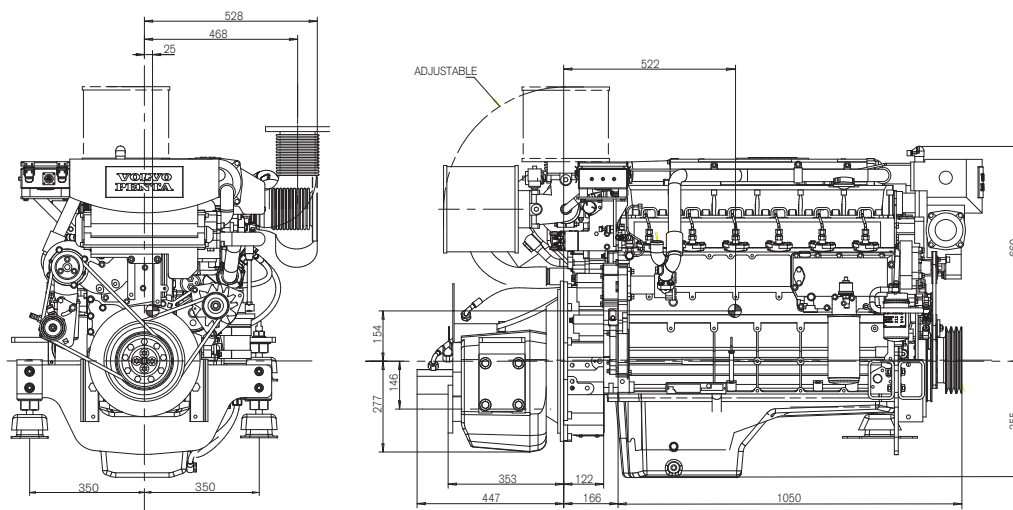
- PTO crankshaft front end, type stub shaft
- Hydraulic pump for steering and other duties

Reverse gear

- ZF280

Dimensions D7C TA/ZF280

Not for installation



More information

Contact your local Volvo Penta dealer for more information regarding Volvo Penta engines and optional equipment/accessories or visit www.volvopenta.com



IOS



Android

Download the Volvo Penta dealer locator App for your iPhone or Android

VOLVO PENTA

AB Volvo Penta

SE-405 08 Göteborg, Sweden
www.volvopenta.com

Not all models, standard equipment and accessories are available in all countries. All specifications are subject to change without notice. The engine illustrated may not be entirely identical to production standard engines.

Appendix K

Emission Results

This appendix extends section 5.4 by showing the effect of dilution strategies on NO and CO emissions at medium and high load conditions (7.1 and 10.6 bar BMEP, respectively). The trends observed were previously discussed in section 5.4.

