

**UNIVERSIDADE FEDERAL DE MINAS GERAIS
ESCOLA DE ENGENHARIA
PROGRAMA DE PÓS-GRADUAÇÃO EM ENGENHARIA MECÂNICA**

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**DEVELOPMENT AND CHARACTERIZATION OF A STRATIFIED
TURBULENT JET IGNITION SYSTEM USING A DEDICATED PRE-
CHAMBER WITH A FOCUS ON REDUCING FUEL CONSUMPTION USING
ALCOHOL FUELS**

Belo Horizonte

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"DEVELOPMENT AND CHARACTERIZATION OF A STRATIFIED TURBULENT JET IGNITION SYSTEM USING A DEDICATED PRE-CHAMBER WITH A FOCUS ON REDUCING FUEL CONSUMPTION USING ALCOHOL FUELS"

NATHÁLIA DUARTE SOUZA ALVARENGA SANTOS

Tese submetida à Banca Examinadora designada pelo Colegiado do Programa de Pós-Graduação em Engenharia Mecânica da Universidade Federal de Minas Gerais, constituída pelos Professores: Dr. José Guilherme Coelho Baêta (Orientador – Departamento de Engenharia Mecânica/UFMG), Dr. Stefan Pischinger (Coorientador - RWTH Aachen University), Dr. Fernando Antônio Rodrigues Filho (Centro Federal de Educação Tecnológica de Minas Gerais/CEFET-MG), Dr. Mario Eduardo Santos Martins (Universidade Federal de Santa Maria/UFSM), Dr. Guenther Carlos Krieger Filho (Universidade de São Paulo/USP) e Dr. Ramon Molina Valle (Professor aposentado do Departamento de Engenharia Mecânica/UFMG), como parte dos requisitos necessários à obtenção do título de **"Doutora em Engenharia Mecânica"**, na área de concentração de "Energia e Sustentabilidade".

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“If I have seen further it is by standing on the shoulders of Giants.”
– Sir Isaac Newton, 1675

ABSTRACT

Engines equipped with pre-chambers in stratified mode have gained prominence by allowing the combustion of ultra-lean mixtures, which reduces NO_x emissions. Pre-chambers also enable higher compression ratio operation, increasing engine thermal efficiency. As a parallel strategy in engine development, biofuels also stand out, especially ethanol and methanol. The fuels have not only greater market availability and renewable character but also interesting properties when used in spark ignition engines, such as higher laminar flame speed and knock resistance when compared to gasoline. Despite being two promising strategies, few studies today address the combination of both. Hence, the main goal of this work is to study an optimal biofuel to operate in an engine with stratified pre-chambers, evaluating its ability to extend the lean limit, increase combustion duration, in-cylinder temperature and heat losses to control emissions and fuel consumption. More objectively, such fuels were experimentally tested in various combinations in a single-cylinder research engine at 6 and 15 bar IMEP at 2000 and 3500 rpm. In addition, 1D simulations performed on GT-Power calibrated with experimental data were used to extend the experimental analysis. Initially the effectiveness of pre-chambers for the lean limit extension was confirmed, also providing faster combustion, especially in its second half. Methanol stood out for being able to achieve at some points of operation $\lambda = 2.5$ at 2000rpm and 15 bar IMEP when compared to ethanol, $\lambda = 2.5$ achieving $\lambda = 2.4$ at the same condition. The biofuels alone enabled engine operation with volumetric compression ratio of 16.4, which was also used for pre-chamber operation. Pressure oscillations were observed both at baseline and pre-chamber operation, and the latter with greater amplitude. Such a phenomenon was analyzed using several knock detection methods and the method of comparing the amplitudes before and after the in-cylinder peak pressure was considered the most adequate to evaluate pre-chamber operation. In addition, the engine with baseline ignition was able to reach $\lambda = 2.0$, necessary for significant NO_x reduction, causing no significant extra pre-chamber emission reduction. The increase of turbulence and heat convection in the cylinder walls due to the pre-chamber installation increased heat losses in this configuration, reducing thermal efficiency in more than 2% when comparing configurations with the same fuel in the main chamber, increasing fuel consumption. The engine equipped with pre-chambers was able to slightly reduce HC and CO emissions at lean limits. However, since the lean limits reached by the engine with spark plug ignition were already higher than $\lambda = 1.7$, in which NO_x emissions decrease to almost zero, pre-chambers had no additional impact in emission control. As a conclusion, the biofuels ability to operate at high compression ratio in an engine with spark ignition should be highlighted. Furthermore, engines equipped with pre-chambers present faster combustion (average of 46,5% closer to stoichiometry and 35,3% at lean limit) and higher knock resistance, being able to operate with even higher engine compression ratios, in which they would possibly have more pronounced benefits in terms of thermal efficiency and fuel consumption.

Keywords: biofuels; ethanol; methanol; pre-chambers; stratification; internal combustion engines

RESUMO

O motor equipado com pré-câmaras, principalmente operando ativamente, com combustível sendo injetado também na pré-câmara, permitindo a combustão de misturas ultrapobres, o que reduz a emissão de NO_x . Além disso, o uso de pré-câmaras possibilita a operação de taxas de compressão mais altas, aumentando a eficiência térmica do motor. Como estratégia paralela no desenvolvimento de motores, destaca-se o uso de biocombustíveis, nos quais o etanol e o metanol são os que mais se aproximam da aplicação comercial. Os combustíveis apresentam não apenas maior disponibilidade no mercado e caráter renovável, mas também propriedades interessantes quando utilizados em motores de ignição por centelha, como maior velocidade laminar de chama e resistência à detonação quando comparados à gasolina. Apesar de serem duas estratégias promissoras, poucos estudos hoje abordam a combinação de ambas. Assim, o principal objetivo deste trabalho é estudar um biocombustível ótimo para operar em um motor equipado com pré-câmara estratificada, avaliando sua capacidade de estender o limite de empobrecimento, acelerar a combustão, controlar a temperatura na câmara de combustão e as perdas térmicas para controlar emissões e consumo de combustível. De forma mais objetiva, tais combustíveis foram testados experimentalmente em várias combinações em um motor de pesquisa monocilíndrico em vários pontos de operação (6 e 15 bar de IMEP a 2.000 e 3.500 rpm). Além disso, simulações 1D realizadas em GT-Power calibradas com dados experimentais foram usadas para estender a análise experimental. Inicialmente foi confirmada a eficácia da utilização de pré-câmaras para a extensão do limite de pobreza, bem como a aceleração da combustão, principalmente em sua segunda metade. O metanol se destacou por conseguir atingir em alguns pontos da operação $\lambda = 2,5$ a 2000rpm e 15 bar de IMEP quando comparado ao etanol, atingindo λ 2,4 na mesma condição. Os biocombustíveis por si só possibilitaram a operação do motor com taxa de compressão volumétrica de 16,4, que também foi utilizada para a operação da pré-câmara. As oscilações de pressão foram observadas tanto na ignição inicial quanto na pré-câmara, sendo esta última com maior amplitude. Tal fenômeno foi analisado por diversos métodos de detecção de detonação e o método de comparação das amplitudes antes e depois do pico de pressão no cilindro foi considerado o mais adequado para avaliar o funcionamento da pré-câmara. Além disso, o motor com ignição convencional conseguiu atingir $\lambda = 2,0$, necessário para redução significativa de NO_x , fazendo com que a redução extra de emissão pré-câmara não fosse muito significativa. O aumento da turbulência e da convecção de calor nas paredes dos cilindros devido à instalação da pré-câmara aumentou as perdas de calor nesta configuração, reduzindo a eficiência térmica em mais de 2% quando comparadas as configurações com o mesmo combustível na câmara principal, aumentando o consumo de combustível. O motor equipado com pré-câmaras foi capaz de reduzir ligeiramente as emissões de HC e CO em limites enxutos. No entanto, como os limites de empobrecimento alcançados pelo motor com ignição convencional já eram superiores a $\lambda = 1,7$, em que as emissões de NO_x diminuem para quase zero, as pré-câmaras não tiveram impacto adicional no controle de emissões. Como conclusão, deve-se destacar a capacidade dos biocombustíveis de operar em alta taxa de compressão em um motor com ignição convencional. Além disso, motores equipados com pré-câmaras apresentam combustão mais rápida (média de 46,5% mais próxima da estequiometria e 35,3% no limite de empobrecimento) e maior resistência à detonação, podendo operar com taxas de compressão do motor ainda maiores, nas quais possivelmente apresentam benefícios mais pronunciados em termos de eficiência térmica e consumo de combustível.

Palavras-chave: biocombustíveis; etanol; metanol; pré-câmaras; estratificação; motores de combustão interna

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LIST OF ABBREVIATIONS AND ACRONYMS

After Bottom Dead Center	aBDC
After Top Dead center	aTDC
Aldehydes	RCHO
Ammonia-Oxidizing Bacteria	AOB
Avalanche Activated Combustion	LAG
Before Bottom Dead Center	bBDC
Before Top Dead Center	bTDC
Benzene, Toluene and Xylene	BTX
Blend with 10% Ethanol and 90% Gasoline	E10
Blend with 25% Ethanol and 75% Gasoline	E25
Blend with 85% Ethanol and 15% Gasoline	E85
Blend with 10% Methanol and 90% Gasoline	M10
Blend with 85% Methanol and 15% Gasoline	M85
Bottom Dead Center	BDC
Brazilian Hydrous Ethanol	E100
Carbon Dioxide	CO ₂
Carbon Monoxide	CO
Centro de Tecnologia da Mobilidade	CTM
Compound Vortex Controlled Combustion	CVCC
Compressed Natural Gas	CNG
Compression Ignition	CI
Computational Fluid Dynamics	CFD
Controlled Auto-Ignition	CAI
Crank Angle at 2% Burned Fuel	pacomb02
Crank Angle at Combustion End (99% of fuel burned)	pacombend
Direct Fuel Injection in Spark Ignition Engines	DISI
Direct Injection	DI
Exhaust Gas Recirculation	EGR
Exhaust-Max Lift Timing Angle	vtmop
Exhaust Open Timing Angle at 0mm Lift	vtvo0
Exhaust Valve Open Timing Angle at 1 mm Lift	(vtvo+vtv0)/2
Exhaust Valve Opening	EVO
Exhaust Valve-Open Timing Angle at Input Reporting Lift	vtvo
Fast Fourier Transfer	FFT
Flexible Fuel Vehicles	FFV
Fuel Science Center	FSC
Full-Scale Output	FSO
Heat Release-Based Calibration	HRBC
Heat Release Rate	HRR
Homogeneous Charge Compression Ignition	HCCI
Hydrocarbons	HC
Hydrogen	H ₂
Hydrogen Assisted Jet Ignition	HAJI
Hydrogen to Carbon Ratio	H/C
<i>Ingenieurgesellschaft Auto und Verkehr</i>	IAV
Intake Valve-Closing Timing Angle at 0mm Lift	vtvc
Intake Valve Closure	IVC

Internal Combustion Engine	ICE
<i>Lehrstuhl für Thermodynamik mobiler Energiewandlungssysteme -</i>	TME
Chair of Thermodynamics of Mobile Energy Conversion Systems (English)	
Liquefied Petroleum Gas	LPG
Low Temperature Combustion	LTC
Maximum Brake Torque	MBT
Measurement Value	MV
Methane	CH ₄
Nitric Acid	HNO ₃
Nitric Oxide	NO
Nitrogen	N ₂
Nitrogen Dioxide	NO ₂
Nitrogen Oxides	NO _x
Main Chamber	MC
Methane	CH ₄
Oxygen	O ₂
Research Octane Number	RON
<i>Rheinisch-Westfälische Technische Hochschule</i>	RWTH
<i>Schichtlade Kammer System - Stratified Chamber System (English)</i>	SK
Single-Cylinder Research Engine	SCRE
Spark Ignition	SI
Sulfur	S
Sulfur Dioxide	SO ₂
Start of cycle	st-cycle+
Start of Combustion	SOC
Port Fuel Injection	PFI
Pre-Chamber	PC
Pre-Chamber Ignition Systems	PCIS
Pre-Chamber Injection System	PCI
<i>Programa de Controle da Poluição do Ar por Veículos Automotores -</i>	Proconve
Air Pollution Control Program for Motor Vehicles (English)	
Proportional–integral–derivative	PID
Three-Pressure Analysis	TPA
Three-Way Catalyst	TWC
Top Dead Center	TDC
Turbulence Generating Pot	TGP
Turbulent Jet Ignition	TJI
Universidade Federal de Minas Gerais	UFMG
Water	H ₂ O
Zero Emission Vehicles	ZEV

LIST OF SYMBOLS

Acoustic Energy Flux	I	[MW/m ²]
Air Density	ρ_a	[kg/m ³]
Air/Fuel Ratio	A/F	[-]
Air Mass	m_a	[mg]
Air Mass Flow	$\dot{m}_a$	[mg/s]
Ambient Temperature	T_A	[K]
Angular Velocity	ω	[rad/s]
Bessel Constant related to the Radial Modes i, j	$B_{i,j}$	[-]
Bore Diameter	D	[mm]
Brake Mean Effective Pressure	BMEP	[bar]
Brake Power	P_b	[W]
Brake Specific Fuel Consumption	BSFC	[g/(kW.h)]
Brake Specific HC	bsHC	[g/(kW.h)]
Brake Specific NO _x	bsNO _x	[g/(kW.h)]
Brake Work	W_b	[J]
Clearance Volume	V_c	[mm ³]
Characteristic Combustion Time	T_c	[s]
Characteristic Turbulence Time	T_t	[s]
Coefficient of Variability of IMEP	COV _{IMEP}	[-]
Combustion Duration between 10-90% of MBF	MFB1090	[°]
Combustion Efficiency	η_c	[%]
Compression Ratio	r_c	[-]
Constant at KI2 calculation	k	[-]
Constant coverage factor t-Student	t	[-]
Convection Multiplier during gas exchange	CM _{GE}	[-]
Convection Multiplier during combustion	CM _{COMB}	[-]
Convection Multiplier difference	b	[-]
Convection Multiplier mean value	a	[-]
Discharge coefficient of the pre-chamber nozzle	C_d	[-]
Displaced Volume	V_d	[mm ³]
Distance	d	[m]
Engine Crank Angle	θ	[°]
Engine Speed	N	[rev/s]
Enthalpy of main chamber, if the flow direction is from pre-chamber to the main chamber and vice versa	$h_{PC,MC}$	[kJ/kg]
Enthalpy of Products	H_p	[kJ]
Enthalpy of Reactants	H_R	[kJ]
Expanded uncertainty	U	[-]
Expansion Ratio	r_e	[-]
Final Volume (Otto Cycle)	V_2	[mm ³]
Force	F	[N]
Frequency of the Resonant Mode i, j	$F_{i,j}$	[kHz]
Friction Mean Effective Pressure	FMEP	[bar]
Friction Power	P_f	[W]
Friction Work	W_f	[J]
Fuel Conversion Efficiency	η_f	[%]
Fuel Mass	m_f	[mg]

Fuel Mass Flow	$\dot{m}_f$	[mg/s]
Geometric nozzle opening area of the pre-chamber nozzle	A_T	[mm ²]
Heat Addition	Q_{in}	[J]
Heat of Vaporization	$\Delta h_{vaporization}$	[J]
Heat Rejection	Q_{out}	[J]
Heat Release Rate	$dQ/d\theta$	[J/°]
Heat Release Rate in the Main Chamber	$dQ_{MC}/d\theta$	[J/°]
In-cylinder Pressure	P	[bar]
In-cylinder Temperature	T_{cyl}	[K]
In-cylinder Volume	V	[mm ³]
Indicated Mean Effective Pressure	IMEP	[bar]
Indicated Power	P_i	[W]
Indicated Specific Fuel Consumption	ISFC	[g/(kW.h)]
Indicated Specific Fuel Consumption - Corrected	ISFC _{corr}	[g/(kW.h)]
Indicated Work	W_i	[J]
Indicated Work per Cycle	W_c	[J]
Initial Volume (Otto Cycle)	V_1	[mm ³]
Intermediate convection multiplier	CM_{inter}	[-]
Isentropic Valve Velocity	U_{is}	[mm/s]
Knock Index 2	KI2	[-]
Knock Intensity 2	I2	[-]
Lambda Factor	λ	[-]
Liquid Fuel enthalpy	h_{liq}	[J/kg]
Local average gas velocity	w	[m/s]
Lower Heating Value	LHV	[MJ/kg]
Lower Heating Value - fuel	LHV _{fuel}	[MJ/kg]
Lower Heating Value - gasoline	LHV _{gasoline}	[MJ/kg]
Main Chamber Pressure Trace	P_{MC}	[bar]
Main Chamber Volume	V_{MC}	[mm ³]
Mass of Exhaust Gas Recycled	m_{EGR}	[g]
Mass Burned Fraction	MBF	[°]
Mass Burned Fraction 5%	MBF5	[°]
Mass Burned Fraction 50%	MBF50	[°]
Mass Burned Fraction 60%	MBF60	[°]
Mass Burned Fraction 90%	MBF90	[°]
Mass of Intake Mixture	m_i	[g]
Mass Transfer between Main and Pre-Chambers	dm/dt	[g/s]
Maximum Amplitude of Pressure Oscillations	MAPO	[bar]
Maximum Amplitude of Pressure Oscillations Threshold	MAPO _{th}	[bar]
Maximum In-cylinder Pressure	P_{max}	[bar]
Maximum In-cylinder Temperature	T_{max}	[K]
Maximum Pressure Rise Rate	$MPRR/(dP/dt)_{max}$	[bar/s]
Mean Effective Pressure	MEP	[bar]
Mechanic Efficiency	η_m	[%]
Molecular Mass of Fuel	MM_f	[g/mol]
Molecular Mass of N ₂	MM_{N_2}	[g/mol]
Molecular Mass of O ₂	MM_{O_2}	[g/mol]
Motored cylinder pressure	p_m	[bar]
Net Work	W_{net}	[J]
Number of Crankshaft Revolutions per Cycle	n_r	[-]

Number of Cycles	N_{cyc}	[-]
Number of Moles of N_2	N_{N_2}	[-]
Number of Moles of O_2	N_{O_2}	[-]
Paddle Wheel Speed - Scaled	n_{p_scaled}	[rps]
Percent of Exhaust Gas Recycled	x_{EGR}	[%]
Polytrophic Coefficient - Ratio of Specific Heats	γ	[-]
Power Output	P_{out}	[W]
Pre-Chamber Pressure Trace	P_{PC}	[bar]
Pre-Chamber Volume	V_{PC}	[mm ³]
Pressure	P	[bar]
Reference enthalpy for liquid fuels	$h_{ref-liq}$	[J/kg]
Reference enthalpy for vapor fuels	$h_{ref-vap}$	[J/kg]
Reference Temperature, 298K	T_{ref}	[K]
Ringing Intensity	RI	[MW/m ²]
Sound Speed	a	[m/s]
Specific Fuel Consumption	SFC	[g/(kW.h)]
Specific Heat at Constant Pressure	c_p	[J/(kg.K)]
Specific Heat at Constant Volume	c_v	[J/(kg.K)]
Standard uncertainty of variable f	u_f	[-]
Standard uncertainty of variable f as function of x	u_x	[-]
Standard uncertainty of variable f as function of y	u_y	[-]
Standard uncertainty of variable f as function of z	u_z	[-]
Swirl Coefficient	C_s	[-]
Temperature	T	[K]
Thermal Efficiency	η_t	[%]
Torque	T	[N.m]
Total Volume	V_t	[mm ³]
Tumble Coefficient	C_t	[-]
Uncertainty Type “A”	u_A	[-]
Uncertainty Type “B”	u_B	[-]
Uncertainty combined	u_C	[-]
Universal Gas Constant	R	[J/(kg.K)]
Vapor Fuel Enthalpy	h_{vap}	[J/kg]
Volumetric Efficiency	η_v	[%]
Working-Fluid pressure	p_r	[bar]
Working-Fluid volume	V_r	[m ³]
Working-Fluid temperature	T_r	[K]

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1 INTRODUCTION

With the rising fuel prices and increased concern towards pollutant and greenhouse gas emissions, the use of internal combustion engines presents new challenges and limitations, mainly regarding the efficient use of natural resources with reduced levels of engine-out emissions. With this in mind, there has been much discussion about the use of electric vehicles to replace current vehicles with internal combustion engines. Such an alternative appears to be feasible when observing the non-emission of pollutants by the vehicle during its operation, leading to the erroneous title of zero emission vehicles (ZEV)[1].

However, when the entire energy chain is evaluated, we recognize from the start an increasing gap between supply and demand of noble metals such as nickel, copper and cobalt, necessary in the manufacture of batteries [2]. Moreover, the optimal use of the existing energy storage system technologies for electric vehicles has not yet been achieved [3]. That added to the current cost of batteries, their durability, need for replacement during the vehicle lifespan and several disposal issues hinder in the current scenario their large-scale application and replacement of internal combustion engines (ICE) [4].

It should also be highlighted the fact that emissions from electric cars depend on the primary source used to manufacture electrical energy. Therefore, if produced from fossil sources, the electric vehicle can indirectly impact the greenhouse effect more than ICE. If we consider internal combustion engines operating on biofuels, carbon emissions from electric vehicles can be even more significant [5]. On the other hand, internal combustion engines (ICE) have an already widespread technology, with a wide network for vehicle maintenance and fuel distribution, parts availability, rapid refueling, among other factors. Thus, the problems related to the use of ICEs are known and can be promptly resolved in order to obtain potential gains in the short and medium term. Hence, the direct transition from ICE to purely electrical systems presents several obstacles to be overcome, making it difficult to replace the technology.

Thus, the development of hybrid technologies, in which the internal combustion engine provides the energy for alternative means of propulsion, has proved to be a viable alternative in the short and medium term. Thereafter, in order to meet these needs, industries and researchers have developed different strategies and technologies such as the use of lean mixtures, biofuels and hybridization associated with biofuels [1, 6, 7].

Conceived by Mr. Harry Ricardo in 1918 for diesel combustion and recently implemented in Otto cycles [8], the PCIS consists of a turbulent jet of extremely reactive gases [9] resulting from combustion in a small chamber, also called pre-chamber (PC), connected to the main chamber by small nozzles [10]. Guided by these nozzles, the jets that leave the PC propagate quickly in the main combustion chamber, acting as scattered ignition kernels, also increasing turbulence [11] and providing more interaction between the fuel and the air molecules [12]. The ignition energy supplied is approximately 1000 times superior than a conventional system, guaranteeing the start of combustion, even in adverse conditions, such as lean mixture operation [11]. Therefore, the system has been exploited for its ability to reduce fuel consumption and emission levels in engines.

One way to classify this ignition method is the fuel injection [13] and in stratified or active pre-chamber systems, the pre-chamber mixture is enriched with additional fuel by means of an auxiliary injection system, increasing the energy available for ignition [14]. Due to this additional energy, such a configuration is ideal for extending the lean limit, especially when compared to homogeneous pre-chambers, in which no additional fuel is injected into the pre-chamber. In that matter, the ignition energy supplied could be up to 1000 times superior than a conventional system [11].

Stadler et al. [15], studying a 0.463L single-cylinder research engine (SCRE) with a side-mounted direct injector operating with gasoline and an active pre-chamber, obtained a lean limit extension from 1.4 to 2.0 when operating with a 14.2 volumetric compression ratio. Such an increase in the compression ratio was enabled due to the higher combustion rate, which mitigates knock occurrence. With the combination of the high compression ratio and ultra-lean combustion, the peak efficiency reached 43%, with NO_x emissions reduced to 0.2 g/KWh. Müller et al. [16] also analyzed a SCRE with direct injection, compression ratio of 13:1, with a different pre-chamber design, injecting natural gas and gasoline in the pre-chamber. In the study, gasoline as auxiliary fuel managed to achieve higher lean limits, reaching $\lambda = 2.9$ at 2000 rpm and 15 bar of IMEP. At such lean levels the NO_x raw emissions reached 0.1 g/kWh.

Despite the good results already achieved by pre-chambers operating with fossil fuels, the benefits of using alternative biofuels should also be highlighted. Biofuels can be produced locally and reduce the fossil fuel dependence [17]. In general the biofuels show the potential to have their availability increased, as they can be produced from various raw materials, also presenting lower carbon dioxide emissions when considering the life cycle of the total biomass [18]. Furthermore, alcohols, such as methanol and ethanol, are especially attractive for spark ignition engines due to their higher-octane rating and latent heat of vaporization when compared to gasoline, improving knock resistance [19]. Furthermore, alcohols present low sulphur contents [20] and present higher laminar flame speed when compared to gasoline, decreasing combustion duration and, consequently, thermal losses [21].

Burkardt et al. [22], investigated a SCRE equipped with a pre-chamber with ethanol and methanol being varied as main fuels and ethanol as an auxiliary fuel. In this study, the compression ratio was increased to 16.4 due to the fuels' higher knock resistance. The configuration in which the main chamber (MC) was operated with methanol reached the highest lean limits and indicated efficiency, respectively $\lambda = 2$ and 48%. This configuration also enabled the lowest specific NO_x emission, of 0.07 g/kWh at $\lambda = 2.3$.

Despite the large volume of studies carried out in recent years in favor of PCIS development, this technology is not yet robust for large-scale use by vehicle manufacturers. Furthermore, although several studies are being produced on the PCIS theme in general, few consider the use of biofuels such as ethanol and methanol, highlighting the Brazilian studies on the theme [23-25]. Thus, the present research project aims to further investigate the stratified pre-chamber ignition system analyzed by Burkardt et al. [22]. In the work, more combinations of biofuels and operating points are explored, seeking to understand different aspects of this system functioning with alternative biofuels and the optimal fuel combination to achieve the best engine performance, mainly regarding fuel consumption and emissions.

1.1 General Objective

To determine the most adequate biofuel/pair of biofuels to operate in an internal combustion engine equipped with a stratified pre-chamber to control fuel consumption and pollutant emissions.

1.2 Specific Objectives

- To determine the lean limit and combustion duration for the engine configurations and fuels tested at different operations points;
- To analyze in-cylinder pressure curves of each fuel configuration, addressing the knock-like pressure oscillations observed in previous studies at pre-chamber operation, pointing to new knock detection parameters;
- To evaluate the thermodynamic results obtained in terms of combustion, in-cylinder temperature control, heat losses, thermal efficiency, fuel consumption and exhaust emissions;
- To consider the characteristics of each biofuels and ignition systems tested throughout the data analysis;
- To point out the next steps of this research considering the difficulties observed and the potential of the present work.

1.3 Work Contributions

Considering the objectives previously defined, this research will represent a further step in the development and characterization of a pre-chamber prototype able to work with biofuels. Thus, the vehicle propulsion system would achieve better performance, translated into reduced fuel consumption and exhaust emissions without representing large additional costs. Furthermore, since the use of biofuels associated with the pre-chamber ignition system is aimed, the fuel supply of commercial vehicles would be more independent from crude oil imports, using local sources of raw material.

2 LITERATURE REVIEW

In this chapter, the concepts and parameters necessary to understand the research are presented, as well as the state of art studies related to the theme. In section 2.1, the main characteristics of methanol and ethanol, the fuels experimentally tested, are discussed. Section 2.2 addresses the exhaust emissions of more concern in spark ignition (SI) engines, as well as strategies for their control. Finally, section 2.3 addresses the technology roadmap, presenting the main strategies for improving the performance and controlling emissions of SI engines. Among them, mixture dilution and use of PCIS are intensively studied in this work, and hence discussed in depth, so that the research gap to be filled with the present research is highlighted.

2.1 Fuels

In the past century, engine development considered almost exclusively the use of fossil fuels, due to petrol-based fuels availability in late 1800s. Currently, crude oil-based fuels represent 95% of all fuels used in the transportation sector. However, according to some local special needs, the use of alternative fuels was also possible after some alterations. As an example, in countries where crude oil must be imported, alternative fuels are needed through supply difficulties. In this context, flexible fuel engines in countries like Brazil are quite common, allowing the use of ethanol and gasoline in all proportions [38,39].

In addition to the quest to fuel quality improvement, its raw material has also been constantly analyzed. Currently, the fossil fuel dependence associated with uncertainties regarding its future extraction, the non-uniformity of its reserve concentration worldwide has raised major concerns. Associated with this, the use of carbon stored underground as engine fuel leads to increased CO₂ in the atmosphere and major environmental impacts, driving the development of renewable energy matrix.

Thus, fuels and energy sources that might be used for transportation should be analyzed according to their CO₂ emissions. While gasoline presents an average of 93 gr CO₂ per MJ produced, ethanol emissions range from 45 to 73 grs of CO₂ per MJ, depending on the biomass feedstock and the process used to produce that fuel. Moreover, contradicting the general belief of electricity and hydrogen as cleaner energy sources, hydrogen obtained from current processes can emit from 100 to 200 gr of CO₂ per MJ, depending on the process used, even though it does not contain carbon. Electricity, which is often generated from mixed sources of carbon materials, such as coal, natural gas and petroleum, can emit from 90 to 160 gr of CO₂ per MJ [5]. It should be highlighted that, there are already options to achieve zero net CO₂ emissions for all pathways.

Hence, the use of combustion engines still presents favorable balances on alternatives to electric energy when considered from well to wheel [26], biofuels can help in an immediate way as they present similar heating value and properties of liquid fossil fuels [17]. In this context, alcohols have being increasingly explored since they can be produced from several types of biomass, present a sustainable nature and can be carbon neutral [27]. Its main types are methanol and ethanol, which are attractive for spark ignition engines due to its higher-octane rating and latent heat of vaporization when compared to gasoline, improving its knock resistance Table 2.1. The ability of a fuel to

resist knock, normally defined by the research octane number (RON), enables engine operation with high compression ratios, highly intake boosted pressure applications, increased engine working pressures and, consequently, increased the engine fuel conversion efficiency [19]. Furthermore, alcohols present low sulphur contents [20] and present higher laminar flame speed when compared to gasoline, decreasing combustion duration and, consequently, thermal losses. However, even though fuel composition can significantly impact HC emissions, many of the organic compounds found in the exhaust are not present in the fuel, and aldehyde emissions (RCHO) are substantially increased [21]. In addition, another problem observed when an alcohol is blended with gasoline is its hygroscopicity. Thus, when an alcohol combines with water, it separates locally from gasoline, resulting in a non-homogeneous mixture [28].

In this specific work, aiming the engine development using environmentally and economically interesting fuels for Brazil, aspects of methanol and ethanol will be addressed in the next subsections. For better comparison, their properties are summarized in Table 2.1, where they are also compared to the gasoline properties.

Table 2.1 - General Fuel Properties [20, 21, 29-31]

	Gasoline	Methanol	Ethanol
Formula	C_8H_{18}	CH_4O	C_2H_6O
Lower Heating Value [MJ/kg]	43	20	26.9
Vapor Pressure [kPa]	5.5	32	5.95
Latent heat of vaporization [kJ/kg]	350	1200	920
RON	91-99	108.7	108.6
Stoichiometric A/F	14.7	6.4	9.0

2.1.1 Methanol

Methanol is considered one of the most promising alternatives to gasoline and has been extensively investigated mainly after the 1973 oil crisis in its pure form or in blends with gasoline [32]. The most common mixtures are the M10 (10% methanol and 90% gasoline) and the M85 (85% methanol and 15% gasoline), which was encouraged in the US by the California State Government, entailing the production of approximately 20,000 units of flexible fuel vehicles (FFV) by 1997 [33]. The fuel stands out for having higher laminar flame speed than gasoline and even ethanol in conditions close to stoichiometric, as shown in Figure 2.1, which leads to shorter combustion duration and greater volumetric efficiency. One problem regarding the use of methanol in gasoline engines is its incompatibility with certain materials. When methanol concentration exceeds 10% in a gasoline blends, corrosion of some metals such as aluminum, magnesium and zinc, and problematic reactions with plastics, gaskets, rubbers, engine oil and greases can be observed [34]. Luckily, these problems can be mitigated if engine components are manufactured from cast iron or steel, which does not significantly impact production costs [35].

Methanol has high heat of vaporization, leading to cold-start problems. Nonetheless, this property provides an effective in-cylinder temperature reduction, which can be used to control knock [36]. Almost all commercial methanol is produced from carbon oxides and H_2 through a syngas production process, with natural gas as feedstock. In addition, two groups of bacteria, methanotrophic and ammonia-oxidizing bacteria (AOB), can convert

methane (CH_4), which is present in natural gas and biogas, into methanol in aerobic conditions [37]. It can be also produced from various techniques and from a broader variety of biomass material than ethanol, including coal, petroleum, natural gas, biomass, wood and landfills. However, for being more chemically toxic, corrosive and presenting lower heating value when compared to ethanol, its commercial use has some challenges in the near future [28]. Even though toxicity is a major issue for the commercial use of all alcohols, this problem is especially true for methanol, being the cause of serious problems due to ingestion, skin or eye contact and inhalation [38].

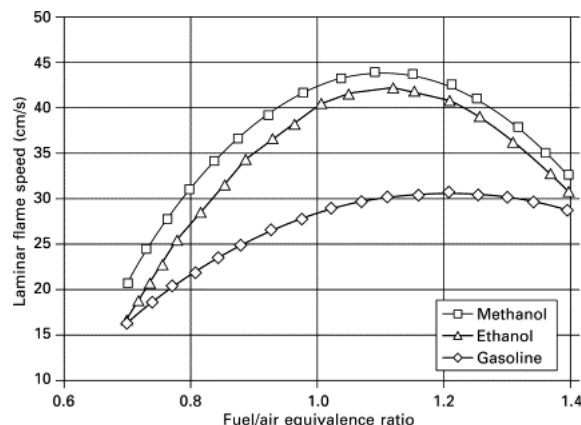


Figure 2.1 - Laminar flame speed of methanol, ethanol and gasoline as a function of fuel-air equivalence ratio (1 atm, 300 K) [39]

2.1.2 Ethanol

Ethanol is an alcohol that can be produced from different kinds of biomass, what is particularly interesting since the carbon dioxide released by its combustion can be fully absorbed by the regrowth of the raw source of biomass used for its production, if the processes are properly optimized [19]. In addition, it presents more well-established production routes and is commercialized on a larger scale when compared to methanol [17]. It presents a higher-octane rating, 108.6 and latent heat of vaporization when compared to gasoline, improving its knock resistance and being a great fuel candidate to a stand-alone fuel. As an example of the ability of ethanol to improve the RON of gasoline, Figure 2.2 presents 4 base gasolines with different RON, blended with ethanol in various volumetric contents. In all cases the gasoline RON was improved and the saturation effect was reached with ethanol percentages above about 30% [40].

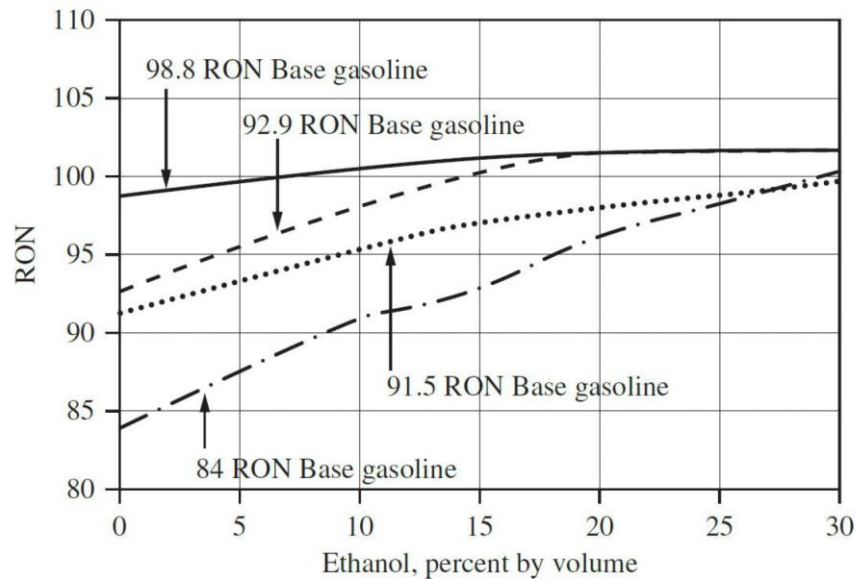


Figure 2.2 - Gasoline research octane number variations as a function of ethanol volumetric content [40]

Despite the increase in RON in ethanol and gasoline blends stabilizing at a given ethanol percentage, studies show that the increased ethanol content in gasoline blends improve engine thermal efficiency and engine power [41, 42]. This effect can be partially attributed to the charge cooling effect of ethanol due to its high latent heat of vaporization, 919 kJ / kg, when compared to gasoline's 350 kJ / kg [43]. Such a process reduces the specific volume of the intake charge, particularly in direct injection systems, in which the fuel has less interaction with the hot engine surfaces in the intake manifold [17].

Even though ethanol presents a lower low heating value of 25.5 MJ/kg when compared to regular gasoline (43 MJ/kg), the superior thermodynamic properties presented by an air-fuel mixture with ethanol manage to supply its reduced lower heating value. Szybist et al. [41] pointed out that stoichiometric mixtures using regular gasoline present LHV of 2.96 kJ/g of air, while blends of gasoline with 10% of ethanol content (E10) and with 85% of ethanol content (E85) presented respectively LHV of 3.02 and 3.06 kJ/g of air.

This can be explained by the thermodynamic effects on the ratio of specific heats (γ), which causes the work balance between air-fuel mixture compression and expansion to be greater for ethanol than for fuels such as isooctane and toluene, present in gasoline [41]. In addition, the mole multiplier effect can be highlighted. The higher the MM , more moles of products are available for heat extraction during the expansion process, resulting in additional pressure during the chemical reactions without any additional heat [44]. Considering stoichiometric combustion, Figure 2.3 shows the MM for various types of hydrocarbons and oxygenated hydrocarbons showing the advantage of using alcohols such as ethanol to increase engine thermal efficiency [41].

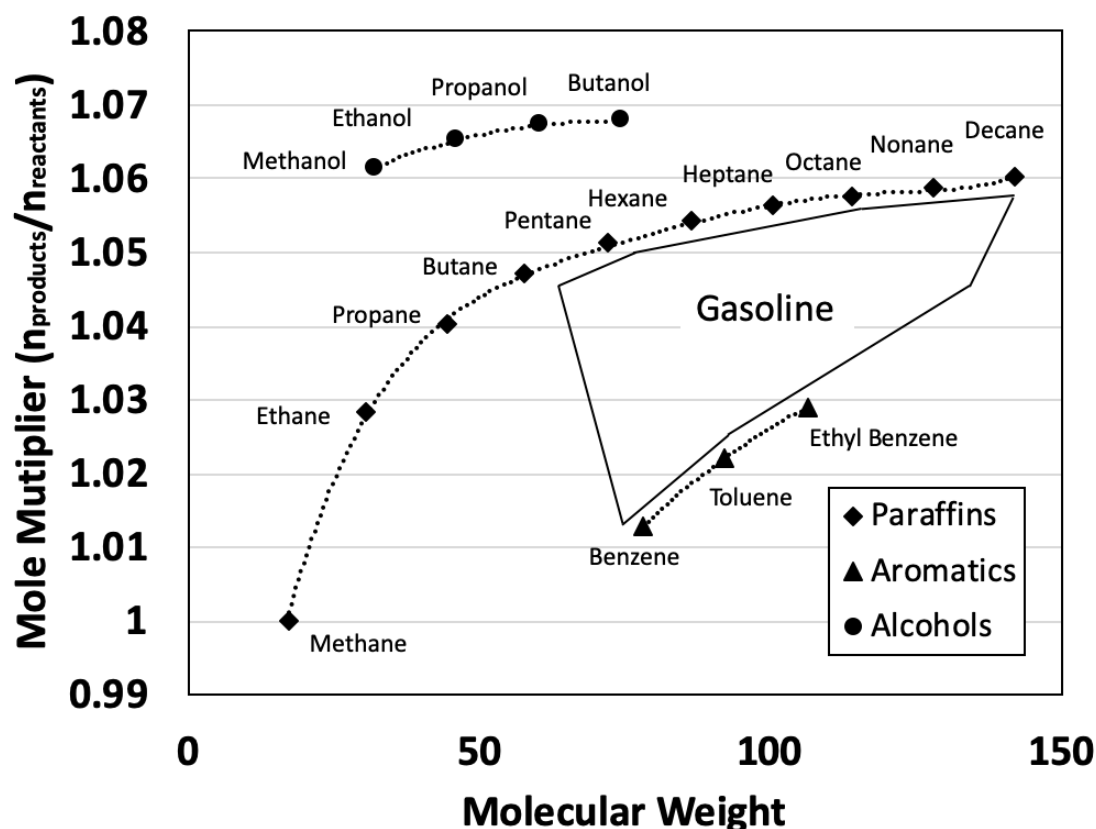


Figure 2.3 - Mole multiplier as a function of molecular weight [17].

Despite all the benefits presented by ethanol use, there is still a lot to develop for its optimal use. First, ethanol operation presents cold start problems and great corrosive potential to engine components and subsystems, such as copper, bronze, aluminum, rubber and plastics. All these problems can be solved with the development of an engine especially for fuel, and not simplified adaptation of engines initially developed for gasoline operation [45-47]. In addition, it is not yet possible for all global fleet to be altered towards this fuel due to its yet insufficient production [48], along with challenges on crop growth and yield and landscape use, problems to be solved in the studies of new generations of biofuels [49].

Due to countless benefits and consolidated production routes, ethanol one of the few biofuels used and commercialized on a large scale, mainly in Brazil and the United States. In Brazil, the fuel production has been extensively incentive by the government as a response to the global oil price increase of 1970 [50, 51]. Since then, the fuel is commercially offered either as hydrous ethanol, with 95.1 to 96% ethanol composition, or as anhydrous ethanol, mixed with gasoline in 27% volumetric percentage. Currently, more than 50% of the automobiles in Brazil can use ethanol due to flex fuel engines. In the US since 2005 gasoline is required to contain 10% renewable fuels such as E10. In 2019, gasoline blends with up to 15 percent ethanol are available [52] and studies to allow the use of ethanol in higher contents are underway, despite some fuel compatibility issues with the existing vehicle fleet [53]. Furthermore, strong incentives and initiatives for the development of the ethanol-producing industry have been seen in regions such as Europe, Asia and the Caribbean Basin [23].

2.2 Engine Emissions

The ideal combustion aims at the oxidation of a highly energetic fuel reagent, generating products with minimal enthalpy, consequently releasing its maximum energetic potential. Therefore, fuels containing carbon in its formulation should ideally produce only CO₂ and H₂O. Despite originating from ideal combustion processes, CO₂ emissions generated in activities related to transportation, industry, power generation and agriculture, have a significant contribution to climate change [54], which in turn is one of the most serious challenges that humanity will face in the coming years [55].

The main impact of CO₂ in the atmosphere is the greenhouse effect, responsible for global warming. Such temperature rises can, for example, result in differences in atmospheric pressures capable of forming cyclones of various intensities [56]. In addition, factors such as combustion anomalies, nitrogen oxidation, air and fuel impurities, fuel quality, mixture λ , among others are a major source of air pollutants, such as NO_x, CO, unburned or partially burned HC and unregulated pollutants, including RCHO, benzene, toluene, xylene (BTX), sulfur dioxide (SO) and others [21, 57]. Such air pollutants are responsible for more than two million premature deaths each year and pointed as cause of pulmonary and cardiovascular diseases due to their inhalation [58].

Hence, as the transport section highly impacts air quality, since it is responsible for 20% of carbon dioxide CO₂ emissions [59] among other emissions, air quality guidelines regarding automobiles are constantly updated, forcing governments to the same [60].

The California government pioneered the adoption of air quality standards and vehicle exhaust emissions in response to protests against smog occurrence in the region, forming a Pollution Control District in 1947. Pollutions problems were already addressed by the state since the 1960, preceding the Clean Air Act of 1963, passed to help control air pollution in the United States, allowing the Environmental Protection Agency (EPA) to set standards for vehicle air pollution [61]. Since then, maximum permissible values for contaminant emissions are constantly set, aiming the preservation of public health [62]

With that concern in mind, Europe introduced in 1970 its first exhaust emissions standard for passenger cars [63]. Currently, the current legislation in the European Union is the European Pathway, which consists of six stages of increasingly stringent emission control requirements on NO_x, HC, CO and particulate matter emissions, comprehending light and heavy duty vehicles, using their weight as a distinction between classes [62]. The G-20 countries are responsible for 90 percent of global vehicle sales, and 17 out of the 20 members have chosen to follow the European regulations. A number of Asian and Latin American countries currently have Euro 2/II, 3/III, and 4/IV standards in force [64].

In Brazil, the *Programa de Controle da Poluição do Ar por Veículos Automotores* (Proconve), in English the Air Pollution Control Program for Motor Vehicles, is responsible for controlling the level of pollutants emitted by new vehicles. Since its creation in 1986, emission limits are progressive stringent, with new standards being established at each stage by the automotive industry, stimulating the development of innovative technologies, such as the use of catalysts and electronic injection systems [65]. Therefore, it is necessary to know the mechanism of formation of such emissions, as well as ways to mitigate them. Thus, this section aims to explain more about the main emissions in SI engines, their causes and some strategies for their reduction.

2.2.1 Carbon Monoxide

Carbon monoxide is a colorless and odorless gas that in low concentrations harms tissues, brain, heart and musculature and can be lethal in high doses due to its affinity with blood hemoglobin, which is 200 to 250 times higher than with oxygen, preventing O_2 absorption [66]. The carbon monoxide formation occurs due to lack of oxygen supply and local inhomogeneities, which leads to incomplete burning of fuels containing carbon and can also be attributed to the dissociation of CO_2 at high temperatures. Thus, the richer the mixture, the greater the CO formation and with mixture enleanment, more oxygen is available in the mixture and less CO is formed. With lean combustion, CO levels are kept low until combustion quality is compromised and misfire occurs [21]. Such behavior is well summarized in Figure 2.4, which also presents the behavior of HC and NO emissions, discussed in the following subsections.