NO emissions

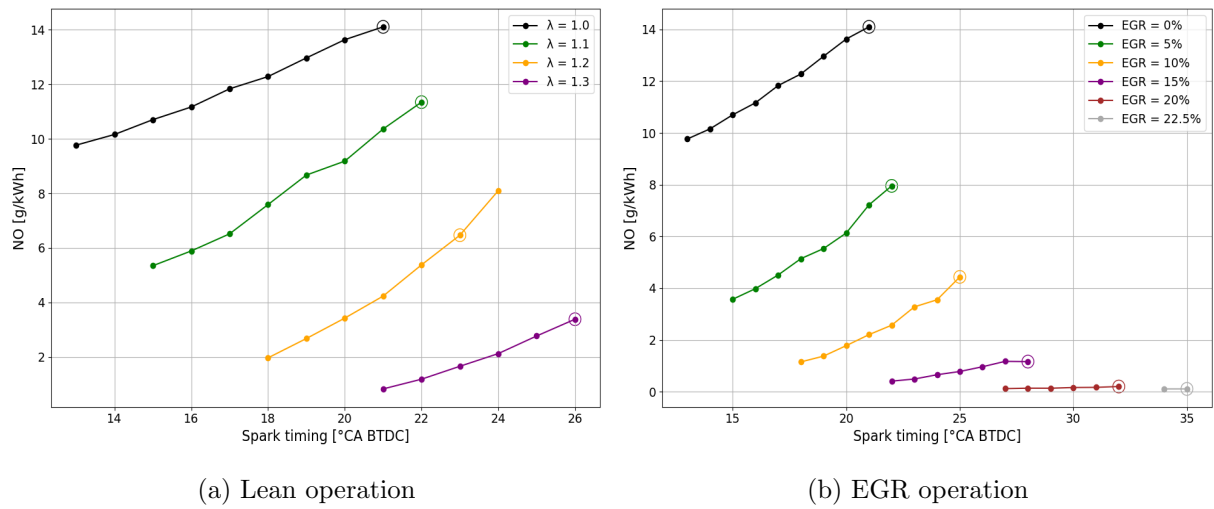
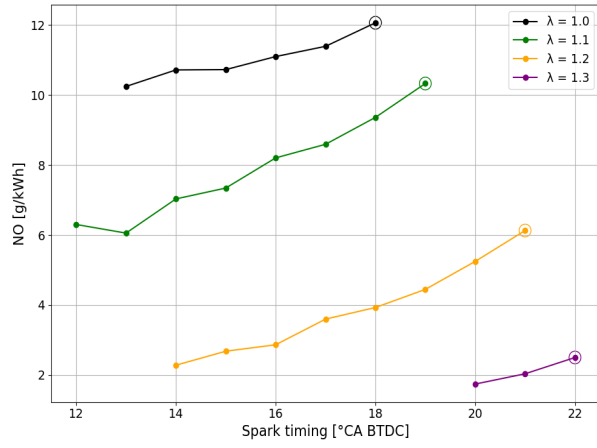
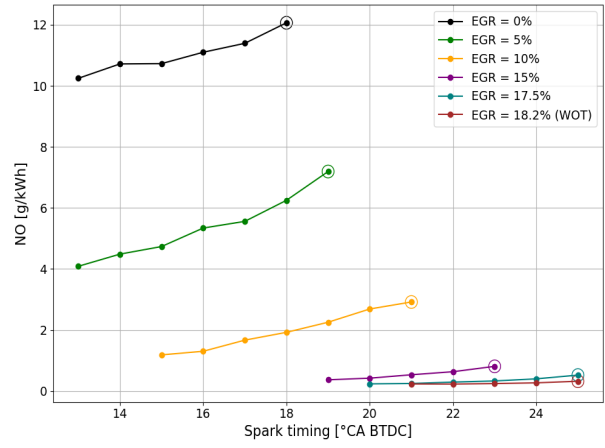