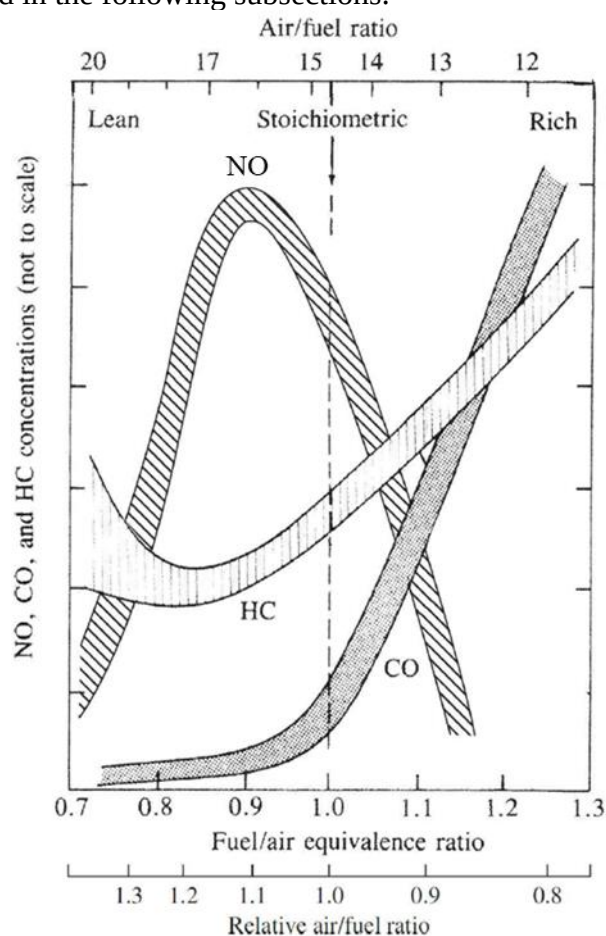


Figure 2.4 - Behavior of CO, HC and NO emissions according to air/fuel ratio variations [21]

2.2.2 Hydrocarbon

Organic emissions, also known as hydrocarbons, are in, general lines, the consequence of incomplete combustion of hydrocarbon fuels. The main environmental impact due to HC emissions is photochemical smog, which occurs when HC react with NO_2 in the presence of sunlight, forming a brown haze [67]. In addition, CH_4 , which is the lightest HC, presents a 21 times higher global warming potential than CO_2 [68]. For the population, exposure to HC emissions is irritating to eyes, nose, skin and upper respiratory tract, can

cause cellular damage, and several hydrocarbons are considered carcinogenic and mutagenic [67].

Hydrocarbon emissions have different sources in the engine. For example, the unburned air fuel mixture trapped into crevices during compression or combustion can leave the cylinder unburned or only partially burned in the exhaust. After the flame is quenched in the combustion chamber walls a quench layer containing unburned or partially burned gases is left at the walls. In addition, the thin film of engine oil on the cylinder liner can absorb hydrocarbons before combustion and unabsorb then after.

Furthermore, as depicted in Figure 2.4, HC emissions are particularly high in rich mixtures because with O_2 depletion the combustion cannot be complete and increases with lean mixtures when combustion is particularly slow, due to bulk quenching. However, the oxidation of the hydrocarbons that were not burned during the main combustion process can occur in the following strokes depending on temperature and oxygen concentration. For instance, HC oxidation in catalytic converters is capable of significantly reducing vehicular emissions.

Fuel composition can significantly impact the composition and magnitude of the HC emissions. However, many of the organic compounds found in the exhaust are not present in the fuel, indicating pyrolysis or synthesis during combustion [21]. The use of alcohol fuels, which contain the hydroxyl functional group (OH), absent in gasoline such as ethanol, increase aldehyde emissions, such as formaldehyde (CH_2O) and acetaldehyde (C_2H_4O). Aldehydes are highly reactive organic compounds that participate in complex chemical reactions in the atmosphere and can be highlighted as significant contributors to photochemical smog [57, 67].

2.2.3 NO_x

Nitric oxide and nitrogen dioxide are usually grouped and called nitrogen oxides, NO_x , being the NO predominant in engine-out emissions. It is highly toxic, very irritant to the eyes and nasal mucous membranes, can cause pulmonary emphysema and be transformed into cancerous substances in the lungs [67]. It is also known to increase susceptibility to respiratory infections and other respiratory problems in general. In addition, it reacts with water in the air and forms one of the main components of acid rain: nitric acid (HNO_3) [67]. Furthermore, in secondary atmospheric reactions, NO_x associated with hydrocarbons is also responsible for the photochemical smog [54].

It is formed by the combination of N_2 and O_2 in the air admitted in two main processes: the thermal NO_x and the prompt NO_x [67]. In the thermal NO_x formation, even though N_2 is supposedly inert at normal atmospheric conditions and in the combustion reaction, it tends to react with oxygen at high temperatures, ranging from 2000-5000 K, in a series of chemical reactions known as the Zeldovich mechanism [69]. This mechanism is responsible for the most NO_x present in engine emissions [70]. The prompt NO_x formation, on the other hand, was first detected by Fenimore and occurs only in combustion of hydrocarbon fuels, such as gasoline, before the formation of the thermal NO_x , in the laminar premixed flame zone [71].

Even though the maximum burned gas temperatures occurs at $\lambda \approx 0.9$, according to Figure 2.4, when mixture is rich the lack of oxygen impedes NO_x formation. In slight lean

mixtures, with $\lambda \approx 1.1$, the NO_x formation is increased since the temperature of the chamber is still high and the excess oxygen favors the combination in N_2 with O_2 . As the mixture is enleaned, the in-cylinder temperature decreases severely, below the optimum temperature for nitrogen oxides formation, diminishing the occurrence of the Zeldovich mechanism. However, the in-cylinder temperature reduction also favors bulk quenching. Thus, the mixture enleanment to reduce NO_x emissions is sustained only until the combustion quality is compromised [21].

In addition to the mixture enleanment strategy, other combustion chamber temperature reduction strategies contribute to the reduction of NO_x emissions. For example, direct fuel injection, which also reduces the average combustion chamber temperature, provides important improvements pollutant emissions, in particular NO_x [72]. Furthermore, aiming to reduce the prompt NO_x , the replacement of fossil fuels with biofuels containing oxygenated hydrocarbons can be also beneficial in that matter [73].

2.2.4 *Emission Mitigating Technologies*

Due to stringent air-pollution regulations, mitigating engine exhaust emissions is essential and can be basically obtained through optimal engine calibrations, emission mitigating strategies and exhaust treatment systems. For instance, the fuel rich operation at high torque to prevent knock, which was historically used, is now much less used since it increases HC and CO emissions. Furthermore, lean combustion or mixture dilution with exhaust gas recirculation (EGR), especially at part load, is being studied since it improves engine efficiency and reduces CO, HC and NO_x emissions when combustion stability is guaranteed. However, combustion instability and bulk quenching resulting from mixture dilution can have a negative impact on CO and HC emission.

Substantial further emission reductions, over 90%, can be conducted by the engine catalytic consisting of a ceramic structure with noble metals built into engine exhaust systems. The exhaust treatment is based on oxidizing catalysts for HC and CO emissions into CO_2 and H_2O and reducing catalysts for NO_x to N_2 . However, for the three-way catalyst (TWC), to be effective, the activation energy for these processes must be reached and the temperature of the catalysts used for HC, CO, and NO_x removal must be at about 250°C for the catalyst to be effective, which is harder to be achieved during lean combustion, when the exhaust temperatures are reduced.

For CO and HC oxidation sufficient oxygen must be available. In a stoichiometric mixture the air associated with CO and unburned HC is on average available. For NO reduction the reducing agents CO, H_2 and HC are generally present. In SI engines the catalysts for CO, HC and NO removal are widely used and highly effective. However, these three-way catalysts are only effective if the exhaust gas is stoichiometric, a condition in which there is plenty oxygen available for oxidation and reducing agents for NO reduction. In diesel engine, where the mixtures are normally lean, selective catalyst reduction of NO_x is carried out using ammonia, NH_3 , as the primary reducing agent, added in form of urea [21].

Despite the benefits of emission reduction, environmental impacts must be observed throughout the catalytic converter life cycle, from the extraction of materials for their manufacture, to their disposal after the end of their shelf life. Hence, even though efficient, the Three-Way Catalyst (TWC) systems present high costs, due to the noble

materials necessary for the reactions to occur, and significant environmental impacts during manufacture and disposal.

2.3 Technology Roadmap

To extend the existence of internal combustion engines in the future, a technology roadmap that presents a flexible strategic plan aiming to match short- and long-term goals with specific technology solutions is introduced in this subsection. Several strategies can be exploited to improve engine performance while meeting the fuel consumption and emission reduction demands, such as downsizing [74], variable valve control [75], mixture dilution [76, 77], engine boosting [19], adaptations to Miller and Atkinson cycles [78, 79] and the use of increased compression and expansion ratios [80]. However, this section will focus solely on air-fuel mixture dilution and the use of engines equipped with PCIS, since this is the objective of the present work.

2.3.1 Mixture Dilution

Mixture dilution can be performed with either excess air, generating lean mixtures, or with recycled exhaust gases. At part-load conditions, mixture dilution can improve the engine's fuel conversion efficiency for the increase in expansion work arising from changes in the thermodynamic properties of the burned gases, for the decrease in pumping work and heat loss reduction, as the burned gas temperatures are decreased. As it represents an important strategy for fuel conversion efficiency improvement and emission reduction, mixture dilution will be addressed in depth in the following subsections, pointing out particularities of lean mixtures, EGR and comparing the two strategies.

2.3.1.1 Lean Mixtures

Since 1908, studies with lean mixture burn in internal combustion engines have pointed out to improvements in thermal efficiency [81, 82]. With the present concern with engine-out emission control and fuel consumption such strategy is being once again analyzed with great interest [83-85]. The lean combustion allows de-throttling, which decreases *SFC* especially at partial loads when pumping losses are greater [86]. In addition, the extremely lean mixture presents a greater polytropic exponent, achieving greater thermal efficiency [72]. The decrease in the average combustion temperature reduces heat losses [87] and knock occurrence and also, more oxygen availability leads to increase in combustion efficiency, as show in Figure 2.5.

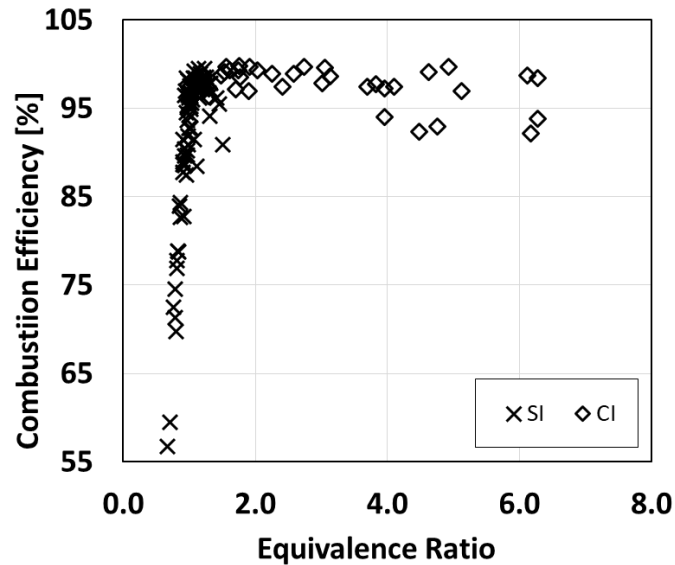


Figure 2.5 - Combustion efficiency as a function of the equivalence ratio. Adapted from [21]

The figure shows how combustion efficiency varies with the fuel-air equivalence ratio for both Diesel and SI internal combustion engines. Due to the type of operation of each engine, it is seen that Diesel engines operate with a wide range of lean mixtures while the spark ignition engines tend to operate ranging from ultra-rich mixtures to slightly lean. In these engines, for lean equivalence ratios, the combustion efficiency is usually between 95 to 98%. For mixtures richer than stoichiometric, lack of oxygen prevents complete fuel combustion. Hence, combustion efficiency steadily decreases as the mixture becomes richer. It is important to highlight that combustion efficiency is little affected by other engine operation and design variables and consequently, Diesel engines, which always operate at leaner conditions, present a slightly higher combustion efficiency (about 98%). Thus, it is possible to state that if the SI operation could remain stable in ultra-lean regimes, great benefits would be obtained from this type of strategy. This strategy would require a careful engine design and technology developments in engines, mainly regarding the ignition system, to overcome some ignition difficulties [73]

Added to the increase of the engine's combustion efficiency, the fuel consumption evidently decreases [88-91] and the exhaust gases profile undergoes notable changes, as emissions on SI engines are particularly sensitive to the air-fuel ratio [21]. The use of slight lean mixtures at part load conditions could lower HC and CO emissions [21]. However, significant oxygen excess benefits NO_x emissions control the most, since most of its formation is associated with high in-cylinder temperatures, which are reduced with mixture enleanment [70]. Nevertheless, extreme mixture dilution can deteriorate combustion quality and increase HC emissions. Hence, without strict engine emission requirements, lean mixtures can be frequently used to mitigate NO_x. However, for tight NO_x, HC, and CO emission control, stoichiometric combustion is preferred as it enables the use of a three-way catalyst. In such cases, mixture dilution with EGR is more appropriate.

2.3.1.2 EGR

The Exhaust Gas Recirculation consists of the reintroduction of burned gases into the cylinder in order to reduce both NO_x emissions and fuel consumption levels at some engine operation points. The burned gas fraction recirculated at the intake produces a significant air-fuel mixture dilution, reducing the in-cylinder temperature and consequently NO_x formation [21]. The reduction of in-cylinder temperature also inhibits knock [72]. Thus, the EGR can potentially substitute mixture enrichment for knock mitigation, reducing CO and HC emissions [92].

In addition, the mixture dilution allows the reduction of both pumping losses at partial loads and heat losses to combustion chamber walls, resulting in an increase in the fuel conversion efficiency and consequent *SFC* reduction [93]. The high dilution rates by EGR increase the cyclic variability of the start of combustion due to air fuel mixture dilution in the vicinity of the spark plug [94]. However, if combustion starts, the EGR promotes more stable process [72]. The combustion quality deterioration with excess EGR dilution presents an almost immediate negative impact for slower burning combustion [95]. However, for fast burning systems, the negative impact is smaller and only significant with EGR levels higher than 15%. At this condition, the *SFC* can decrease up to 5% relative to zero EGR combustion [21].

The percent of exhaust gas recycled (x_{EGR}) is defined according to Equation 2.1, in which m_i is the total mass of intake mixture and m_{EGR} is the mass of exhaust gas recycled. Depending on operating conditions, in SI engines up to about 30% of the exhaust can be recycled.

$$x_{EGR} = \frac{m_{EGR}}{m_i} \quad 2.1$$

The internal EGR is performed by altering the intake and exhaust opening and closing timings. At partial loads, due to reduced engine speed, the late exhaust valve closing promotes the return of exhaust gases from the exhaust manifold to the intake manifold, which increases the fresh mixture dilution. A similar effect can be obtained with early closing of the exhaust valve before TDC (bTDC), since large amounts of burned gases are retained in the cylinder at the end of the exhaust stroke [96]. Delaying the exhaust valve closure while maintaining the valve overlap leads to intense back flow during the intake valve closure (Miller reverse cycle), which associated with the internal EGR, achieves high levels de-throttling, resulting in pumping loss and *SFC* reduction at various operation conditions [97].

The exhaust gas recirculation can also be conducted externally, with a cooled or uncooled systems. In systems with uncooled external EGR, part of the exhaust gases are removed from the exhaust manifold and introduced directly into the intake manifold through a control valve [72]. In systems with cooled external EGR, most used today for engines operating at high loads, the reintroduction of part of the exhaust gases is dosed by a control valve and the induced charge is cooled by heat exchangers at the intake manifold. The exhaust charge cooling increases the cooling effect of the combustion chamber, and further reduces fuel consumption and pollutant emission levels, in addition to being more efficient for auto-ignition suppression [72]. The knock mitigation enables proper ignition timing calibration to achieve MBT, increasing the fuel conversion efficiency [98]. The increased volumetric expansion ratio associated with the use of cooled external EGR showed gains in thermal efficiency of 2.1 to 3.5% [77]. Simulations of engines equipped

with a cooled EGR system presented, in some cases, a 5.9% reduction in specific fuel consumption, and also peak in-cylinder temperature reduction [99].

2.3.1.3 Comparison of Dilution Strategies

The mixture dilution with air or with recycled exhaust gases presents analogous but not identical effects on ignition. The use of both diluents must be limited by combustion quality and stability as excess dilution can lead to slow combustion process and increased cycle to cycle variability. As load decreases, less dilution of the fresh mixture is tolerated because with lower gas speeds and unfavorable valve control, cylinder scavenging is limited, what increases internal dilution with residual gas. Hence, at idle conditions, EGR is usually not tolerated and the mixture should be stoichiometric or even rich to achieve adequate combustion stability [21].

The EGR dilution increases combustion variability more than air dilution. The behavior comparison between both types of dilution is presented in Figure 2.6, which depicts combustion stability, $bsNO_x$ (brake specific NO_x) and $bsHC$ (brake specific HC) emissions and BSFC experimental data obtained in a 2-liter four-valve engine, at 2000 rpm and 2 bar BMEP, with calibration aiming at MBT. EGR is extremely effective at reducing NO_x emissions. For example, 15% EGR reduces NO_x by about a factor of 3. In addition, the use of EGR at lower dilution rates (35%) presents more significant NO_x emission reductions when compared to lean combustion. In addition, when combustion is degraded, HC emissions from operation using EGR is slightly less than lean operation. However, the coefficient of variability of IMEP (COV_{IMEP}) increases significantly, EGR levels move beyond 20%, and BSFC decreases to its minimum. On the other hand, lean operation can reach 40% dilution at the same COV_{IMEP} . Hence, even though lean combustion is less effective for emission reduction, it is more effective at BSFC reduction (from about 6% to about 10%) [100].

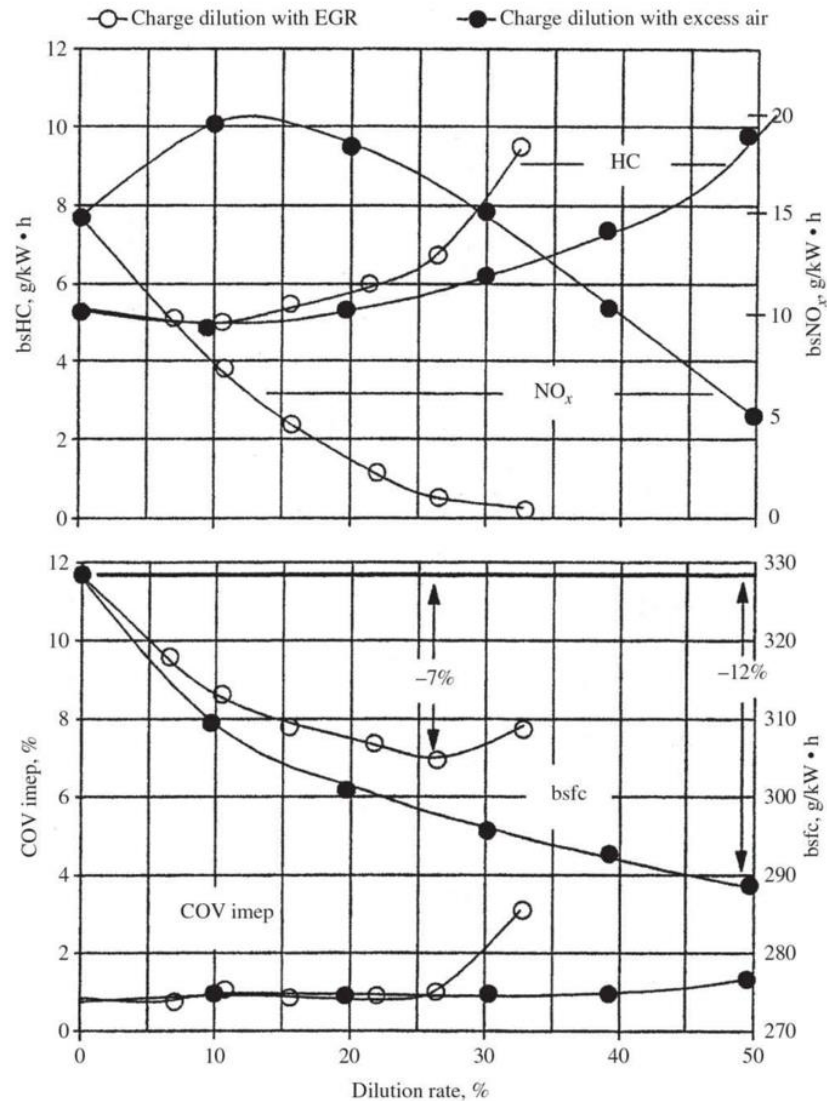


Figure 2.6 - Comparison between charge dilution with excess air and with EGR [100]

At low loads, the air dilution limit is almost twice the dilution limit with EGR, as it presents greater impact on the mixture burning rate [101]. Only with low dilution ($\lesssim 10\%$ EGR), the difference in the ignition system behavior for EGR and air dilution can be disregarded [101].

Despite its benefits, lean combustion also worsens combustion quality and requires a highly effective lean NO_x catalyst as the TWC cannot be used with lean engine operation. Thus, assessments of the potential benefits of both dilution strategies should be analyzed [21]. One approach extensively explored utilizing both strategies is a DISI engine that operates with lean and stratified charge at light load with some EGR, homogeneous lean mixtures with EGR at mid-load, and stoichiometric mixtures at high load, as shown in Figure 2.7. The EGR role, especially at stratified charge, is to reduce peak burned gas temperatures [102].

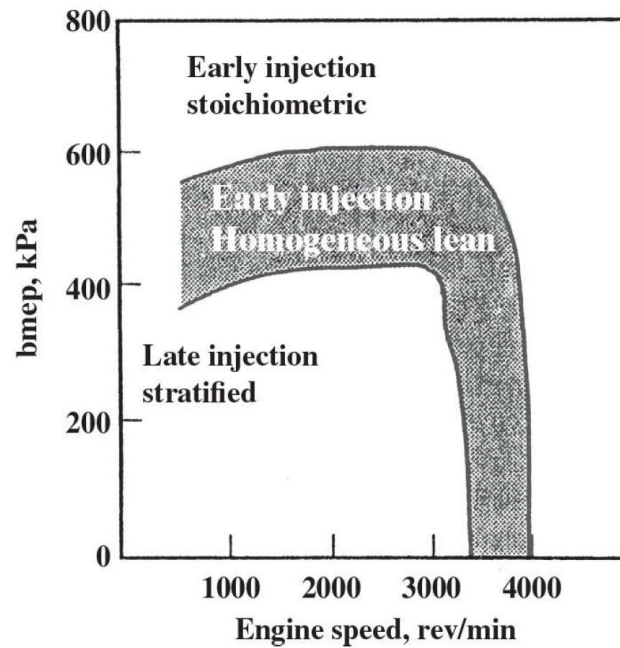


Figure 2.7 - Suggested DI engine operation showing different regimes for lean stratified, lean homogeneous (both with EGR), and stoichiometric (higher load) regimes [102]

2.3.2 Pre-chamber Ignition Systems

2.3.2.1 General Aspects

Due to the difficulties observed with lean mixture burn, an alternative to extend the mixture lean limit is the pre-chamber ignition system, capable of reducing cyclic variability, and enhancing both the ignition energy and the burning speeds observed in lean mixtures, still guaranteeing the benefits of the lean burn regarding emissions and fuel consumption [65].

In this system, a small pre-chamber, as exemplified by Figure 2.8, is connected to the main chamber through interconnection holes. Combustion starts in the pre-chamber, by means of an electric spark, where pressure is increased, leading the combustion gases to be expelled towards the main chamber through the interconnection holes. Each jet of combustion gases represents a source of high ignition energy for the lean unburned mixture in the main chamber, extending the mixture lean limit [9]. The jets also promote high turbulence, leading to more efficient and faster combustion [11, 12, 103]. In addition, the optimal combustion start and turbulence are able to reduce the in-cylinder temperature, minimizing knock occurrence, improving fuel consumption efficiency and reducing pollutant emissions such as NO_x and CO [104]. Furthermore, the system is very attractive because it is associated with a relatively low cost and easy installation when compared to other systems [13].



Figure 2.8 - Pre-Chamber Ignition System Example [105]

2.3.2.2 History

In 1918, Sir. Harry R. Ricardo developed and patented the first engine with a PCIS, shown in Figure 2.9 [8] and from the first prototype, many other models were conceived. In 1926, Caleb E. Summers proposed the use of two carburetors and auxiliary rocker to control a third valve responsible the entrance of mixture in the pre-chamber [106]. In 1963, L.A. Gussak patented the forced initiation of ignition by avalanche activation process of combustion of a working mixture in the combustion chamber (LAG-Process) [107] obtaining significant improvements in relation to its predecessors.

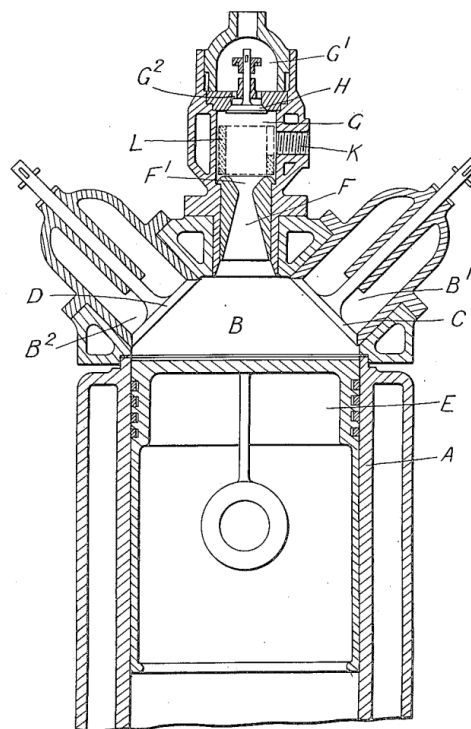


Figure 2.9 - Ricardo's Pre-Chamber Design [8]

In the 1970s, with progressively stricter laws to regulate vehicle emission levels, Honda developed the CVCC system, which was used for many years in its vehicles, achieving low fuel consumption and reduced rates emissions of polluting gases [108]. The CVCC was used in the Honda Civic, and tested in engines from other automakers, such as the GM Vega and Chevrolet Impala. In Honda Civic, the system managed to keep emissions in g/mi below the maximum levels established by the legislations of 1975, reducing CO, HC and NO_x emissions respectively by 22%, 20% and 44% when compared to the engine not equipped with CVCC. For the engines of other automakers, the system managed to reduce CO, HC and NO_x emissions by up to 85%, 88% and 69%, respectively, without however achieving reductions in BSFC and only reaching the 1975 standards for HC emissions [109].

At the same time, studies such as Turkish et al., in 1974 [110], and Fuji et al., in 1985 [111], developed systems similar to CVCC, aiming to increase the lean limit and combustion stabilization. However, these systems still depended on carburetors and various mechanical components to function. Wyczalek et al. in 1975 [112], studying the same system, proposed for the first time the use of electronic fuel injection in the pre-chamber, reducing mechanical components, increasing the reliability of the system, improving the control of mixture formation, besides reducing the maintenance costs and the complexity. Prototypes similar to the proposed Wyczalek were developed by Volkswagen, with the Pre-chamber Injection System (PCI) [113], by Toyota, with the Turbulence Generating Pot (TGP) [114] and by Porsche, with the *Schichtlade Kammer System* (SK – in English stratified chamber system) [115].

Currently, with the popularization of electronic injection and ignition, the PCIS implementation by automakers is possible and even draws attention due to the increasingly strict regulations that drive the automotive industry. In this context, the study of pre-chamber ignition systems has become increasingly frequent, being used even in Formula 1 engines [116].

2.3.2.3 Pre-Chamber Main Classifications

The PCIS can be classified into two main types: homogeneous and stratified [13]. In the homogeneous PCIS, also called passive, the mixture is introduced into the cylinder through intake and directed by the piston to the pre-chamber during compression, maintaining the same air-fuel ratio in both chambers [117]. In the stratified PCIS, also called active, the mixture introduced in the pre-chamber is enriched with additional fuel, usually supplied by an auxiliary injection system, increasing the energy available for ignition [14].

For stratified systems, studies with liquid fuels have not been so successful since, due to the small volume in which fuel is injected, too much fuel is trapped on the walls, leading to increased HC emissions. With this, gaseous fuels have been studied, such as liquefied petroleum gas (LPG), compressed natural gas (CNG), and H₂ [62, 118, 119] and among them, hydrogen can be highlighted for its faster flame speed and higher heating value [120].

In addition, the systems can be classified according to the diameter, quantity and arrangement of the interconnection holes, since these parameters determine part of the combustion process in the main chamber, such as pre-chamber filling in the compression

stroke, the fraction of residual gas in the pre-chamber and the jets outlet speed [121]. Among these parameters, the interconnection hole diameter can be highlighted as one of the most important. When the pre-chamber flame front moves towards such holes, it can be extinguished or not according to its diameter and the heat exchange between combustion gases and nozzle walls. In fact, several studies point out to a critical nozzle diameter, which determines the flame extinguishing and, consequently, the ignition mechanism in the main chamber. This diameter is determined according to the engine design and the fuels used, being dependent also on parameters such as on the air/fuel ratio and pressure [122]. Generally, if the nozzle diameter is above the critical, the flame is not extinguished, entering the main chamber in form of a torch, promoting a torch ignition in the main mixture. However, for pre-chambers with interconnection holes with diameter smaller than the critical, in which the jets speed reaches supersonic values capable of extinguish the flame, jet ignition is observed [122].

In the case of jet ignition, the ignition does not occur immediately in the main chamber as time is needed to allow the pre-chamber gases to interact with air-fuel portions of the main chamber, locally raising pressure and temperature, promoting auto-ignition spots. In this case, the jets become more reactive with the presence of active radicals from rich mixture combustion, ensuring the highest concentration of auto-ignition points, which together will compose a flame front [123]. In addition, the distribution of active radicals in the main chamber facilitates and accelerates the end-gas combustion, increasing the speed and quality of lean combustion. It should be noted that in pre-chamber operation, characteristics of the two types of ignition can overlap, with flame extinction after nozzle exit, partial flame extinction, among others [124].

2.3.2.4 Pre-Chamber Use and Knock Occurrence

Even though pre-chambers are normally implemented in ICE aiming ultra-lean combustion, the system operation close to stoichiometry is also interesting since it increases ignition energy and reduce the combustion duration through turbulence, decreasing the knock tendency. Hence, the engines can be configured with higher compression ratios, which increases the thermal efficiency of the engine and, consequently, reduces fuel consumption.

Despite the benefits of its use, pre-chambers may influence the range of operation in which knock occurs [125]. In addition, the use of pre-chambers can lead to new combustion anomalies since the resonance modes can be altered. Santos et al. [126], studying a pre-chamber representing 2.2% of the main combustion chamber volume with 5 interconnecting holes, highlighted limitations in the operation with mixtures close to the stoichiometric with homogeneous and stratified pre-chambers, using hydrogen (H_2) as auxiliary fuel. The in-cylinder pressure curves, presented in Figure 2.10, were acquired at 1500 rpm and 4 bar of indicated mean effective pressure (IMEP). In this work, the pre-chamber ignition system (PCIS) pressure peak profiles show fluctuations, in contrast to the baseline ignition profile. It is pointed out that, even though a similar pressure pattern occurs at knock events, that is improbable since the burn rate is accelerated with PCIS, which can even be used to mitigate knock [13]. Hence, given that no knock was detected in the engine with baseline ignition, it can be inferred that the engine equipped with PCIS did not experienced knock in the studied condition [10].

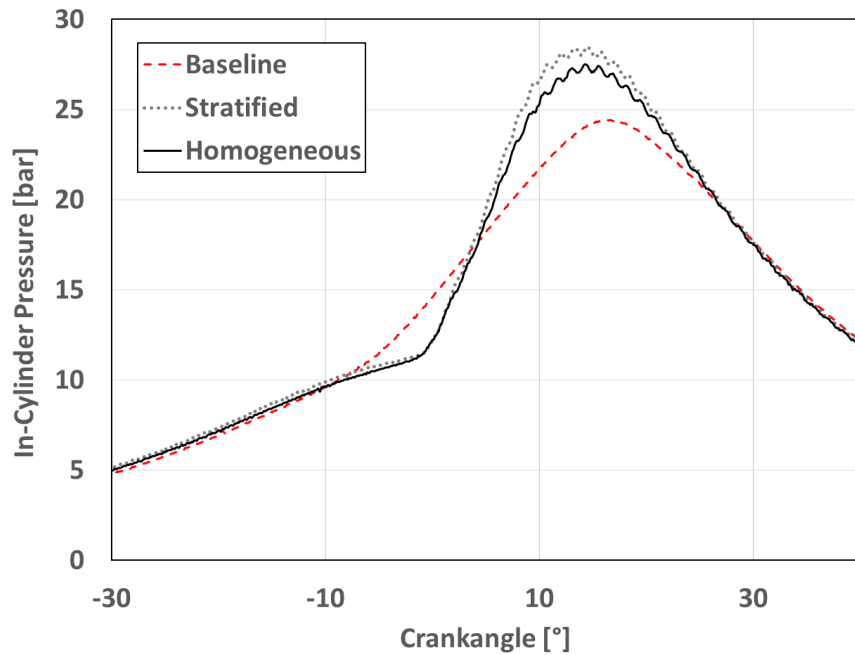


Figure 2.10 - In-cylinder pressure with the baseline and homogeneous and stratified PCIS [126]

Figure 2.11b presents the in-cylinder pressure, shown in Figure 2.11a, treated with a high-pass filter, in which homogeneous pressure oscillations are observed. These fluctuations do not occur after a sudden and sharp pressure rise, as common in knock events, and can hence be associated with the jet oscillations between the main and pre-chambers. This phenomenon may be justified by the excess pre-chamber pressure that propagates through the main chamber and is reflected from the cylinder walls, transmitting negative pressure waves between the chambers.

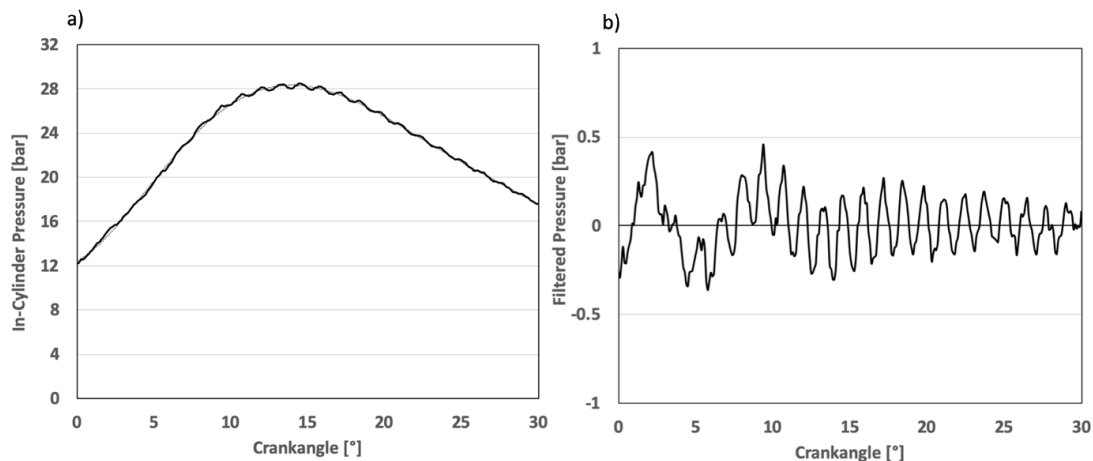


Figure 2.11 - Pressure oscillation in the pre-chamber systems

Sens et al. [25] also analyzed the operation with pre-chambers, comparing an engine operating with spark plug and pre-chambers at 2000 rpm and 10 bar brake mean effective pressure (BMEP), as shown in Figure 2.12. In the tests, knock was observed in the baseline engine and pressure oscillation in the engine equipped with PCIS. The phenomenon of pressure fluctuations is similar to that described by Attard et al [127] as jet knock, in which small pressure oscillations similar to conventional end-gas knock are

observed, only normally smaller in magnitude and initiated just after the start of ignition instead of towards the end of combustion.

The jet knock was also observed in other studies in which divided chamber combustion systems were studied [16, 128, 129] and occurs due to the relatively small throat areas and the large pressure ratio between the pre- and main chamber, which generates high pre-chamber jet velocities [130]. When the jet velocity equals or exceeds sonic velocity, a hemispherical or conical shock wave can be created, being detected as pressure fluctuations. According to Gruden and Karlheinz (apud Müller et al. [16]) in 1974, one of the first works to report the phenomenon, the pressure oscillation amplitude is dependent on the combustion chamber heat release rate (HRR). Differently from what as observed by Attard et al. [127], however, Muller et al. [16] reports that the oscillation frequency range is very similar in baseline knocking cycles and during PCIS operation, which hinders an automated distinction of these two phenomena.

Returning to the study by Sens et al. [25], despite all the theories explaining the jet knock occurrence as a common phenomenon in pre-chamber operation, the results obtained must be analyzed very carefully, since knock is detected at the same operating point with baseline operation. In the study, the initial pressure fluctuations are not considered to represent conventional knock and the authors believe, instead, they result solely from gas-dynamic effect. Nonetheless, it is hard to determine whether or not the observed gas phenomenon is accompanied by harmful rise in temperature at a local level and possible engine damage nor is masking the knock phenomenon.

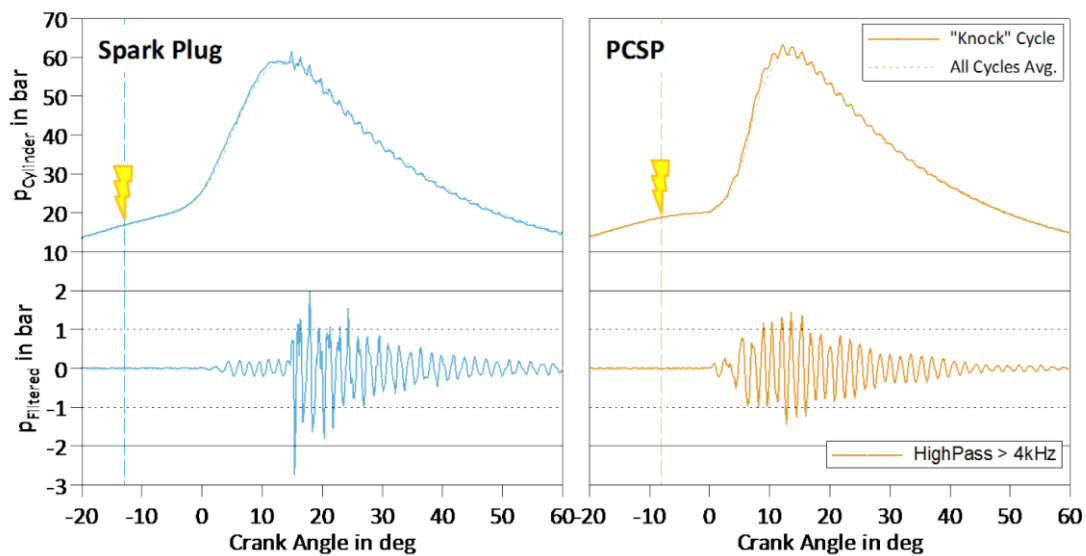


Figure 2.12 - Cycles with strong pressure oscillations at MBF50 = 8°CA for spark plug (left) and PCIS (right) at 2000 rpm and 10 bar BMEP [10]

It should be noted, however, that pre-chamber operation is not always limited to jet ignition, and when torch ignition is observed, the pressure oscillations must be interpreted differently. Tanoue et al. [131], working in a rapid compression machine (RCM) with a pre-chamber with 4.17% of the clearance volume with 1 and 2 interconnection holes, observed knock during PCIS operation. For this study a high-speed camera was used to record the combustion pattern. Figure 2.13 and Figure 2.14 show the combustion development observed through different crank angle positions for the 1-hole and 2-hole configurations, respectively, and indicate the occurrence of torch ignition in both cases.

In Figure 2.13 it is seen that the flame front released by the pre-chamber collide with the cylinder wall and is directed towards the pre-chamber holes and the upper and lower wall surfaces. Hence, the rapid flame dispersion, which is theoretically sufficient to suppress knock was, in fact, its cause, since the flame front that returns towards the interconnecting hole collides with the flame that still leaves the pre-chamber, resulting in knock at approximately 12° aTDC. In the 2-holes configuration, according to Figure 2.14, two flame fronts are developed in the main chamber, collide with the cylinder wall and, before auto-ignition is developed in the end-gas, the 2 flame fronts collide resulting in knock at 7.7° aTDC. For both configurations, after the flame is released to the main chamber, knock is marked by typical pressure oscillations and after several data analysis, the 1-hole PCIS design presented the maximum amplitude of pressure oscillation (MAPO) of 5.8 ± 1.8 bar, while MAPO for 2-hole design was 4.7 ± 1.2 bar. Therefore, although it was initially believed that the 1-hole configuration presented a greater knock intensity, this statement cannot yet be made due to uncertainties of test and measurements.

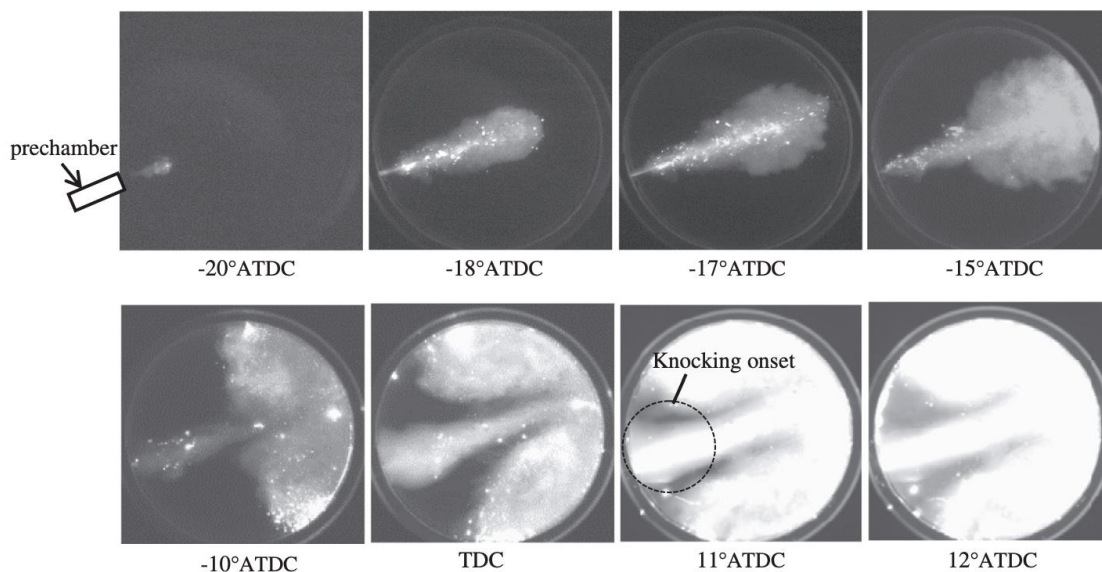


Figure 2.13 - Photographs of the flame development (1-hole configuration) [131]

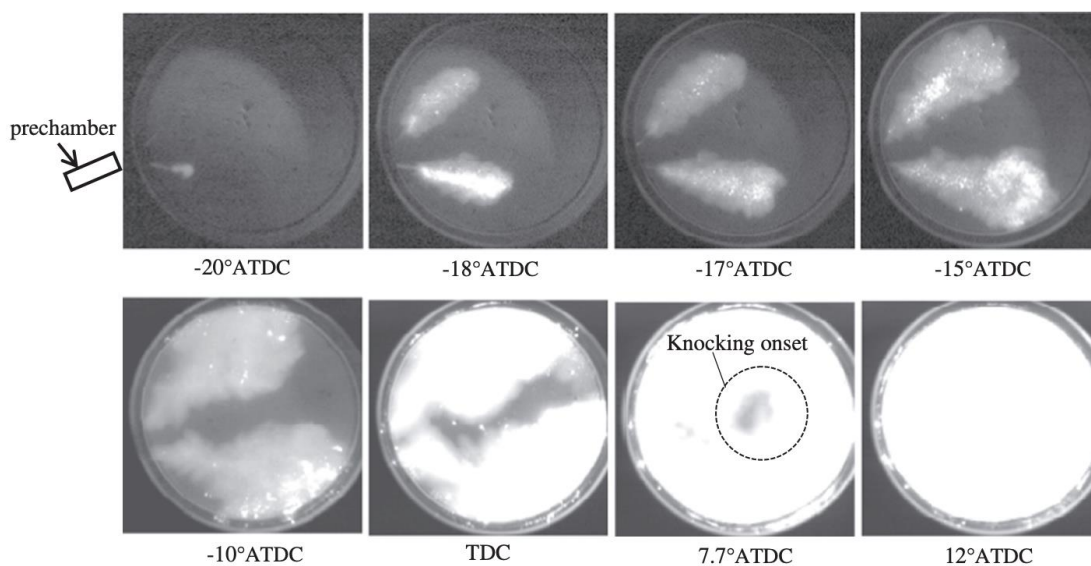


Figure 2.14 - Photographs of the flame development (2-hole configuration) [131]

Since the occurrence of torch and jet ignition may influence the determination of the knock occurrence, it is initially necessary to determine which phenomenon is dominant in PCIS operation to evaluate the pressure traces. In addition, suitable criteria for knock detection for operation with normal pressure fluctuations should be explored, so that the knock is not masked or misinterpreted. Such a topic will be further analyzed in the present work.

2.3.2.5 Pre-Chamber Geometry and Design

The performance of engines equipped with PCIS is highly dependent on their geometry, volume, nozzle diameter and number, and all these parameters have been widely explored by researchers. Adams et al. [132] pointed out that the turbulence generated by the pre-chamber is highly dependent on its volume, and hence, larger PC volumes can promote total combustion in the main chamber. In the 1970s, PC volumes corresponding to 8 - 15% of the total clearance volume were frequently studied and the authors pointed out that for a 5 L engine, the PC volume should be at least 12%. Corroborating with Adams et al., Nakazono et al. [133] pointed out that the combustion stability increases and the combustion period decreases with an increase in the pre-chamber volume. Wall and Heywood [134], experimenting two pre-chambers of 8.7% and 12% of the clearance volume, maintaining the total air-fuel ratio, pointed out that this volume variation had little impact on NO, HC or CO emissions. However, NO levels in the main chamber were reduced since the mixture was leaner.

On the other hand, Nakazono et al. [133] observed that, since the pre-chamber mixture is generally rich, its NO_x emissions increase. Hence, considering emissions produced by the PC combustion, a smaller pre-chamber volume is recommended. Furthermore, the greater the pre-chamber volume, the greater the increase in the combustion chamber surface and, consequently, the greater the thermal losses. Thus, current studies are performed with smaller pre-chamber volumes. More recently, Gentz et al. [135] tested pre-chamber with volumes between 1.1% and 3.7%, the best results were obtained with an intermediate volume of 2.4%. This result is according to previous literature, since intermediate volumes are pointed out as the best for reducing the emission of pollutants, fuel consumption and ensuring stability between cycles [9].

As previously mentioned, the pre-chamber nozzle diameter influences the flame extinction due to heat transfer and flow limitations. Furthermore, Sakai et al. [136] investigated the influence of nozzle diameter in the PCIS performance at partial and full load at MBT. The nozzle diameter varied from 4mm to 13mm and as the diameter decreased, the ignition timing was retarded and the combustion speed was increased. At partial loads the increase in combustion speed decreased exhaust temperatures and ISFC. However, at full load conditions, as nozzle diameter decreased, knocking phenomena and throttling losses were increased as the nozzle diameter decreases, increasing ISFC.

Wall and Heywood [134] also tested different nozzle sizes evaluating combustion parameters as well as emissions. It was observed that by decreasing the orifice diameter, a significant drop in peak pressure difference between the PC and the main chamber (MC) was observed, leading to an increase in cycle-to-cycle variation. This can be explained by Shah et al. [137], which highlights that smaller orifice diameters cause greater flow restrictions through the chambers, resulting in greater pressure losses and the development of long-lasting jets. Consequently, larger nozzle diameters reduce pressure

drop, developing jets of short duration. However, for very large orifices great variability between cycles and an increase in NO_x emissions can be observed [138].

Although according to Mavinahally et al. [139] the nozzle diameter has more influence on combustion, the quantity and arrangement of the interconnecting orifices, also determining part of the combustion process, such as pre-chamber filling in the compression stroke and the residual gas fraction in the pre-chamber. Moreover, the greater the number of holes, the greater the thermal efficiency, especially in ultra-lean mixtures, in addition to reduced cyclical variability, reducing HC and CO emissions.

Recently, as part of the EU funded EAGLE project, FEV together with the Institute for Internal Combustion Engines at the *Rheinisch-Westfälische Technische Hochschule* (RWTH) Aachen University and other partners developed a pre-chamber project with the objective of promoting efficient jet injection for use with ultra-rich mixtures. Müller et al. [140] then developed a pre-chamber geometry using 3D simulation of the flow and mixture formation to obtain information on pre-chamber turbulence and mixing levels. Such data was an input for OD simulations so that basic parameters such as nozzle size diameter, number of holes and chamber volume were predetermined.

After preliminary research, it was decided to work with pre-chambers with an internal volume equivalent to 3% of the total volume of the combustion chamber to avoid excessive thermal losses. When defining the nozzle positioning, there is no presence of a central hole, as shown in Figure 2.15, presenting the geometries considered. That is explained since the aim was to promote jets that move through the longest path possible without touching the cylinder walls, and a central hole would reach the piston very quickly. In addition, the rotation of gas flow entering the pre-chamber with low significant losses by dissipation was prioritized. This promotes a better mixture formation in the pre-chambers, leading to a faster combustion and consequently greater jet penetration power into the main combustion chamber.

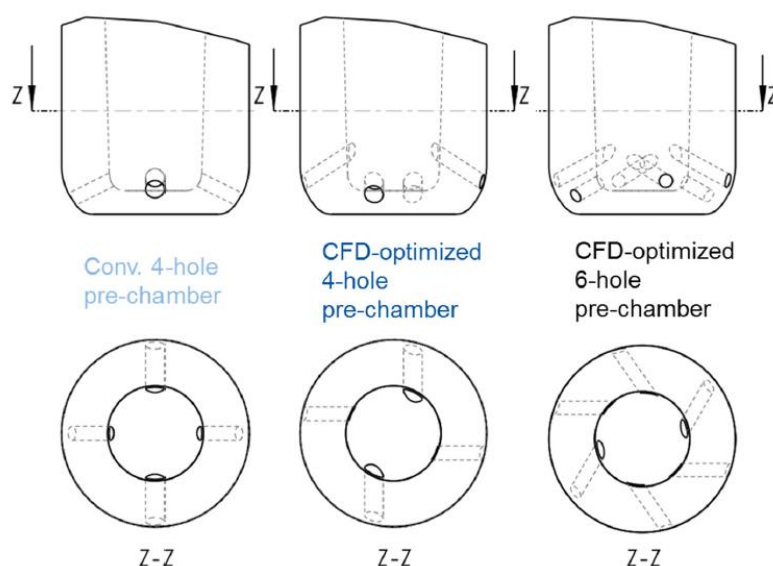


Figure 2.15 - Tested pre-chamber jet nozzle designs of Müller et al. [140]

Some design optimizations have introduced a small side and significant height offset to promote swirl into the pre-chambers and improve the mixture formation even further. This increased the turbulent kinetic energy level at the spark plug during the charging

phase of the PC, while the turbulent jets have a long free travel to ignite the mixture in the main combustion chamber as homogeneously as possible. With the optimizations, it was decided to use a 6-hole pre-chamber, in which a peak turbulent energy level of roughly 50% higher than the other analyzed configurations was observed, as shown in Figure 2.16. In the study, the amount of fuel injected into the pre-chambers was also analyzed, so that in tests carried out with 6 orifices, the injection of 0.2 kg/hr of CNG showed better results in terms of burn delay and combustion duration when compared to the injection of 0.15 kg/hour. However, such tests need to be carried out individually with each fuel used.

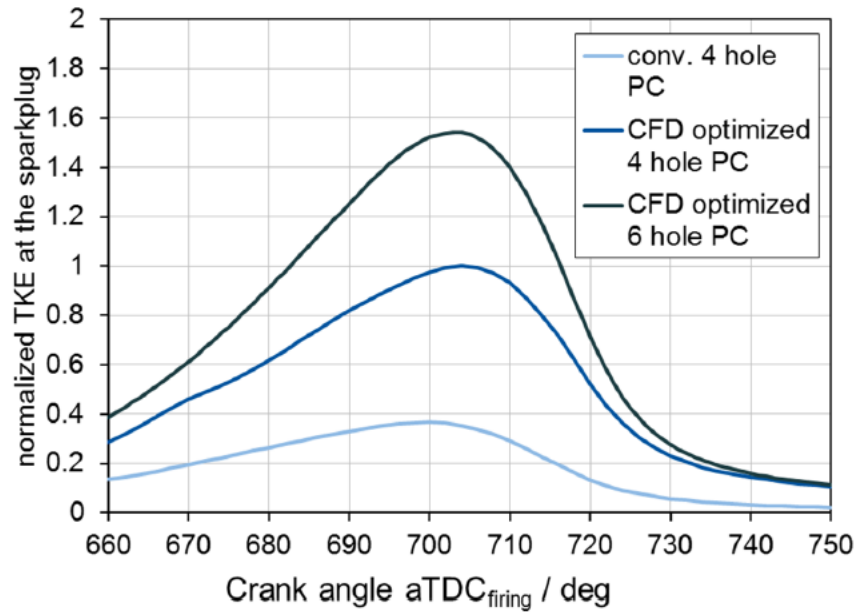


Figure 2.16 - Normalized turbulent kinetic energy at the sparkplug for the tested designs of Müller et al. [140]

2.3.2.6 Latest Pre-Chamber Developments

Over time, environmental restrictions have increased while electronic fuel-air mixture management technologies have developed. Consequently, the pre-chamber technology has been increasingly explored, both in academic research and automotive industry. Hence, the main works developed lately will be briefly discussed.

In the 1990s, the Hydrogen Assisted Jet Ignition (HAJI) system was developed by Harry C. Watson et al. [141] at the University of Melbourne. The system aimed to associate the active radicals from the rich hydrogen combustion in the pre-chamber with turbulence, providing ignition energy of two orders of magnitude higher than the spark. Such a system is ideal for initiating ultra-lean combustion in the main chamber. The pre-chamber is installed in the spark plug bore, has a small volume, 1% of the clearance volume, and uses hydrogen as an auxiliary fuel, corresponding to 2% of the total amount of fuel injected. The jet ignition occurrence is confirmed since flame quenching is observed in optical combustion images. The system can extend lean limit up to $\lambda = 2$, keeping $COV_{IMEP} < 2\%$ even though the lean limit can be extended to $\lambda = 5$. HAJI is able to reduce combustion temperature, reducing NO_x emissions and dissociations such as CO_2 to CO and H_2O to H_2 , increasing fuel conversion efficiency, which also increases due to reduced combustion speed.

Operating with ultra-lean limits, mainly with $\lambda > 2$, an increase in HC emissions is observed. Studying the sources of HC emissions, Lawrence et al. [141] concluded that the majority (72%) of the HC emissions were originated at piston ring crevices at engine conditions where adequate combustion stability was still maintained. Despite its good performance, the use of H_2 as an auxiliary fuel is currently linked to high production costs and/or the need to reform part of the gasoline for its production. Therefore, Toulson et al. [142] studied the possibility of operating with alternative fuels such as numerous alternative pre-chamber fuels, including CNG, LPG, reformed fuel mixtures and carbon monoxide in the pre-chamber and main chamber using the HAJI system. In the study it was highlighted that the effectiveness of the auxiliary fuel is highly dependent on the generation of chemical species, which varies greatly according to the properties of each fuel, and not only the energy contained in the jets. Furthermore, it was observed that the chemical properties achieved by operating with rich mixtures in the pre-chamber is essential to promote the quality combustion of leaner mixtures, and that the thermal and turbulent effects are secondary. Moreover, it is shown that, although hydrogen is considered the most suitable fuel to aid ignition and combustion, other alternative fuels can be used for its replacement without much compromise.

Following this research line, the Turbulent Jet Ignition (TJI) was developed by MAHLE Powertrain [143, 144]. This system was developed to operate on readily available commercial fuels such as gasoline, propane and natural gas. In TJI, the pre-chamber volume represents 2% of the clearance volume to reduce HC emissions, the crevice volume was reduced and the diameter of the interconnecting nozzles are kept small (1.25 mm) to enable the pre-chamber flame front to be extinguished. Operating with TJI, an improvement of 11% in peak efficiency was obtained in relation to the baseline ignition system. Furthermore, the system shows the potential to exceed 45% of indicated net thermal efficiency (19% relative improvement) with a r_c increase to 14. Comparing TJI with boosted Homogeneous Charge Compression Ignition (HCCI) using the same engine, the thermal efficiency achieved with the TJI exceeds the HCCI peak thermal efficiency of 43% [145, 146]

The *Ingenieurgesellschaft Auto und Verkehr* (IAV) has developed its own pre-chamber design with a volume representing 2-3% of the clearance volume and with 1mm interconnecting nozzle diameter. It is reported that the layout optimization of the passive PCIS is able to reduce the knock tendency during high-load operation. The use of PCIS can also enable the increase of compression ratio by approximately two units. Using an active PCIS, IAV achieved expanded the lean limit to $\lambda = 2.2$ with $COV_{IMEP} < 2\%$, while the passive pre-chamber was able to reach $\lambda = 1.2$. With the passive pre-chamber, fuel consumption considering the whole operation map is reduced by 2-4% and ultra-low specific NO_x emissions of approximately 0.3 g/kWh can be achieved. By adding an addition fuel metering system to the PCIS, the passive system can turn into an active PCIS, achieving 7-8% reduction in fuel consumption and also great NO_x reduction, even though an exhaust after treatment will probably be necessary [10]. Despite the benefits, the study shows additional heat losses from additional pre-chamber wall surface and at higher loads, additional wall heat losses are observed due to excessive turbulence in the main chamber. Furthermore, the study discusses the occurrence of pressure pulsations with pre-chamber operation, requiring a new definition of the conventional knock criterion [10].

In the latest Mahle study [147], several pre-chamber designs were evaluated, varying the number of nozzles and their diameters. The results indicate that nozzle designs can induce different engine responses at different operating conditions and that the main impact parameter on engine responses is the overall nozzle orifice area. At low speeds and loads, the smallest orifice areas tend to produce better engine performance, as it is able to extend the lean limit to $\lambda = 2.3$. This happens because, with a small orifice area, the jets are better distributed in the main chamber, in addition to having greater velocity and, consequently, greater penetration. With leaner combustion, greater thermal efficiency is observed, resulting in lower ISFC and lower HC, CO and NO_x emissions. It should be noted that the TJI causes exponential decrease in NO_x emissions with mixture enleanment. However, for high loads, engine operation is limited due to knock occurrence, which makes thermal efficiency more dependent on knock than nozzle design. This poses a difficulty, since the optimized configuration at full load impairs the combustion efficiency for larger lambdas and vice versa. In the study, however, it is shown that TJI is an effective technology for ultra-lean operation. Peak brake thermal efficiency of 41.5% were achieved while BSFC < 200 g/kWh were observed at many steady-state conditions. Considering a drive-cycle, the BSFC reduction obtained with the system is approximately 20% when compared to modern production engines of similar displacement and equivalent peak torque output.

Müller et al. [140] not only determined an optimized pre-chamber geometry to operate with ultra-lean mixtures but also tested some fuels in the process. In the study, gasoline was used as the main fuel and CNG as the auxiliary fuel and the amounts injected were varied, along with the number of interconnecting nozzles and their disposition. The cylinder head design is favorable to turbulence in the primary flow structure, which may have helped the PCIS performance. In the study, the pre-chamber configurations were tested at 15 bar IMEP and 2000 rpm and at 16 bar and 4000 rpm. For both operating points, the optimized pre-chamber with 6 nozzles with the injection of 0.2 kg/h of CNG achieved lower NO_x and HC emissions, higher combustion speed, with consequent lower ignition delay, higher indicated efficiency, and extended lean limit with $\lambda = 3$ with $COV_{IMEP} < 2$ at lower speed and $\lambda = 2.5$ for high speed. The study also highlighted that although gaseous fuels have a great advantage when injected into the pre-chamber, due to the ease of homogenization of the mixture, their use requires a second fuel tank with a specific gas injection system in the vehicle. Therefore, from the fuel supply infrastructure point of view, the injection of liquid fuels is the most obvious and should be considered and studied in greater depth. This however also presents challenges when the mixture formation and for the very unusual boundary conditions for the layout of the direct fuel injection system.

Stadler et al.[15], studied the 0.463L single-cylinder research engine (SCRE), with two compression ratios, 10.5 and 14.2, operating with gasoline and active and passive pre-chamber configurations. It should be noted that, unlike the conventional pre-chamber operation, in which PFI is used, a side-mounted direct injector was installed in the main chamber to ensure knock mitigation. The increase in turbulence was considered important for knock control, which allowed the operation with highest compression ratios. The active pre-chamber enabled the lean limit extension from 1.4 to 2.0 when operating with a 14.2 volumetric compression ratio. With the combination of the high compression ratio and ultra-lean combustion, the peak efficiency reached 43%, with NO_x emissions reduced to 0.2 g/KWh. In addition, lean burn reduced pumping losses for partial loads. However,

the increase in heat losses due to additional PC surface area and the increase in particulate emissions, were pointed out as challenges.

Despite the good results already achieved by pre-chambers operating with fossil fuels, the use of alternative biofuels must be prioritized due to all the previously mentioned benefits. Burkardt et al. [22], understanding the benefits to the efficiency of the internal combustion engines brought by both the use of biofuels and PCIS, associated the two strategies in a SCRE, using the pre-chamber developed at the study of Müller et al. [140]. In their study, passive and active pre-chamber concepts were used and two alcohol-based fuels, ethanol and methanol, were analyzed as main fuels, and ethanol as an auxiliary fuel, with direct injection of both fuels. In addition, the compression ratio was increased to 16.4 due to the fuels' higher knock resistance. The fuels were tested in part and high load conditions, at 5 and 16 bar of IMEP, and it was concluded that the configuration operating with methanol on the MC was able to operate with higher excess ratios and indicated efficiency. Specifically, the highest thermal efficiency, of 48%, was achieved at $\lambda = 2$. Combustion with methanol also resulted in lower specific HC emissions due to higher laminar flame velocity and high oxygen content of the fuel. This configuration also enabled the lowest specific NO_x emission, of 0.07 g/kWh at $\lambda = 2.3$. It was also observed that, even with the benefits of faster combustion, the pre-chambers operation presented at some specific points, from $\lambda = 1.4$ to 1.8, accentuated wall heat losses due to the additional surface area of the PC. The presence of knock was also detected, which should be further investigated, since this phenomenon prevented the correct calibration of the engine, in which the MBF50 had to be delayed.

The latest prototypes of stratified pre-chamber ignition systems developed have already shown great benefits in terms of fuel consumption and emissions. However, there is still a large gap in the use of pre-chambers with liquid fuels, which would greatly facilitate their implementation in the commercial vehicular fleet. In addition, a very small portion of the studies that consider the use of liquid fuels in such a system consider the use of biofuels, despite the promising results regarding efficiency and emissions. Thus, the present doctoral thesis aims to explore and further investigate some not fully understood aspects of PCIS, further investigating the stratified pre-chamber ignition system analyzed by Burkardt et al. [22]. The methodology section presented below will present details of how the experiments will be conducted, as well as which parameters will be analyzed exactly.

3 METHODOLOGY

This work was divided into two methodological parts. In the first part, experimental tests were carried in a single cylinder research engine to evaluate two ignition systems: the baseline spark plug ignition and a stratified pre-chamber ignition system using biofuels. The experimental data are analyzed according to different metrics and to extend the data analysis, 1D numerical simulations are performed, using the experimental data to calibrate the model. In this section, the equipment used, as well as the procedures performed in each phase of the research are described in detail.

3.1 Materials

The experimental tests are performed in a four-stroke DISI SCRE, whose main geometric characteristics are presented in Table 3.1. The intake and exhaust valve timings are presented in Figure 3.1 The large stroke/bore ratio enables operation with a high volumetric compression ratio by promoting a large tumble motion through the intake runner and pent roof design [148]. Furthermore, it should be noted that the high volumetric compression ratio is possible since the fuels to be tested, ethanol and methanol, show high knock resistance.

The cylinder head is designed modularly and capable of exchanging the ignition concept and for the present investigation, the cylinder head is equipped with an active pre-chamber positioned in the center of the pentroof. It can be highlighted that the r_c is determined when the spark plug is positioned in place of the pre-chamber, and as the pre-chamber installation increases the clearance volume, the engine compression ratio when pre-chambers are tested decreases 3.2%.

Table 3.1 - Engine geometric parameters [149]

	Unit	Value
Displacement	cm ³	500
Bore	mm	75
Stroke	mm	113.2
Stroke/bore ratio	-	1.5
Connecting rod length	mm	220
Valve per cylinder	-	4
Intake valve event length (at 1 mm valve lift)	°CA	178
Exhaust valve event length (at 1mm valve lift)	°CA	178
Maximum intake valve lift	mm	8.8
Maximum exhaust valve lift	mm	8.4
Compression ratio (at SI configuration)	-	16.4:1
Injector position MC	-	Lateral
Injector position PC	-	Central

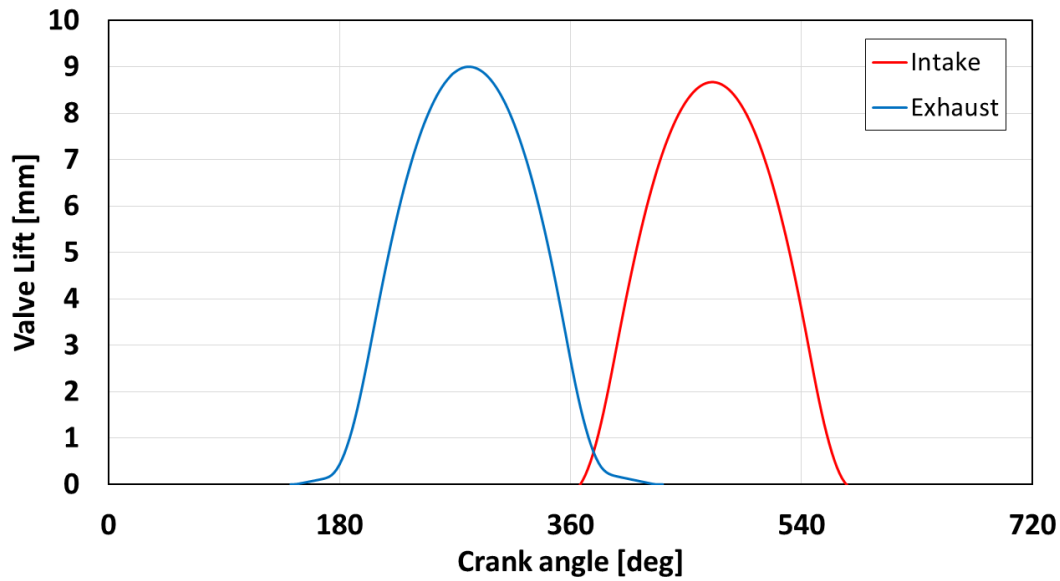


Figure 3.1 - Intake and Exhaust Valve Timings

The pre-chamber used in the research was also explored in the studies of Müller et al.[140] and Burkardt et al. [22], in which it was optimized, and its main geometric characteristics are presented in Table 3.2. The pre-chamber is equipped with a spark plug and a custom-made centrally located single-hole direct injector to prevent excess fuel impingement on pre-chamber walls, as shown in Figure 3.2. The main chamber fuel injection occurs through a laterally mounted direct injector between the intake valves.

Table 3.2 - Pre-Chamber geometric parameters [149]

Variable	Unit	
Number of orifices	6	-
Orifice offset	1.4	mm
Orifice diameter	0.85	mm
Orifice area	3.4	mm ²
Orifice length	2.72	mm
PC volume	1080	mm ³
PC volume to clearance volume	3.33	%
Orifice area to PC volume	0.0031	1/mm

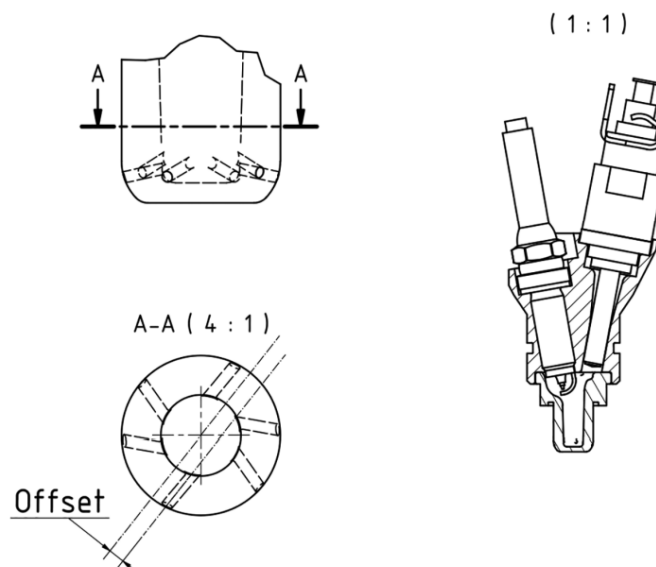


Figure 3.2 - Pre-chamber geometry [149]

An electric dynamometer and an eddy-current brake were used to impose engine load and engine speed to the SCRE, which was reached with accuracy of ± 1 1/min, independent of the load [150]. The SCRE is equipped with an external boosting device model A2710902680 327492 from Eaton and an exhaust gas backpressure valve to achieve respectively the required boost pressure and the targeted cylinder pressure difference. The maximum boost achieved is 3.5 bar [150]. The intake air mass flow was measured using a hot-film air mass flow meter upstream of the external boosting device. The MC and PC fuel mass flows were measured by means of specific Coriolis-type mass flow sensors. The in-cylinder pressures of MC and PC and the dynamic intake and exhaust pressures were measured respectively by the pressure transducers Kistler 6044A, Kistler 6054B, Kistler 4045A and 4045A. The main chamber pressure was in fact measured by two pressure transducers of the same type, installed oppositely between the intake and exhaust valves in the cylinder head. Hence, the indicated pressure data analyzed is an average of both MC pressure transducers. The crankshaft position was measured by an optical encoder with a resolution of 0.1° CA. Combustion data from each operation point were obtained by 200 cycles average and these data were processed alongside with the crankshaft position using the FEV combustion analysis system FEVIS. The static pressures and temperatures were measured with pressure transmitters and thermocouples, respectively, and averaged over an interval of 30 s. The exhaust emissions were sampled before the backpressure valve and within these gaseous emissions, HC, O₂, CO, CO₂, NO_x and soot emissions are measured. The HC is measured with a flame ionization detector, the O₂ using a paramagnetic oxygen analyzer, the CO and CO₂, with an infrared gas analyzer, using the equipment Rosemount NGA 2000. NO_x was measured using a chemiluminescence analyzer model Eco Physics 700 EL ht and soot was determined through the smoke meter AVL 415s.

Table 3.3 shows the sensors used for performance and combustion analysis and its accuracy.

Table 3.3 - Specifications of the used measurement instruments [151]

Instrument	Model	Measurement range	Accuracy
Engine dynamometer	Schorch	0–10,000 1/min	≤0.04%
Air mass flow meter	FEV air rate	0–1000 kg/h	≤1.00%
Fuel mass flow meter	Emerson micro motion	0–34.92 kg/h	≤0.054%
In-cylinder pressure transducer	Kistler 6044A	0–300 bar	≤0.13% @30 bar
			≤0.01% @200 bar
Intake pressure transducer	Kistler 4045A	0–5 bar	≤0.3%
Exhaust pressure transducer	Kistler 4045A	0–5 bar	≤0.3%
Flame ionization detector (HC)	Rosemount NGA 2000	0–25,000 ppm	≤1%
Chemiluminescence analyzer (NO _x)	Eco physics CLD 700 EL ht	0–10,000 ppm	≤0.5%

3.2 Experimental Procedure

In the current study, the performance of a SCRE is evaluated considering the use of ethanol and methanol and with the analyzed fuels, 5 configurations are tested and Table 3.4 presents their nomenclature and explanation.

Table 3.4 - Nomenclature of the configurations tested

Nomenclature	Explanation
SI-MeOH	Methanol used as main fuel in baseline ignition
SI-EtOH	Ethanol used as main fuel in baseline ignition
MC-MeOH/PC-MeOH	Methanol used as main fuel and as auxiliary fuel in pre-chamber ignition
MC-MeOH/PC-EtOH	Methanol used as main fuel and ethanol as auxiliary fuel in pre-chamber ignition
MC-EtOH/PC-EtOH	Ethanol used as main fuel and as auxiliary fuel in pre-chamber ignition

To ensure the good performance of the pre-chamber and consequently of the fuel pairs, 4 operating points are tested, at low and medium speeds and at low and high loads, as presented:

- 2000 rpm and 6 bar of IMEP;
- 2000 rpm and 15 bar of IMEP;
- 3500 rpm and 6 bar of IMEP and
- 3500 rpm and 15 bar of IMEP.

For each operating condition, the ignition timing is varied in order to achieve maximum brake torque (MBT). In general, this range occurs when half of the mass burned fraction (MFB50) is observed at 8° aTDC. Furthermore, the calibration aims to achieve the smallest cyclical variation of the IMEP (COV_{IMEP}), which will be kept lower than 5%. This criterion is used to reach the lean limit for each set of fuels is reached [126]. Injection timing in the main chamber is 420° bTDC and in the pre-chamber is 60° bTDC. Injection pressure is 200 bar for the pre-chamber injector and 5 bar for the main chamber injector.

Once the calibration step has been completed, the engine is ready to start data recording. After reaching the desired condition, one minute must be waited before data recording to guarantee the stability of the analyzed condition. In these tests, combustion and emission parameters are analyzed. A special analysis is also carried out considering knock occurrence, also determining the knock threshold for each operation point analyzed, as

described in the next subsection. After carrying out the experimental tests, the measurement uncertainties is conducted and described in Appendix E of this document, according to the guidelines of Albertazzi and Souza [152].

3.3 Pressure Oscillation Analysis

The in-cylinder gas resonance modes become more evident during knock, as more energy is supplied. Therefore, one knock-detecting method is the analysis of the in-cylinder wave's resonance modes. As the excitation of resonant modes can indicate knock occurrence, a Fast Fourier Transform (FFT) analysis is performed in the pressure data. The modes to be calculated are presented in Figure 3.3.

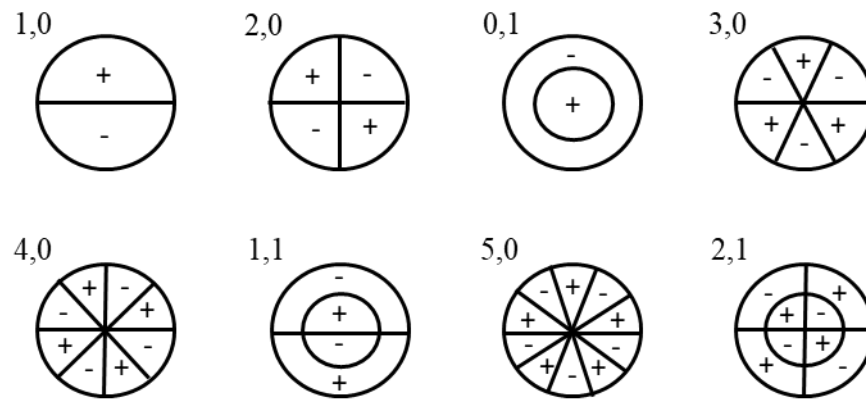


Figure 3.3 - Oscillation Modes Calculated. Adapted from [131]

To determine the engine resonant modes, the mathematical model developed by Draper [153] is used. In Equation 3.1, the frequency of the mode i, j , $F_{i,j}$, is calculated, in which a is the sound speed of the burned gas, D is the bore diameter and $B_{i,j}$ is a Bessel constant related to the radial modes i, j [154]. The Bessel function is used in this case since according to experimental observation, it accurately represents the velocity profile in an engine [155].

$$F_{i,j} = \frac{a \cdot B_{i,j}}{\pi \cdot D} \cdot 10^{-6} \quad 3.1$$

However, it should be highlighted that combustion chamber resonance can be excited by phenomena other than knock [156] and pressure fluctuations can be detected in some other operating regimes, such as HCCI. Thus, other knock metrics are analyzed in the present study. Bradley and Kalghatgi [157] showed the relationship between the autoignition wave velocity and the amplitude of a pressure pulse subsequent to an auto-ignition occurrence. Furthermore, since the first waves of the pressure oscillation are responsible for engine structure excitation, this excitation is proportional to the acoustic energy flux, I , associated to this oscillations, as presented in Equation 3.2, in which ΔP is the pressure wave amplitude [158]. Thus, the knock index can be measured as the maximum amplitude of pressure oscillation (MAPO) of the knock signal, as used by the Müller et al. [9].

$$I = 10 \cdot \frac{(\Delta P)^2 \cdot \sqrt{\gamma R T}}{2 \gamma P} \quad 3.2$$

As one of the main knock indicators, the MAPO analysis is used in the present study, being defined with a band pass filtered cylinder pressure with a pass band from 3-30 kHz. The 200 MAPO values obtained from this calculation for each operating point collected and analyzed are sorted in ascending order. With this new distribution, the MAPO values are analyzed using log-normal distribution plots, as pointed out by Iwata et al. [159], who show in their study that non-knocking and knocking data sets follow log-normal distributions, but with different slopes. Therefore, it is possible to determine the knock threshold of any operation by analyzing the data collected at each point. This strategy was used by Corrigan et al. [156] in the determination of new knock thresholds in engines operating by pre-chambers, as shown in Figure 3.4, in which the knock threshold is determined as soon as the data set no longer presents a linear behavior.

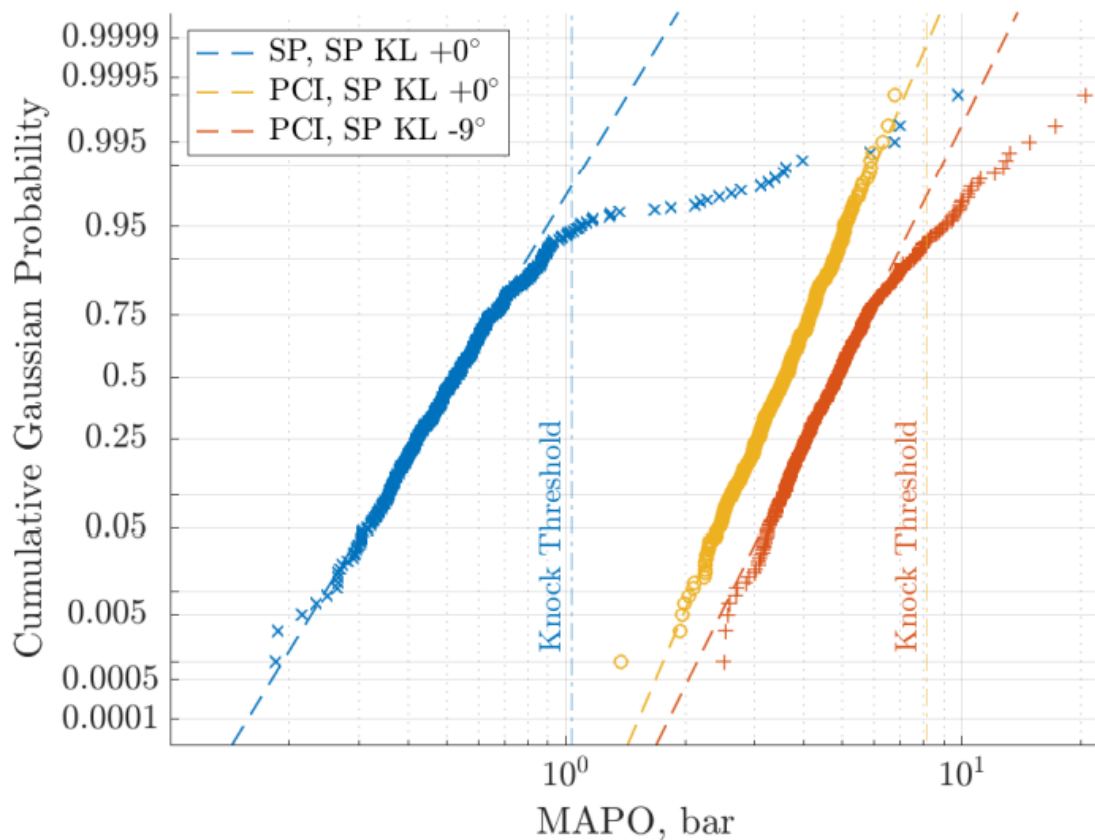


Figure 3.4 - Knock threshold determination using a method suggested by Iwata et al. [159] Source [156]

The MAPO threshold determination presented in Figure 3.4 is also used in the present study. Thus, it is possible to determine a threshold for each operating point of the engine operating with pre-chambers. It should be noted that this strategy, despite being a first response to the problem of knock detection in engines operating with pre-chambers, still presents limitations, because the threshold is determined only after the collection and processing of engine data for each point of operation. Furthermore, the threshold changes not only for each operating point, but also for different calibration conditions, such as the location of the MBF50. From this threshold, however, it is possible to perform an equivalence so that the threshold is found for other knock detection parameters, such as the resonance peaks observed in the FFT analysis [156].

It should be noted, however, that for pre-chamber ignition analysis, MAPO is found to be influenced by the jet oscillation, leading to knock overprediction [160]. Therefore, the data analyzed by these parameters will be compared with the parameter knock index 2 (KI2), suggested by Galloni et al [161]. One of the main differences between spark knock and pre-chamber jet-induced pressure oscillations is the fact that pressure oscillations in pre-chamber operation start from the start of combustions, and not from the peak of cylinder pressure [162]. Therefore, the KI2 method aims to develop a threshold to assess the knock occurrence on a case-by-case basis, analyzing the moment when pressure fluctuations occur. The MAPO threshold $MAPO_{th}$ is defined based on the maximum pressure fluctuation in the first stage of combustion, from mass burned fraction (MFB) 5-60%, according to Equation 3.3. In the equation, k is a constant and in this study the factor used is 1 and j is the cycle number analyzed [160].

$$MAPO_{th,j} = k \cdot \max(MAPO_{MFB\ 5\%}^{MFB\ 60\%}) \quad 3.3$$

This threshold is then compared with the maximum oscillation calculated in the second stage of combustion, from MFB60 until MFB90. Therefore, the knock event is only detected if large-amplitude oscillations occur prominently after 60% of combustion. Hence, the knock intensity, $I2$, is calculated according to Equation 3.4, in which $I2$ equals zero whenever the oscillations before MFB 60 are greater than at the second part of the combustion [160].

$$I2_j = \frac{\max[(MAPO_{MFB\ 90\%}^{MFB\ 60\%} - MAPO_{th,j}), 0]}{MAPO_{th,j}} \quad 3.4$$

From the knock intensity, the KI2 is defined according to Equation 3.5, in which N_{cyc} is the number of cycles acquired in the analyzed acquisition point [160].

$$KI2 = \frac{\sum_{j=1}^{N_{cyc}} I2_j}{N_{cyc}} \cdot 100 \quad 3.5$$

3.4 Numerical Analysis

As stated previously, the experimental tests are carried out with the aid of numerical analysis and the main goal of numerical analysis is the heat and mass transfer between main and pre-chambers. This is of utmost importance, since the cylinder characterization is somewhat particular when modeling an engine equipped with pre-chambers.

Based on the information provided by the model, it is also possible to intensify the analysis carried out previously, more focused on the study of pressure fluctuations. With the model validation through experimental results, the in-cylinder sound speed and the average velocity of the jets in each of the pre-chamber interconnection holes can be calculated.

The internal combustion engine computer model developed in the GT-power software to represent the experiments conducted in the present study is shown in Figure 3.5. The model is made of blocks, which represent the parts and systems that make up the engine and are found in the software's internal library. Each object is associated with a set of

equations that describe its operation and, for the modeling of each part or system, it is necessary that each block receives various information as input. Once modeled, the objects and systems are interconnected according to a logic flow of engine fluids. Thus, the engine simulation is carried out by solving the set of equations associated with each object and system in the model. Further explanations of the model are presented in Appendix A.

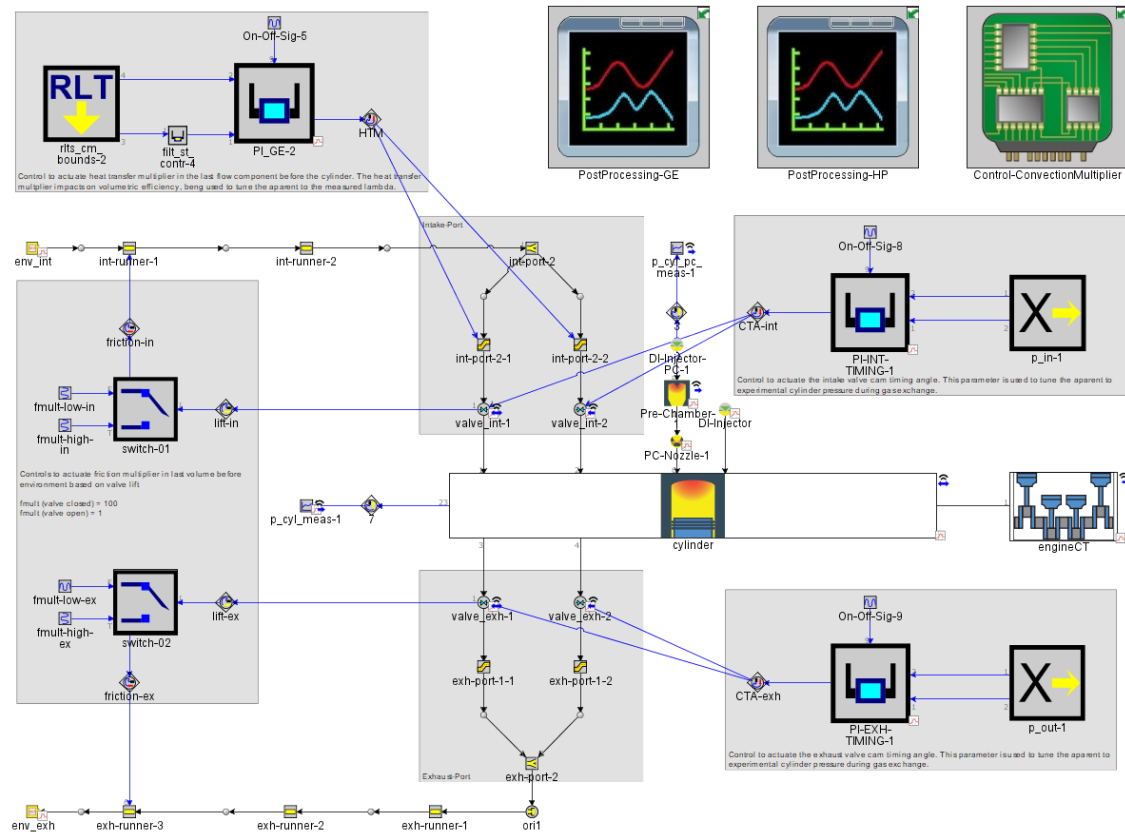


Figure 3.5 - GT-Power Model

4 RESULTS

In the current section, the results obtained experimentally and through simulations are discussed. It should be noted that there is no separation between the two types of analysis since they are complementary and help in a deeper analysis of the object of study. Initially, the first impressions obtained from the experimental tests are discussed, followed by discussions of the extent of the lean limit and combustion speed in each configuration tested. Next, discussions are held on the in-cylinder pressure in each configuration, as well as analysis on the observed pressure oscillations. This is followed by a discussion on the possibility of knock occurrence in such an operation. After such an analysis, the main analyzed engine efficiencies are discussed together with the combustion chamber temperature and fuel consumption. Furthermore, the exhaust emissions for each configuration are discussed. Finally, simulation results of extrapolations of the studied model are presented, in which higher compression ratios are analyzed for both the baseline engine and the engine with PCIS.

4.1 Preliminary Discussions on the Tested Fuels

During experimental tests, the pre-chamber fuel injection is controlled in order to inject as little fuel as possible, limiting the amount of fuel in the PC walls. For this to be possible, an injector with only one orifice was used. The latest injector design developments were a reduction of injector diameter for better control of fuel amount. However, that became a problem during these specific tests since deposits are formed within the injector hole and on the outside of the injector tip, especially when methanol is injected. These deposits reduce the hydraulic diameter in the nozzle, reducing consequently the amount of fuel injected and spray quality, leading to a fast degradation of methanol operation [163]. Hence, the injectors must be cleaned with ultrasound baths, as shown in Figure 4.1. However, in some cases a complete injector replacement is required.



Figure 4.1 - Pre-chamber injector holes after A) 13-hour operation and B) 9-hour operation before and after ultrasonic bath

The deposits observed are certainly not related to carbonization, since this phenomenon is associated with fuels of high viscosity, density, lower heating value and lower cetane number [164], or the presence of additives such as zinc in the fuel [163]. Since the use of alcohols in conjunction with high density and viscous fuels to reduce deposits and injector coking [165], the observed deposits occur probably due to gum formation.

The gum occurrence on the injectors tip can be caused by oxidation during fuel storage and polymerization processes at high temperatures and pressures in the combustion chamber [166]. The presence of alcohol generally dissolves gums in storage and transport vessels and thus, even though they are not likely to form gums, they can carry gum into the engine fuel system. Furthermore, impurities in the fuel, such as copper, can contribute to gum formation [167]. In the present study, rubbers and seals were replaced to be resistant to alcoholic fuels. However, the fuel pump has not been replaced and may be the source of gum. For this, a filter is used after the pump to limit the passage of any contaminants. In addition, the pre-chamber gasket was also changed to prevent lubricant leakage into the pre-chamber, and its consequent polymerization.

This problem continued to be observed, especially with the operation with methanol, which is justified due to the greater corrosive power of the fuel, requiring greater precautions when handling it [168]. Experimental tests with methanol were delayed since it was necessary to send the tested injectors to the manufacturer to identify the source of the problem. Considering current ethanol relevance and in view of the delay brought due

to methanol injection, the data obtained with methanol was considered enough to be used as a comparison.

The operation points that are evaluated are presented in Table 4.1. It is noteworthy that the difficulty of operating with methanol, especially at 15 bars of IMEP, in which more fuel should be injected, both in the MC and the PC leads to more gum and deposits to be brought by the fuel into the injector, making it currently impossible to operate.

Table 4.1 - Operation points discussed in the Results section

Engine Speed [rpm]	IMEP [bar]	SI-MeOH	SI-EtOH	MC-MeOH/PC-MeOH	MC-MeOH/PC-EtOH	MC-EtOH/PC-EtOH
2000 rpm	6	x	x	x	x	x
	15		x	x		x
3500 rpm	6	x	x	x	x	x
	15		x	x		x

After the initial problem and discussion, all experimental tests were performed, as well as the simulation of such cases in GT Power software. The simulations were performed to extract additional information relevant to the thermodynamic analysis performed in this chapter. Thus, results obtained through simulations are analyzed together with experimental results throughout this chapter.