Figure K.1: Influence of dilution strategies on NO emissions at 7.1 bar BMEP



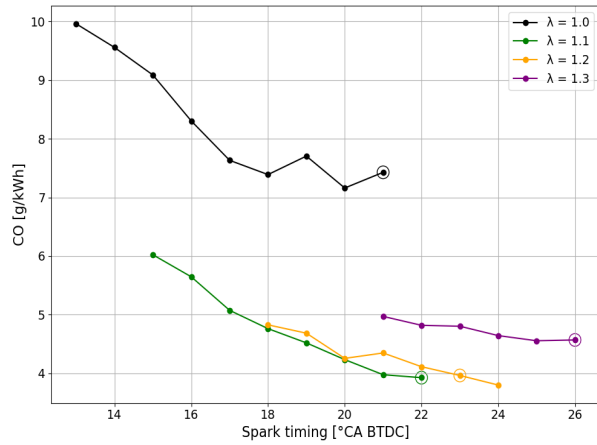
(a) Lean operation



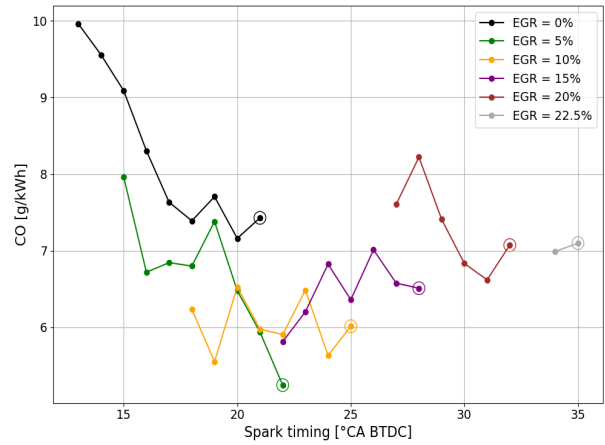
(b) EGR operation

Figure K.2: Influence of dilution strategies on NO emissions at 10.6 bar BMEP

CO emissions

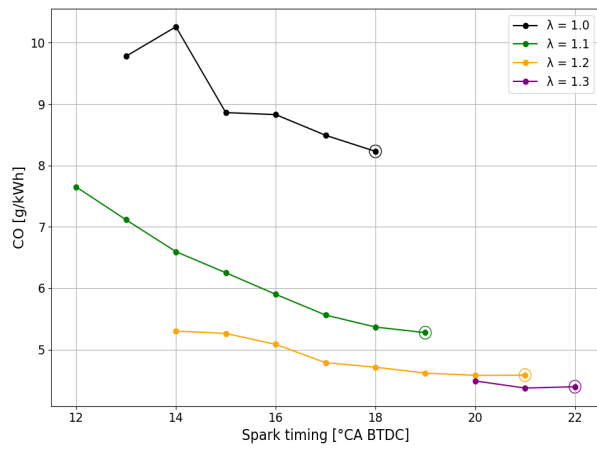


(a) Lean operation

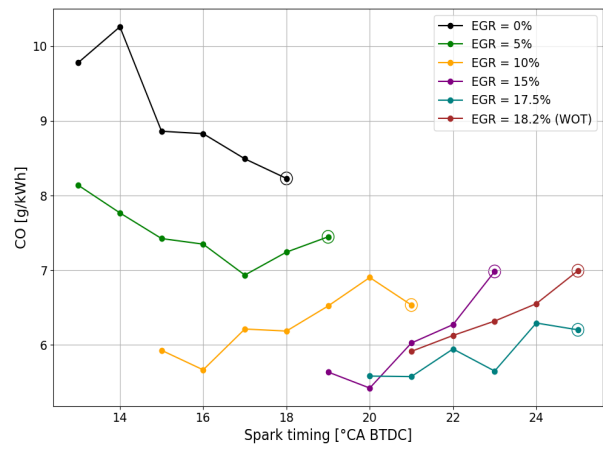


(b) EGR operation

Figure K.3: Influence of dilution strategies on CO emissions at 7.1 bar BMEP



(a) Lean operation



(b) EGR operation

Figure K.4: Influence of dilution strategies on CO emissions at 10.6 bar BMEP

Bibliography

- [1] United Nations Climate change. The paris agreement, Oct 2020.
- [2] Kate Whiting. How 6 heavy-emitting industries are working to decarbonize, Sept 2024.
- [3] M.J. Wernke. Diesel exhaust. In Philip Wexler, editor, *Encyclopedia of Toxicology (Third Edition)*, pages 111–114. Academic Press, Oxford, third edition edition, 2014.
- [4] Ahmad Fayyazbakhsh, Michelle L. Bell, Xingbao Zhu, Xueyi Mei, Marek Koutný, Nima Hajinajaf, and Yexin Zhang. Engine emissions with air pollutants and greenhouse gases and their control technologies. *Journal of Cleaner Production*, 376:134260, 2022.
- [5] Panshuo Li, Anh-Tu Nguyen, Haiping Du, Yan Wang, and Hui Zhang. Polytopic lpv approaches for intelligent automotive systems: State of the art and future challenges. *Mechanical Systems and Signal Processing*, 161:107931, 2021.
- [6] Sebastian Verhelst Quinten Dejaegere. Renewable alternatives for fossil fuels in non-road mobile machinery: A multicriteria analysis. *SAE Technical Paper*, 2023.
- [7] DNV.GL. Comparison of alternative marine fuels, 2019.
- [8] R.J. Pearson and J.W.G. Turner. 2 - the role of alternative and renewable liquid fuels in environmentally sustainable transport. In Richard Folkson, editor, *Alternative Fuels and Advanced Vehicle Technologies for Improved Environmental Performance*, pages 19–51. Woodhead Publishing, 2014.
- [9] Nancy W. Stauffer. Designing better batteries for electric vehicles, August 2021.
- [10] Hongli Diao, Hang Yang, Tan Tan, Gui Ren, Minhua You, Longyue Wu, Mingxiang Yang, Yabin Bai, Shibin Xia, Shaoxian Song, Mildred Quintana, Lei Liu, and Qiang Xue. Navigating the rare earth elements landscape: Challenges, innovations, and sustainability. *Minerals Engineering*, 216:108889, 2024.
- [11] Zachary J. Baum, Robert E. Bird, Xiang Yu, and Jia Ma. Lithium-ion battery recycling-overview of techniques and trends. *ACS Energy Letters*, 7(2):712–719, 2022.
- [12] Zongao Xie, Qihang Jin, Guanli Su, and Wei Lu. A review of hydrogen storage and transportation: Progresses and challenges. *Energies*, 17(16), 2024.
- [13] Debora Mignogna, Paolo Ceci, Claudia Cafaro, Giulia Corazzi, and Pasquale Avino. Production of biogas and biomethane as renewable energy sources: A review. *Applied*

Sciences, 13(18), 2023.