In addition to the convergence imposed by the TPA model in the main chamber, lambda and net IMEP criteria are also used as convergence, and to validate the model for all the cases mentioned, the experimental and simulated lambda and IMEP must present at least 95% accuracy. Appendix B shows the measured and simulated pressure curves in 3 different lambdas (1, 1.5 and the lean limit) for all simulated cases with the simulated IMEP and lambda, used for validation.

4.2 Lean Limit and Combustion Development

As an initial analysis, Figure 4.2 presents the net IMEP and the CoV_{IMEP} for all configurations tested and as initial comments, for all operating points, ignition types (baseline and PCIS) and fuel combinations, different lean limits are observed. Table 4.2 summarizes this observation, presenting the lean limit for all cases, in which the lean limit is further extended with the use of pre-chambers instead of baseline ignition and generally using methanol over ethanol. As for the fuel combinations used, the MC-MeOH/PC-MeOH reaches mainly the leaner limits, followed closely by MC-MeOH/PC-EtOH. The lower lean limit of the four pre-chamber combinations is the one operating solely with ethanol. The only exception to this rule is at 2000 rpm and 6 bar of IMEP, in which the MC-MeOH/PC-EtOH configuration presents a leaner limit than MC-MeOH/PC-MeOH. At this operating point lambda 2.6 was reached with CoV_{IMEP} 5.3, which is slightly above the established operation threshold. It should be noted, however, that the pre-chambers operating solely or partially with ethanol present a much higher CoV_{IMEP} after the limits indicated in the table are reached, as presented in Figure 4.3, which shows the greater capacity of methanol to maintain combustion stability in regimes of leaner operation.

In addition, it is interesting to observe the lean limit behavior with the variation of loads and engine speeds. In general, the lean limit is extended at lower speeds and loads, what was not expected. Theoretically, high loads would increase the in-cylinder temperature,

causing the flame to be quenched later on and higher speed would increase combustion chamber turbulence, also accelerating the mixture combustion, extending the lean limit.

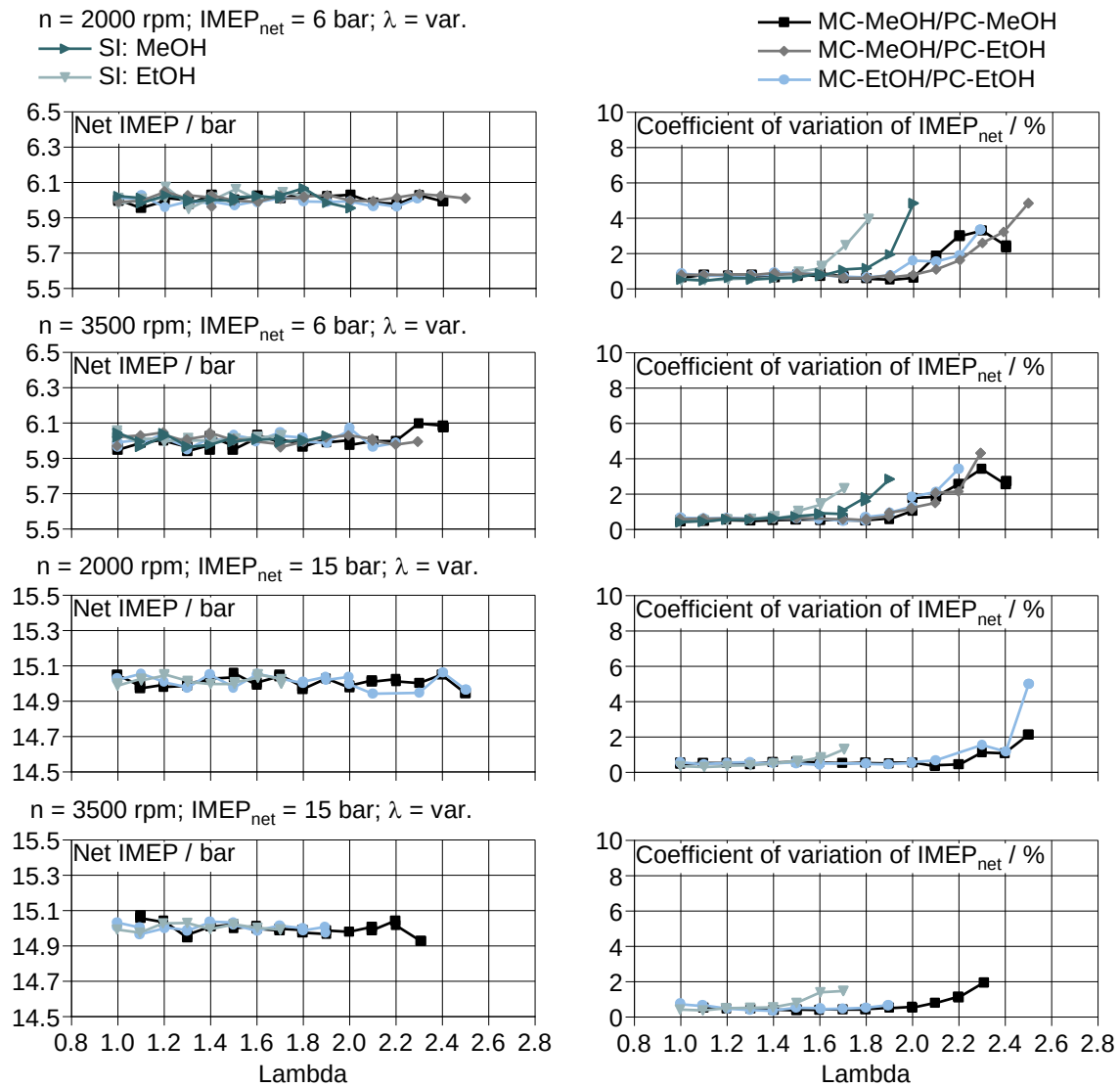


Figure 4.2 - Net IMEP and coefficient of variation of IMEP at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and MeOH)

Table 4.2 - Lean Limit at methanol (2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and MeOH)

	2000 rpm		3500 rpm	
	6 bar IMEP	15 bar IMEP	6 bar IMEP	15 bar IMEP
SI-MeOH	2.00	-	1.90	-
SI-EtOH	1.81	1.71	1.71	1.70
MC-MeOH/PC-MeOH	2.40	2.50	2.41	2.31
MC-MeOH/PC-EtOH	2.49	-	2.29	-
MC-EtOH/PC-EtOH	2.29	2.50	2.20	1.89

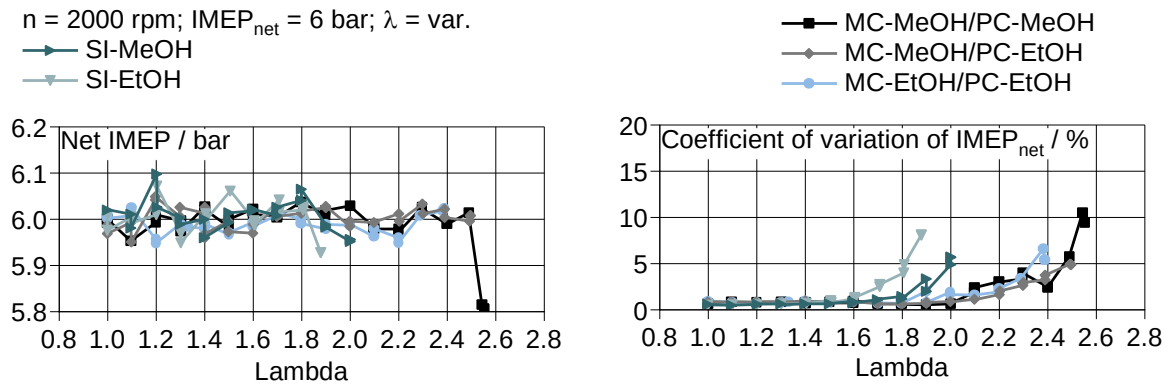


Figure 4.3 - Net IMEP and coefficient of variation of IMEP at 2000 rpm and 6 bar of IMEP using several combinations of ethanol (EtOH) and MeOH). Observation of the behavior after $\text{CoV}_{\text{IMEP}} 5\%$

Before conducting such investigations, however, the reasons for the behavior of the different fuels and ignition systems in relation to the mixture enleanment is explained in more detail and for that, all cases will be analyzed using Figure 4.4, Figure 4.5, Figure 4.6 and Figure 4.7, which show the combustion duration at different phases.

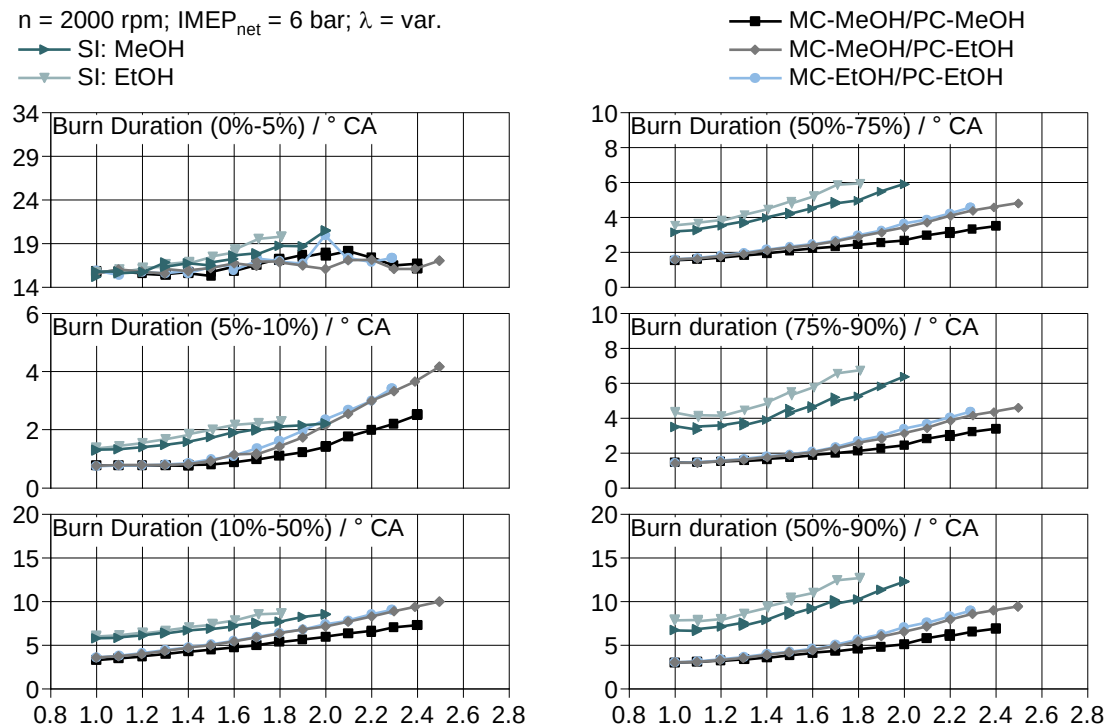
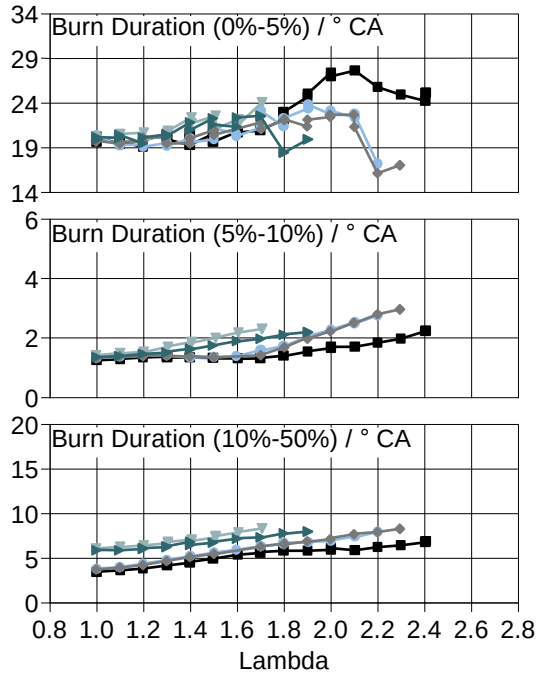


Figure 4.4 - Burn duration at 2000 rpm and 6 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

$n = 3500 \text{ rpm}$; $\text{IMEP}_{\text{net}} = 6 \text{ bar}$; $\lambda = \text{var.}$

— SI-MeOH

— SI-EtOH



— MC-MeOH/PC-MeOH

— MC-MeOH/PC-EtOH

— MC-EtOH/PC-EtOH

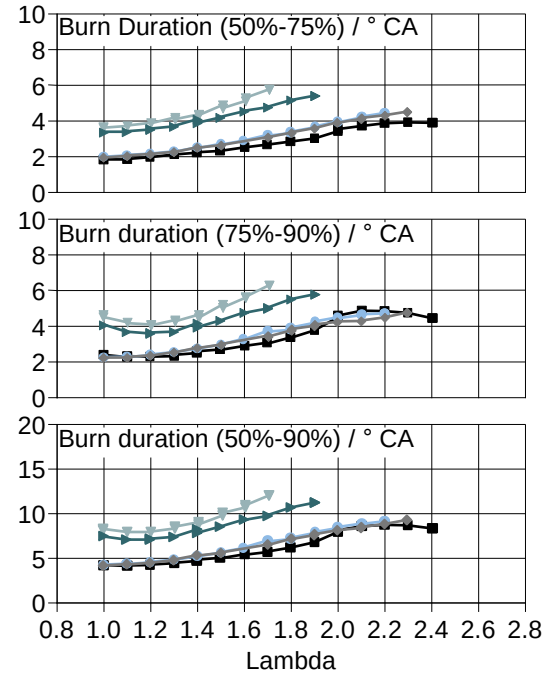
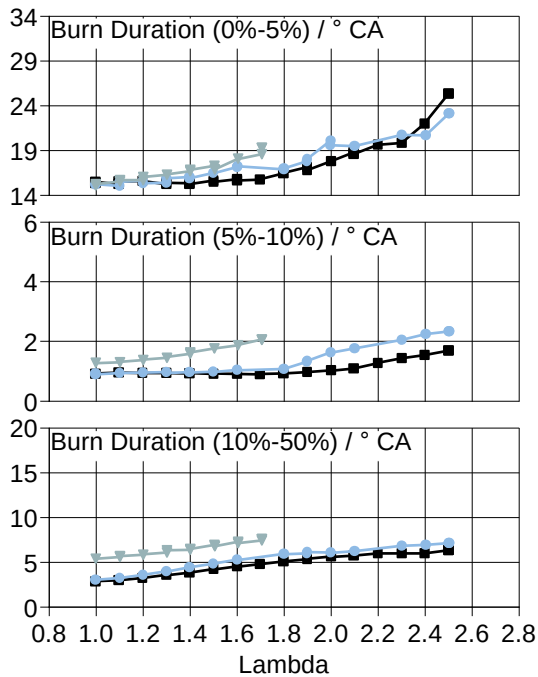


Figure 4.5 - Burn Duration at 3500 rpm and 6 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

$n = 2000 \text{ rpm}$; $\text{IMEP}_{\text{net}} = 15 \text{ bar}$; $\lambda = \text{var.}$

— SI-MeOH

— SI-EtOH



— MC-MeOH/PC-MeOH

— MC-MeOH/PC-EtOH

— MC-EtOH/PC-EtOH

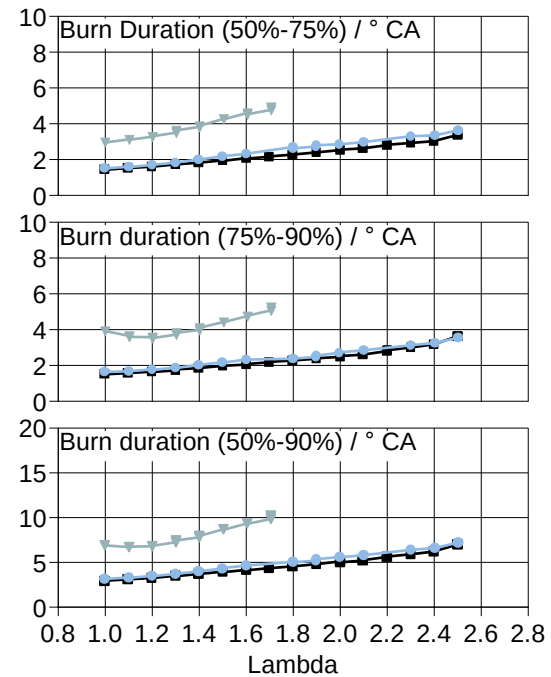


Figure 4.6 - Burn Duration at 2000 rpm and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

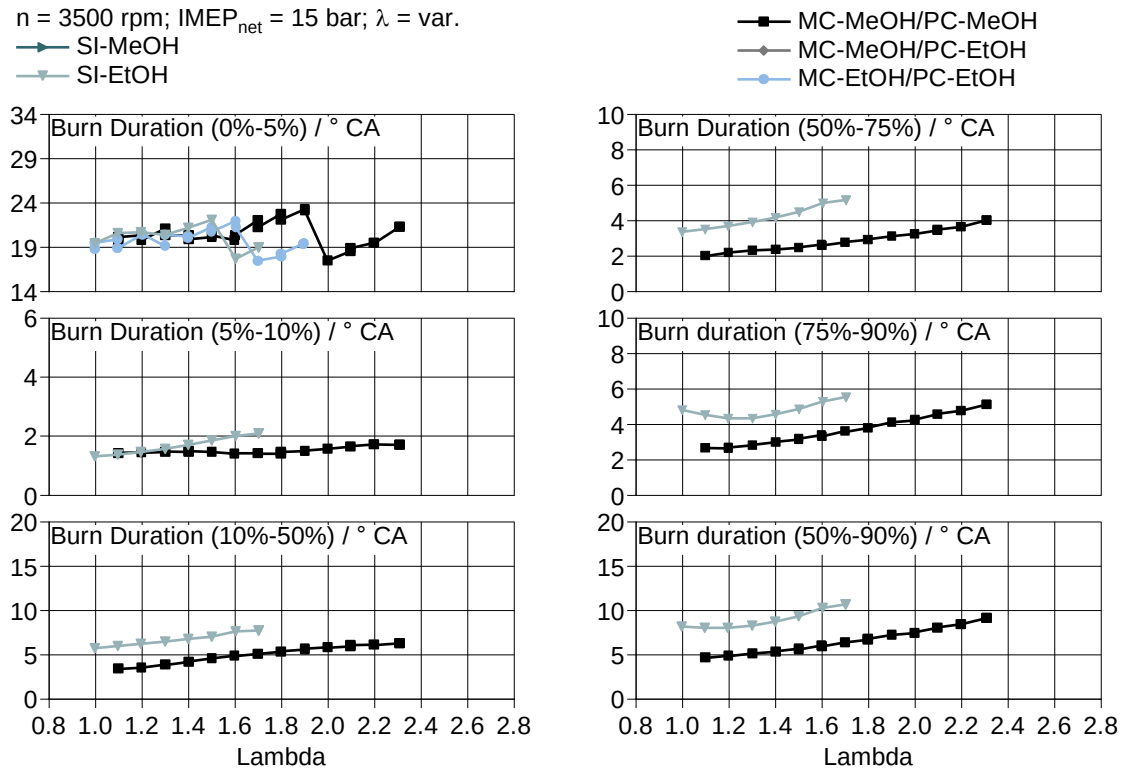


Figure 4.7 - Burn Duration at 3500 rpm and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

The burn duration MBF0-5 is the flame development period, in which ignition energy and laminar flame speed are the determining factors. During this period, a flame kernel is formed and developed into a self-sustaining flame front, which is initially laminar [34]. Moreover, such duration is also slightly shorter when using pre-chambers, which is justified by the richer mixture being burned in the pre-chambers. Such duration, although quite similar for each load and speed for all configurations analyzed in conditions close to stoichiometry, is always shorter for methanol in relation to ethanol with the mixture enleanment. This is justified both by the faster laminar flame speed and by the lower minimum ignition energy in air of methanol [29]. The latter is a justification for the leaner limit of methanol, since misfires are less frequent for more easily ignitable mixtures.

It is also observed that pre-chamber operation presents a more stable MBF0-5 duration in relation to baseline ignition. This shows the relevance of using an ignition system that provides greater ignition energy for lean mixtures. In general, for leaner mixtures MFB 0-5 duration has a more pronounced increase for ethanol compared to methanol. Such observation is interesting because with the increase of lambda the temperature in the combustion chamber decreases, increasing the ignition delay of methanol in relation to ethanol, as shown in Figure 4.8 [169]. However, increasing lambda reduces the laminar flame velocity of ethanol more than that of methanol, as shown in Figure 4.9. Therefore, it is concluded that increasing the ignition delay of methanol has less impact on the start of combustion than reducing the laminar flame speed of ethanol.

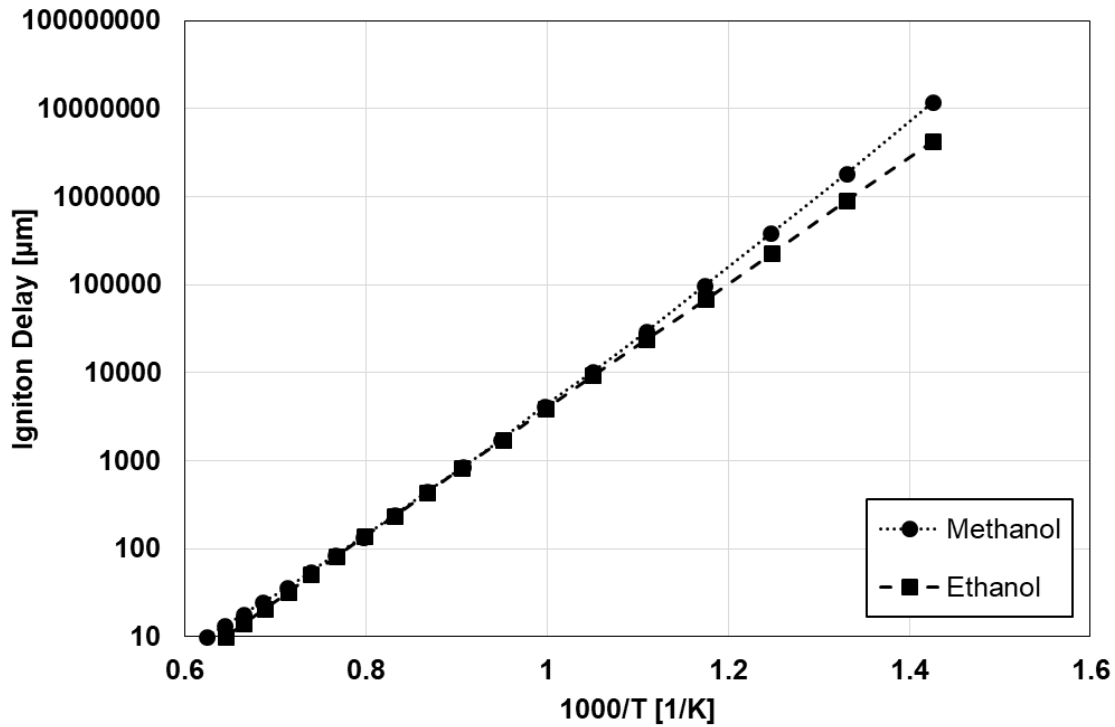


Figure 4.8 - Ignition delay of ethanol and methanol as a function of temperature at 5 atm. Adapted from [169]

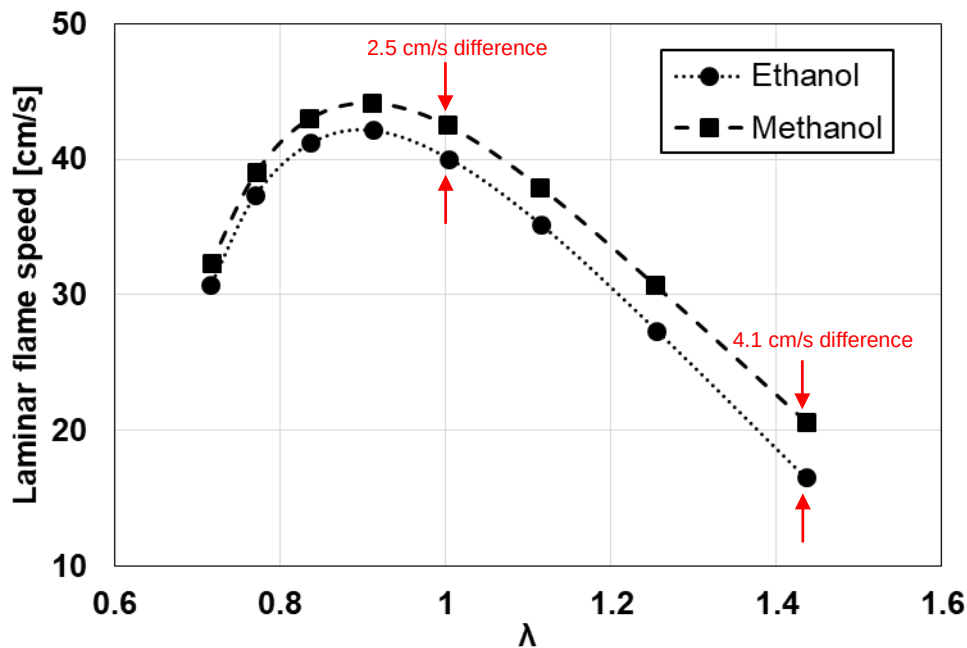


Figure 4.9 - Laminar flame speed of ethanol and methanol as a function of air-fuel equivalence ratio and the speed difference with mixture enleanment (1 atm, 300K). Adapted from [170]

It should be noted, however, that the configuration PCIS 1 at the operating point 3500 rpm and 15 bar of IMEP presents an abnormal behavior as the mixture is enleaned, and up to λ 1.7 the MBF0-5 is much longer than expected, which may be justified by lack of fuel injected into the pre-chamber. Anomalies at this point will also be observed in other analysis and hence, a further discussion on this topic will be conducted in a future subsection.

In addition, it is worth mentioning that the duration of the MFB 0-5 is not at all altered for load variation. However, it theoretically increases with the engine speed from 2000 to 3500 rpm, being approximately 15.5 °CA at 2000 rpm and 20°CA at 3500 rpm close to the stoichiometry, representing a 29% increase in duration in crank angle degrees. Nonetheless, considering that each CA at 3500 rpm occurs at a 42.9% shorter time when compared to 2000 rpm, the flame development period reduces in time approximately 26.5% with increasing rpm. Such reduction is probably justified since the in-cylinder temperature increases with engine speed due to the reduced time available for heat transfer [171]. As laminar flame speed increases with increasing in-cylinder temperature, less time is required for this initial phase of combustion, strongly dominated by the laminar regime [172].

The period in which combustion is between 5-10% has an initial predominance of the laminar regime. Therefore, in conditions close to the stoichiometry, the MBF5-10 is similar between the baseline and PCIS ignition systems. However, as the flame is developed, it starts to be affected by scales of turbulence [173], what is observed from MBF 5-10 onwards, in which combustion with baseline ignition tends to last longer than PCIS ignition, a system that significantly increases turbulence in the main chamber.

The difference in duration of combustion between ignition systems is much more prominent than the difference between main chamber fuels and is already visible from stoichiometry, increasing as the mixture is leaner. Initially it is possible to notice that the combustion durations before and after the MBF 50 behave differently.

In general MBF 50–90% is significantly more affected by the use of pre –chambers since the system is capable of maintaining a high turbulence level for an extended period of time [116]. However, as the mixture is leaner and the in-cylinder temperature reduces, the greater is the difference between MBF 50-90 and MBF10-90 in general.

The use of pre-chamber more at lower loads and speeds. This occurs because with increasing speed, the inertia of the admitted gases increases, increasing the turbulence level and in-cylinder temperature, what is capable to reduce combustion duration. The load increase is also capable of accelerating combustion duration, but not as effectively as increasing engine speed. This can be explained since pressure increase is responsible for reductions in the laminar flame speed [172]. Furthermore, since at higher loads a higher amount of fuel is to be burned, the combustion process takes more time.

A slightly abnormal behavior, however, is observed for loads of 15 bar of IMEP at both speeds, mainly when observing the MBF 10-50. As the load increases, the PCIS configurations present a further combustion duration decrease. This indicates that more fuel than necessary was injected into the PC in these two cases, what may have created region of enriched mixture near the interconnection holes in the main chamber, accelerating the fast half of combustion.

Considering the use of both fuels, comments should be made regarding their differences. The superior laminar flame speed of methanol, extensively reported in the literature [29], is evidenced by the faster combustion duration in general when comparing the SI operation of methanol with that of ethanol. Although the laminar regime is dominant only at the beginning and end of combustion, it influences the combustion of methanol in a general way. However, the combustion duration of methanol is shorter than that of

ethanol, especially when the mixture is lean and in the second half of the combustion, where additional turbulence has a greater effect. Therefore, it can be inferred that the faster propagation of the flame front increases the end-gas pressure, producing more turbulence in the combustion chamber and, consequently, a faster reaction.

When the engine is equipped with pre-chambers, the impact of the main chamber fuel is more pronounced than the pre-chamber fuel, according to using Figure 4.4, Figure 4.5, Figure 4.6 and Figure 4.7. Close to stoichiometry a very similar combustion duration is observed for all configurations using pre-chambers, but as the mixture becomes leaner, especially up to λ 1.7 there is a clear increase in combustion duration for MC-MeOH/PC-EtOH and 4. As stated previously, during MFB10-90 combustion is predominantly governed by turbulence, but laminar speed still has some impact. Such a difference is more observed when the mixture is leaner, since the laminar flame speed of the ethanol decreases more evidently with the increase of the λ , as shown in Figure 4.9 [170]. The configuration MC-MeOH/PC-MeOH, in which methanol is injected into the pre-chamber has slightly faster combustion. This is justified since when methanol is burned in the PC, the jets reach the main chamber with higher speed, leading to a slight increase in turbulence and, consequently, combustion speed. The additional turbulence generated by injecting methanol on the PC also justifies the extension of the lean limit compared to PCIS4.

Although discussed separately, the impact of the use of pre-chambers in all stages of combustion is evident, especially in the MFB10-90 period, when combustion occurs predominantly in a turbulent regime. Such an impact is summarized in Figure 4.10, which shows that the average combustion duration 46.5% lower for the engine equipped with pre-chambers close to stoichiometry and 35.3% for leaner mixtures when compared to the configuration with baseline ignition. Table 4.3 presents in detail the average MBF10-90 for each of the situations used to perform this calculation. In addition, Figure 4.10 shows a faster combustion with the use of methanol as main fuel, especially with baseline ignition.

Table 4.3 - Average MBF10-90 in CA° for PCIS and SI for several operation points

Engine Speed	Engine Load	PCIS		SI	
		Stoichiometry	Lowest lean limit	Stoichiometry	Lowest lean limit
2000 rpm	6 bar IMEP	6.6	10.7	13.1	19.3
	15 bar IMEP	6.0	9.2	12.2	16.9
3500 rpm	6 bar IMEP	7.9	12.1	13.6	17.9
	15 bar IMEP	13.6	17.9	14.1	15.5

In Figure 4.10 it is also possible to observe the in-cylinder peak pressure. Considering an analysis of several operating configurations, but with the same IMEP, it is logical to think that the condition with the shortest duration will have the highest peak pressure. According to the figure, the engine with baseline ignition presents pressure peaks from 6 to 16 bar lower than the pressure peaks presented by the PCIS configurations. It is observed that the difference between pressure peaks is greater up to 2000 rpm, which can be explained by the low initial turbulence levels present in the cylinder, making the use of pre-chambers more prominent, as previously discussed.

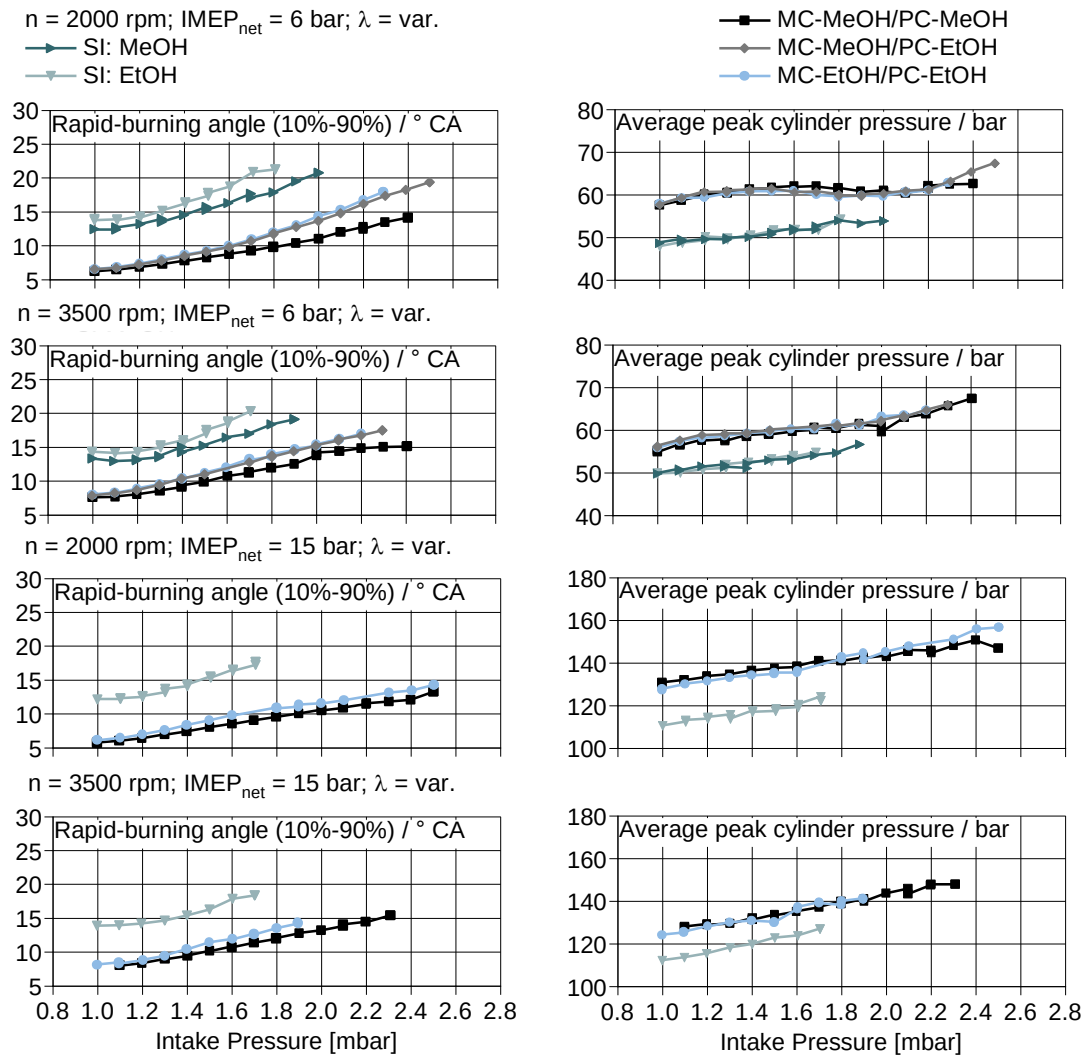


Figure 4.10 - MBF10-90 and mean peak cylinder pressure at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and MeOH)

It is also observed in Figure 4.10 that although the MFB-10-90 increases with lambda, which is normal due to the reduction of the in-cylinder temperature, the mean peak cylinder pressure also increases. This happens due to the increase in intake pressure. For IMEP maintenance with mixture enleanment, it is necessary to increase the amount of air admitted to the combustion chamber, which is only possible with the increase of the intake pressure performed by the supercharger. Therefore, as the initial in-cylinder pressure increases, it is to be expected that during combustion higher pressures are observed, as well as the pressure peak, as shown in Figure 4.11. More in-depth analyzes of in-cylinder pressure, however, are carried out in the next subsection. As a final comment regarding Figure 4.10, it should be noted that since spark-timing was different for every fuel, pressure and temperature at the moment of ignition are different and that influences combustion and its development.

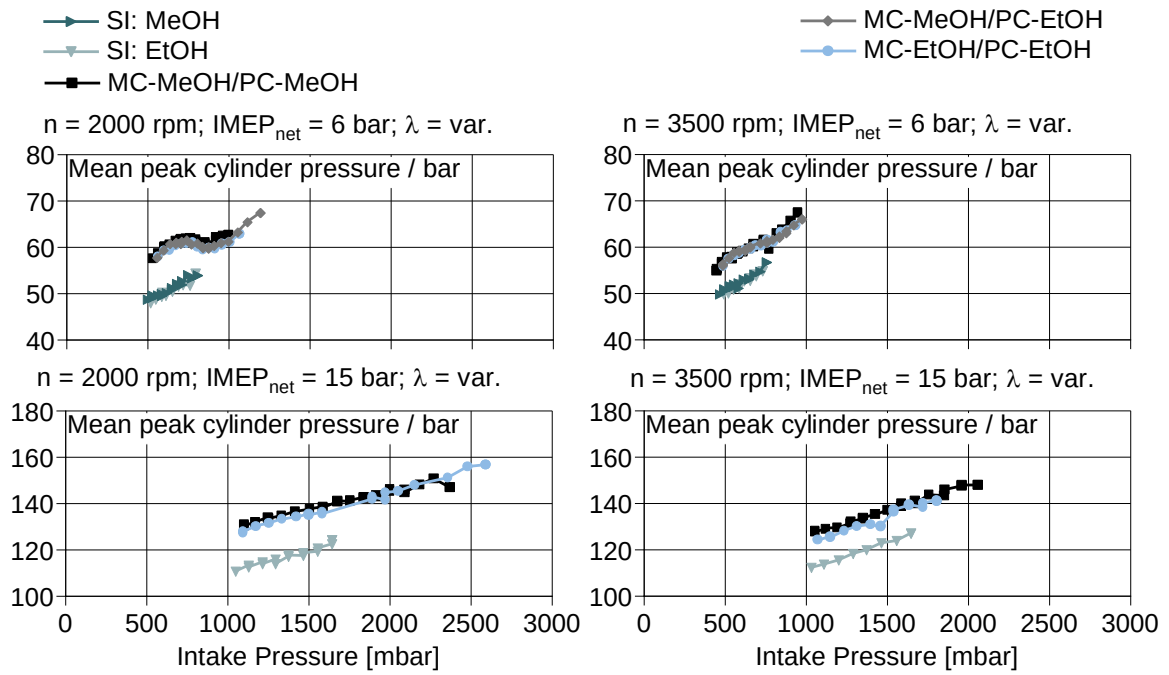


Figure 4.11 - Relation between mean peak cylinder pressure and intake pressure

Still considering the duration of combustion, another interesting analysis can be made. The ignition timing is adapted to the combustion duration in order to achieve MBT. Therefore, as the lambda increases and the burning becomes slower, an earlier ignition is needed in order for the MBF50 to occur at approximately 8° aTDC , as shown in Figure 4.12.

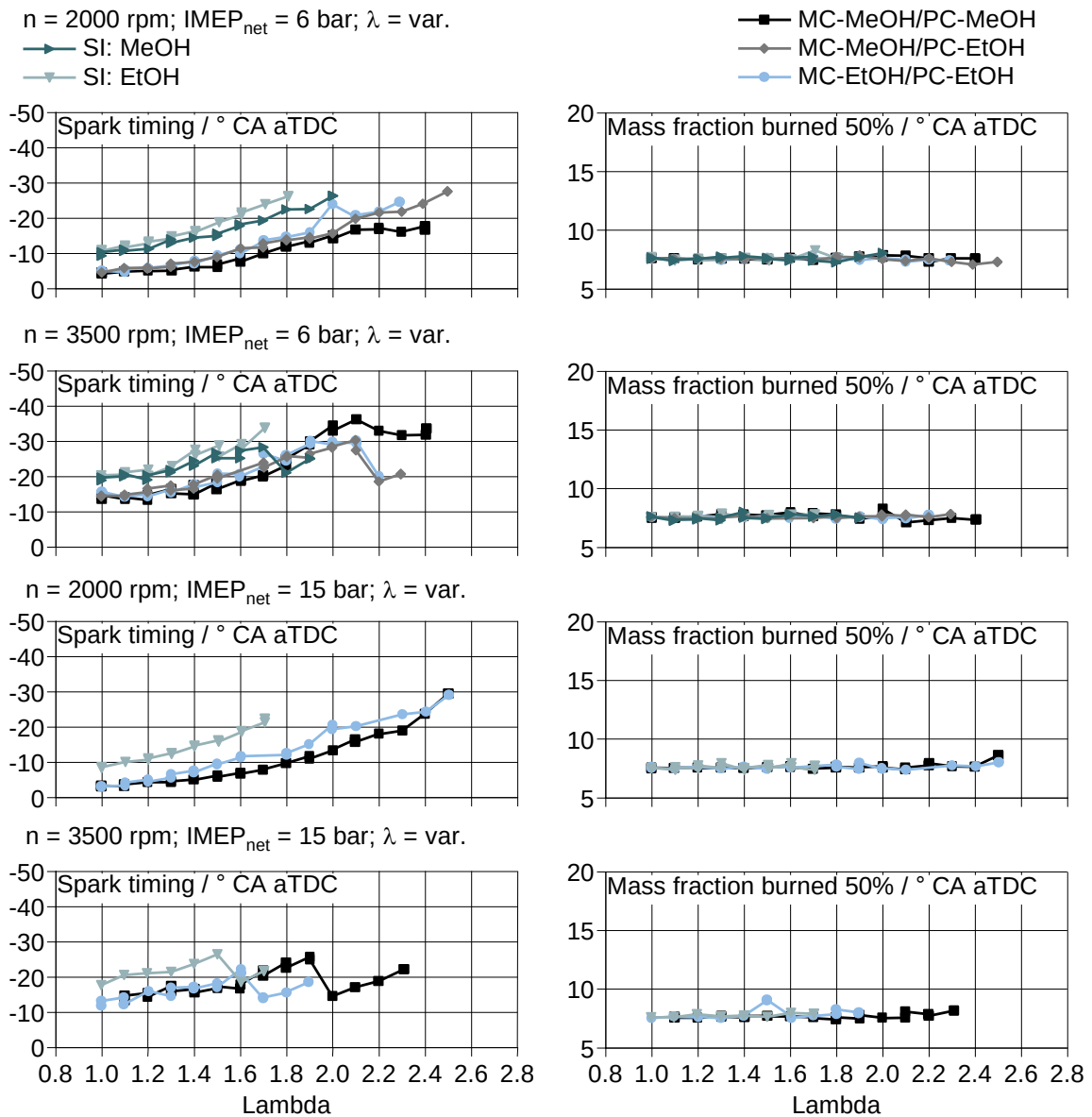


Figure 4.12 - Spark timing and MBF50 at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and MeOH

4.3 In-Cylinder Pressure and Pressure Oscillations

For general comments regarding the differences between combustion with the two ignition systems, Figure 4.13 is presented, in which the SI-EtOH and MC-EtOH/PC-EtOH (only ethanol used as fuel) configurations at 2000 rpm and 6 bar of IMEP are observed. The pressure curves analyzed are those of the most representative cycle (MRC) among the data package, determined according to the methodology proposed by Herzer et al. [174].

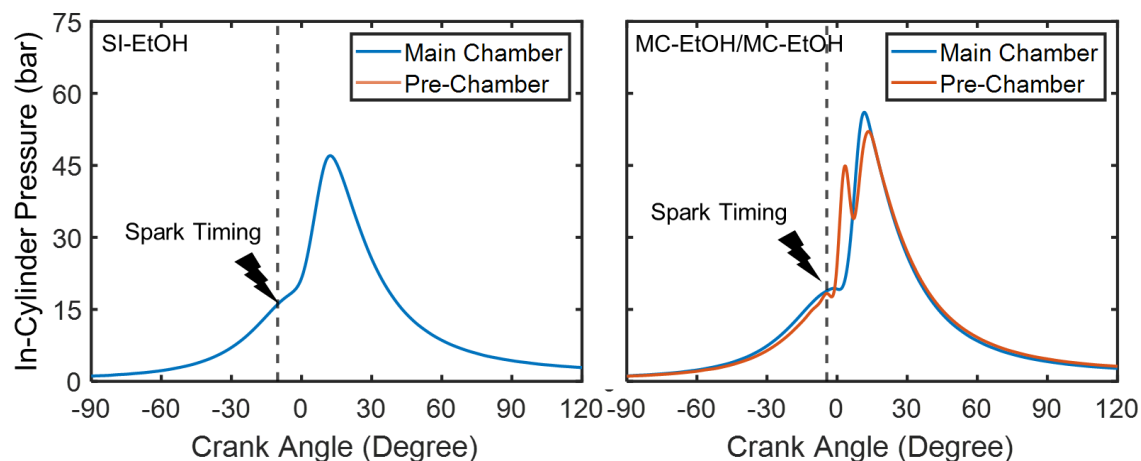


Figure 4.13 - Pressure trace for baseline and pre-chamber ignition systems at 2000 rpm and 6 bar of IMEP using ethanol

As previously discussed, a higher pressure peak is observed when operating with pre-chambers, associated with a shorter combustion duration. Furthermore, it is possible to observe pressure fluctuations in the main and pre-chamber pressure traces for the MC-EtOH/PC-EtOH configuration. Due to the longer combustion duration associated to baseline ignition, a longer ignition advance is required. Up to TDC, an in-cylinder pressure increase occurs due to the compression stroke and soon after the spark timing, the pressure quickly increased due to the start of combustion for both configurations. It should be noted, however, that the ignition delay is much higher using baseline ignition, since the ignition assisted by a pre-chamber accelerates the combustion development.

Differently than when using baseline ignition, PCIS operation is more complex, what can be better explained by observing the pressure traces in the two combustion chambers. To this explanation easier, Figure 4.14, obtained using GT-Power simulations, is also used. In the figure, the average pressure traces of the main and pre-chambers and the simulated mass flow passing through the pre-chamber interconnection holes are presented.