- [14] J. Rochussen, N.S.B. Jaeger, H. Penner, A. Khan, and P. Kirchen. Development and demonstration of strategies for ghg and methane slip reduction from dual-fuel natural gas coastal vessels. *Fuel*, 349:128433, 2023.
- [15] Ebrahim Nadimi, Grzegorz Przybyła, Michał T. Lewandowski, and Wojciech Adamczyk. Effects of ammonia on combustion, emissions, and performance of the ammonia/diesel dual-fuel compression ignition engine. *Journal of the Energy Institute*, 107:101158, 2023.
- [16] L. Bromberg and W.K. Cheng. Methanol as an alternative transportation fuel in the us: Options for sustainable and/or energy-secure transportation. *Massachusetts Institute of Technology*, November 2010.
- [17] Yanhui Zhang, Yunhao Zhong, Shengsen Lu, Zhiqing Zhang, and Dongli Tan. A comprehensive review of the properties, performance, combustion, and emissions of the diesel engine fueled with different generations of biodiesel. *Processes*, 10(6), 2022.
- [18] Rebecca Moore. Stena line orders two methanol dual-fuelled ferries, may 2023.
- [19] Martyn Wingrove. World’s first methanol-fuelled tug named in port of antwerp-bruges, may 2024.
- [20] A.P. Moller Maersk. Maersk to deploy first large methanol-enabled vessel on asia-europe trade lane, december 2023.
- [21] Thomas Vanderbeken Sheldon Demiddeleer. Experimental evaluation of the performance of methanol in a marine diesel engine retrofitted to si operation. Master’s thesis, Ghent University, 2024.
- [22] Ben Gyselinck Kwinten Bracke. Exploring the potential of a heavy-duty ci diesel engine retrofitted to methanol si operation. Master’s thesis, Ghent University, 2023.
- [23] Hardikk Valera and Avinash Kumar Agarwal. *Methanol as an Alternative Fuel for Diesel Engines*, pages 9–33. Springer Singapore, Singapore, 2019.
- [24] Jeroen Dierickx, Jip Verbiest, Tom Janvier, Jens Peeters, L. Sileghem, and Sebastian Verhelst. Retrofitting a high-speed marine engine to dual-fuel methanol-diesel operation: A comparison of multiple and single point methanol port injection. *Fuel Communications*, 7:100010, 03 2021.
- [25] Xudong Zhen and Yang Wang. An overview of methanol as an internal combustion engine fuel. *Renewable and Sustainable Energy Reviews*, 52:477–493, 2015.
- [26] J. Vancoillie, J. Demuynck, L. Sileghem, M. Van De Ginste, S. Verhelst, L. Brabant, and L. Van Hoorebeke. The potential of methanol as a fuel for flex-fuel and dedicated spark-ignition engines. *Applied Energy*, 102:140–149, 2013. Special Issue on Advances in sustainable biofuel production and use - XIX International Symposium on Alcohol Fuels - ISAF.
- [27] Sebastian Verhelst, James WG Turner, Louis Sileghem, and Jeroen Vancoillie. Methanol as a fuel for internal combustion engines. *Progress in Energy and Combustion*