After ignition, while the pressure difference between the two chambers is still small and there is a large portion of the air-fuel mixture to be burned in the pre-chamber, small amounts of fluid are exchanged between them, leading to pressure oscillations, which are present throughout the process. With the rapid development of the flame front and the increase of pre-chamber pressure, the mass transfer between the two chambers begins. As a convention, the positive direction of air mass flow is from the pre-chamber to the main chamber. Therefore, with the initial increase in pressure in the pre-chamber due to combustion, a peak of mass flow towards the main chamber is observed.

As jets of intermediate combustion products with high temperatures, pressures and active radicals enter the main chamber, the mixture is ignited and combustion starts, increasing the main cylinder pressure. Soon after this pressure increase, there is an increase in the pressure difference between the chambers, leading part of the combustion products from the main chamber to expand into the pre-chamber, increasing the pre-chamber pressure for the second time. At that moment, the inverse path of the gases is observed, leaving the main chamber, going towards the pre-chamber [175].

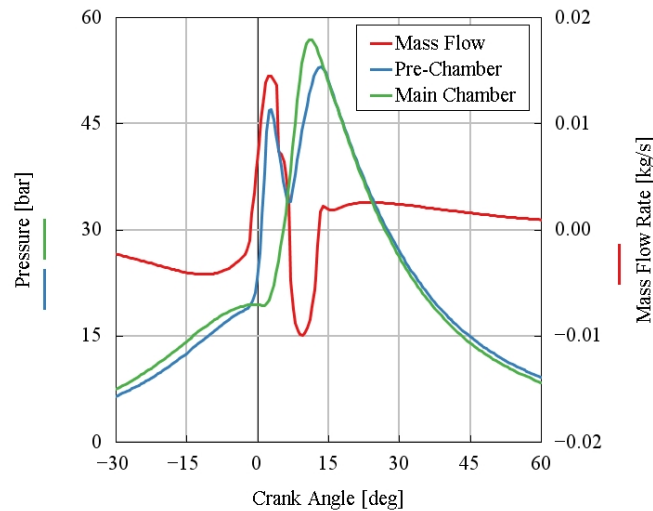


Figure 4.14 - Mass flow rate through pre-chamber nozzles during pressure variations in main and pre-chamber at 2000 rpm and 6 bar of IMEP for configuration MC-EtOH/PC-EtOH obtained using GT-Power

Once the general combustion mechanism has been discussed, the development of main and pre-chamber pressure is shown from Figure 4.15 to Figure 4.20. In the figures the in-cylinder pressure is presented at the top and at the bottom, using a high-pass filter on the pressure curve, the pressure fluctuations are shown. Three frames are used for the pressure analysis: with the stoichiometric mixture, with lambda 1.5, a point that can be compared in all cases and still slightly distant from the lean limit, and the lean limit point.

In general, an increase in the cylinder pressure is observed with mixture enleanment in both ignition systems due to intake pressure boosting, in order to maintain the same IMEP. Furthermore, as previously discussed, a higher-pressure peak is observed when operating with pre-chambers. As a rule, close to stoichiometry, the moment after ignition is more evident as a disruption in the pressure slope. This occurs because the in-cylinder pressure before combustion is lower, which makes the increase of pressure after ignition more representative. It is also worth mentioning that, during stoichiometry, the combustion duration is shorter, marked by a steeper increase in pressure. As the mixture becomes leaner, in general, since the in-cylinder pressure is already higher and the ignition must occur earlier due to the larger MBF10-90, the ignition timing is not clearly evidenced.

It is also observed that under stoichiometric conditions, the pre-chamber use causes a great impact on the pressure curve shape, being steeper and with a more prominent pressure peak. However, as the mixture is enleaned, the PCIS pressure curve starts to resemble the obtained with baseline ignition. This occurs because PCIS at this condition does not lead to additional pressure increase, but rather, just enough ignition energy for a leaner mixture, increasing the lean threshold.

Since the first pre-chamber pressure peak is related to its combustion, it is observed that as the mixture becomes leaner, this peak increases, which probably means an increase in the pre-chamber fuel injection. This is necessary because a richer mixture is capable of producing more active radicals and at higher temperatures and pressures, enabling the ignition of leaner mixtures [176].

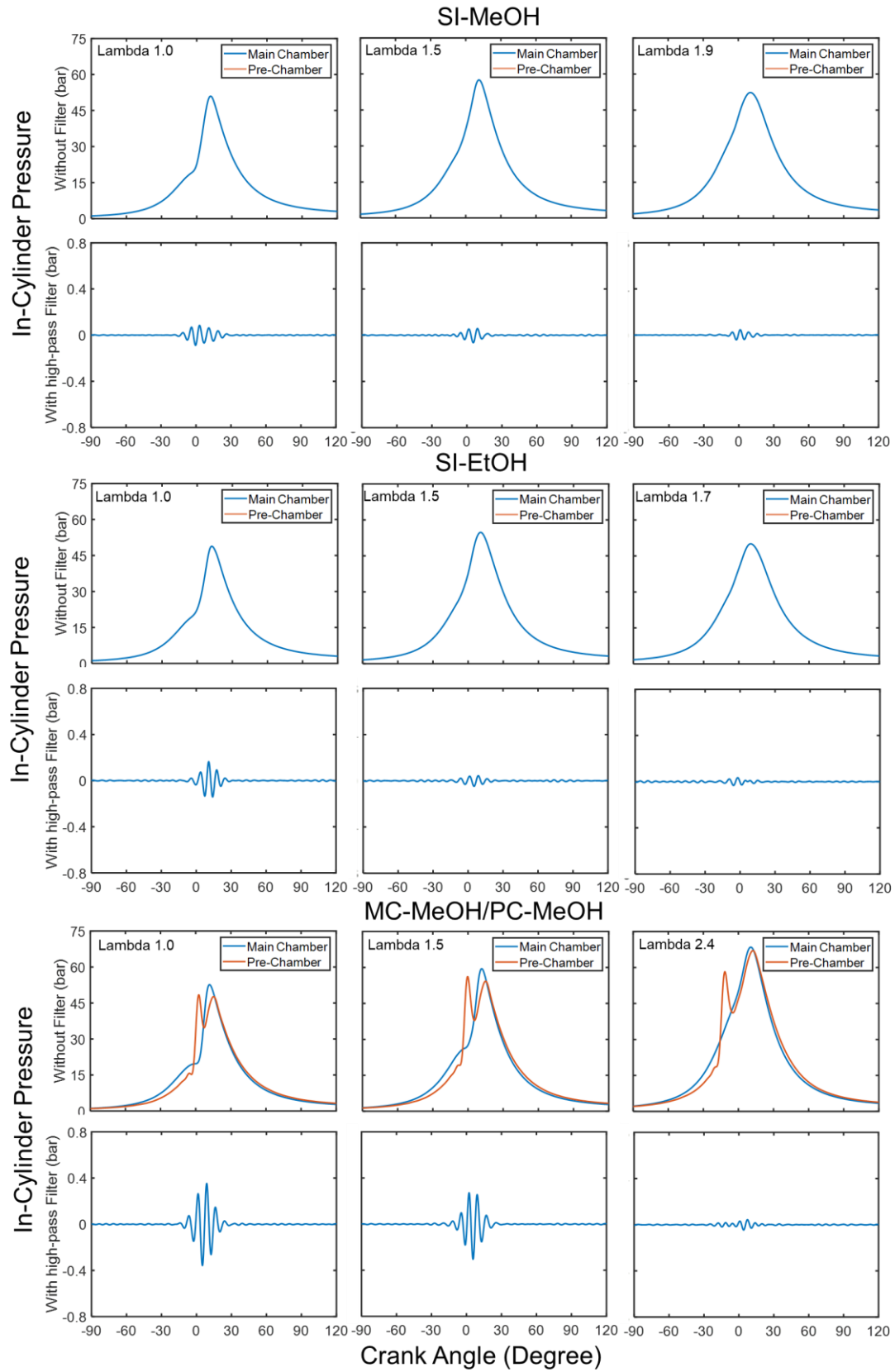


Figure 4.15 - In-Cylinder Pressure and pressure oscillations for SI-MeOH, SI-EtOH and MC-MeOH/PC-MeOH at 2000 rpm and 6 bar of IMEP

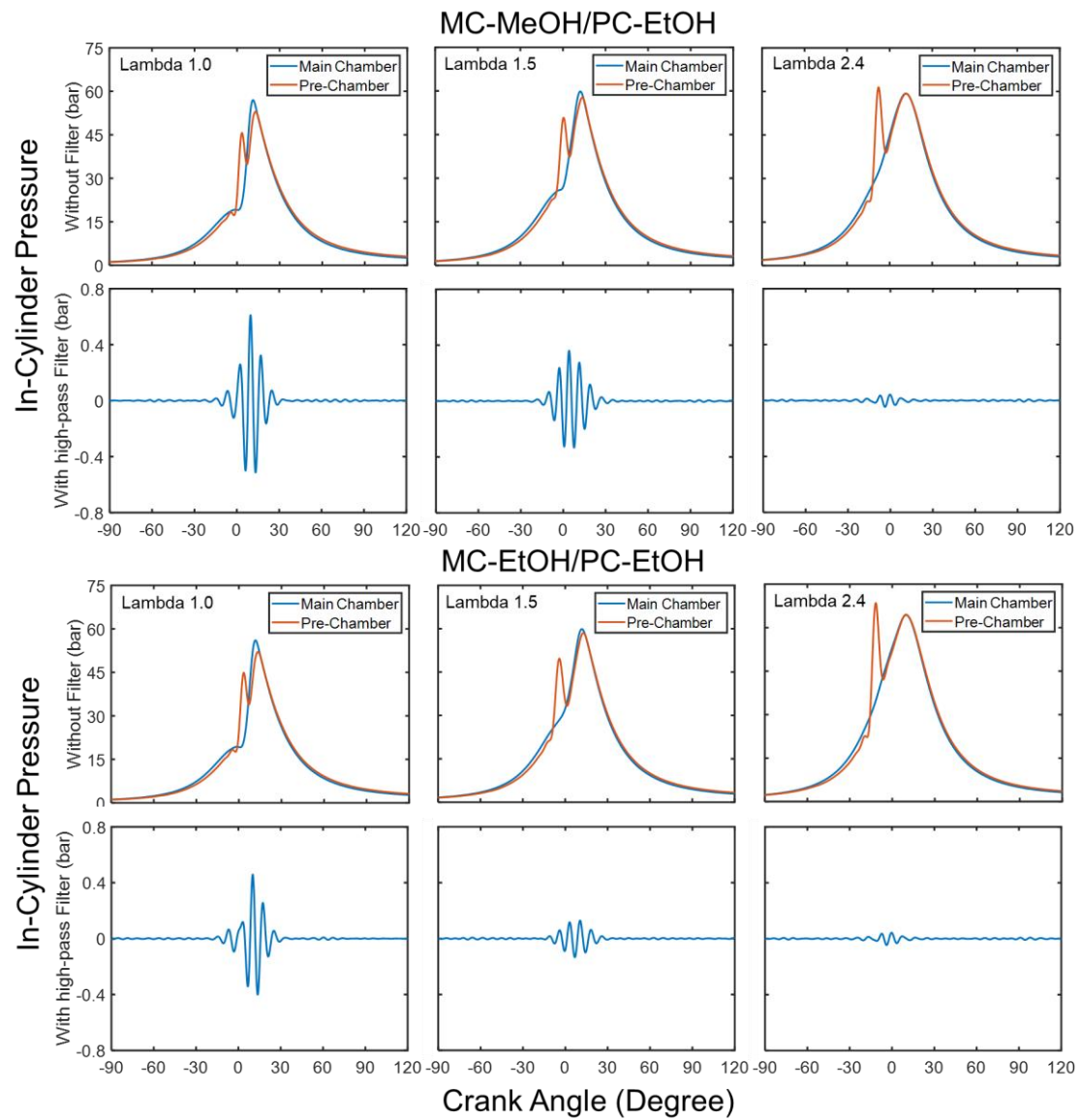


Figure 4.16 - In-Cylinder Pressure and pressure oscillations for MC-MeOH/PC-EtOH and MC-EtOH/PC-EtOH at 2000 rpm and 6 bar of IMEP

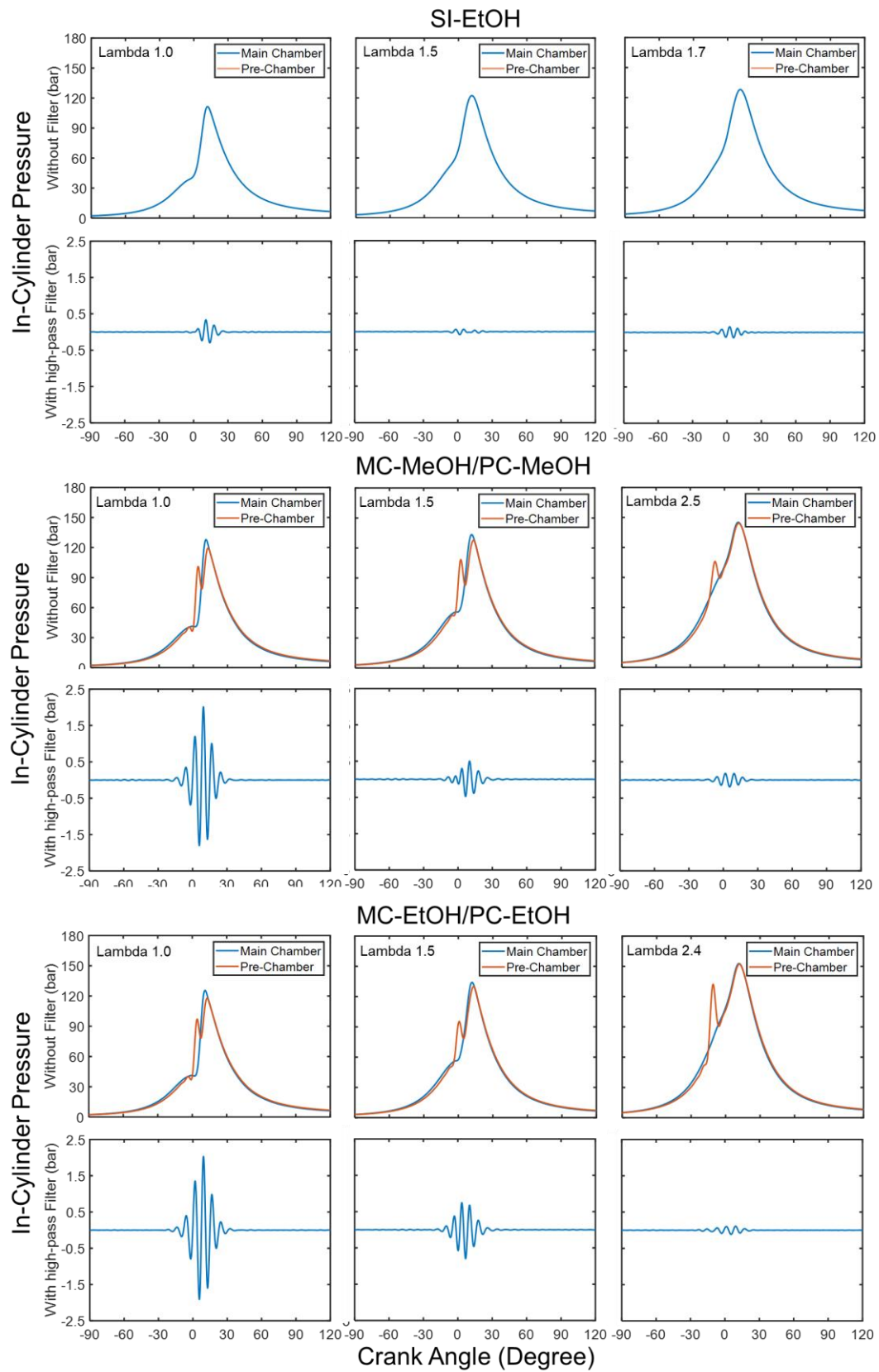


Figure 4.17 - In-Cylinder Pressure and pressure oscillations for Si-EtOH, MC-MeOH/PC-MeOH and MC-EtOH/PC-EtOH at 2000 rpm and 15 bar of IMEP

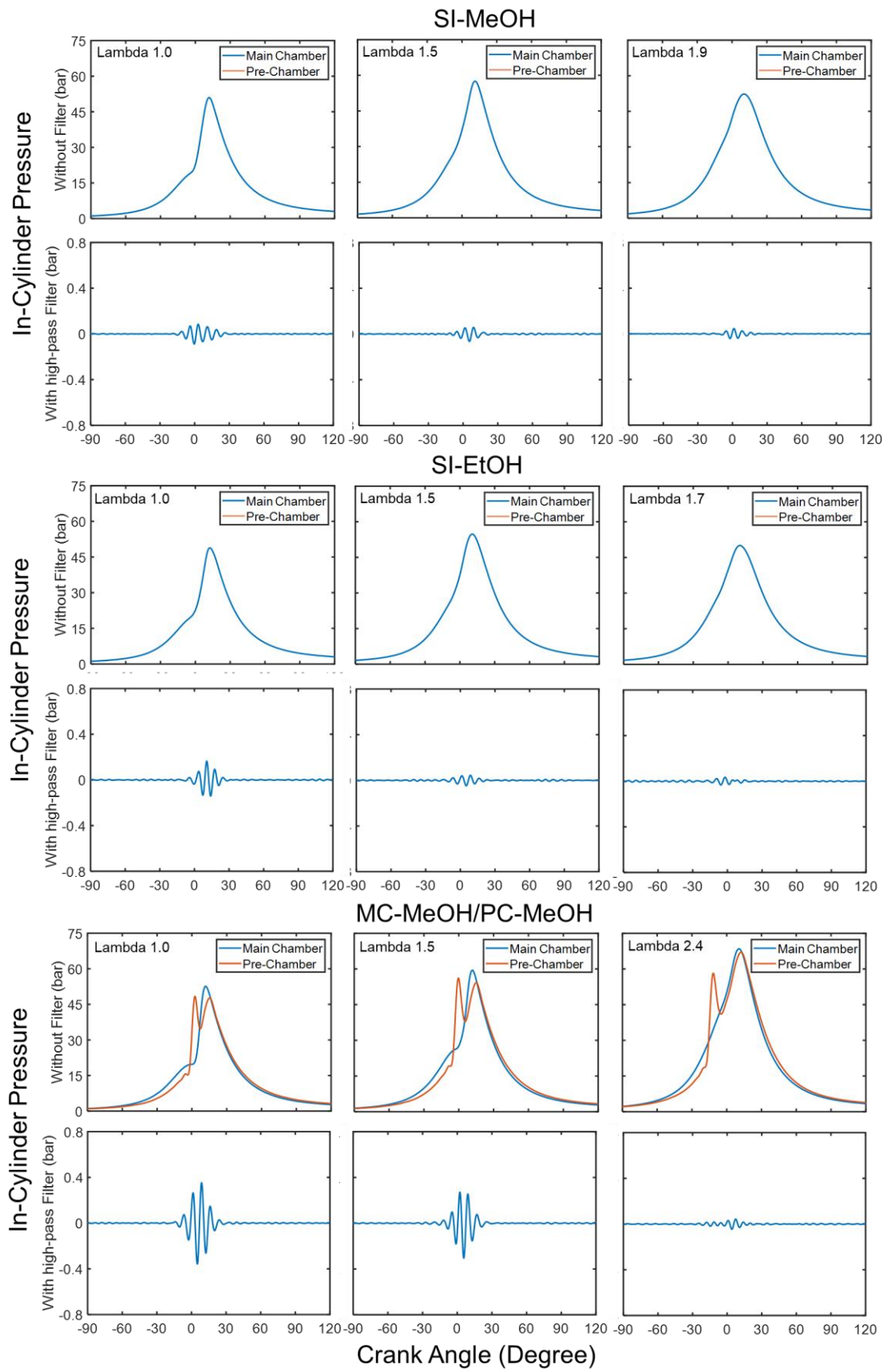


Figure 4.18 - In-Cylinder Pressure and pressure oscillations for SI-MeOH, SI-EtOH and MC-MeOH/PC-MeOH 1 at 3500 rpm and 6 bar of IMEP

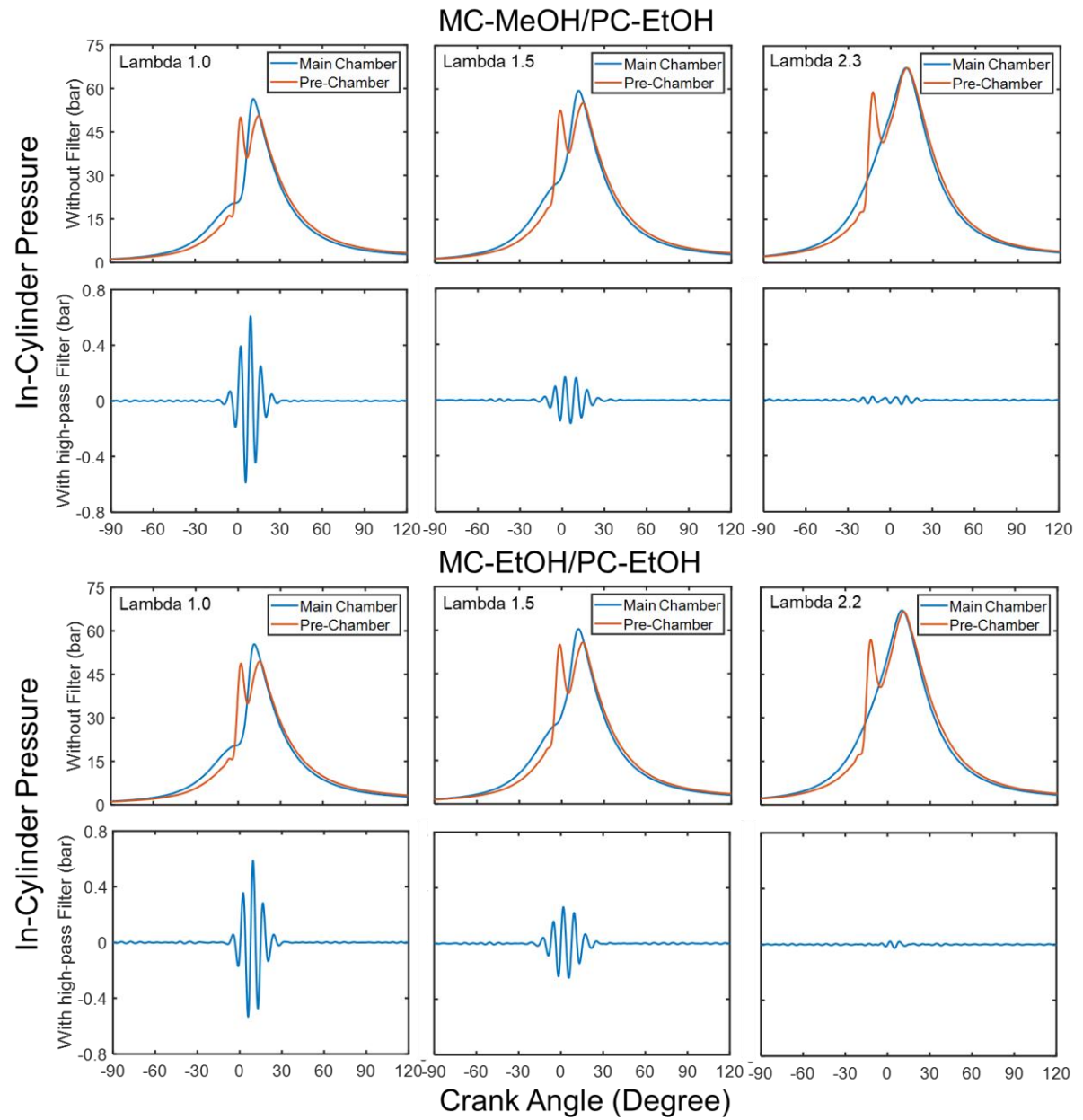


Figure 4.19 - In-Cylinder Pressure and pressure oscillations for MC-MeOH/PC-EtOH and MC-EtOH/PC-EtOH at 3500 rpm and 6 bar of IMEP

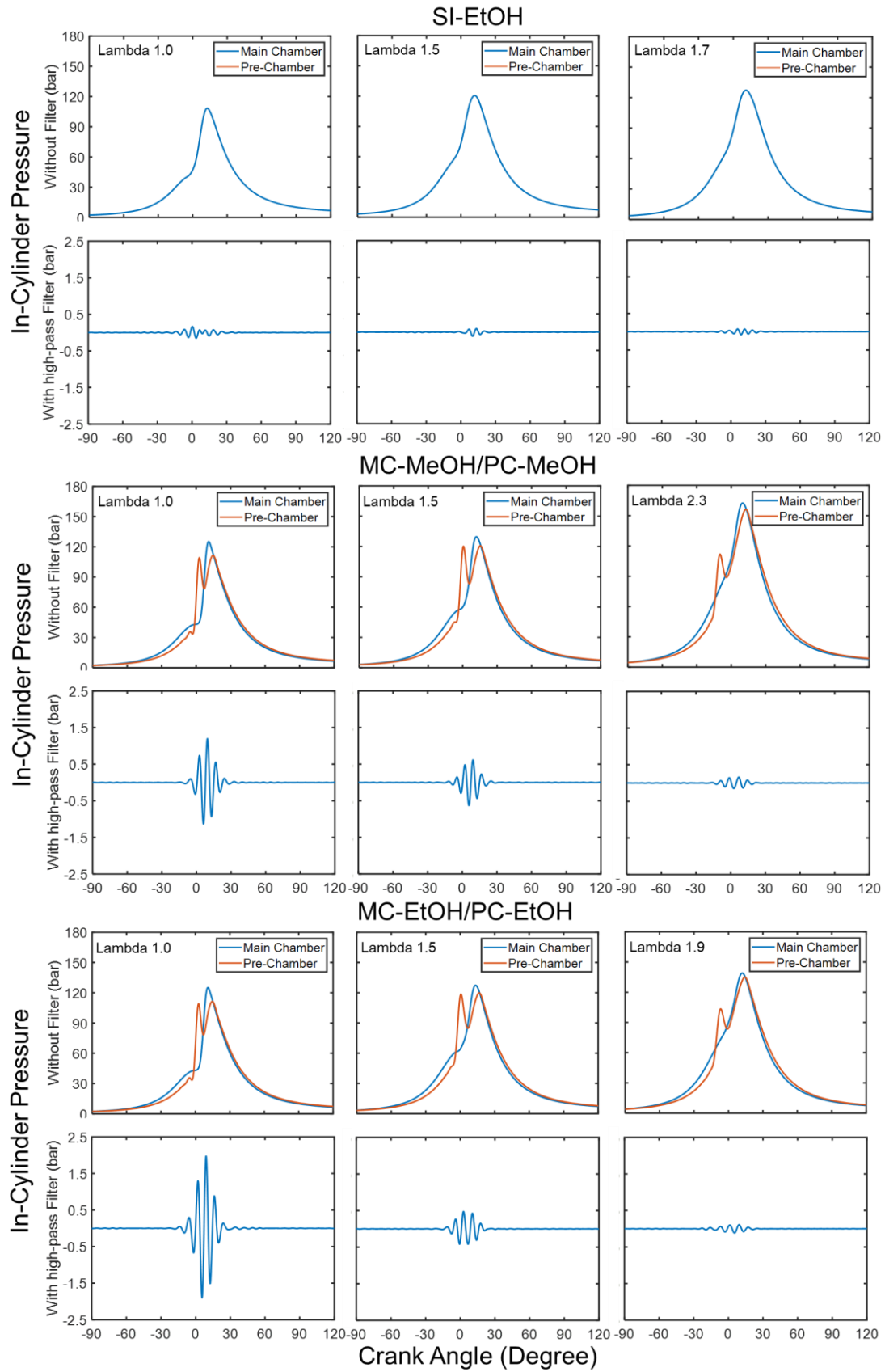


Figure 4.20 - In-Cylinder Pressure and pressure oscillations for SI-EtOH, MC-MeOH/PC-MeOH and MC-EtOH/PC-EtOH at 3500 rpm and 15 bar of IMEP

In the figures it is still interesting to observe the pressure fluctuations present in the pressure curves. It is very important to make a detailed analysis of this parameter since it can be, or be confused with, knock. The occurrence of such a phenomenon is very harmful to the engine integrity and therefore, exploring the phenomenon and ensuring its differentiation from knock is extremely important for a safe and durable operation of the engine.

Even in SI engines pressure oscillations are inherent to the combustion process. With the mixture ignition and flame front propagation, the end-gas is compressed, causing gas movement in the combustion chamber [177]. Proof of this is the fact that the amplitude of the oscillations of the SI-MeOH configuration, fueled with methanol, is greater than the amplitude observed for the SI-EtOH under the same conditions, due to the faster laminar flame speed of methanol. However, it is observed that the maximum amplitude of oscillation, or MAPO, always remains below 0.5 bar, what is a common threshold used baseline ignition knock detection [178]. These oscillations naturally decrease in amplitude as the mixture becomes leaner, since the flame propagation speed reduces with the temperature. Therefore, less pressure is applied to the end-gas, which has less movement inside the cylinder, resulting in less oscillations in the pressure curve.

However, with the use of pre-chambers, a more intense fluctuation is observed. To explain such a phenomenon, several reasons can be listed. Initially, it is observed that the pressure oscillations start as soon as the pre-chamber pressure starts to increase. Since combustion starts inside a very small combustion chamber, a pressure wave is formed inside the pre-chamber itself, which reflects and establishes a compression and refraction pressure wave that goes sinusoidally until the pressure reached is enough to pass through the pre-chamber nozzles. From this moment on, there is always a pressure imbalance between the two chambers, leading to an exchange of gases. This movement can be observed through the oscillations in the pressure traces of both chambers. Even without the presence of two chambers, the oscillations would also be justified by the highest pressure peaks. In fact, the highest pressure amplitude is always observed close to the pressure peak of both chambers. Furthermore, it should be noted that the amplitude of pressure oscillations is proportional to the in-cylinder pressure peaks, so that for 6 bar of IMEP the average oscillation amplitude with the stoichiometric mixture is 0.6 bar and 2 bar amplitude for the load of 15 bar of IMEP. Some smaller amplitudes are observed for the MC-MeOH/PC-MeOH configuration for loads of 15 bar, which is probably justified by the amount of fuel injected in the pre-chamber, which is analysed in a subsequent section.

Moreover, Attard et al.[127] points out that the main reason for the present pressure oscillations is due to the transfer of mass from the pre-chamber to the main chamber in the form of high-speed jets, generated by the relatively small interconnection hole diameter. When such jets equal or exceed sonic velocity, a hemispherical or conical shock wave is developed in the main chamber, which is detected as a pressure oscillation. To better illustrate the phenomenon, using the GT-Power software, the Mach number of the jets coming out of the pre-chamber holes is shown in Figure 4.21, Figure 4.22, Figure 4.23 and Figure 4.24.

It is observed that in all the analyzed configurations the Mach number is equal to or greater than 1 at lambda 1, indicating a supersonic gas regime, which explains the pressure fluctuations. However, it is observed that in situations of lambda 1.5 and

methanol injected into the pre-chamber, the Mach number is higher. Such an observation can lead to two conclusions. First, it can be understood that in cases in which ethanol is injected into the main chamber, the pressure oscillations occur mostly due to the supersonic regime of the pre-chamber jets. In addition, in the case of methanol injected into the pre-chambers, the pressure oscillations are less dependent to this fact and greater justified by the turbulence generated in both chambers by the movement of gases, which becomes more intense due to the higher laminar velocity of the methanol flame. From this first hypothesis, it is inferred that pressure fluctuations are a combination of several factors related to the dynamics of gases when operating with an engine with two combustion chambers.

Furthermore, as a second hypothesis, the inability of the GT-Power Software to describe in detail the jets exiting the pre-chambers should be highlighted. The software runs with a very simplified pre-chamber geometry, in which injector, spark plug positions and interconnecting holes are not properly specified. It should be highlighted that previous studies and even the studies regarding pre-chamber currently studied point out important variations in combustion from the variation of such parameters [140]. Therefore, it is possible that with simulations performed on 3D models, with more detailed pre-chamber geometry, more precise information about the jet is provided and jet speeds and even higher Mach numbers are detected.

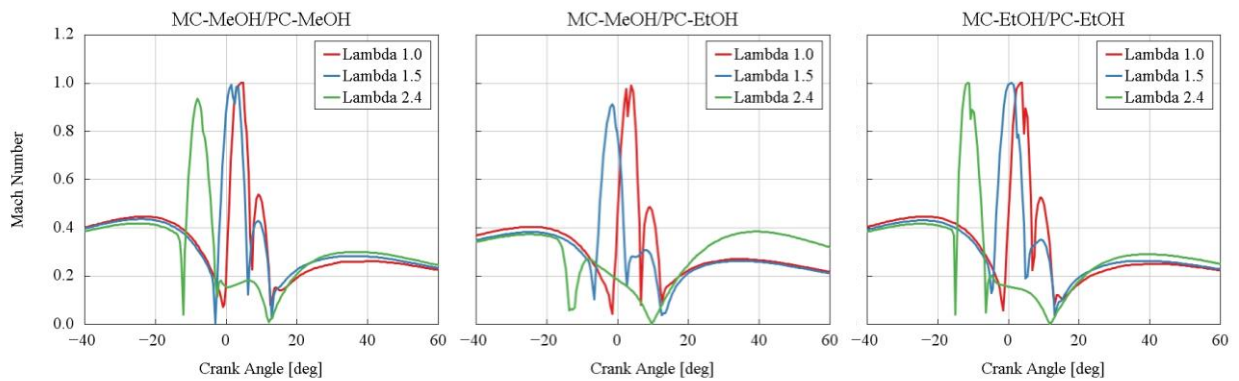


Figure 4.21 - Mach Number measured at the pre-chamber nozzles in different fuel configurations at 2000 rpm and 6 bar of IMEP

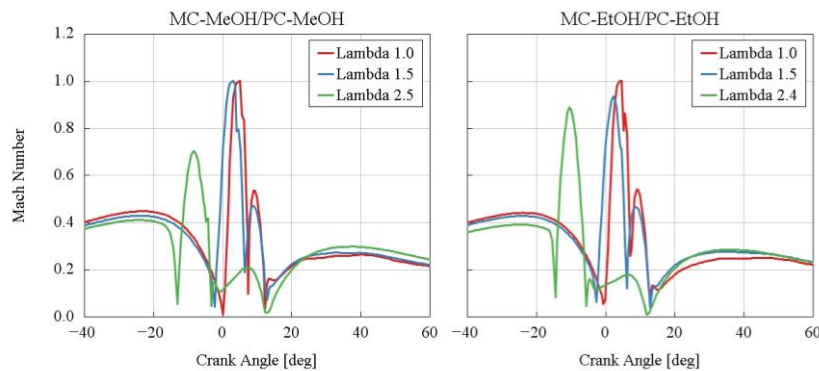


Figure 4.22 - Mach Number measured at the pre-chamber nozzles in different fuel configurations at 2000 rpm and 15 bar of IMEP

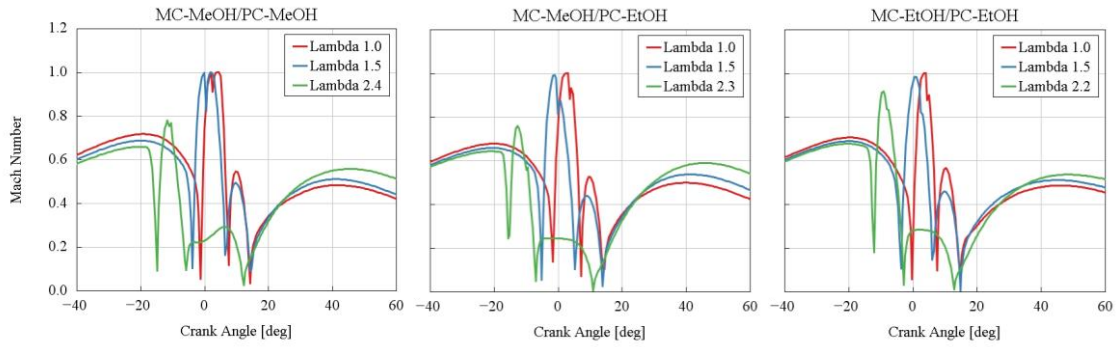


Figure 4.23 - Mach Number measured at the pre-chamber nozzles in different fuel configurations at 3600 rpm and 6 bar of IMEP

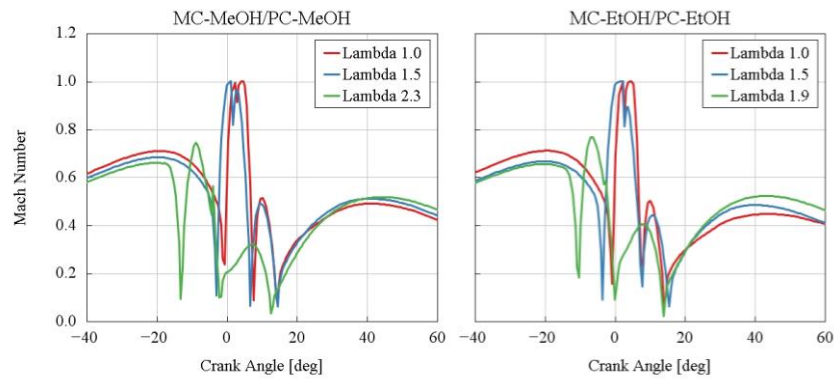


Figure 4.24 - Mach Number measured at the pre-chamber nozzles in different fuel configurations at 3600 rpm and 15 bar of IMEP

Furthermore, due to the observed pressure oscillations, the in-cylinder gas resonance modes are calculated, determined as indicated in the methodology section. According to Tanoue et al. [131], the frequency range for each knock mode was found by calculating the frequency with the adiabatic flame temperature, T_{ad} , as the upper limit and the mean of the T_{ad} and the wall temperature, T_{al} , as the lower limit. For the calculations, the wall temperature was considered 420 K since the average temperature used for temperature modeling is within the range of 410-420 K depending on the wall depth [179]. The polytropic coefficient used in the calculations was calculated using data with Matlab. Although the polytropic coefficient and the temperatures vary with the lambda, this variation is not significant for the purpose of identifying the frequency modes, which lie within an interval. Therefore, the lower and upper limits are calculated for a stoichiometric configuration of methanol and another of ethanol are used as a reference. The calculated mode frequencies for methanol are presented in Table 4.4 and in Table 4.5 for ethanol.

Table 4.4 - Mode frequencies for Methanol

i,j		(1,0)	(2,0)	(0,1)	(3,0)
B_{i,j}		1.84	3.05	3.83	4.20
F_{i,j} [kHz]	(Considering T_{al})	7.7	12.8	16.0	17.6
	(Considering T_{ad})	8.3	13.8	17.3	19.0
i,j		(4,0)	(1,1)	(5,0)	(2,1)
B_{i,j}		5.32	5.33	6.42	6.71
F_{i,j} [kHz]	(Considering T_{al})	22.2	22.3	26.8	28.0
	(Considering T_{ad})	24.1	24.1	29.0	30.3

Table 4.5 - Mode frequencies for Ethanol

i,j		(1,0)	(2,0)	(0,1)	(3,0)
B_{i,j}		1.84	3.05	3.83	4.20
F_{i,j} [kHz]	(Considering T _{al})	7.6	12.5	15.7	17.3
	(Considering T _{ad})	8.2	13.6	17.0	18.7
i,j		(4,0)	(1,1)	(5,0)	(2,1)
B_{i,j}		5.32	5.33	6.42	6.71
F_{i,j} [kHz]	(Considering T _{al})	21.8	21.9	26.4	27.6
	(Considering T _{ad})	23.6	23.7	28.5	29.8

To assess whether any of the resonance modes are being excited, fast fourier transfer (FFT) analysis is conducted over the resonance frequency band. Figure 4.25, Figure 4.26, Figure 4.27 and Figure 4.28 show the result obtained when analyzing all sets of data. As expected, the configurations with baseline ignition did not present any prominent excitation. On the other hand, the engine equipped with PCIS presented a very different behavior.

In general, an excitation close to 2 kHz is observed and, at this specific frequency, excitations occur with more intensity at 15 bar f IMEP. At first, it is possible to discard this as the engine natural frequency, since such frequencies are generally lower than any possible engine disturbances frequencies of idle, avoiding excitations leading to resonance [180]. However, it should be noted that in addition to determining the knock current, FFT analysis can also be used in the field of mechanical maintenance [181] to detect failure, such as unbalance, bearing defect, excess clearance and misalignment, among others. Since the same engine has been tested with and without pre-chambers, problems with bearings and clearances can be ruled out as they would appear in all data. In addition, misfires can be detected through engine's instrumentation. Therefore, the excitation at 2 kHz may be raising the effect of the imbalance. In addition, for the calculation of resonance frequencies, a perfect cylindrical combustion chamber is considered. However, with the pre-chamber addition, other resonance modes can be developed, such as the resonance mode due to the transfer between chambers, which can be better explored in future research.

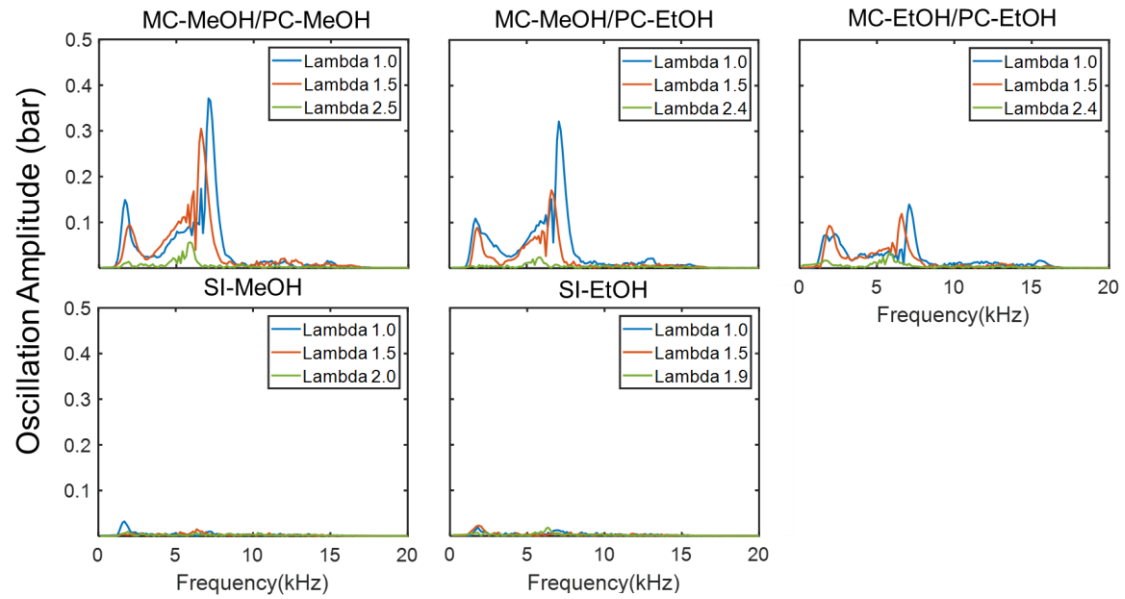


Figure 4.25 - FFT spectrum of the in-cylinder pressure acquired with the engine configurations studied at 2000 rpm and 6 bar of IMEP

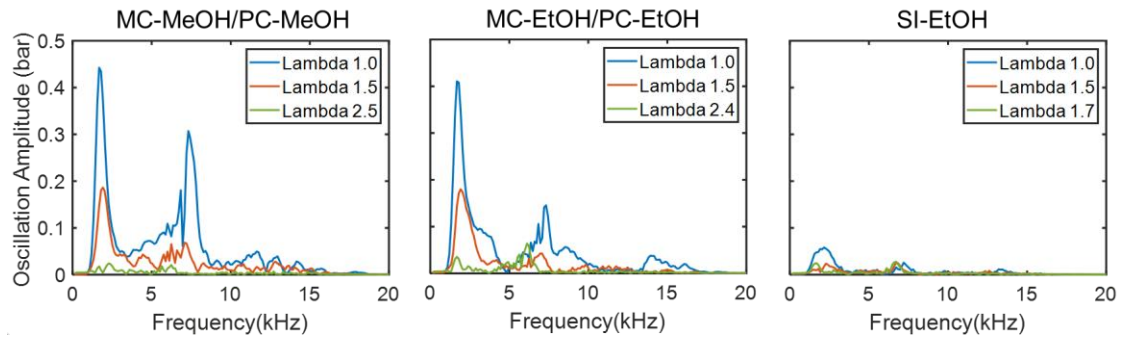


Figure 4.26 - FFT spectrum of the in-cylinder pressure acquired with the engine configurations studied at 2000 rpm and 15 bar of IMEP

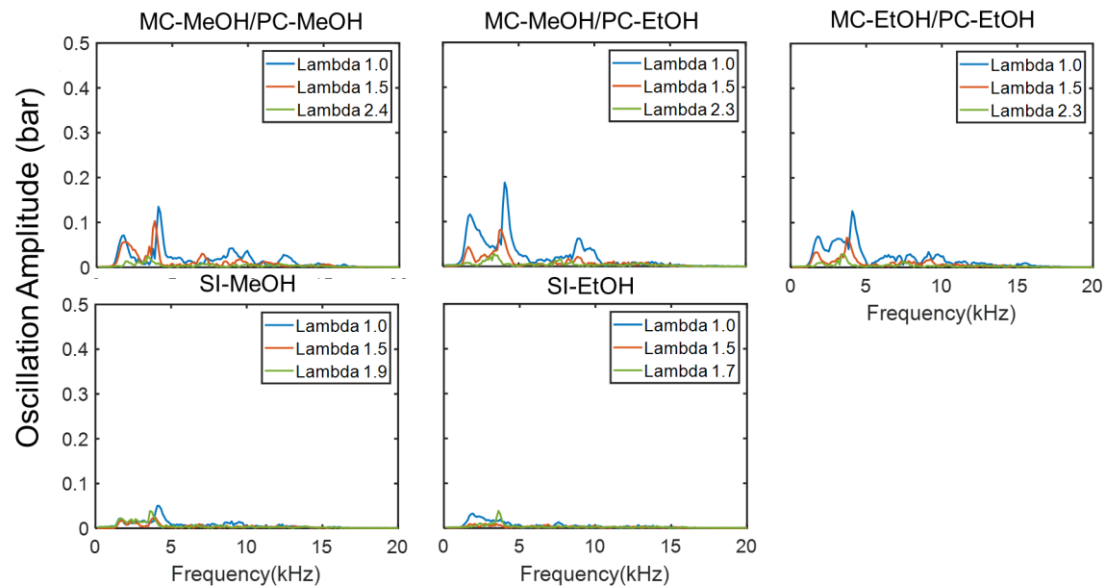


Figure 4.27 - FFT spectrum of the in-cylinder pressure acquired with the engine configurations studied at 3500 rpm and 6 bar of IMEP

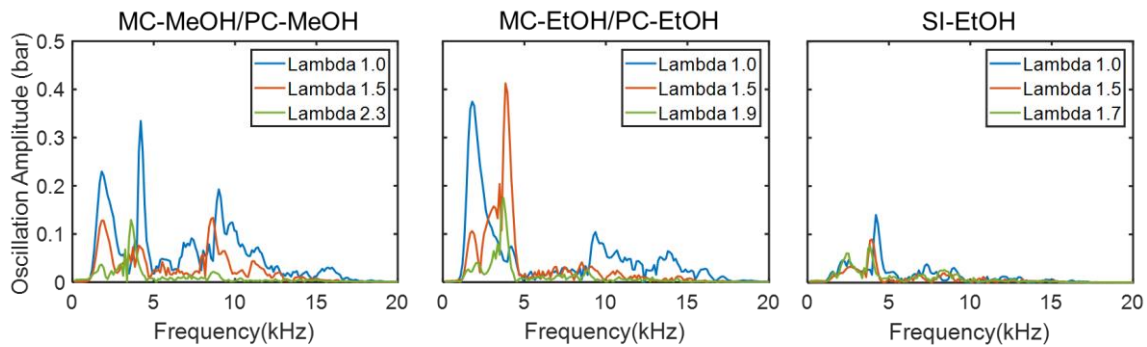


Figure 4.28 - FFT spectrum of the in-cylinder pressure acquired with the engine configurations studied at 3500 rpm and 15 bar of IMEP

Furthermore, it is observed especially for the speed of 2000 rpm excitation at a point between 7 and 8 kHz. It should be noted that an excited frequency corresponds to the oscillation mode 1,0, according to Table 4.4 and Table 4.5, which is common since the majority of the wave energy is typically centered in the lowest mode [182]. Zhang et al. [182] proposed an online knock detection method in which the average energy over resonance frequency band of mode (1,0) is calculated as the knock intensity. However, even though the first oscillation mode is excited, this excitation may be related to knock occurrence, since knock is marked for increased oscillation amplitude of several modes, not just one. Furthermore, the oscillations amplitude is significantly lower than the in-cylinder pressure peaks.