- tion Science*, 70:43–88, 2019.
- [28] Datta Bharadwaz Yellapragada and Govinda Rao Budda. *Methanol—A Sustainable Fuel for SI Engine*, pages 103–137. Springer Singapore, Singapore, 2021.
 - [29] Maritime Page. What is the lifespan of a cargo ship? <https://maritimepage.com/lifespan-of-cargo-ships/>, n.d. Accessed: 2024-11-18.
 - [30] MAN Energy Solutions. Agreement for methanol main engine retrofit solutions signed. <https://www.man-es.com/company/press-releases/press-details/2023/07/06/agreement-for-methanol-main-engine-retrofit-solutions-signed>, 2023. Accessed: 2024-11-18.
 - [31] Jianqin Fu, Banglin Deng, Jingping Liu, Linjun Wang, Zhengxin Xu, Jing Yang, and Gequn Shu. Study of si engine fueled with methanol vapor and dissociation gas based on exhaust heat dissociating methanol. *Energy Conversion and Management*, 79:213–223, 2014.
 - [32] Ankur Kalwar and Avinash Agarwal. *Overview, Advancements and Challenges in Gasoline Direct Injection Engine Technology*, pages 111–147. Advanced Combustion Techniques and Engine Technologies for the Automotive Sector, 01 2020.
 - [33] Joseph O. Amaechi, Chinujinim Godstime Thomas, and Chukwumela Obulor Ochogba. Comparative factors in spark ignition (si) and compression ignition (ci) engines for a sustainable technological economy. *International Journal of Science and Research (IJSR)*, 4(12):67–72, 2015. Licensed under Creative Commons Attribution CC BY.
 - [34] Nguyen, Duc-Khanh and Stepman, Bram and Vergote, Viktor and Sileghem, Louis and Verhelst, Sebastian. Combustion characterization of methanol in a lean burn Direct Injection Spark Ignition (DISI) engine. In *SAE TECHNICAL PAPER SERIES*, page 11. Society of Automotive Engineers (SAE), 2019.
 - [35] Jun Li, Chang-Ming Gong, Yan Su, Hui-Li Dou, and Xun-Jun Liu. Effect of injection and ignition timings on performance and emissions from a spark-ignition engine fueled with methanol. *Fuel*, 89(12):3919–3925, 2010.
 - [36] Matthew Brusstar, Mark Stuhldreher, David Swain, and William Pidgeon. High efficiency and low emissions from a port-injected engine with neat alcohol fuels. *SAE Technical Paper Series*, 2002-01-2743, 2002. Presented at the 2002 SAE Powertrain and Fluid Systems Conference and Exhibition.
 - [37] Senthil Krishnan Mahendar, Anders Erlandsson, and Ludvig Adlercreutz. Challenges for spark ignition engines in heavy duty applications: A review. *SAE Technical Paper*, 2018. © 2018 SAE International. All Rights Reserved.
 - [38] Zhi Wang, Hui Liu, and Rolf D Reitz. Knocking combustion in spark-ignition engines. *Progress in Energy and Combustion Science*, 61:78–112, 2017.
 - [39] Zengqiang Zhu, Zhiqiang Mu, Yanju Wei, Ruiheng Du, and Shenghua Liu. Experimental evaluation of performance of heavy-duty si pure methanol engine with egr.

Fuel, 325:124948, 2022.

- [40] Duc-Khanh Nguyen, Bram Stepman, Viktor Vergote, Louis Sileghem, and Sebastian Verhelst. Combustion characterization of methanol in a lean burn direct injection spark ignition (disi) engine. *SAE Technical Paper Series*, 2019. Published 02 Apr 2019.
- [41] Zhi Wang, Hui Liu, and Rolf D Reitz. Knocking combustion in spark-ignition engines. *Progress in Energy and Combustion Science*, 61:78–112, 2017.
- [42] Nishchay Sharma. Knock model evaluation - gas engine. Master of science thesis, KTH Industrial Engineering and Management, Machine Design, August 2018. Commissioned by AVL, Södertälje. Examiner: Dr. Andreas Cronhjort; Supervisor: Dr. Johan Fjällman.
- [43] P. Bares, D. Selmanaj, C. Guardiola, and C. Onder. A new knock event definition for knock detection and control optimization. *Applied Thermal Engineering*, 131:80–88, 2018.
- [44] Ward Suijs, Stijn Broekaert, Thomas De Cuyper, and Sebastian Verhelst. The sensitivity of pressure-based knock threshold values to alternative fuels: A comparison of methanol vs. gasoline. *Fuel*, 362:130850, 2024.
- [45] Enzo Galloni. Dynamic knock detection and quantification in a spark ignition engine by means of a pressure based method. *Energy Conversion and Management*, 64:256–262, 2012. IREC 2011, The International Renewable Energy Congress.
- [46] Zengqiang Zhu, Zhiqiang Mu, Yanju Wei, Ruiheng Du, Wei Guan, and Shenghua Liu. Cylinder-to-cylinder variation of knock and effects of mixture formation on knock tendency for a heavy-duty spark ignition methanol engine. *Energy*, 254:124197, 2022.
- [47] Stephen Russ. A review of the effect of engine operating conditions on borderline knock. *SAE Technical Paper 960497*, 1996.
- [48] L. Patterson, W. Sutherland, M. Hsu, and L. Smith. A study of the effects of air-fuel ratio and egr on knock in a si engine. *SAE Technical Paper 983026*, 1998.
- [49] Pascal Granier Maxime Jean and Thomas Leroy. Combustion stability control based on cylinder pressure for high efficiency gasoline engines. *Energies*, 2022.
- [50] Bahram Bahri, Azhar Abdul Aziz, Mahdi Shahbakhti, and Mohd Farid Muhamad Said. Understanding and detecting misfire in an hcci engine fuelled with ethanol. *Applied Energy*, 108:24–33, 2013.
- [51] Eric Randolph, Kelsey Fieseler, Graham Conway, Terrence Alger, and Christopher Chadwell. The effects of egr composition on combustion performance and efficiency. *SAE International Journal of Advances and Current Practices in Mobility-V130-99EJ*, 2020.
- [52] Changming Gong, Baoqing Deng, Shu Wang, Yan Su, Qing Gao, and Xunjun Liu. Combustion of a spark-ignition methanol engine during cold start under cycle-by-cycle control. *Energy & Fuels - ENERG FUEL*, 22, 07 2008.