It is important, however, to point out that the first resonance mode (1,0) corresponds to resonance by means of mostly radial pressure waves, with a compression and a refraction wave, according to Figure 4.29. This is a strong indicator that pressure oscillations especially at 2000 rpm occur also due to positive pressure generated by the transfer of mass from the pre-chamber to the main chamber. This hypothesis corroborates the explanation of pressure fluctuations presented by Attard et al. [127]. Moreover, this result can also indicate the flow direction of the injectors [183, 184]. Such pressure oscillations also represent the increase of turbulence level in the combustion chamber, and with that, air fuel mixtures are able to homogenize more quickly, being burned more effectively and quickly. This fact prevents and delays the occurrence of misfire and early flame quenching, helping to extend the lean limit.

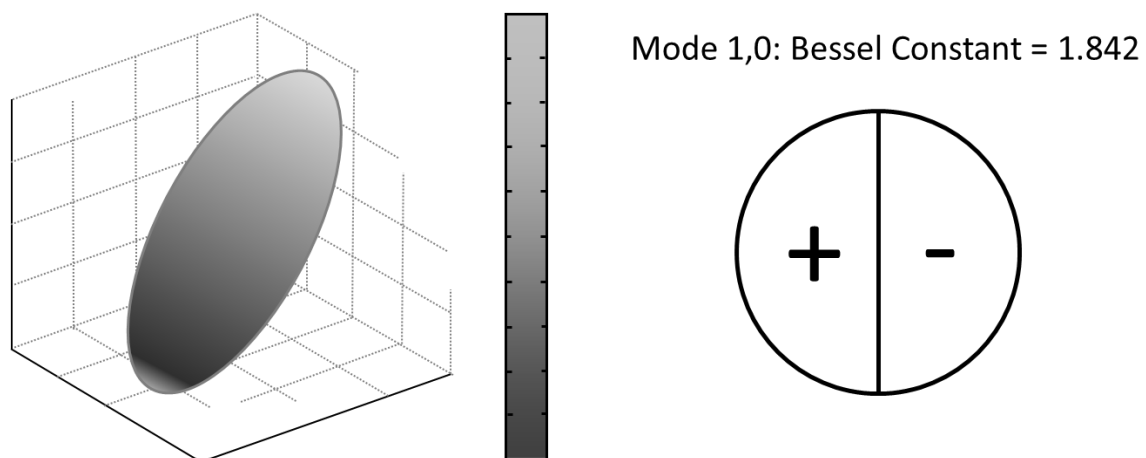


Figure 4.29 - First Oscillation Mode. Adapted from [183]

Therefore, a possible reason why, with the use of pre-chambers at a speed of 3500 rpm, the first oscillation mode was not excited could be the non-formation of additional turbulence by the pre-chambers under such conditions. This even justifies the fact that at the rotation of 2000 rpm the lean limit was further extended.

4.4 Knock Analysis

According to the discussion in the previous section, the in-cylinder pressure curves in the present study show pressure fluctuations, which present greater amplitudes than those normally observed in SI operation and may be interpreted as knock. However, as the operation with pre-chambers differs from the operation with baseline ignition in several aspects and a certain fluctuation in the pressure curves is expected both by the higher pressure peaks and by the gas transition between two chambers, the knock detection must be carried out more specifically.

It is known that the greater the flame propagation speed, the greater the ability to absorb auto-ignition points before they develop into a new flame, occasionally leading to knock. Thus, it is expected that knock is increasingly suppressed in conditions of greater turbulence, and consequently shorter combustion duration, what is achieved in pre-chamber operation. However, the higher combustion speed enabled by the pre-chambers does not necessarily mean total knock mitigation and the typical pressure fluctuations of pre-chamber operation, especially at stoichiometry, can mask knock occurrence.

Liu et al. [177] points out that the flame propagation speed is one of the main factors that affect the knock intensity, since the greater the flame front speed, the greater its ability to compress the end-gas to higher temperature and pressure. Thus, knock should be observed in pre-chamber operation at higher intensities, and consequently, the threshold used to ensure engine integrity must be higher, not to compromise engine calibration with false knock detection.

Thus, knock is investigated in the present study through two parameters, MAPO Threshold and KI2, which are capable of determining knock thresholds suitable for each type of engine configuration, as explained in the methodology section. The results of the two methodologies are presented in Figure 4.30.

The threshold established in the figure is determined according to the methodology of Itawa et al. [159] and the plots used to obtain the results are presented in Appendix C. The thresholds determined show conformity with what is already expected in the literature. The difference between the baseline and pre-chamber ignition configurations is clearly observed, especially considering the operation close to stoichiometry, in which the combustion speed is higher when using pre-chamber. There is also a higher threshold for the engine operating only with methanol, explained by the higher laminar flame speed of the fuel. As the mixture becomes leaner, the turbulence of the jet leaving the pre-chamber is not enough to maintain the main chamber burn duration, at the same time not intensively compression the end-gas. Therefore, the mixture is still susceptible to knock, but at lower pressures, requiring a lower MAPO threshold. In addition, it is generally observed that the MAPO knock threshold increases with the load (from 6 to 15 bar of IMEP), since the amplitude of the pressure fluctuations increases with in-cylinder pressure peaks.

Despite presenting coherent data, the MAPO Threshold methodology can lead to knock overestimation, and it even indicates the knock occurrence in almost all series of data collected from the engine for operation with and without PCIS, what is observed in Appendix D in the data above the threshold determined. A great evidence that no knock was observed in the present study is the fact that at all operating points and for all configurations tested it was possible to maintain the MBF50 point at 8°C_A. However, even without the presence of knock, pressure fluctuations hinder knock detection. It is important to determine a suitable knock detection parameter for such and so that parameters determine to control engine operation with baseline ignition are not mistakenly used. With adequate knock thresholds, engines equipped with pre-chambers could be able to operate safely, better exploiting their operating limits. Hence, the parameter KI2 is also analyzed [160].

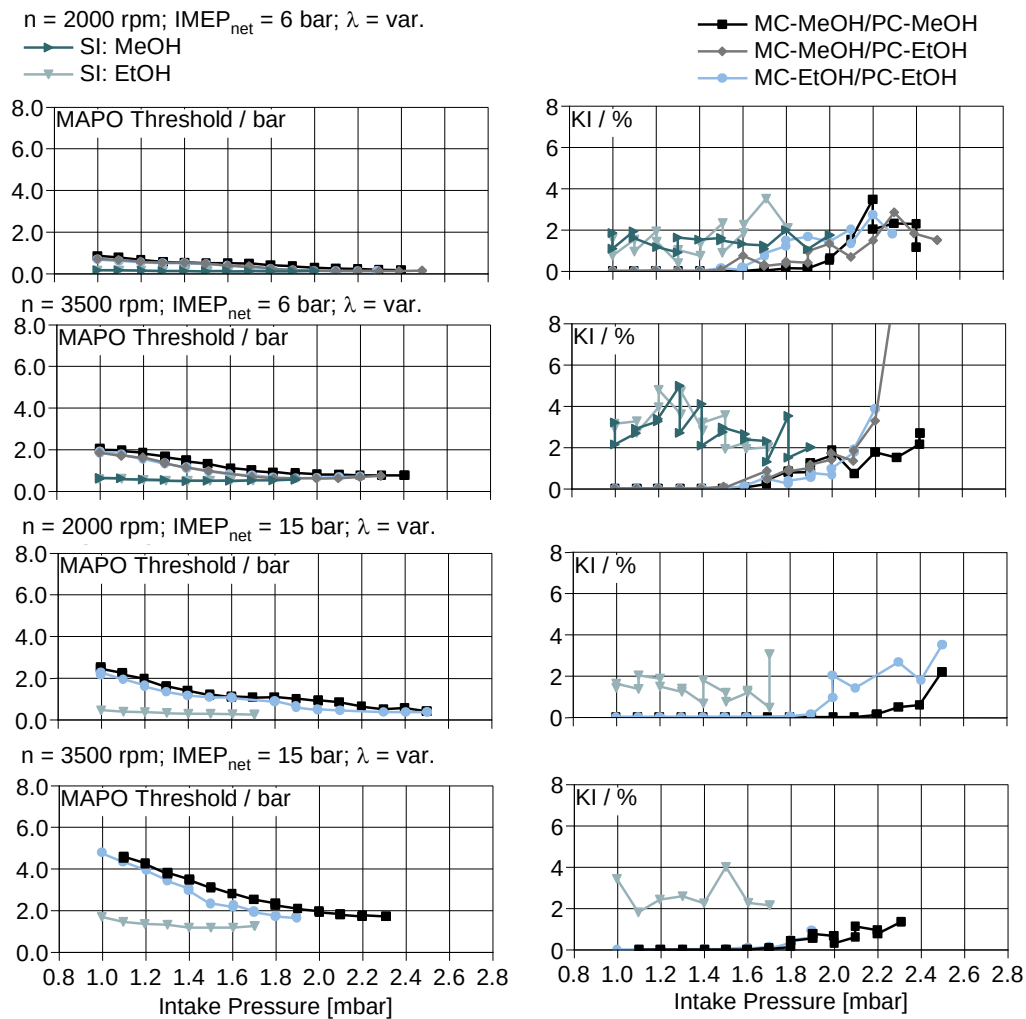


Figure 4.30 - MAPO Threshold and KI2 at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

The KI2 parameter is based on a comparison between the $MAPO_{th}$, determined before the maximum cylinder pressure, and the $MAPO_{end}$, determined after the maximum pressure, as better explained in the methodology section. The method considers that if the $MAPO_{end}$ is greater than $MAPO_{th}$, significant pressure oscillations may have occurred only after peak pressures. This may indicate that the end-gas was compressed to the point of generating auto-ignition points that developed, leading to knock occurrence [162]. Appendix C contains plots showing the $MAPO_{th}$, in black and the $MAPO_{end}$, in blue. The

greater the $MAPO_{end}$ in relation to the $MAPO_{th}$, the greater the knock intensity, represented in red in the graphics.

The parameter KI analyzed in Figure 4.30 is an average of the knock intensity in each of the 200 cycles acquired in each case in Appendix D. A large difference in behavior of the configurations with and without pre-chamber close to stoichiometry is initially observed, and also a similarity of the data in the baseline configuration with the pre-chamber close to the lean limit.

Analyzing the pre-chamber operation no knock is observed close to stoichiometry, which is in accordance with the hypothesis that technologies that provide higher combustion speed are more capable to suppress knock. The fact also agrees with the knock overestimation when using MAPO thresholds. However, for diluted mixtures, a very large reduction in MAPO is generally observed, as also observed by MAPO threshold. Hence, the oscillation amplitude is so small that often the amplitude after the pressure peak is slightly greater, leading to the false interpretation of knock after lambda 1.4. The same also happens while analyzing almost all SI data, justifying why the knock intensity appears to be much higher close to stoichiometry.

Hence, the KI2 method proves to be effective for being able to make more direct knock analysis for PCIS configurations at stoichiometry, which can be conducted directly in the pressure acquisition and interpretation system. However, for SI and PCIS operation close to the lean limit, data should be analyzed with conventional knock detection methods to avoid false knock prediction. Thus, KI2 should be used closer to stoichiometry, where pressure oscillations are more evident and damaging.

4.5 In-Cylinder Temperature, Engine Efficiencies and Indicated Specific Fuel Consumption

In the previous subsections, the amount of fuel injected into the main and pre-chambers was listed as a possible reason for some observed anomalous behavior. However, to perform the fuel injection and consumption analysis, the engine thermal analysis must be previously analyzed. For this, the temperature curves obtained through engine's operation, shown in Figure 4.31, Figure 4.32, Figure 4.33 and Figure 4.34, are analyzed. Such figures were obtained by the GT-Power software.

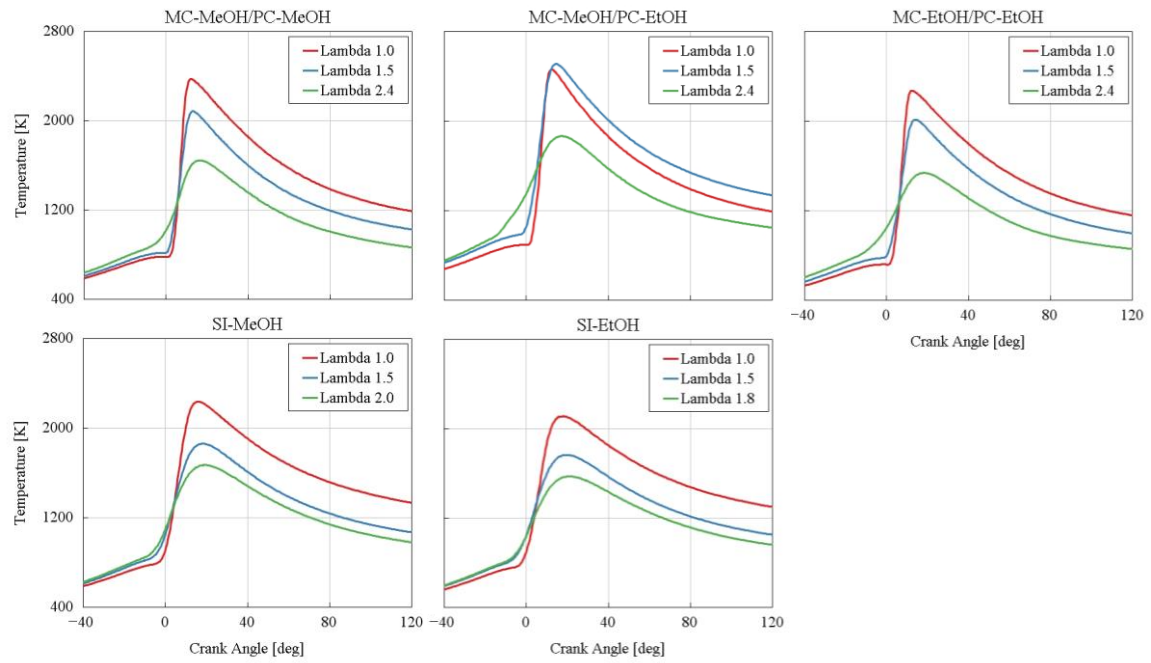


Figure 4.31 - In-Cylinder temperature for several combinations of ethanol (EtOH) and methanol (MeOH) at lambda 1, 1.5 and lean limit at 6 bar of IMEP and 2000 rpm

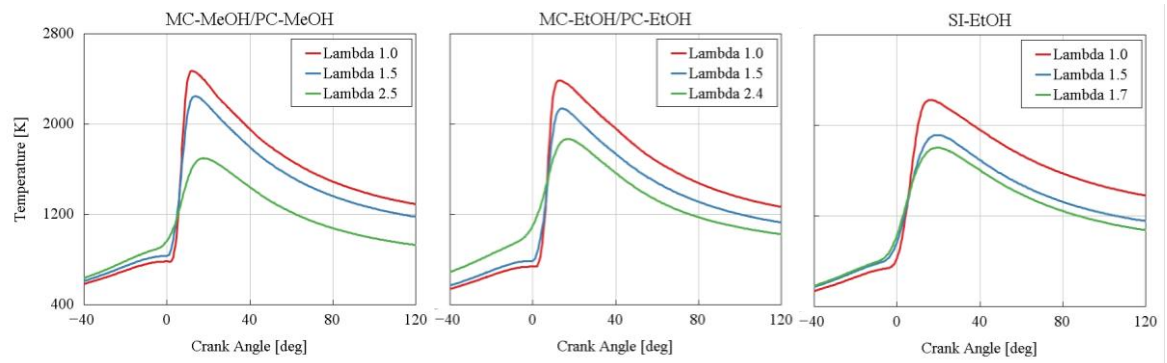


Figure 4.32 - In-Cylinder temperature for several combinations of ethanol (EtOH) and methanol (MeOH) at lambda 1, 1.5 and lean limit at 15 bar of IMEP and 2000 rpm

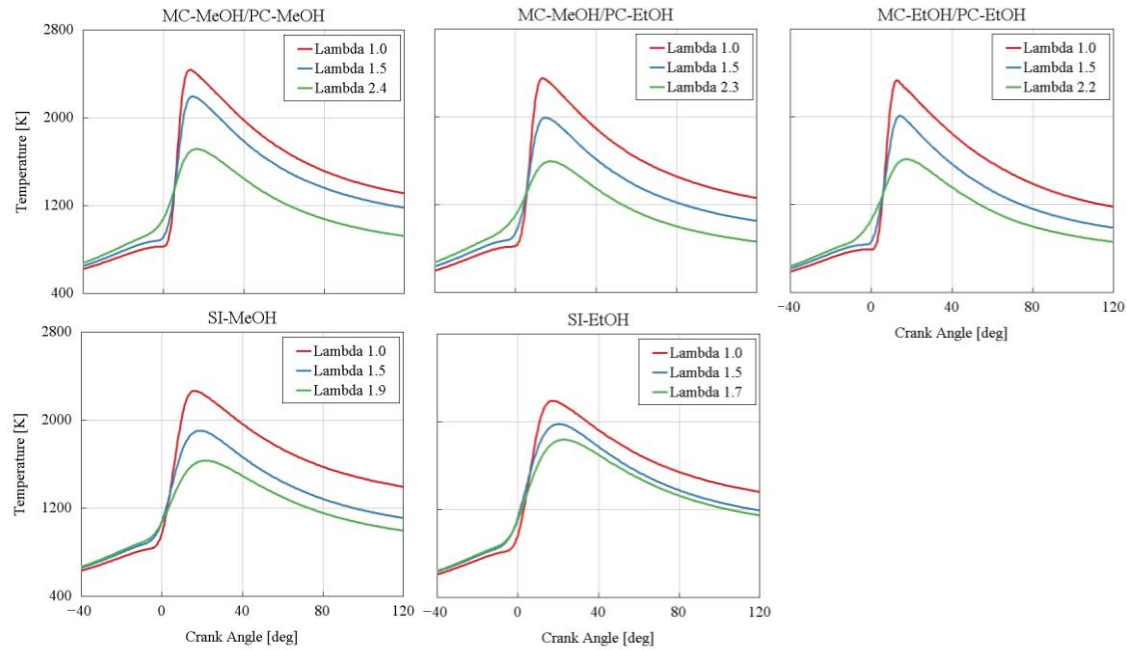


Figure 4.33 - In-Cylinder temperature for several combinations of ethanol (EtOH) and methanol (MeOH) at lambda 1, 1.5 and lean limit at 6 bar of IMEP and 3500 rpm

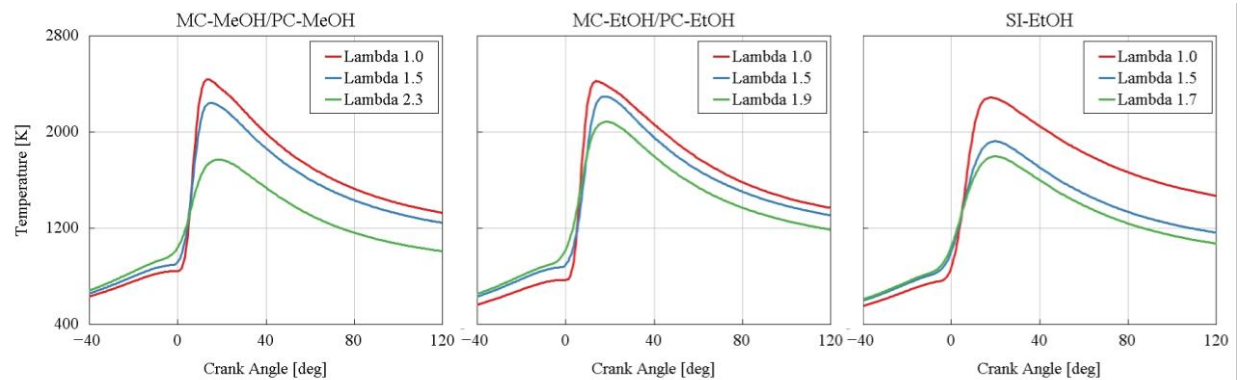


Figure 4.34 - In-Cylinder temperature for several combinations of ethanol (EtOH) and methanol (MeOH) at lambda 1, 1.5 and lean limit at 15 bar of IMEP and 3500 rpm

In the figures some interesting points can be observed. Initially, it should be noted with mixture dilution, the in-cylinder temperature is reduced for all conditions. It is also observed that with pre-chamber operation, lower temperature peaks are reached when compared to configurations with baseline ignition, since configurations with pre-chambers, in general, are able to further extend the lean limit, causing the temperature to drop. This factor is very important both for the control of thermal losses and NOx formation. However, it should be noted that pre-chamber operation is associated with high pressure peaks and such peaks lead to high temperature peaks, especially in operation close to stoichiometry. At 2000 rpm and 6 bar IMEP, it is observed that even though SI-MeOH and SI-EtOH present very similar mean peak pressure, the lowest temperatures are observed with the SI-MeOH operation, in which methanol is used, due to its higher latent heat of vaporization. However, such a trend is not observed for baseline ignition at 3500 rpm and 6 bar of IMEP and in general for MC-MeOH/PC-MeOH, MC-MeOH/PC-EtOH, MC-EtOH/PC-EtOH, in which the configuration operating with methanol reaches the highest temperature peak. This is explained by the higher flame speed of methanol fuel associated with a higher engine speed or combustion with pre-chambers, which increases the in-cylinder temperature. This causes an amount of fuel to

be burned in less time, releasing a larger amount of energy, increased in-cylinder temperature. Therefore, even though the average in-cylinder temperature can be lower due to the greater heat absorption capacity of the fuel, in a short period of time the combustion chamber reaches much higher temperatures. Such a situation is not necessarily positive as it can lead to peaks of increased thermal losses, as well as a greater NO_x formation.

The analysis of in-cylinder temperatures, however, is not enough for the thermal efficiency analysis since it also depends on the combustion efficiency. Thus, a fuel injection analysis will be conducted in parallel. Figure 4.35 and Figure 4.36 represent the mass amount of fuel injected into each of the chambers and the percentage of fuel injected into the pre-chamber in relation to the amount injected into the main chamber.

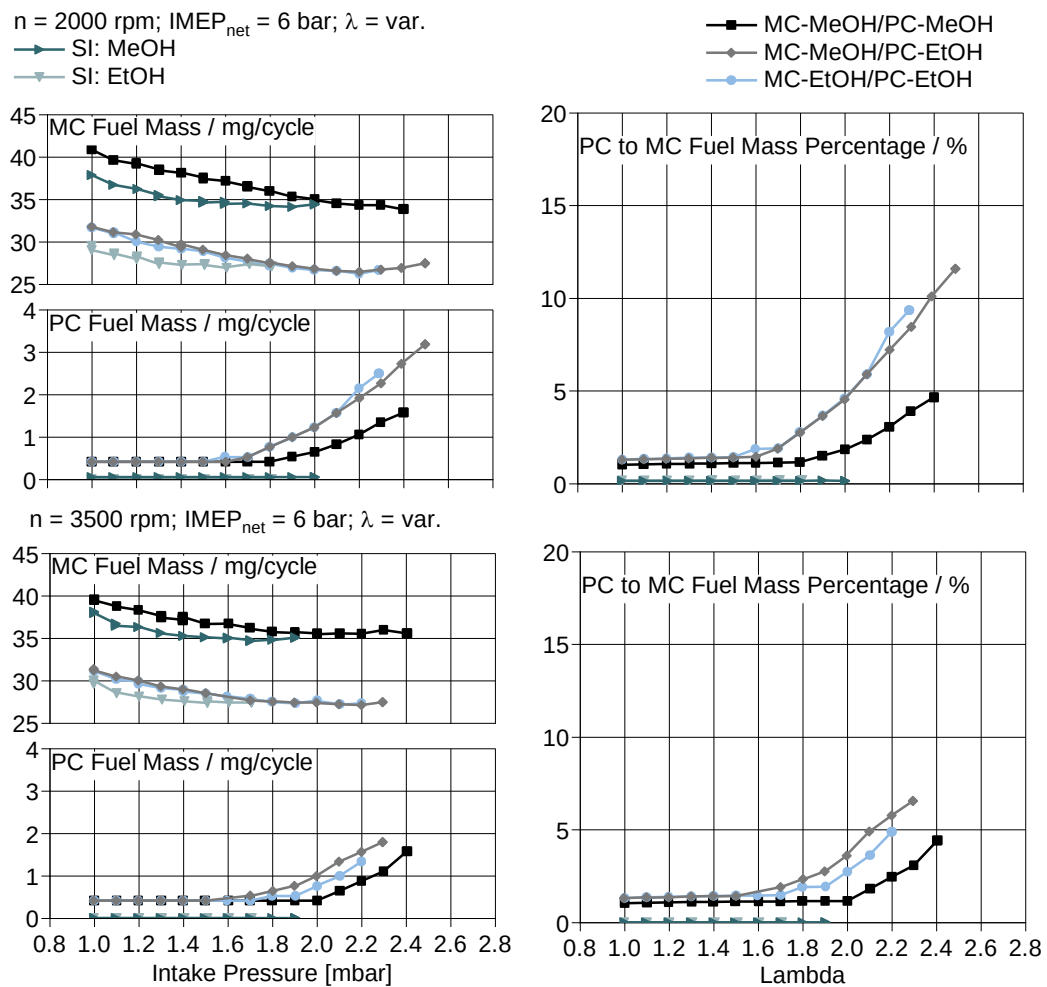


Figure 4.35 - Mass fuel injection at the main and pre-chambers and the percentage of pre- to main chamber injected fuel at 2000 and 3500 rpm and 6 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

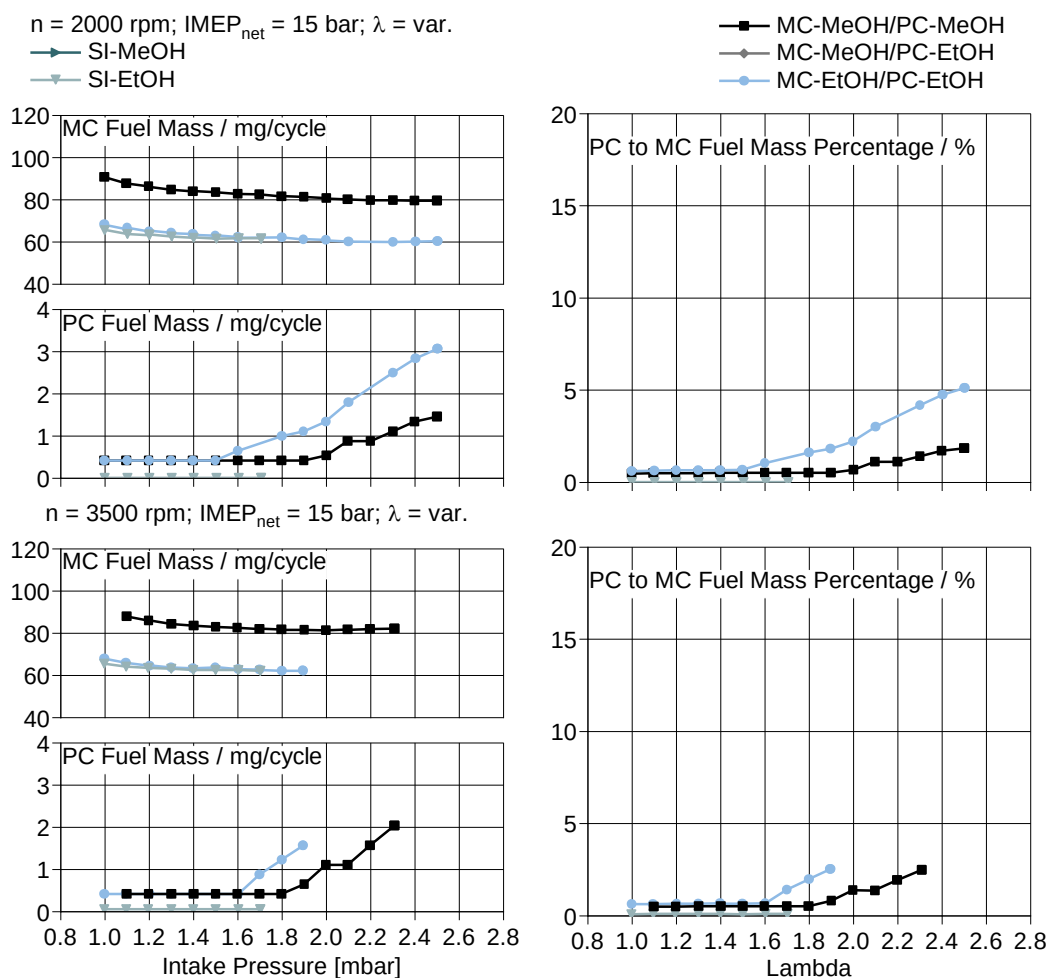


Figure 4.36 - Mass fuel injection at the main and pre-chambers and the percentage of pre- to main chamber injected fuel at 2000 and 3500 rpm and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

Initially, considering the main fuel injection, it is observed that despite load maintenance, in all configurations and operating points, as the mixture becomes leaner, less fuel is injected. Such behavior is justified by the increase in thermal efficiency due to the in-cylinder temperature reduction, which reduce thermal losses and is discussed hereafter. Also due to thermal losses, it is observed that for the same main fuel, less fuel is injected with baseline ignition. Such differences are reduced and even leveled out when the mixture leans and the heat losses decrease. It should be noted that for the SI-MeOH, MC-MeOH/PC-MeOH, and MC-EtOH/PC-EtOH configurations, the fuel injection in the main chamber is much higher, since the main fuel is methanol, with a lower LHV.

Table 4.7 shows the average percentage of fuel injected into the pre-chamber under conditions close to stoichiometry (until lambda 1.5). In general it is observed that the amount of pre-chamber fuel required for the load of 6 bar of IMEP is higher at all conditions and configurations. This fact is explained by the lower temperature in the combustion chamber for such loads, which requires more energy to ignite the mixture. Such pattern is also observed when the mixture is leaner and proportionally more fuel is required in such conditions.

Despite the lower LHV of methanol when compared to ethanol, the amount of methanol injected into the PC is approximately 30% lower for methanol relative to ethanol in stoichiometry, a ratio that increases near the lean limit. The reduced amount of methanol

injected into the PC causes the first peak pressure in the pre-chamber to be lower in these cases, as shown in Figure 4.37. However, due to the high laminar flame speed of methanol, the jets coming out of the pre-chamber are enough to ignite the mixture in the main chamber and cause the pressure curve oscillations to have greater amplitudes than when using ethanol. Therefore, it can be concluded that the amount of required PC fuel to stabilize main chamber combustion, and consequently COV_{IMEP} , depends on its laminar flame speed. This is an important fact, because when less fuel is injected into the pre-chamber, a lower fluid film is formed on pre-chamber walls, controlling consequently, the formation of unburned hydrocarbon emissions.

Table 4.6 - Heat Losses in the Main and Pre-Chambers

	Engine Speed (rpm)	IMEP (bar)	Main Chamber		Pre-Chamber	
			In-Cylinder heat transfer (kW)	Heat Transfer, Total Fuel Energy (%)	Average Heat Transfer (kW)	Cumulative Heat Loss (kJ)
SI-MeOH	2000	6	2.44	20.66		
			1.97	18.23		
			1.77	16.51		
SI-EtOH			3.80	29.89		
			2.81	24.15		
			2.83	23.99		
MC-MeOH/PC-MeOH			3.58	27.8	0.18	3.27
			2.75	23.2	0.18	3.20
			2.21	20.1	0.16	2.86
MC-MeOH/PC-EtOH			2.31	16.14	0.11	2.17
			3.03	23.66	0.17	3.24
			2.63	20.67	0.18	3.65
MC-EtOH/PC-EtOH			4.38	31.64	0.25	4.53
			3.46	27.4	0.25	4.53
			2.56	19.63	0.25	4.29
SI-EtOH		15	5.93	21.03		
			4.48	16.88		
			4.53	17.04		
MC-MeOH/PC-MeOH			5.21	18.37	0.28	5.16
			4.71	18.04	0.29	5.25
			3.48	13.78	0.27	4.87
MC-EtOH/PC-EtOH			7.83	26.44	0.44	7.85
			5.65	20.62	0.36	6.51
			4.35	16.06	0.35	6.38
SI-MeOH	3500	6	4.41	21.23		
			3.69	19.38		
			3.72	18.98		
SI-EtOH			6.53	29.22		
			3.49	16.92		
			3.24	15.80		
MC-MeOH/PC-MeOH			3.53	16.16	0.18	1.88
			3.25	16.05	0.20	2.09
			2.69	13.78	0.18	2.06
MC-MeOH/PC-EtOH			3.49	19.93	0.17	1.72
			2.64	16.54	0.17	1.79
			2.88	18.09	0.18	1.85
MC-EtOH/PC-EtOH			7.41	31.14	0.36	3.63
			6.31	28.95	0.44	4.33
			5.63	25.98	0.47	4.66
SI-EtOH		15	8.37	16.87		
			7.81	16.60		

MC-MeOH/PC-MeOH			8.09	17.24		
			6.36	13.22	0.32	3.67
			6.17	13.58	0.38	3.97
			5.22	11.38	0.42	4.43
MC-EtOH/PC-EtOH			10.29	20.01	0.60	6.06
			5.96	12.33	0.42	5.23
			5.64	11.95	0.39	4.24

Table 4.7 - Average amount of fuel injected until lambda 1.5

	6 bar of IMEP		15 bar of IMEP	
	2000 rpm	3500 rpm	2000 rpm	3500 rpm
MC-MeOH/PC-MeOH	1.05	1.06	0.47	0.48
MC-MeOH/PC-EtOH	1.34	1.35		
MC-EtOH/PC-EtOH	1.35	1.36	0.62	0.63

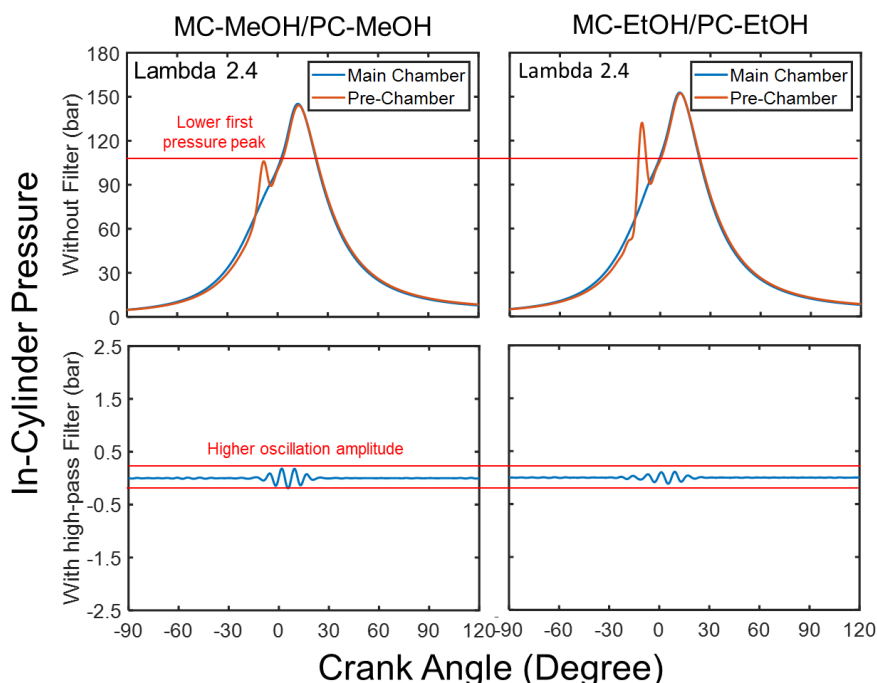


Figure 4.37 - Comparison between pressure curves of MC-MeOH/PC-MeOH and MC-EtOH/PC-EtOH configurations at 2000 rpm and 15 bar of IMEP in lambda 2.5

Previously, the hypothesis was raised that insufficient amount of fuel at 3500 rpm and 15 bar of IMEP is related to the increase in combustion duration of the MBF 50-90. This hypothesis is refuted with the analysis of Figure 4.35 and Figure 4.36 since the amount of fuel injected at this operating point is similar to the amount of fuel injected at the speed of 2000 rpm and 15 bar of IMEP. However, it can be argued that the increase in turbulence generated by the combination of PC use with the increase in engine speed and load may have negatively influenced combustion. This would also explain why at this point the lean limit was not extended. Corroborating this hypothesis, it is highlighted in the FFT analysis the first resonance mode, which associated with the radial direction of jets in the pre-chamber, was only excited at 2000 rpm. Therefore, it is possible that with the excess turbulence obtained by other sources, the directed turbulence provided by the pre-chambers had a much smaller effect at 3500 rpm.

Finally, the point 3500 rpm and 6 bar of IMEP is highlighted. In it, more methanol is injected into the PC than ethanol, which this time may be justified by the higher LHV of ethanol. It is observed in this case that the first pressure peaks in the two cells are similar and however, there is more amplitude of pressure fluctuations when methanol is injected into the main chamber, as observed in Figure 4.38. Since the hypothesis is raised that excess turbulence, especially at higher engine speeds, causes the lean limit not to be extended, perhaps for future calibrations it is recommended that less fuel is injected into the PC at this point, as at the others.

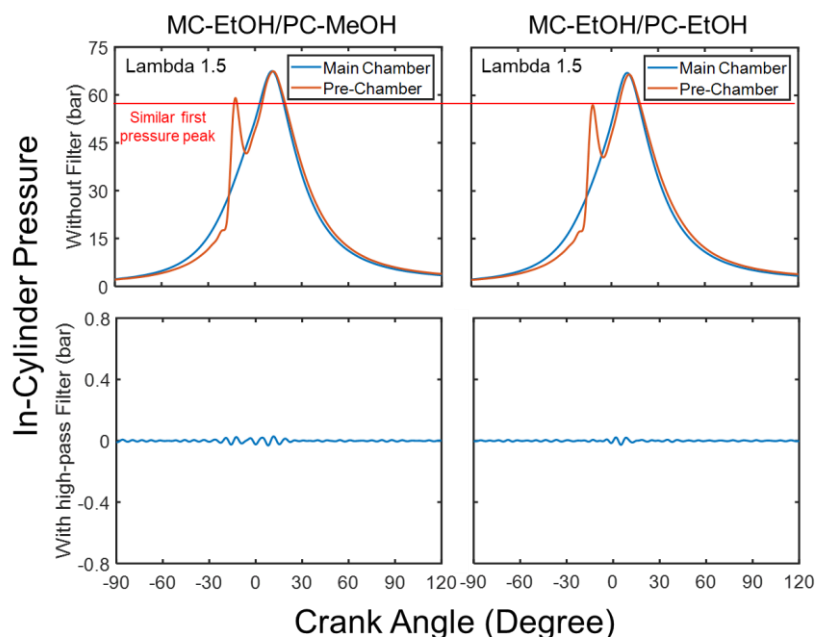


Figure 4.38 - Comparison between pressure curves of MC-MeOH/PC-EtOH and MC-EtOH/PC-EtOH configurations at 3500 rpm and 6 bar of IMEP in lambda 1.5

Although interesting conclusions can be drawn from the analysis of fuel mass injected per cycle, especially regarding the amount of fuel injected in each chamber, it is important to consider engine performance, which can be analyzed with the specific fuel consumption. To understand this parameter, one should consider both the fuel properties and the characteristics of the ignition system used. In this sense, it is interesting to observe the indicated combustion and thermal efficiencies of the engine, presented in Figure 4.39.

Combustion efficiency measures how much energy was extracted from the reaction between fuel and air in relation to the maximum energy potential. This maximum potential is reached through the complete combustion reaction, in which the fuel, with high enthalpy, is transformed into the products CO_2 and H_2O , the chemical species with the lowest possible enthalpy.

As shown in Figure 4.39, in stoichiometric operation, the combustion efficiency is always very high, close to 100%, for all operating conditions due to high in-cylinder temperatures [185]. It should be noted, however, that even under these conditions, the pre-chamber operation presents higher combustion efficiency because the extra turbulence generated is able to create a more homogeneous mixture of the gases and, consequently, a faster and more efficient reaction.

As the mixture is diluted, the in-cylinder temperature reduces, as previously discussed, causing the flame front to be quenched more easily, resulting in the formation of unburned or partially burned products, HC and CO [186], resulting in a general lower combustion efficiency. However, the faster the combustion reaction, the more fuel is burned before the flame is quenched, which leads the trend of the combustion efficiency to be directly opposite to the combustion duration. Therefore, when the combustion is supported by PCIS, the combustion efficiency does not present a reduction as significant as that presented by the configurations with baseline ignition.

In addition, a slightly lower combustion efficiency is observed for the engine speed 3500 rpm, which can be explained, despite the greater turbulence, with the fact that the combustion duration is lower. This can reduce the amount of reaction products that are completely oxidized into the lowest enthalpy molecules. Furthermore, the hypothesis of excess turbulence at 3500 must also be highlighted, which may have also hindered complete combustion. Such a hypothesis can be better analyzed in the section, in which exhaust emissions are discussed.

It should be noted, however, that the present calculation of combustion efficiency is based on the chemical species measured in the exhaust ports. Due to the high temperatures of the gases leaving the combustion chamber, it is possible that the in-cylinder combustion products continue to be oxidized in the exhaust ports.

In general, it is observed that the trend of thermal efficiency is opposite to that of combustion efficiency. Under lean combustion, the engine thermal efficiency is generally improved owing to the high ratios of specific heat and low heat transfer losses due to lower in-cylinder temperatures [187].

Despite the higher-pressure peaks and shorter combustion times, the indicated thermal efficiency for pre-chamber operation is generally lower. This is due to the fact that the addition of pre-chambers increases turbulence in the main chamber, increasing heat convection in the cylinder walls, increasing thermal losses in relation to the engine with baseline injection [188]. In addition, it should be noted that with the addition of pre-chambers, the engine has a slightly lower compression ratio, which also contributes to the reduction of thermal efficiency [13]. Moreover, it is possible that there are higher heat losses in the period in which the higher pressure and temperature peaks observed in the operation with pre-chambers occur. Furthermore, with higher in-cylinder turbulence there is also a greater possibility of heat transport to the walls, and however, increased thermal losses.

However, as combustion is faster, less time is available for the heat produced during combustion to be converted into thermal losses, so that more intake heat is converted into work. Therefore, when comparing the two engine configurations with baseline injection, the highest thermal efficiency is observed for methanol combustion. Analogously, when analyzing the configurations with PCIS, the highest efficiency is observed by the engine operating solely with methanol (MC-MeOH/PC-MeOH) and the lowest, solely ethanol (MC-EtOH/PC-EtOH). Additionally, methanol presents higher latent heat of vaporization, which reduces the in-cylinder temperature and decreases thermal losses.