- [53] Zhaohui Li, Changming Gong, Xiang Qu, Fenghua Liu, Jingzhen Sun, Kang Wang, and Yufeng Li. Critical firing and misfiring boundary in a spark ignition methanol engine during cold start based on single cycle fuel injection. *Energy*, 89:236–243, 2015.
- [54] DieselNet. Eu nonroad emission standards, 2024. Accessed: 2024-11-24.
- [55] James W. G. Turner, Andrew G. J. Lewis, Sam Akehurst, Chris J. Brace, Sebastian Verhelst, Jeroen Vancoillie, Louis Sileghem, Felix C. P. Leach, and Peter P. Edwards. Alcohol fuels for spark-ignition engines: Performance, efficiency, and emission effects at mid to high blend rates for ternary mixtures. *Energies*, 13(23), 2020.
- [56] Xin Wang, Yunshan Ge, Linlin Liu, and Huiming Gong. Regulated, carbonyl emissions and particulate matter from a dual-fuel passenger car burning neat methanol and gasoline. *SAE Technical Papers*, 2015, 04 2015.
- [57] Chung Song Ho, Jianfei Peng, UnHyok Yun, Qijun Zhang, and Hongjun Mao. Impacts of methanol fuel on vehicular emissions: A review. *Frontiers of Environmental Science and Engineering*, 16(9):121, 2022.
- [58] Erik Svensson, Changle Li, Sam Shamun, Bengt Johansson, Martin Tuner, Cathleen Perlman, Harry Lehtiniemi, and Fabian Mauss. Potential levels of soot, no_x, hc and co for methanol combustion. Technical Paper 2016-01-0887, SAE International, 2016.
- [59] Arne Güdden, Stefan Pischinger, José Geiger, Benedikt Heuser, and Martin Müther. An experimental study on methanol as a fuel in large bore high speed engine applications – port fuel injected spark ignited combustion. *Fuel*, 303:121292, 2021.
- [60] Yajie Zhang, Zhiqiang Mu, Yanju Wei, Zengqiang Zhu, Ruiheng Du, and Shenghua Liu. Comprehensive study on unregulated emissions of heavy-duty si pure methanol engine with egr. *Fuel*, 320:123974, 2022.
- [61] Xiaoyan Li, Xudong Zhen, Yang Wang, Daming Liu, and Zhi Tian. The knock study of high compression ratio si engine fueled with methanol in combination with different egr rates. *Fuel*, 257:116098, 2019.
- [62] G.H. Abd-Alla. Using exhaust gas recirculation in internal combustion engines: a review. *Energy Conversion and Management*, 43(8):1027–1042, 2002.
- [63] Ward Suijs and Tijl Jooris. Improving part-load performance of a methanol-fueled engine by means of exhaust gas recirculation, 2019. Master of Science in Electromechanical Engineering.
- [64] S. Verhelst, J. Vancoillie, K. Naganuma, M. De Paepe, J. Dierickx, Y. Huyghebaert, and T. Wallner. Setting a best practice for determining the egr rate in hydrogen internal combustion engines. *International Journal of Hydrogen Energy*, 38(5):2490–2503, 2013.
- [65] Haiqiao Wei, Tianyu Zhu, Gequn Shu, Linlin Tan, and Yuesen Wang. Gasoline engine exhaust gas recirculation – a review. *Applied Energy*, 99:534–544, 2012.
- [66] Sham Radhey, Rajesh Saluja, Gurwinder Singh, Vineet Kumar, and Shubham Parmar. Experimental investigation to study the effects of using hot and partially cold

egr in direct injection diesel engine. *TECHNOLOGY*, 04:1–4, 08 2016.

- [67] Xiongbo Duan, Yiqun Liu, Jingping Liu, Ming-Chia Lai, Marcis Jansons, Genmiao Guo, Shiheng Zhang, and Qijun Tang. Experimental and numerical investigation of the effects of low-pressure, high-pressure and internal egr configurations on the performance, combustion and emission characteristics in a hydrogen-enriched heavy-duty lean-burn natural gas si engine. *Energy Conversion and Management*, 195:1319–1333, 2019.
- [68] David B Roth, Philip Keller, and Michael Becker. Requirements of external egr systems for dual cam phaser turbo gdi engines. Technical report, SAE Technical Paper, 2010.
- [69] Cinzia Tornatore, Fabio Bozza, Vincenzo De Bellis, Luigi Teodosio, Gerardo Valentino, and Luca Marchitto. Experimental and numerical study on the influence of cooled egr on knock tendency, performance and emissions of a downsized spark-ignition engine. *Energy*, 172:968–976, 2019.
- [70] Yangyang Li, Peng Wang, Shuqian Wang, Jingping Liu, Yunkun Xie, and Wenjun Li. Quantitative investigation of the effects of cr, egr and spark timing strategies on performance, combustion and nox emissions characteristics of a heavy-duty natural gas engine fueled with 99 *Fuel*, 255:115803, 2019.
- [71] Bin Wu, Lejun Wang, Xinwei Shen, Rongbin Yan, and Peng Dong. Comparison of lean burn characteristics of an si engine fueled with methanol and gasoline under idle condition. *Applied Thermal Engineering*, 95:264–270, 2016.
- [72] Xinlei Liu, Priybrat Sharma, Mickael Silva, Abdullah S. AlRamadan, Emre Cenker, Qinglong Tang, Gaetano Magnotti, and Hong G. Im. Computational investigation of methanol pre-chamber combustion in a heavy-duty engine. *Applications in Energy and Combustion Science*, 15:100192, 2023.
- [73] Bin Wang, Fangxi Xie, Wei Hong, Jiakun Du, Hong Chen, and Yan Su. The effect of structural parameters of pre-chamber with turbulent jet ignition system on combustion characteristics of methanol-air pre-mixture. *Energy Conversion and Management*, 274:116473, 2022.
- [74] R. Novella, J. Gomez-Soriano, I. Barbary, and P.J. Martinez-Hernandez. Exploring the passive the pre-chamber ignition concept for spark-ignition engines fueled with natural gas under egr-diluted conditions. *Energy*, 294:130909, 2024.
- [75] Ponnya Hlaing, Manuel Echeverri, Paula Burgos, Emre Cenker, Moez Ben Houidi, and Bengt Johansson. Analysis of fuel properties on combustion characteristics in a narrow-throat pre-chamber engine. *SAE International Journal of Advances and Current Practices in Mobility*, 3, 04 2021.
- [76] Sipeng Zhu, Sam Akehurst, Andrew Lewis, and Hao Yuan. A review of the pre-chamber ignition system applied on future low-carbon spark ignition engines. *Renewable and Sustainable Energy Reviews*, 154:111872, 2022.
- [77] Yi-Hao Pu, Quinten Dejaegere, Magnus Svensson, and Sebastian Verhelst. Renewable methanol as a fuel for heavy-duty engines: A review of technologies enabling single-