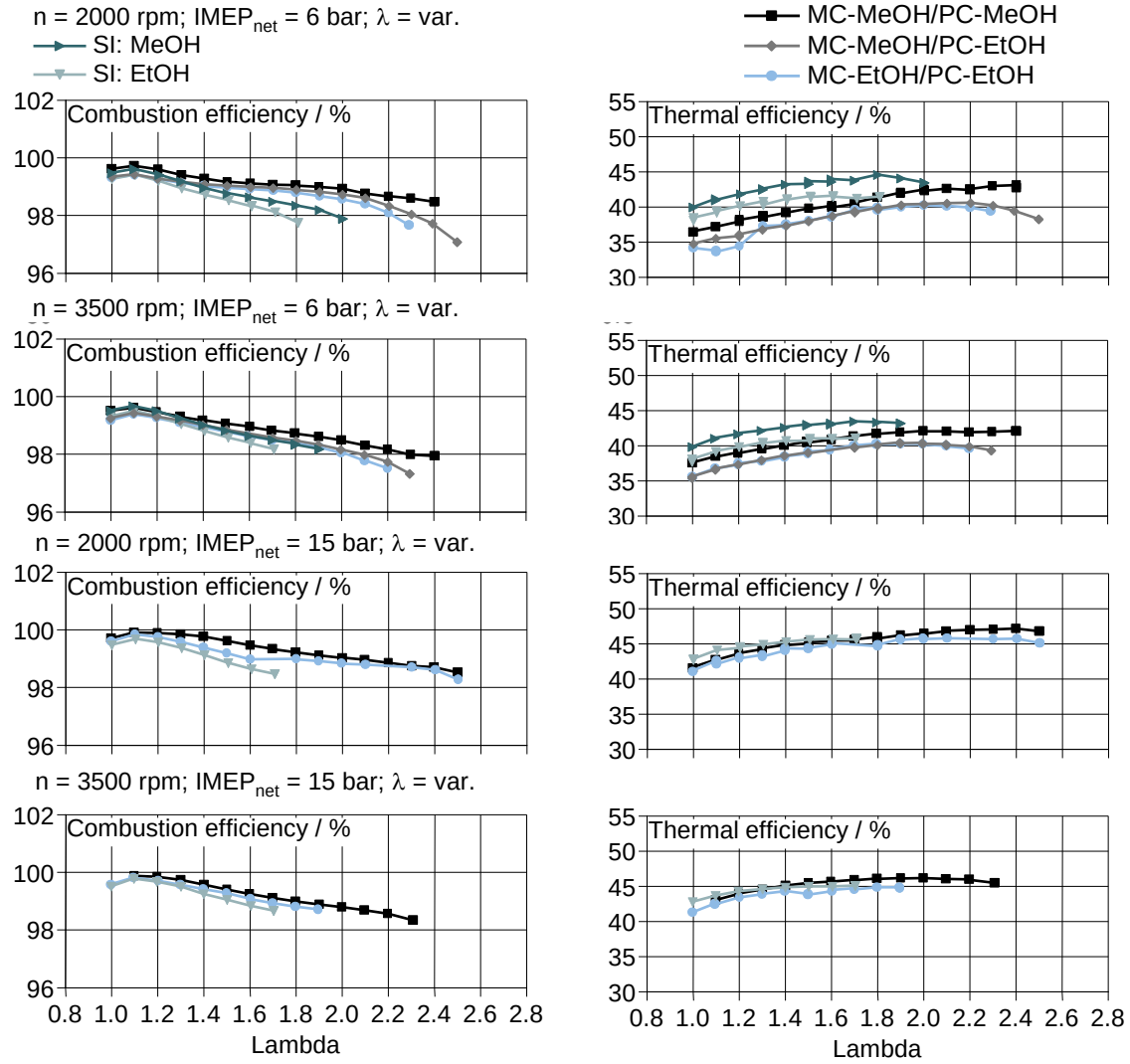


Figure 4.39 - Combustion and thermal efficiency at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

Since the association of thermal and combustion efficiency results in the fuel conversion efficiency, with the analysis carried out with Figure 4.39 it is also possible to analyze Figure 4.40.

The figure shows the IFSC and ISFC corrected. The ISFC is always higher for in the configurations fueled with methanol, making this comparison not very fair, due to the lower LHV of this fuel. Thus, to compare fuels with different LHVs, the $IFSC_{corr}$ is used, being calculated according to Equation 4.1. In the equation LHV_{fuel} represents the LHV calculated considering the proportion of fuels used in the operation of a cycle, as shown in Figure 4.40 and $LHV_{gasoline}$ is the LHV of gasoline, arbitrarily chosen so that the $IFSC_{corr}$ unit remains the same as the ISFC.

$$IFSC_{corr} = \frac{ISFC \cdot LHV_{fuel}}{LHV_{gasoline}} \quad 4.1$$

Figure 4.40 clearly shows that the ISFC is lower for the SI configurations, showing the dominant importance of the thermal efficiency in this parameter, since the combustion efficiency, although lower for the SI configuration, remains throughout dilution at all

operating conditions and for all configurations above 97%. The lower thermal efficiency of PCIS configurations, the higher the ISFC, which only presents similar values to those presented by the SI engine in very lean conditions, in which the SI engines cannot reach. Thus, it is noteworthy that for the configuration studied, the benefits presented by pre-chamber operation are mainly to the increase in the lean limit, controlling emissions such as CO, HC and NOx.

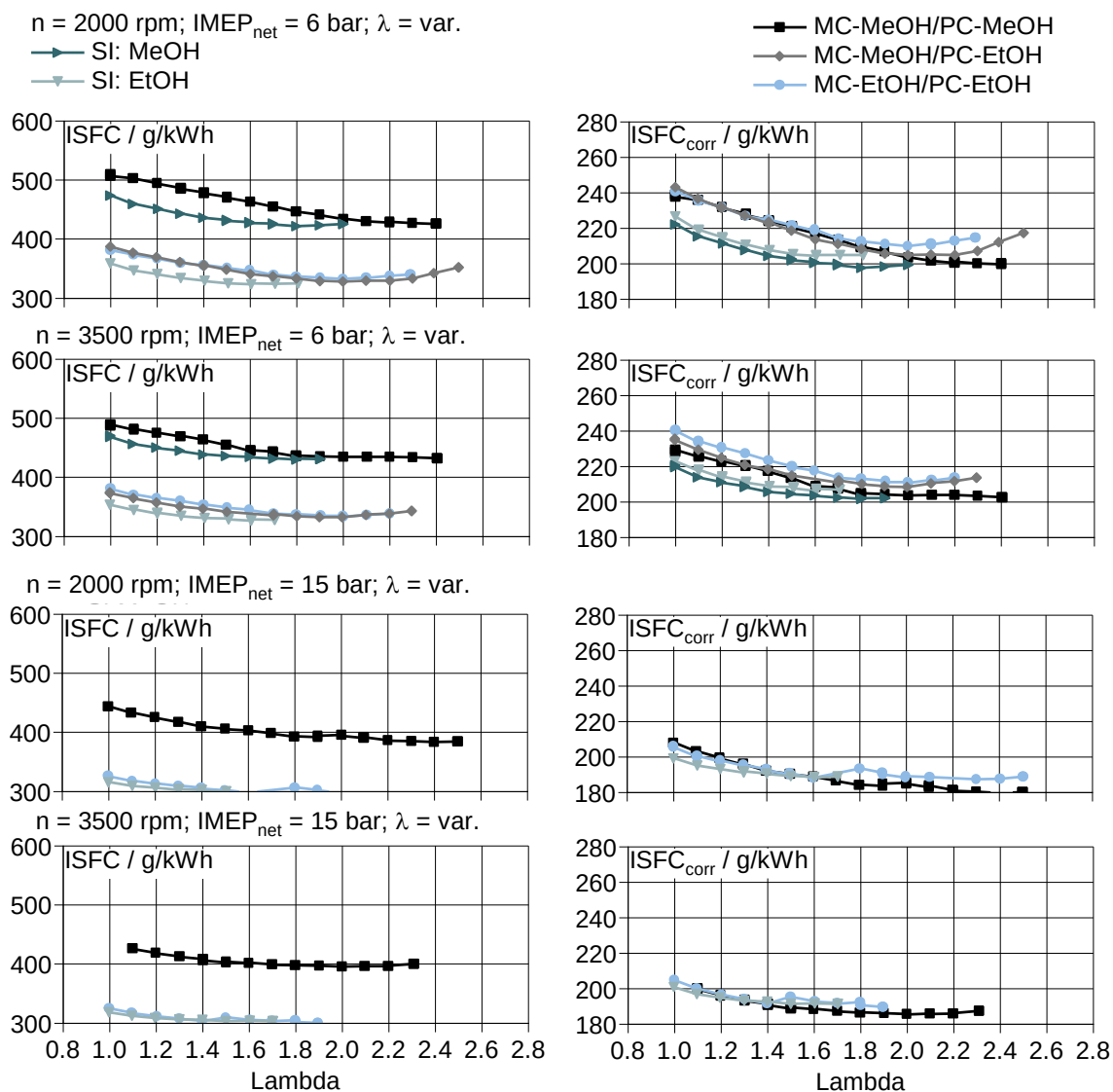


Figure 4.40 - ISFC at 2000 and 3500 rpm and 6 and 15 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

4.6 Exhaust Emissions

As previously mentioned, the main aspect that benefits pre-chamber operation would be the reduction of emissions, since the operation in high compression ratios with ethanol and methanol allows great efficiency and, consequently, greater reduction of ISFC. Thus, to better understand the effect of pre-chambers operation on exhaust emissions, Figure 4.41 and Figure 4.42 are analyzed. As the complete fuel combustion should result solely on CO₂ and water, the decrease of specific CO₂ emissions can have two meanings: lower ISFC or that lower CO₂ formation at the expense of CO and HC, that is, lower combustion efficiency. The difference among the indicated specific CO₂ emission is not that great as

the ISFC difference between baseline and pre-chamber ignition systems. This is precisely due to the increase in HC and CO emissions presented for SI configurations, which, due to their lower combustion speed, suffer to a greater degree the effects of flame quenching. Hence, the lower combustion efficiency observed when using the system, as previously observed, can be explained. It is generally observed that HC emissions increase more significantly before the lean limit is reached, which indicates an earlier flame quenching.

The formation of NO_x is intricately linked to the in-cylinder temperature and N_2 and O_2 the availability [73]. Therefore, all configurations observed in Figure 4.41 and Figure 4.42 follow the same trend. At lambda 1 the NO_x formation is great because the in-cylinder temperature is the highest. However, as there is no mixture dilution, less O_2 and N_2 are available for the endothermic reaction of NO_x formation. Thus, lambda 1.1 produces the highest NO_x formation for all configurations since the excess air is combined with high combustion chamber temperature. From that point onwards the in-cylinder temperature decreases, considerably reducing the NO_x emissions until they reach values very close to 0 in lambda 1.8.

It is observed that for the same load the emissions do not vary significantly, as observed in the ISFC graphs. However, for the increase of the load from 6 to 15 bar of IMEP, the HC and CO emissions decrease due to the increase of in-cylinder temperature, resulting in the complete oxidation of a greater amount of combustion products. Consequently, at such loads an increase in NO_x emissions is also observed, especially before lambda 1.6.

It should be noted that this occurs for all configurations, also the ones with baseline ignition. It is possible that this lambda can be reached without the help of pre-chambers. This occurs since the fuels used, ethanol and methanol, present high laminar flame speed and high knock resistance, which allow them to be operated with high compression ratios, consequently providing faster and more efficient combustion. Consequently, it can be concluded that pre-chambers do not provide great benefits in relation to NO_x emission reduction when operated with high compression ratios and alcohol fuels. Hence, the benefits brought by PCIS operation are justified in this case by the slight reduction of HC and CO emissions.

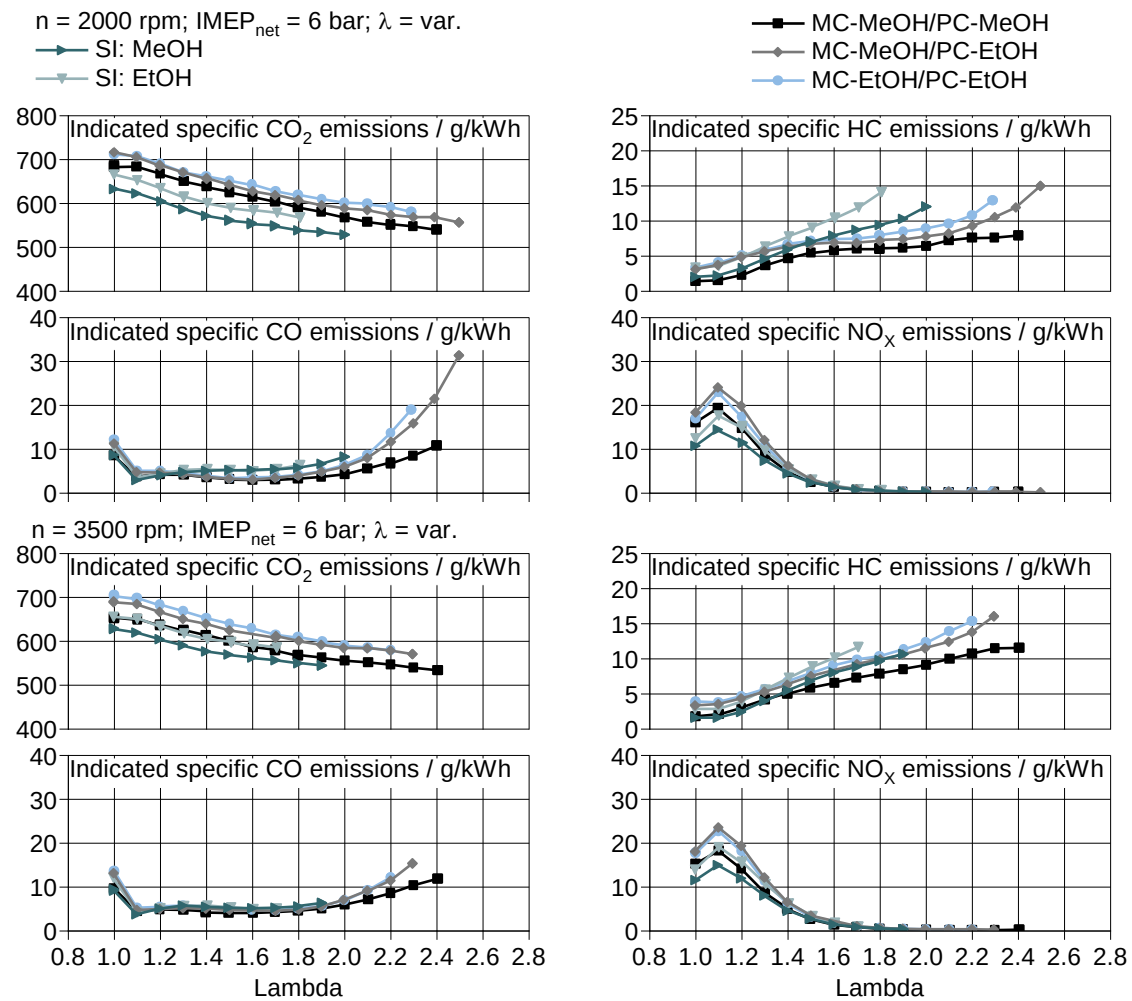


Figure 4.41 - Indicated CO_2 , CO , HC and NO_x emissions at 2000 and 3500 rpm and 6 bar of IMEP using several combinations of ethanol (EtOH) and methanol (MeOH)

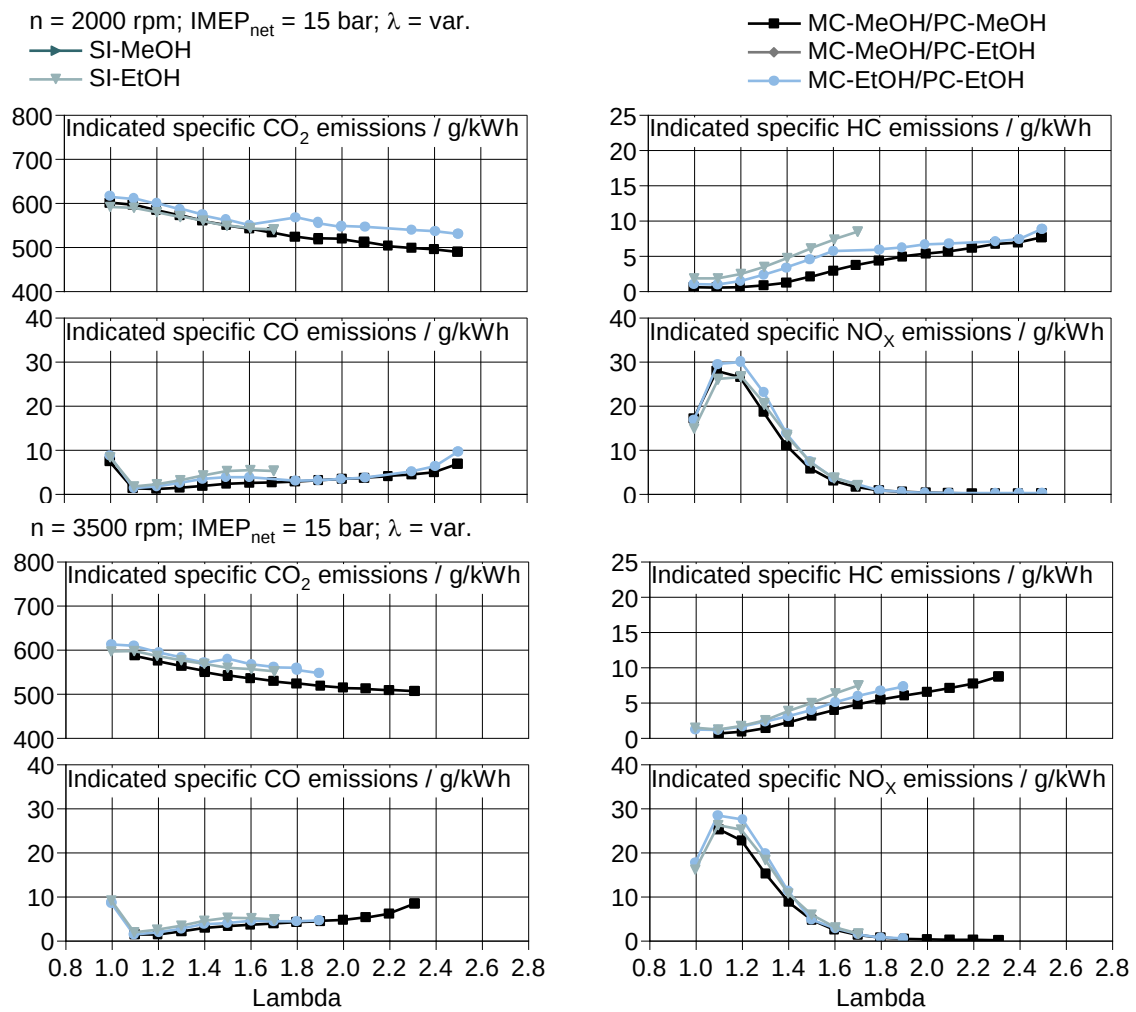


Figure 4.42 - Indicated CO_2 , CO, HC and NO_x emissions at 2000 and 3500 rpm and 15 bar of IMEP using several

5 FINAL CONSIDERATIONS

In general, it is considered that the work was successful in carrying out the activities and the analysis planned. Moreover, with this research, many additional characteristics of engines equipped with stratified pre-chambers using biofuels were acknowledged and discussed in greater depth. Therefore, this research can be considered the starting point for future research in which the theme is further developed. More specifically, from the results obtained in this study, and the corresponding discussions, it was possible to reach the following conclusions:

- Methanol both with baseline or with PCIS operation obtained the most extended lean limits, highlighting $\lambda=2.0$ with baseline operation at 2000 rpm and 6 bar of IMEP and $\lambda=2.5$ with PCIS operation at 2000 rpm and 15 bar of IMEP. However, due to operational difficulties, mainly due to injectors' degradation, methanol was not so prioritized in the study. Thus, it is noteworthy that ethanol obtained a lean limit of $\lambda=1.8$ with baseline operation at 6 bar of IMEP and 2000 rpm and $\lambda=2.5$ with PCIS operation at 15 bar of IMEP and 2000 rpm. It can be also highlighted that the lean limit throughout the study is further extended at lower speeds and loads, which was not expected. According to further analysis, this may occur due excessive turbulence caused by the pre-chamber, which negatively impacts the combustion. To conclude, it is important to highlight the already impressive lean limit extension achieved using baseline ignition. This is due to both the high engine compression ratio and the knock resistance and faster laminar flame speed of the biofuels explored.
- It is generally highlighted that the use of PCIS greatly influences combustion duration, in which MBF10-90 reaches 6°C_A for methanol at 15bar of IMEP and 2000 rpm, while the same fuel in the same operation point reached MBF10-90 is 12.2 for baseline ignition. Furthermore, the PCIS has a much greater impact on the duration of combustion in its second half, since the system is capable of maintaining a high turbulence level for an extended period of time.
- By analyzing the pressure curves, pressure fluctuations were observed in both configurations tested, which is considered inherent to the combustion process, caused by the compression of the end-gas and its subsequent movement. However, for the pre-chambers, due to the higher combustion speed, there is also a greater end-gas compression, leading the pressure oscillations to have greater amplitude. Such oscillations were explored from various methods and it was determined that, although they often have higher amplitudes than those determined by conventional knock thresholds, they do not represent knock. Thus, the use of the KI2 knock detection method and the determination of a new MAPO threshold for operation with pre-chamber were suggested so that PCIS operation is not limited by not adequate thresholds.
- Higher temperature and pressure peaks are observed in pre-chamber operation when compared to baseline in stoichiometric conditions, since combustion is faster and higher energy release peaks are observed. Thus, a greater heat amount is converted into work, resulting in greater thermal efficiency, and less heat is dissipated by the end-gas, which has in general lower temperatures in pre-chamber operation. However, the thermal efficiency when operating with pre-chambers is in general lower than that of the baseline operation due to increase turbulence generated by the technology, which increases heat losses.

- The fuel conversion efficiency is a combination of thermal and combustion efficiency and justifies the indicated fuel consumption behavior. Combustion efficiency is in general higher for pre-chamber combustion since the combustion at higher rates is more effective. However, the indicated fuel consumption is higher for pre-chamber operation, what indicates that the thermal efficiency had more impact on this specific parameter.
- Although the indicated fuel consumption represents a general sum of pre-and main chamber fuel, the role of fuel injected into the pre-chamber is very important. Despite the lower LHV of methanol when compared to ethanol, the amount of methanol injected into the PC is approximately 30% lower for ethanol in stoichiometry, a ratio that increases near the lean limit. This occurs due to the higher laminar flame speed of the fuel, which provides higher energy jets for the main chamber. Hence, the amount of PC fuel required to stabilize main chamber combustion, and consequently COV_{IMEP} , depends on its laminar flame speed.
- It is observed that between the baseline and pre-chamber operation, the difference among the indicated specific CO_2 emission is not that great as the ISFC difference, which is explained by the higher HC and CO emissions presented for SI configurations. For the same load, HC and CO emissions vary slightly between baseline and pre-chamber operation, since the pre-chamber enable a higher quality combustion. Furthermore, when the load increases from 6 to 15 bar of IMEP, the HC and CO emissions decrease due to the increase of in-cylinder temperature, resulting in the complete oxidation of a greater amount of combustion products. Consequently, at such loads an increase in NO_x emissions is also observed. Furthermore, it can be concluded that in the configuration tested, pre-chambers do not provide great benefits in relation to NO_x emission reduction when operated with high compression ratios and alcohol fuels. Hence, the benefits brought by PCIS operation are justified in this case by the slight reduction of HC and CO emissions.
- As a final conclusion, from the experimental results obtained in this study, it was observed that the engine equipped with pre-chambers presents lower thermal efficiency due to the increased heat losses intricate increase in cylinder surface for potential thermal losses and due to its 3% lower engine compression ratio in comparison to the baseline ignition. This fact affected the specific fuel consumption, which remains lower for the baseline configuration. It should be noted that inadequate knock thresholds were used in the calibration of the engine equipped with pre-chamber, which hindered the technology to reach its full potential. Furthermore, it is believed that a fair comparison between the two ignition systems would occur ideally with the engine equipped with pre-chamber at much higher compression ratios. This would be possible since the technology enables faster combustion and it is more knock resistant. Thus, the PCIS engine would achieve higher thermal efficiency and lower fuel consumption, which would justify its implementation in an internal combustion engine.

6 SUGGESTIONS FOR FUTURE WORKS

From the present work, some points can be addressed in greater depth so that pre-chamber technology achieves maturity. Among these points, the main ones are highlighted:

- The gum formation in the methanol injection tips of the pre-chamber should be further analyzed so that the fuel can be exploited commercially, since it presents very interesting characteristics for the operation in internal combustion engines. It must be determined whether this formation is due to the orifice diameter, the presence of some incompatible material in the fuel supply line or even in the methanol production.
- The lean limit extension in all configurations explored in this work was greater for lower loads and speeds. Such behavior can have many explanations, such as the excess of turbulence provided by the pre-chambers at higher loads and speeds. Therefore, a deeper investigation should be conducted, considering parameters such as turbulence and temperature to explain the phenomenon.
- As discussed in the results and discussions section, the determination of a new MAPO threshold is interesting because it considers the different thermodynamic aspects of the operation of an engine operated with pre-chambers. However, since the MAPO parameter leads to knock overestimation, this method has limitations. In addition, the KI2 method, despite being very suitable for operation close to stoichiometry, has limitations when evaluating combustions of extremely lean mixtures. Therefore, it would be important to combine the two analyzed parameters to define a new knock parameter, which addresses this effect throughout the entire engine operation.
- The increase in turbulence brought by the pre-chambers used in the present study increased significantly the heat losses, which resulted in much lower thermal efficiency for the configurations with pre-chambers. In addition, the hypothesis was raised that the excess of turbulence could have affected the lean limit extension at operating points with higher rotations and loads. Therefore, a more detailed research of the gaseous movement inside the main combustion chamber is suggested due to the use of the pre-chambers, studying ways in which turbulence does not negatively affect combustion
- As one of the main conclusions of the present study, pre-chamber operation can be justified only for volumetric compression ratios in which baseline injection operation is not possible. Thus, it would be interesting that further research to be conducted in this regard. The experimental data used in this work could be used to calibrate a turbulence combustion model in GT-Power in order to predict combustion with both ignition systems while the compression ratio is increased. Furthermore, a knock prediction model should also be used.
- Complementing the previous suggestion, a next step would be the experimental testing of the two ignition systems tested at the maximum volumetric ratio possible, in which knock occurrence, thermal efficiency, engine-out emissions and fuel consumption should be evaluated.

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Appendix A. GT-Power Model

The main objects to model an engine operating with pre-chambers are the *EngCylinder*, the *EngineCrankTrain* and the *PreChamber*, which characterize respectively the properties of the main-chamber and the admitted fluid; the geometric characteristics of the engine and the cylinder; and all the geometric and thermodynamic properties regarding the pre-chamber. The intake and exhaust ports and runners were modeled with the GEM 3D software, contained in the GT-ISE package. In the software, geometries described in STL files for their construction are converted into basic flow components. Furthermore, some controllers and optimizations were used to approximate the experimental and simulated results, so that the model is validated and information extracted only from it can also be used. The model used for this study is presented in Figure 3.5 and the next subsections describe how the GT-Power model is configured, detailing each object used.

Cylinder Characterization

Main Chamber Characterization

The cylinder is characterized through the *EngCylinder* object. A reference object for the initial simulation conditions must be established. Hence, the reference object *FluidInitialState* was used to describe the pressure, temperature and composition of the fluid in the initial simulation state. The inlet pressure and temperature were measured individually for each experimental test and the initial composition of the gases was determined by the reference object *FluidMixture* named “init-int”, which represents the air as a mixture of 76.7% N₂ and 23.3% O₂. Furthermore, the relative air humidity is used as input according to the measurements carried out during experimental tests, and it was assumed that the only humid species on the fluid mixture was water in a vapor state.

The temperature of the cylinder structures was imposed for the cylinder head, piston and liner. Such temperatures were measured in previous experimental tests and the data collected was then used to create a RLT Dependence file with a *XYZMap*, in which the wall temperatures are dependent on both average engine speed and BMEP. The cylinder head, piston and liner temperatures are respectively presented in the Table A.1, Table A.2 and Table A.3.

Table A.1 - Cylinder Head Temperature in K according to the engine speed (rpm) and in-cylinder pressure (bar)

		Engine speed (rpm)						
		1000	1500	2000	3000	4000	5000	6000
In-cylinder pressure (bar)	2	434.0	443.0	453.0	464.0	469.0	475.0	480.0
	5	439.0	448.0	458.0	469.0	474.0	480.0	485.0
	8	444.0	453.0	463.0	474.0	479.0	485.0	490.0
	10	447.0	456.0	466.0	477.0	482.0	488.0	493.0
	14	457.0	466.0	476.0	487.0	492.0	498.0	503.0
	18	467.0	476.0	486.0	497.0	502.0	508.0	513.0
	20	470.5	479.5	489.5	500.5	505.5	511.5	516.5
	22	474.0	483.0	493.0	504.0	509.0	515.0	520.0
	25	484.0	492.0	500.0	508.0	515.0	520.0	525.0

Table A.2 - Cylinder Piston Temperature in K according to the engine speed (rpm) and in-cylinder pressure (bar)

		Engine speed (rpm)						
		1000	1500	2000	3000	4000	5000	6000
In-cylinder pressure (bar)	2	445.0	452.0	459.0	468.0	477.0	486.0	493.0
	5	451.0	458.0	465.0	474.0	483.0	492.0	499.0
	8	457.0	464.0	471.0	480.0	489.0	498.0	505.0
	10	461.0	468.0	475.0	484.0	493.0	502.0	509.0
	14	471.0	478.0	485.0	494.0	503.0	512.0	519.0
	18	482.0	489.0	496.0	505.0	514.0	523.0	530.0
	20	486.5	493.5	500.5	509.5	518.5	527.5	534.5
	22	491.0	498.0	505.0	514.0	523.0	532.0	539.0
	25	497.0	504.0	511.0	520.0	529.0	538.0	545.0

Table A.3 - Cylinder Liner Temperature in K according to the engine speed (rpm) and in-cylinder pressure (bar)

		Engine speed (rpm)						
		1000	1500	2000	3000	4000	5000	6000
In-cylinder pressure (bar)	2	393.0	398.0	404.0	410.0	419.0	425.0	434.0
	5	402.0	407.0	413.0	419.0	428.0	434.0	443.0
	8	412.0	417.0	423.0	429.0	438.0	444.0	453.0
	10	419.0	424.0	430.0	436.0	445.0	451.0	460.0
	14	430.0	435.0	441.0	447.0	456.0	462.0	471.0
	18	440.0	445.0	451.0	457.0	466.0	472.0	481.0
	20	444.5	449.5	455.5	461.5	470.5	476.5	485.5
	22	449.0	454.0	460.0	466.0	475.0	481.0	490.0
	25	457.0	462.0	468.0	474.0	483.0	489.0	495.0

A model of heat transfer between the fluid and the walls is required in the cylinder object through a reference object *EngCylHeatTr*. Among the available heat transfer models, the WoschiniGT was chosen. The model calculates the in-cylinder heat transfer similarly to the classical Woschni correlation without swirl [21], differing by the treatment of heat transfer coefficients. The Woschini model assumes a correlation for spatial average coefficients according to Equation A.1

$$Nu = 0.035Re^m \quad A.1$$

Considering B the cylinder bore and w as the local average gas velocity in the cylinder, which should be proportional to the main piston speed during intake, compression and exhaust, $k \propto T^{0.75}$, $\mu \propto T^{0.62}$, and $p = \rho RT$, the Equation A.2 can also be written as:

$$h_c = CB^{m-1}p^m w^m T^{0.75-1.62m} \quad A.2$$

The average cylinder gas velocity can be determined according to Equation A.3, in which p is the instantaneous cylinder pressure, p_r , V_r , T_r are working- fluid pressure, volume and temperature at the referend state and p_m is the motored cylinder pressure at the same crank angle as p .

$$w = \left[C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} (p - p_m) \right] \quad A.3$$

The C1 coefficients are calculated for the gas exchange, compression and combustion and expansion period for high-speed engines with swirl. However, in the model Woschini GT, since the swirl data of the engine test is not available, the heat transfer coefficient is increased by inflow velocities through the intake valves and also by backflow through the exhaust valves.

The cylinder head to bore area ratio and piston to bore area ratio must be informed, being respectively 1,425 and 1,044 for the engine model studied. Such data were obtained from the analysis of SLT. files containing the combustion chamber geometry.

To have in-cylinder information on swirl, tumble and turbulence in general it is necessary to create a flow object through a reference object *EngCylFlow*. In this object all possible options are modeled as default and a Piston Cup Object must be modeled according to the piston geometry shown in Figure A.1. **Fehler! Verweisquelle konnte nicht gefunden werden.**

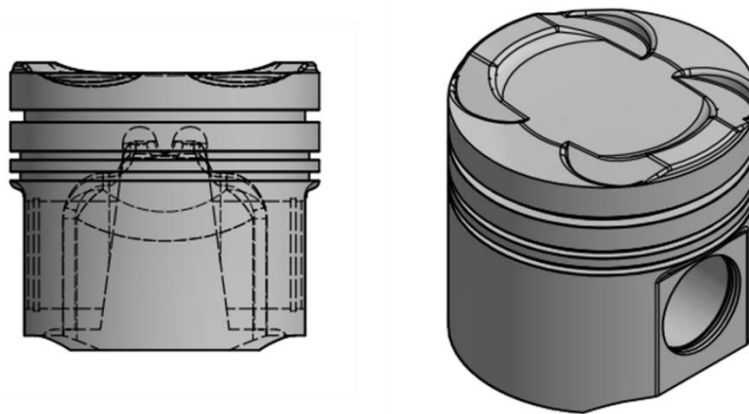


Figure A.1 - Piston head and cup geometry

Since the work is based on data obtained experimentally, the **Three-Pressure Analysis (TPA)** method was used that more complex mathematical analyzes could be conducted. The method calculates the burn rate from the measured cylinder pressure, requiring instantaneous measurements of pressure in the cylinder and in intake and exhaust ports, in addition to temperature and mass flow measurements where the intake port pressure is measured. The model works in iterative manner and cycles repeat until the imposed and current in-cylinder pressure parameters converge, which helps in validating the model [189]. In addition, in the simulation run the folder *RLT convergence* is enabled to impose steady state criteria that depend on RLT variables in a simulation, so that convergence only happens when these criteria are also respected. In the current model, the simulated lambda and IMEP parameters are set as targets, so that the simulation only converges if the simulated data reaches the measured results with maximum 5% tolerance.

For such an analysis to be effective, no cylinder pressure analysis is conducted since it is already imposed. To perform the combustion analysis on the *EngBurnRate* reference object, the parameters related to combustion are configured as following:

- Measured data object: object in which the measured cylinder pressure is provided with 0.1° AC increments.

- Start of combustion: defined as the moment when the heat release becomes different from zero in the main chamber, or MFB 0.1. Such a parameter can also be roughly defined as the pre-chamber ignition timing.
- Imposed end of measured burn rate: set as 180° CA for all cases so that the combustion can be analyzed in its entirety.
- End of analysis: defined as end of the measured burned rate.
- Number of temperature zones: defined as two zones for more accurate results.
- Entrained Fuel-Air Option: defined as homogeneous, indicating that the air-fuel ratio remains constant throughout the combustion process.

In the advanced cylinder modeling options, a reference object to describe the evaporation of the fraction of the liquid charge entering the combustion chamber must be created. The *EngCylEvaporation* object was used to model the fuel vaporization, being a phenomenological evaporation model, which calculates the fraction evaporated at each time step from a relationship between time required to evaporate 50 % of the in-cylinder at 600K and 4000 rpm. These values vary according to the fuel used and are obtained from the literature. It is possible to adjust evaporation curves through exponential parameters to temperature and pressure but since no literature data was found on these exponent parameters, they were set with the software default values.

In addition, a fluid object *FluidLiqincompress* was created, in which the heat of vaporization, the density and the absolute entropy at 298K were defined. The exponents for enthalpy calculation are defined, as well as the fluid transport properties such as dynamic viscosity array and thermal conductivity as a function of temperature and pressure. The transport properties of air are already known inside the GT-Power code and used as default. However, if another fluid is used, such information must be provided, especially since dynamic viscosity and thermal conductivity are highly dependent on pressure and temperature.

After defining the incompressible liquid, it is necessary to define a Vapor Fluid object, in which the number of carbon, hydrogen and oxygen atoms per molecule, the lower heating value, the critical temperature, pressure and absolute entropy at 298 K should be stated. Furthermore, the enthalpy coefficients for these calculations are defined. The transport properties of the evaporated fluids were defined as identical to air properties.

Also, in the output section of the *EngCylinder* object the lambda should be defined, since it is used in the calculation of various RLTs, plots, and sensors. The **lambda** definition is recommended and used instead of **lambda-old**, especially when working with oxygenated fuels, since not only the amount of C, H, and Sulphur (S) atoms are counted, but also the necessary oxygen to get H₂O, CO₂, and sulfur dioxide (SO₂) as a product is determined.

In order that the simulation results are tuned with the experimental results, some optimizations are discussed later in this section, after the simulation is fully characterized.

Pre-Chamber Characterization

The pre-chamber is characterized by the *PreChamber* Object. The initial state object used to describe the initial conditions of the simulation in the system is the “init-int”, being the same initial state assigned to the main camera. Although it is known that the states of the

main and pre-chambers are different, if enough cycles is run, the model should converge to the same result of the initial conditions. Thus, no attempt has been made to define this initial condition so accurately. Regarding the heat transfer analysis, the heat transfer coefficient model and the wall temperature model are assumed to be the same as that of the main chamber. The heat transfer multiplier used was the software default, set to 1.

The pre-chamber was simulated in “Heat Release-Based Calibration (HRBC) mode”, in which the apparent heat release rates were calculated from the measured pressure traces and applied to the prediction of combustion progress in the chamber. With this, it is necessary to establish the measured *In-Cylinder Pressure Analysis Object*, in which the pre-chamber pressure measured as a function of the crank angle is given.

For such an analysis to be effective, no cylinder pressure analysis is conducted since it is already imposed. To perform the combustion analysis on an *EngBurnRate Reference Object* similar to the one used to model the main combustion chamber is used, in which the measured cylinder pressure is provided with 0.1° CA increments and the start of combustion is defined as MFB 0.1. Such a parameter can also be roughly defined as the pre-chamber ignition timing. The end of the combustion analysis is defined as the end of the measured burned rate and a two-temperature zone is and homogeneous entrained Fuel-Air Option is considered. As no oscillations are observed in the pre-chamber pressure curves, no filter was applied to such data.

The combustion object used is the *EngCylCombSIWiebe*, a template which imposes the burn rate for spark-ignition engines using a Wiebe function. The Wiebe Function represents well the mass burned fraction evolution in time and, in computational models such as GT-Power, the mass burned rate is imposed, not characterizing a predictive model [190].

For the Wiebe Model to be defined, some variables must be determined, such as an Anchor Angle, which is normally defined as MBF50. When this anchor angle is chosen, the specified angle defined should be the number of crank angle degrees between TDC and the 50 % combustion point of the Wiebe curve. The duration of combustion, defined by default as MBF1090, must also be reported, as well as the Wiebe exponent and the Fraction of Fuel Burned. If a value less than 1.0 is specified, the unburned fuel will be removed proportionately from the combustion rate so that the burn rate will have identical shape, but a lesser scale.

For the simulation results determined by the Wiebe combustion model to be tuned with the measured results, an Excel file named *WiebeComb.xls* included in the installation directory of GT-SUITE is used to determine the Wiebe constants that are used in each model.

To determine the pre-chamber flow, all options for turbulence modeling and outflow compositing are set to either standard or default. The pre-chamber geometry is determined in an advanced way. The determination of the surface area and volume of the pre-chamber were obtained from the Inventor software, where the 3D model of the pre-chamber was design.

The pre-chamber nozzles are represented with the *PreChamberConn* object. With this template, the nozzle diameter, which is 0.85 mm, the number of nozzle holes, 6, and the forward and reverse nozzle discharge coefficients, which assume values between 0.6 and 0.85, are determined.

Crank Train Characterization

The *EngCrankTrain* object was used to characterize the crank train, specifying the attributes used to calculate the kinematic and dynamics model of rigid bodies. Thus, the models contained in this object calculate the torque on the crankshaft due to the pressure forces acting on the piston, at each time step, allowing the modeling of friction, inertia of moving parts and application of external loads.

In this same object the *Engine Type* is determined, which in this case is **4-stroke**. As all the tests conducted at a given speed varying lambda, the speed was specified in every case. The engine friction object or friction mean effective pressure (FMEP) was determined according to the experimental data. The start of the cycle is the same as used in the calibration of in-cylinder pressure between IVC and EVO interval.

The cylinder geometry is described in the *EngCylGeometry* object with the values presented in Table A.4. In the same table the **Wrist Pin to Crank Offset** is also determined when characterizing the piston-to-crank offset.

Table A.4 - Engine Cylinder Geometry Data used in the GT-Power Model

Bore [mm]	75
Stroke [mm]	113.2
Connecting Rod Length [mm]	220
Compression Ratio	16.4
TDC Clearance Height [mm]	1
Wrist Pin to Crank Offset [mm]	0.4

In the RLT norms tab, it is necessary to determine the reference state for volumetric efficiency, assumed as the ambient condition or “init-int”, the state of the air at the intake ducts. With this it is possible to calculate the reference density for calculation of the volumetric efficiency RLT variable in *EngCylinder* parts and the *EngineCrankTrain*. It is also necessary to set the **Part Name for Manifold Volumetric Eff. Reference**, which corresponds to the name of the part to be used to provide the reference density for calculating **Manifold Volumetric Efficiency** RLT variables parts and the *EngineCrankTrain*, which is the first part of the intake system, the “int-runner-1”.

No inertia data or external load applied to the engine was characterized, since these are used only for load mode tests, where the speed is calculated for a given load imposed on the motor. Furthermore, since the bearing load output is not desired at this study, this folder was left empty assigning "ign" and "def" values in the first column.

Fuel Injection System Characterization

In the present model, two injectors adequate for direct injection were used, one attached both to the main and the pre-chamber. The fuel injection systems were characterized

through the *InjPulseConn* object. To characterize the injection fuel rate 2 out of 3 attributes requested by the software must be provided, either the **Injection delivery rate**, the **Injection pulse width** and the **Injected mass**. In both injectors, the injection pulse width and the injected mass were chosen and defined from measurements performed at the test bench.

For both the pre- and main chambers, the default configuration of the injector was used, which means a nozzle diameter of 1 mm, with 8 holes per nozzle and with a nozzle discharge coefficient of 0.7.

Also for both injectors, the **Source of Angle** for the injection profile lookup is defined as *attached to the cylinder*. The **Injection timing flag** is defined as the *injection-start* and the **Injection flag** is then defined with the ECU data. Since the **Injection location** only needs to be defined for PFI injections, this field was filled with "ign", meaning the information was ignored. The **Injection fluid temperature** was also defined according to the test bench measurements, the **Vaporized fuel fraction** was defined as 0 and the **Fluid object** was used to define the fuel injected in each case studied.

In order to define the fuel studied, incompressible liquids were created using the *FluidLiqIncompress* template. For this, in the main settings it is necessary to determine the **Vapor Fluid Object**, the **Heat of Vaporization at 298K**, the **Fuel density** and the **Absolute Entropy at 298K**. To define the enthalpy of the fluid according to the temperature T of the combustion chamber, some fit coefficients are used to determine the liquid fuel enthalpy, h_{liq} , from Equation A.4. In the equation, T_{cyl} is the in-cylinder temperature, T_{ref} is considered 298K and the reference enthalpy for liquid substances, $h_{ref-liq}$, is defined by Equation A.5. In Equation A.5 $h_{ref-vap}$ is the enthalpy of the vapor fluid object at 1 bar and 298 K and $\Delta h_{vaporization}$ is heat of vaporization at 298 K.

$$h_{liq} = h_{ref-liq} = a_1 \cdot (T_{cyl} - T_{ref}) + a_2 \cdot (T_{cyl} - T_{ref})^2 + a_3 \cdot (T_{cyl} - T_{ref})^3 \quad A.4$$

$$h_{ref-liq} = h_{ref-vap} - \Delta h_{vaporization} \quad A.5$$

All parameters and coefficients necessary for the characterization of liquid methanol and ethanol, in addition to the coefficients for the determination of enthalpy are presented in Table A.5.

The dynamic viscosity and thermal conductivity of the fuel are determined according to temperature and pressure in the transport properties tab. The transport properties of ethanol and methanol are shown respectively in Table A.6 and Table A.7. When determining the **Vapor Fluid Object**, the *FluidGas* template is needed, and it defines the number of carbon, hydrogen, oxygen, nitrogen, sulfur and argon per molecule. The lower heating value, the substance's critical temperature and pressure, and the absolute entropy up to 298K are determined. In addition, the vapor fuel enthalpy, h_{vap} , is determined according to Equation A.6.

$$\begin{aligned}
h_{vap} &= h_{ref-vap} \\
&= b_1 \cdot (T_{cyl} - T_{ref}) + b_2 \cdot (T_{cyl} - T_{ref})^2 \\
&+ b_3 \cdot (T_{cyl} - T_{ref})^3 + b_4 \cdot (T_{cyl} - T_{ref})^4 \\
&+ b_5 \cdot (T_{cyl} - T_{ref})^5
\end{aligned} \tag{A.6}$$

All properties used for the determination of the vapor fuel are presented in Table A.8. For the vapor fluid objects, transport properties were defined as default, which indicates that air properties were used, representing a simplification of the model.