- fuel solutions. *Energies*, 17(7), 2024.
- [78] Karpov V. Gussak L. and Tikhonov Y. The application of lag-process in prechamber engines. *SAE Technical Paper 790692*, 1979.
 - [79] Ashish Shah, Per Tunestal, and Bengt Johansson. Effect of pre-chamber volume and nozzle diameter on pre-chamber ignition in heavy duty natural gas engines. *SAE Technical Papers*, 2015, 04 2015.
 - [80] Wei Li, Junfang Ma, Tao Zhu, Haiqiao Wei, and Jiaying Pan. Nozzle design of plug-and-play passive pre-chamber ignition systems for natural gas engines. *Applied Sciences*, 13(16), 2023.
 - [81] Elisa Toulson, Harold Schock, and William Attard. A review of pre-chamber initiated jet ignition combustion systems. *SAE Technical Papers*, 10 2010.
 - [82] K. Christianen J. Dierickx, S. Verhelst. *Study of the combustion of hydrogen and methanol in dual-fuel engines*. PhD thesis, Ghent University, 2023.
 - [83] S. Devos B. De Cock. Effect of intake conditions on the performance of a diesel-methanol dual-fuel engine. Master’s thesis, Ghent University, 2022.
 - [84] J. Verbiest and T. Janvier. Optimisation of a methanol-diesel dual-fuel marine engine towards a sustainable shipping industry. Master’s thesis, Ghent University, 2018.
 - [85] Q. Dejaegere. The effect of intake air conditions and hydrous methanol on combustion in a diesel-methanol dual-fuel engine. Master’s thesis, Ghent University, 2021.
 - [86] A. Van Gijzeghem. Ontwerp en implementatie van een egr systeem op een diesel-methanol dual-fuel motor. Master’s thesis, Ghent University, 2021.
 - [87] Vancoillie, Jeroen and Verhelst, Sebastian and Demuynck, Joachim and Galle, Jonas and Sileghem, Louis and Van De Ginste, Maarten. Experimental evaluation of lean-burn and EGR as load control strategies for methanol engines. In *SAE TECHNICAL PAPER SERIES*, page 18. Society of Automotive Engineers, 2012.
 - [88] John B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill Education, New York, 2nd edition, 2018.
 - [89] Zhongjie Zhang and Zhaolei Zheng. Numerical simulation of the effects of the egr ratio and ignition timing on a supercharged and high compression ratio hybrid gasoline engine. *Fuel*, 341:127695, 2023.
 - [90] Xiaolong He, Ramalingam Ramanujam, and Paul Miles. Characterization of the tumble flow in a spark-ignition engine using particle image velocimetry and planar laser-induced fluorescence. *International Journal of Engine Research*, 8(2):109–124, 2007. Impact of Tumble on Combustion in SI Engines.
 - [91] Mehmet Kaplan. Influence of swirl, tumble, and squish flows on combustion characteristics and emissions in internal combustion engine: A review. *European Mechanical Science*, 3(3):94–102, 2019.
 - [92] DieselNet. International: Imo marine engine regulations. <https://dieselnet.com/standards/inter/imo.php#nox>, 2025. Accessed: 2025-05-22.

- [93] José Manuel Luján, Héctor Climent, Ricardo Novella, and Manuel Eduardo Rivas-Perea. Influence of a low pressure egr loop on a gasoline turbocharged direct injection engine. *Applied Thermal Engineering*, 89:432–443, 2015.
- [94] Diming Lou, Yedi Ren, Yunhua Zhang, and Xia Sun. Study on the effects of egr and spark timing on the combustion, performance, and emissions of a stoichiometric natural gas engine. *ACS Omega*, 5(41):26763–26775, 2020. PMID: 33111003.
- [95] Tim Watling and Julian Cox. Factors affecting three-way catalyst light-off: A simulation study. *SAE International Journal of Engines*, 7:1311–1325, 05 2014.
- [96] S. Zhu, S. Akehurst, A. Lewis, and H. Yuan. A review of the pre-chamber ignition system applied on future low-carbon spark ignition engines. *Renewable and Sustainable Energy Reviews*, 154:111872, 2022.
- [97] Elisa Toulson, Harold J. Schock, and William P. Attard. A review of pre-chamber initiated jet ignition combustion systems. *SAE Technical Paper*, 2010(01-2263), 2010.
- [98] Ashish Shah, Per Tunestal, and Bengt Johansson. Effect of pre-chamber volume and nozzle diameter on pre-chamber ignition in heavy duty natural gas engines. *SAE Technical Paper*, 2015(01-0867), 2015.
- [99] Ko Onodera, Tomohiro Kato, Satoshi Ogano, Kosuke Fujimoto, Katsuyoshi Kato, and Toyoharu Kaneko. Engine oil formulation technology to prevent pre-ignition in turbocharged direct injection spark ignition engines. In *SAE Technical Paper Series*. SAE International, 2015.