Table A.5 - Incompressible Liquid Properties for Ethanol and Methanol

	Ethanol	Methanol
Heat of Vaporization at 298K [MJ/kg]	0.928	1.18
Density [kg/m ³]	787.00	786.32
Absolute Entropy at 298 K [J/(kg.K)]	3159.09	3972.02
a ₁	2074	1837.8
a ₂	-0.17129	0
a ₃	7.6231E-1	0

Table A.6 - Transport Properties of Ethanol at 1 bar

Temperature [K]	Dynamic Viscosity [Pa.s]	Thermal Conductivity [W/(m.K)]
180.0	0.033192	0.2008
200.0	0.014058	0.1948
220.0	0.0069454	0.189
240.0	0.0038427	0.1833
260.0	0.0023147	0.1778
280.0	0.0014877	0.1725
298.15	0.0010405	0.1679
300.0	0.0010052	0.1674
320.0	7.0623E-4	0.1624
340.0	5.1154E-4	0.1577
360.0	3.7949E-4	0.1531
380.0	2.8684E-4	0.1486
400.0	2.1998E-4	0.1444
420.0	1.7061E-4	0.1403
440.0	1.3343E-4	0.1364
460.0	1.05E-4	0.1327

Table A.7 - Transport Properties of Methanol at 1 bar

Temperature [K]	Dynamc Viscosity [Pa.s]	Thermal Conductivity [W/(m.K)]
240.0	0.0015696	0.2144
260.0	9.5985E-4	0.206
280.0	6.69E-4	0.1977
298.15	5.2015E-4	0.1902

300.0	5.0848E-4	0.1895
320.0	4.077E-4	0.1813
340.0	3.3622E-4	0.1733
360.0	2.7961E-4	0.1654
380.0	2.3087E-4	0.1575
400.0	1.8693E-4	0.1498
420.0	1.4693E-4	0.1421
440.0	1.1119E-4	0.1346
460.0	8.0458E-5	0.1271
480.0	5.5359E-5	0.1197

Table A.8 - Vapor Properties for Ethanol and Methanol

	Ethanol	Methanol
Carbon atoms per molecule	2	1
Hydrogen atoms per molecule	6	4
Oxygen atoms per molecule	1	1
Nitrogen atoms per molecule	0	0
Sulfur atoms per molecule	0	0
Argon atoms per molecule	0	0
Lower Heating Value [MJ/kg]	26.84	19.9
Critical temperature [K]	516	513
Critical pressure [MPa]	6.38	7.95
Absolute Entropy at 298 K [(J/kg.K)]	5804.13	7484.33
b ₁	1566.5	1369.10
b ₂	1.43	1.15
b ₃	-4.15E-4	3.85E-4
b ₄	6.08E-8	-8.76E-7
b ₅	-3.49E-12	3.73E-10

Intake and Exhaust Flow Components Characterization

The characterization of the intake and exhaust systems is divided into two, since the geometry of the systems was first imported to the GEM-3D software and converted into a GT-ISE configuration. After the model is exported to GT-Power, all flow components must be characterized. Hence, the GEM-3D conversion and the GT-Power system characterization will be divided into two subsections.

GEM-3D Conversion

When a STL file is imported into the GEM-3D software, a mesh shape component is created. Using the *separate by curves* operation, the geometry is separated into different meshes based on changes in the surface of the object. In addition to these, manual separations can be performed manually by creating cutting planes so that similar cross-section shapes can be modeled more accurately.

After splitting the meshes, they must be converted into flow components. In order for the flow direction to be respected, when each mesh is converted into a shape, it is necessary to identify the inlet and outlet ports. In Figure A.2

gefunden werden. and Figure A.3 it is possible to observe the intake and exhaust flow components respectively, already divided and converted, as well as their names.

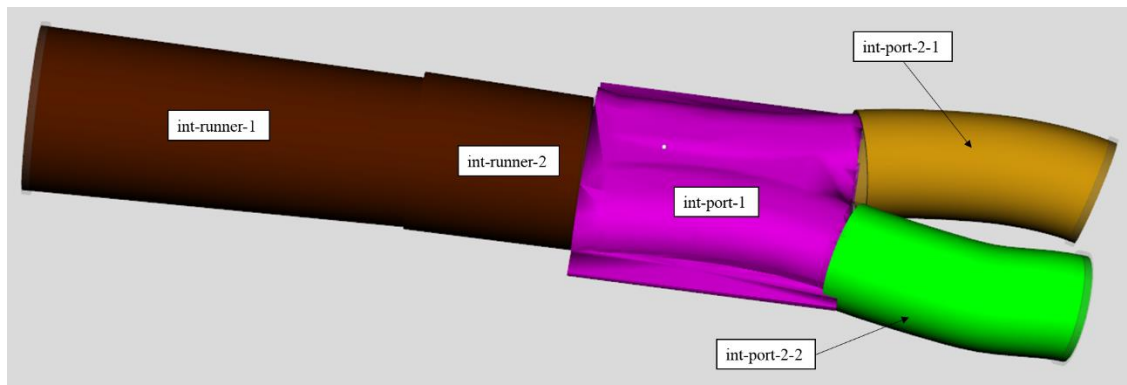


Figure A.2 - Intake Flow Components in GEM-3D

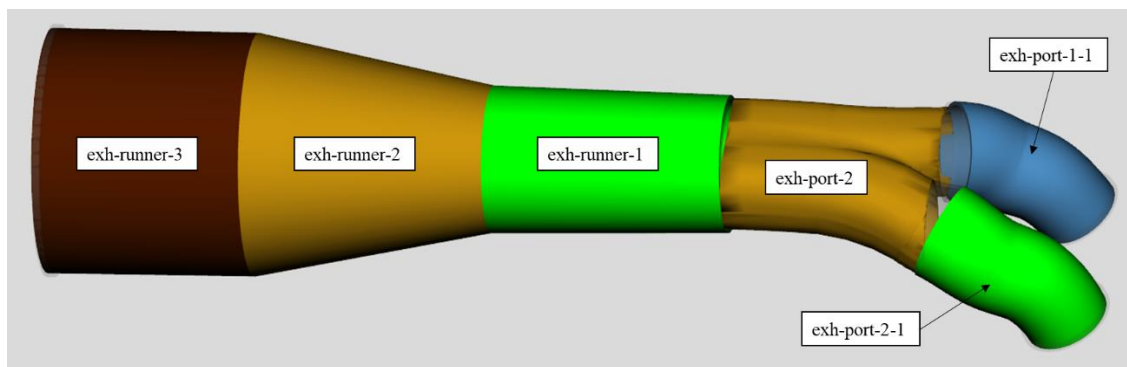


Figure A.3 - Exhaust Flow Components in GEM-3D

The “int-runner-1” is the first part of the intake runner, modeled as a straight pipe with the cross section varying according to linear interpolation between the pipe ends. It should be noted that in the original design, the component cross section had different geometries, but in the software, when pipes are modeled, the geometries of the cross-sections are approximated by circles, keeping the same cross-sectional area. The “int-runner-2” is a flange positioned between the intake runner and the intake port, modelled as the “int-runner-1”. The “exh-runners-1, -2 and 3” are also modelled as the two intake-port-pipes.

The “int-port-1” is the first part of the intake port, modeled as a *single general flowsplit*, which can have any number of inlet or outlet ports. The mesh flow split’s ports are initially defined and after that, the software converts such flow component into a very accurate geometry. The outer edges shown in the figure are fittings for a tumble generator plate, which was not used for these experiments. The “exh-port-3” is modeled just like the “int-port-1”.

After the division performed by the flowsplit, two pipes were formed, each one being modeled as multiple bend pipes, whose cross section is defined from linear interpolation between ends, originating the “int-port-2-1” and “int-port-2-2”. The “exh-port-2-1” and “exh-port-2-2” flow components are modelled likewise. In both cases, the intake and exhaust valve openings are removed and the mesh hole left with this removal is patched with the *patch mesh holes* function, so that the section can be modeled as a pipe.

During the process of converting meshes to components, several pipes created do not align perfectly and to solve this, the *flow connection* operator was used to join the components, which causes the pipes to be positioned slightly differently than before. Such an operation is not problematic as the flow volumes are maintained. When such flow connections promote a very large change in the pipe arrangement, the most recommended is to build *extruded connections*. In addition to the flow connections between pipes and flowsplits, the subassembly connection ('GEMSubAssExtConn') was created to impose connection direction in the boundaries. Hence, in the “int-runner-1”, “exh-port-1-1” and “exh-port-1-2”, the subassembly connection was created indicating an inlet flow direction. Likewise, in the “int-port-3-1”, “int-port-3-2” and “exh-runner-3”, a subassembly connection indicating an outlet flow direction was created. By making these connections, the model can be discretized and exported to GT-ISE.

System Characterization at GT-ISE

Figure A.4 and Figure A.5 show the geometries converted by GEM-3D and exported to GT-ISE. It should be noted that in exhaustion, due to the cross-section difference between “exh-runner-1” and “exh-port-3”, when performing an extruded connection, the software developed an orifice, named “ori9”, to represent a better flow transition between components than a default orifice connection.

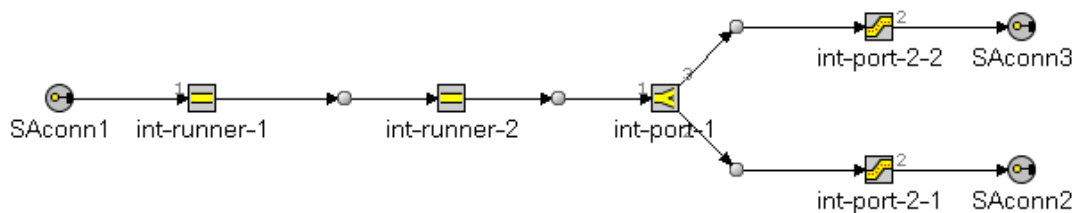


Figure A.4 - Intake Flow Components in GT-ISE

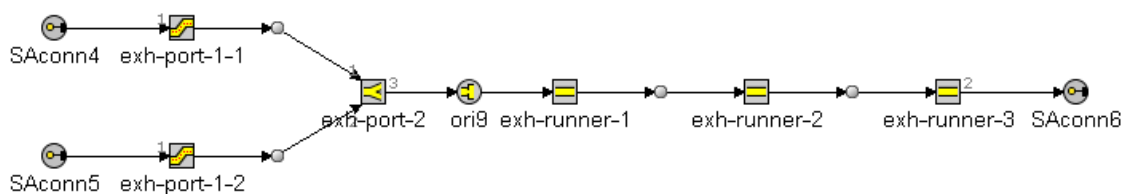


Figure A.5 - Exhaust Flow Components in GT-ISE

Once exported to GT-ISE, the intake and exhaust must be defined as objects in the engine performance model. Thus, several conditions for each flow component must be defined, such as the initial state conditions, the discretization length, the surface finish, in addition to the wall temperatures.

For the intake runners and ports, the set pressure and temperature are measured at the intake runner and for the exhaust runners and ports, the pressure and temperature are defined according to experimental measurements at the exhaust runner. The initial state for the intake runners and ports is named “inital-int”, represented by the reference object *FluidMixture*, considering air as a mixture of 76.7% N₂ and 23.3% O₂. The initial state for the exhaust system is named “initial-exh” and is represented as a *FluidMixtureBurned*,

considering the correct proportions of air and fuel, calculated according to stoichiometry and the lambda measured, prior to combustion. In addition, the relative humidity measured during the tests was provided for the model, considering water vapor as the only humidity species.

The discretization length is the approximate length of each sub-volume where solution quantities such as pressure, temperature and mass fractions are calculated. Generally, for TPA engine analysis, a discretization length should be reduced to about half of the normal recommended values (or about 20% of cylinder bore in intake and 25% of cylinder bore in exhaust ports. Hence, the assumed discretization lengths for intake and exhaust to achieve more accuracy were set at 15 and 20mm.

To determine the surface finish, the roughness of the extruded aluminum contained in the software database was used for the intake runner, and similarly, the roughness of the steel was used as the roughness for the exhaust runner. For the intake and exhaust ports, a smooth surface was considered.

The wall temperatures are a strong function of speed and load and since the frequency of the pressure waves is sometimes critical to the engine breathing, it is important to have the wall temperatures changing with operating condition. To best estimate wall temperatures at the exhaust system and at the intake runners, in which environmental and operation conditions have great impact on wall temperatures, the wall temperature solver is used. This feature is also capable of determining the resulting heat transfer rate to the fluid. The calculation requires the *WallLayerProperties* object, which provides the surface emissivity, layer thickness and number of sublayers. The material used is also described, determining its conductivity and specific heat in relation to temperature variation and density, established as constant. At intake runner and exhaust ports, Aluminum 2024 is used and stainless steel is the exhaust runner material. In addition, it is necessary to provide the *WallExternalBoundaryCondition* object, which requests the external convection temperature and external radiation sink temperature, both set as the ambient temperature during the tests, in addition to the external convection coefficient, defined as 5 W/m².K. For the calculations, the initial temperature for the intake and exhaust systems are set to 320 K. The wall temperature for the intake port were imposed according to experiments, being defined as 450 K.

For modeling the runners, besides the requested inputs described earlier in this subsection, all other parameters were set as default. However, for ports the default settings needed to be altered. Since, typically, the flow coefficients for cylinder valves are obtained from a flow bench, they inherently contain all the flow losses present in the ports. Hence, friction pressure losses and pressure losses in the ports are disregarded by selecting the **No Friction Pressure Losses** and **Zero Pressure Losses from Bends and Tapers**.

Since a TPA model is used, in which the experimental intake, exhaust and cylinder pressure profiles are used as input, static pressure is being imposed and hence, when the valves are closed, pressure fluctuations are spuriously generated within the ports. In order for the actual pressure behavior is as similar as possible to the measure pressure, the magnitude of the fluctuations are reduced during the valve closed periods. This is possible by altering the friction multiplier within the last pipe attached to each *BoundaryPressureTPA* objects, which are the “int-runner-1” and “exh-runner-3”. The friction multiplier is normally set to 1 when the non-smooth option is selected for surface

finish. However, during valve closed periods, this multiplier is increased to 100 to damp these fluctuations, which is only possible using a control system. A *SensorConn* measures the lift of the intake and exhaust valves. This sensor is connected to a *Switch* actuator, which operates when the lift is lower than 0.001, that is, when the valve closes. This *Switch* actuator is connected to two *SignalGenerators*. When the valve is closed (**Output If Threshold Equation True**) and there is no air flow into the cylinder, the “fmult-high-in” or “fmult-high-ex” signal are sent to the switch and the actuator “friction-in” or “friction-ex” alter the friction multiplier from 1 to 100, damping the pressure fluctuations. When the valve is open (**Output If Threshold Equation False**), the “fmult-low-in” or “fmult-low-ex” signal are sent to the switch and the friction multiplier remains 1. The Figure A.6 and Figure A.7 **Fehler! Verweisquelle konnte nicht gefunden werden.** show the friction multiplier control scheme of the in the intake and exhaust systems, respectively.

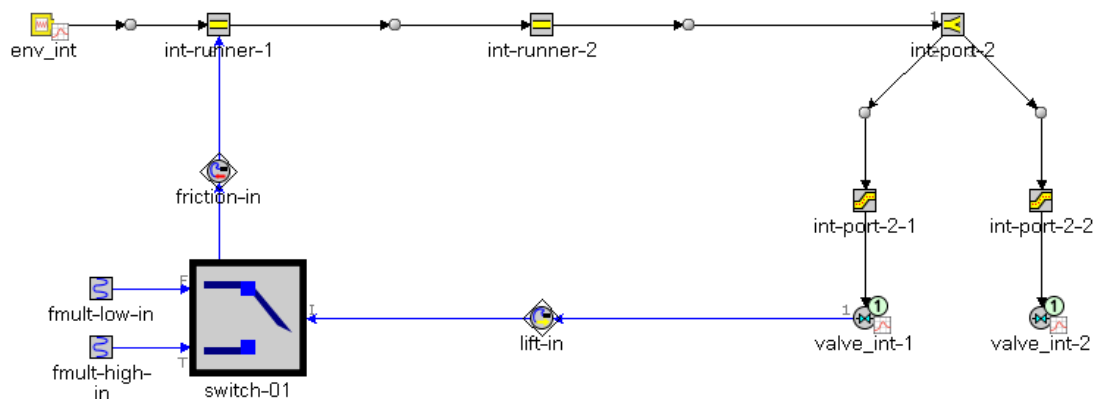


Figure A.6 - Friction Multiplier control system at intake

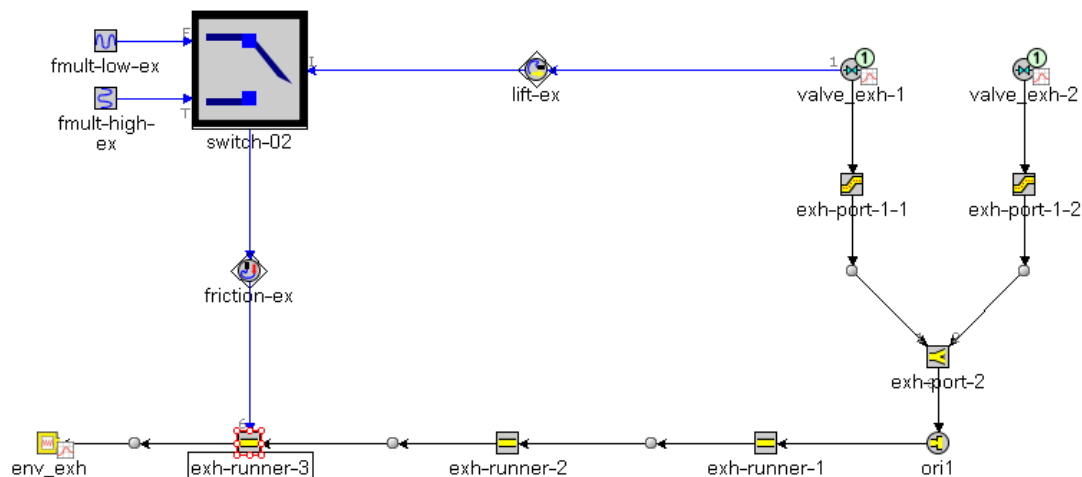


Figure A.7 - Friction Multiplier control system at exhaust

The heat transfer coefficient multipliers for the runners were set to 1, since when non-smooth surfaces were defined, additional heat transfer over a smooth pipe is already calculated. For intake and exhaust ports, the heat transfer coefficient of 2 was defined, as the surfaces were considered smooth.

Valve Characterization

Valves actuated through cam shafts are characterized using *ValveCamConn* objects. As the engine has four valves (two for intake and two for exhaust), four *ValveCamConn* objects are created.

The valve lift was experimentally measured and is characterized by a set of angles and lift arrays, in which the valve lift is measured at intervals of 2° CA. The *Cam Timing Anchor Reference* defined for the intake and exhaust valves is the firing top dead center (TDCFiring). For the intake valves, the *Cam Timing Lift Array Reference* is determined as an **arrayend**, positioning the valve profile in relation to its closure, and hence, the *Cam Timing Angle* corresponds to IVC. For the characterization of exhaust valves, the *Cam Timing Lift Array Reference* is the **startarray**, thus EVO is the *Cam Timing Angle*. The profile of the intake and exhaust valves, as well as the *Cam Timing Anchor* and *Cam Timing Lift Array References* are shown in Figure A.8 and Figure A.9, respectively.

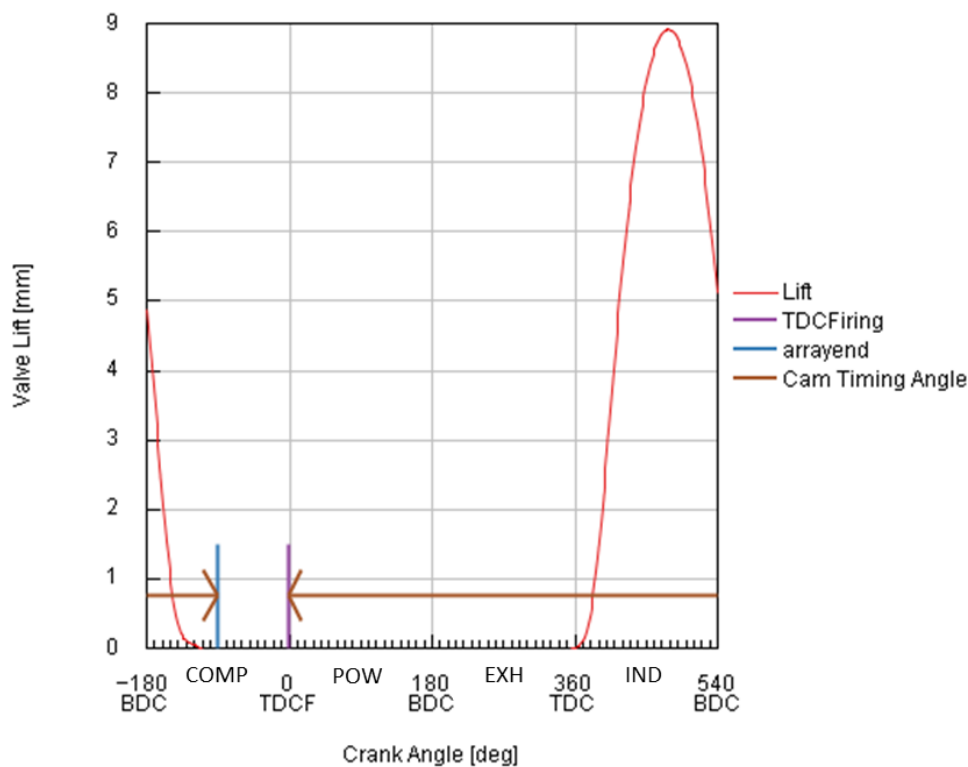


Figure A.8 - Intake valve lift

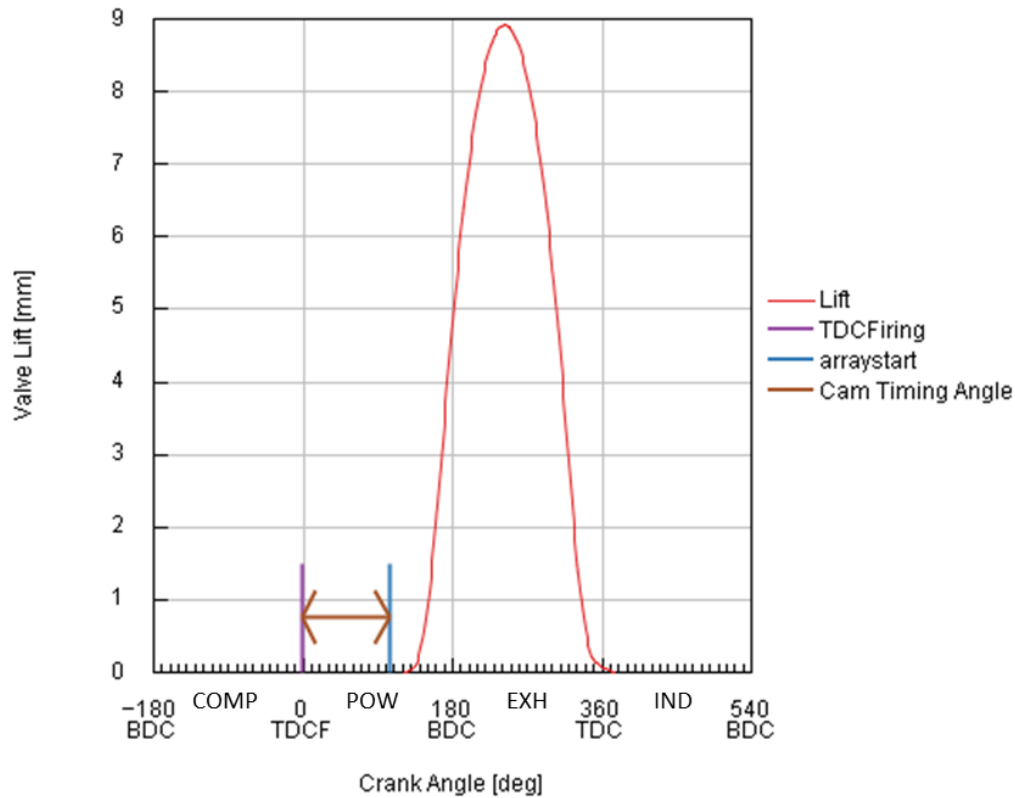


Figure A.9 - Exhaust valve lift

The intake and exhaust valve lash and timing are determined according to what was experimentally set, as described previously in Table 3.1 and Figure 3.1.

When providing the discharge coefficients, it is also important to inform the valve reference diameter, used to calculate the effective flow area from the discharge coefficients. This diameter does not need correspond to any physical valve attribute and, for the present model, the cylinder bore is used. The reference array to inform the forward and reverse discharge coefficients corresponds to the valve lift divided by *Valve Reference Diameter*. The intake and exhaust forward and reverse discharge coefficients were determined in previous studies by CFD simulations and are considered default for this engine, presented in Figure A.10 and Figure A.11, respectively. Such coefficients are normalized by the piston area

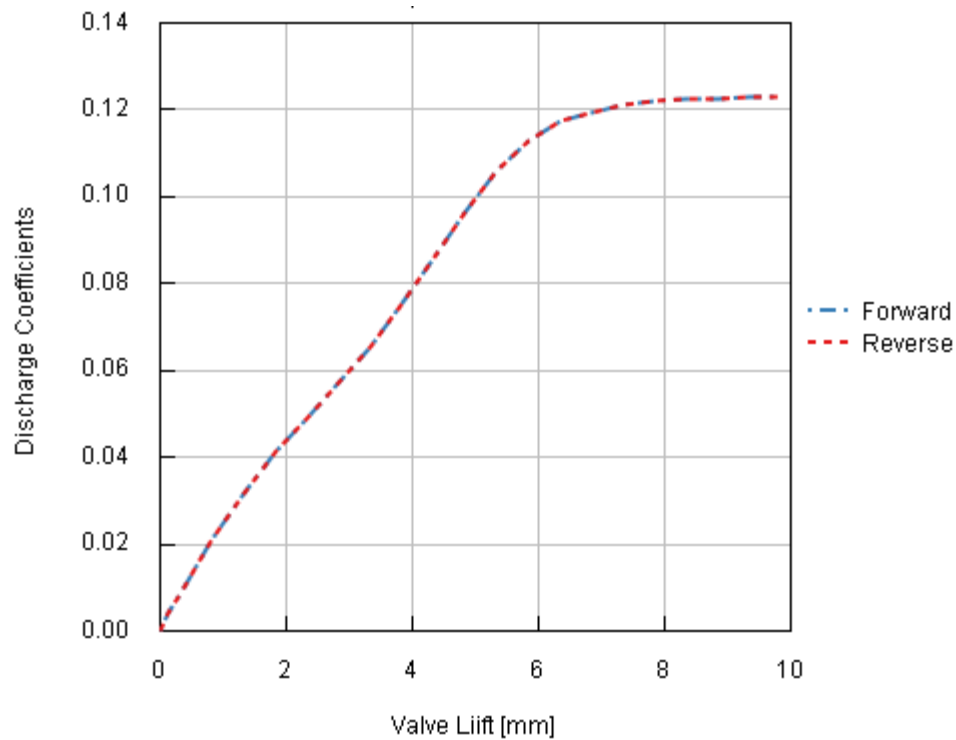


Figure A.10 - Intake valves discharge coefficients

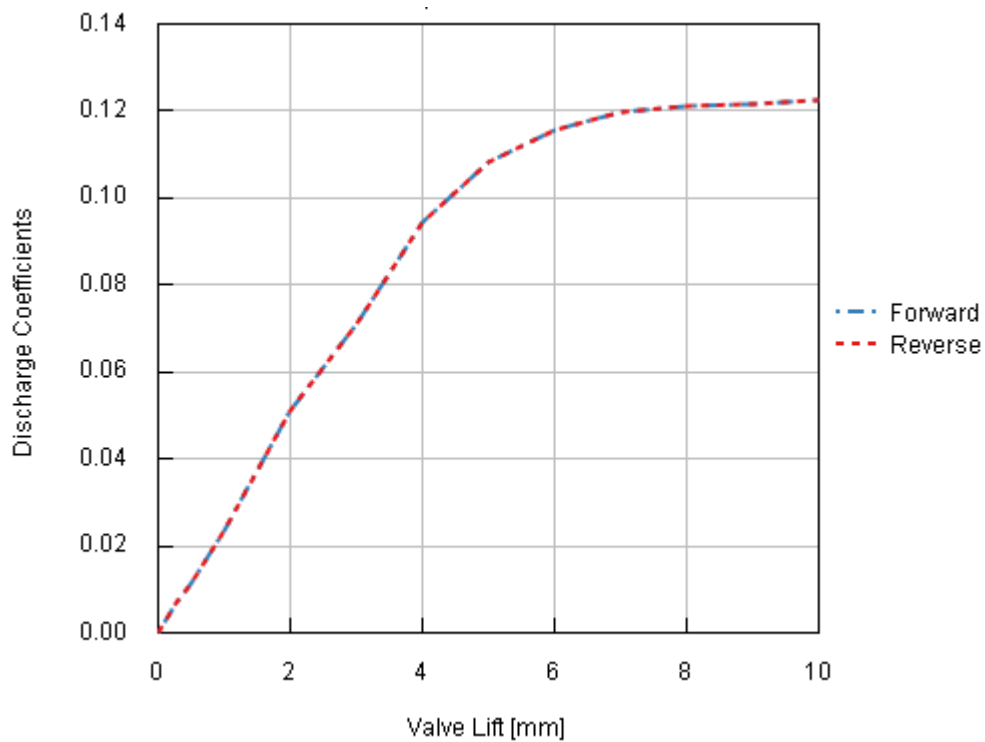


Figure A.11 - Exhaust valves discharge coefficients

Experimental and simulated data match in TPA analysis

One of the main points of attention in the TPA analysis are the experimental pressure data and it is important to consider the presence of pressure fluctuations due to pre-chamber operation also in the main combustion chamber modeling. In order for the cylinder pressure not to be over-predicted and prior to EVO, the noise in the cylinder pressure data

should be eliminated since it leads the burn rate calculation to oscillate above and below zero after combustion has ended. Since no negative burn rates are allowed in the calculations, the negative portions are truncated by the software, which effectively creates a spurious tail in the burn rate profile, as shown in Figure A.12.

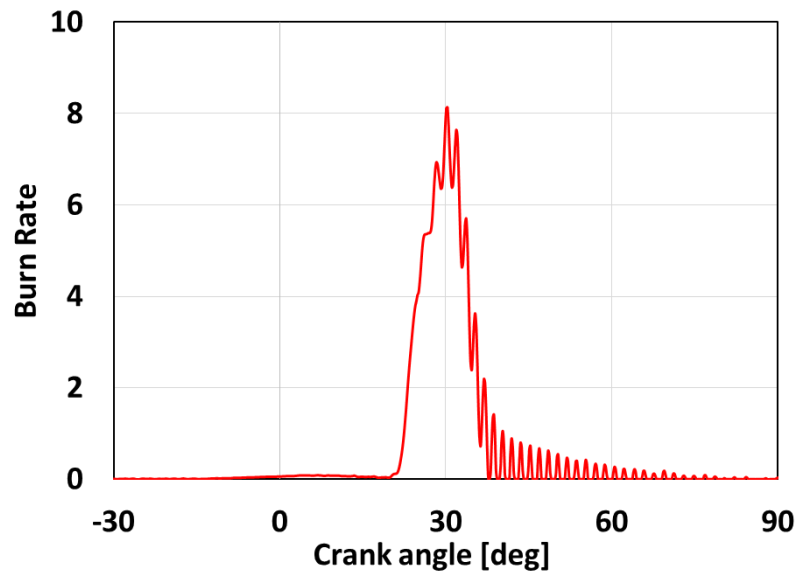


Figure A.12 - Burn Rate normalized by total fuel mass without filtering

Hence, in this modelling, the cylinder pressure was smoothed by using a Low Pass Filter with a cutoff frequency of 3 kHz, eliminating the pressure oscillations. Hence, the match between simulated and measured cylinder pressure during expansion stroke is significantly improved, as presented in Figure A.13. In addition, it should be noted that the pressure data used for the simulations are the average of the 200 cycles recorded by the pressure analysis system.

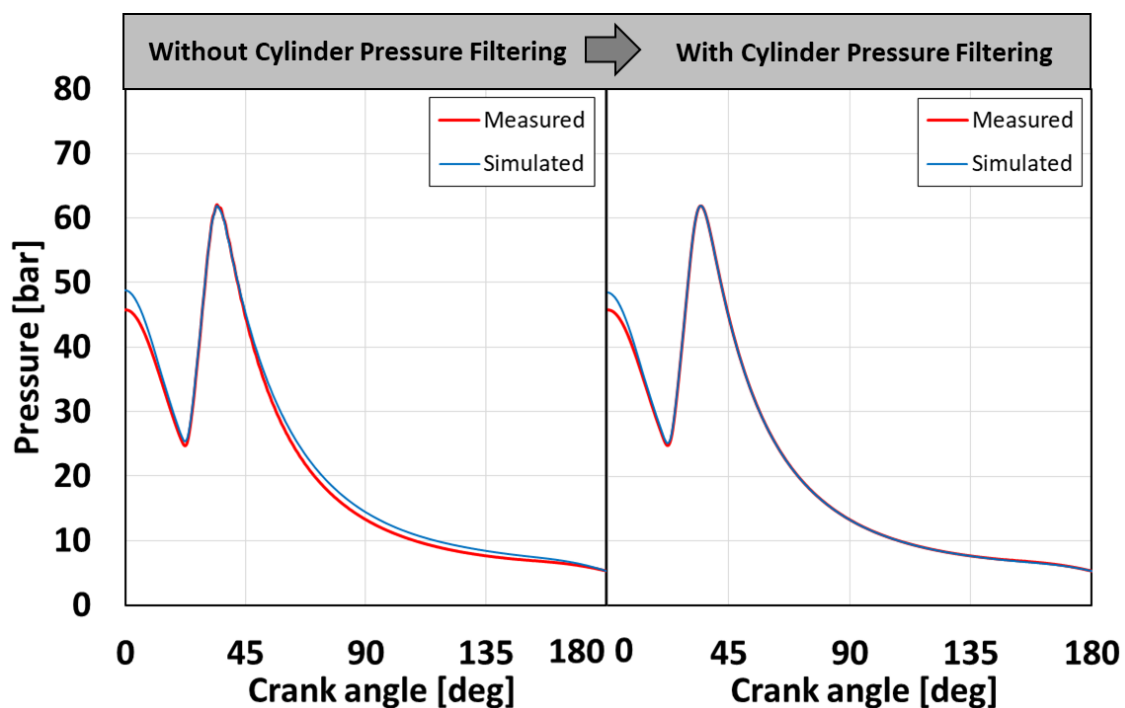


Figure A.13 - In-cylinder pressure before and after the cylinder pressure filtering

In order for the pressure curves calculated by the software to be as close as possible to the experimentally measured curves in the TPA analysis, it is necessary to calibrate the model. For this calibration, the problem is divided into two steps: calibration of the gas exchange process (from EVO to IVC) and the high pressure process (from IVC to EVO). An example of this calibration step is presented in Figure A.14. In addition, another necessary calibration is related to the lambda. The 3 calibration steps are described in the next subsections.

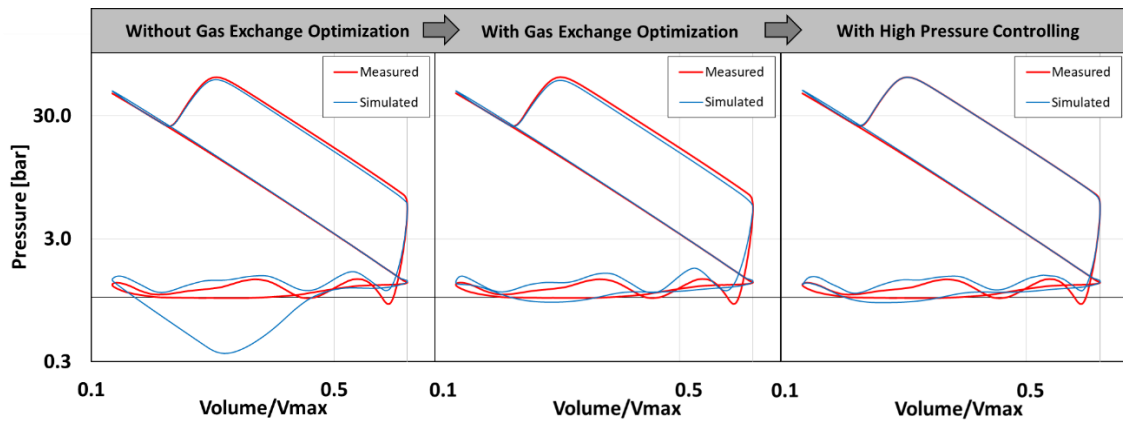


Figure A.14 - Steps of cylinder pressure calibration

Calibration of in-cylinder pressure during gas exchange

For the engine tested, the valve timing is manually determined when assembling the engine for the experiment, but due to uncertainties arising from the assembly process, a difference of a few crank angle degrees between the previously determined and the operating is observed. Thus, for the simulated and measured results to be tuned, the valve timing was optimized, in which adjustments by a few degrees were carried out manually, so that the mass of air admitted and the pressures in the cylinder were adequate. Figure A.15 is shown as an example of the valve timing adjustments, in which the pressure drops are tuned for exhaust valve timing and pressure rises are also tuned to adjust the intake valve timing.

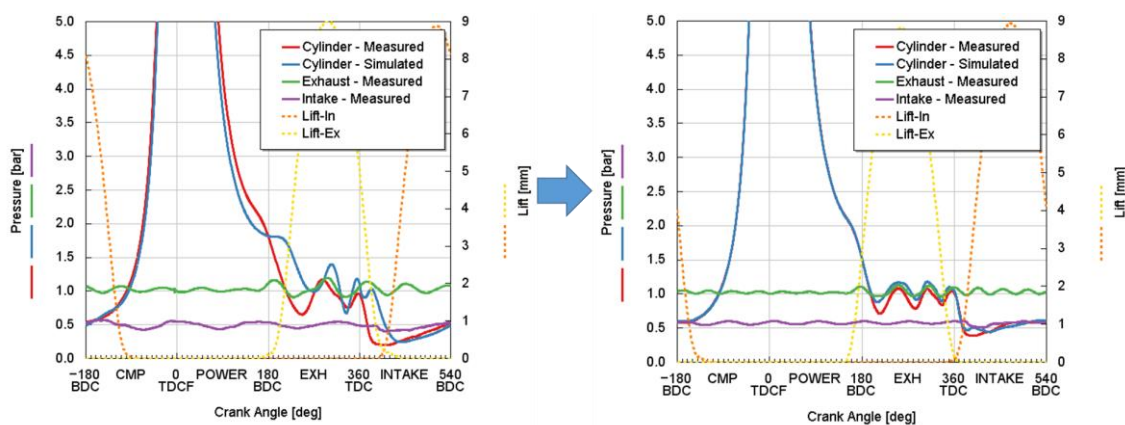


Figure A.15 - Effect of valve timing adjustments

Furthermore, for finer adjustment of the cam timing angle, controllers were implemented, since they can impact volumetric efficiency, lambda and in-cylinder pressure. For this control, a subassembly within the GT-power model is created for a more detailed analysis of the gas exchange close to EVO and IVC. The signals to be received and controlled are

obtained through a *ReceiveSignal* object, the “rec_post_processing_1”, which receives wireless signals from the main tab, where the engine model and its adjacent systems are presented. In this object, 3 output signals are selected, the local cylinder angle (wrapping at start of cycle), the cylinder pressure measured during experimental tests and the apparent cylinder pressure, acquired through a *Lookup 1D table* “p_cyl_meas-1” connected to the cylinder. In the *Lookup 1D table*, the cylinder pressure used to characterize the Measured Cylinder Pressure Analysis Object is presented as an independent variable.

Through “rec_post_processing_1” two sets of signals are formed, the first with the crank angle associated with the measured cylinder pressure and the second with the crank angle associated with the apparent cylinder pressure. Each set of signals is used as input in 16 *SignalHold* objects, named “DISCRETE-VAL*”. The *SignalHold* template is used to capture and hold a certain value of a signal, capturing some sensible quantity over a specific time period or angle duration. Through the *SignalHold* objects it is possible to discretize the cylinder pressure during gas exchange in several intervals, allowing a deeper analysis of each one of them.

In order to characterize the “DISCRETE-VAL*” objects, in addition to the inputs (crank angle and pressure), it is necessary to determine the Signal-Pass Minimum Threshold and the Signal-Pass Maximum Threshold, which represent the lower and upper bounds of the signal-pass range applied to the Control Input signal respectively. The Signal-Pass Minimum Threshold is set individually for each variable described through the output signals set. The Hold Style on the *SignalHold* object is defined as **first_value**, which indicates that the Input will be passed to the Output at the first timestep when the Control Input enters the signal-pass range. Since the **first_value** option is chosen, the Signal-Pass Maximum Threshold is set for all cases to 720, just to guarantee that any Signal-Pass Minimum Threshold angles would be analyzed.

A signal generator defines the start of the gas exchange process, which takes place before EVO, at 180° aTDC. This signal is used as the Signal Pass Minimum Threshold for the first two sets of discretized variables, in the same way that the end of the gas exchange process, which occurs after IVC, at 70° bTDC is used as the output for the last set of discretized variables. Since the gas exchange cylinder pressure is discretized into 16 variables, the difference between the duration of the gas exchange is divided into 15 parts, thus calculating the gain to be incremented between discretized variables. Therefore, each discretized variable analyzes the pressure in the cylinder at specific points at intervals equivalent to the gain, during gas exchange.

The tab which discretizes the in-cylinder pressure during gas exchange is presented by Figure A.16. The post processing of the gas exchange data is represented in the main GT-Power subassembly object “PostProcessing-GE”, presented in the figure’s left upper corner.

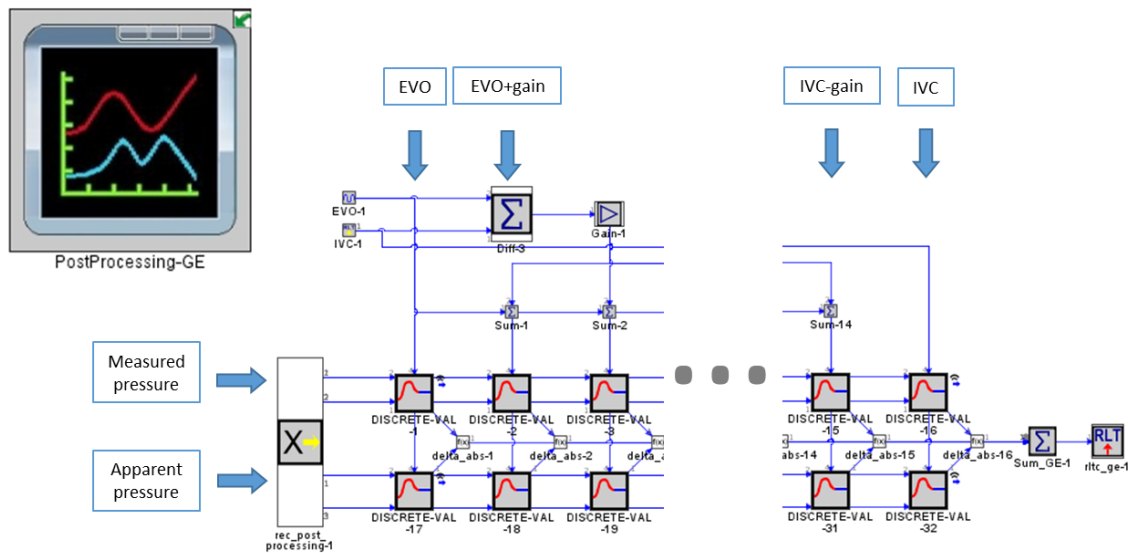


Figure A.16 - Post-processing subassembly of the cylinder pressure during the gas exchange process

The measured and apparent pressures analyzed in the last discretized variable before the IVC, “DISCRETE-VAL-16” and “DISCRETE-VAL-32”, are used as Target for Input Signal and input signal for the PID controller, respectively. A PID controller is then used so that the input reaches the target input by means of varying the cam timing angle, the controller's output, as shown in Figure A.17. The same control is performed in exhaust, but in this case, the first discretized variables after EVO, “DISCRETE-VAL-1” and “DISCRETE-VAL-17, are considered, presented in Figure A.18.

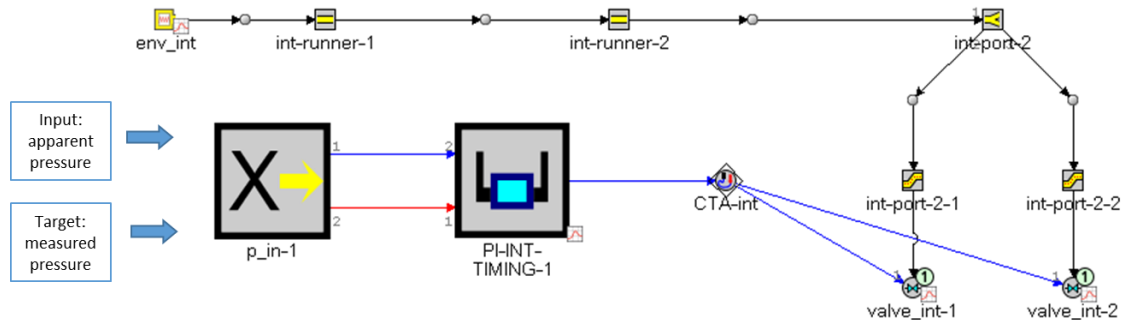


Figure A.17 - PID control of the intake cam timing angle

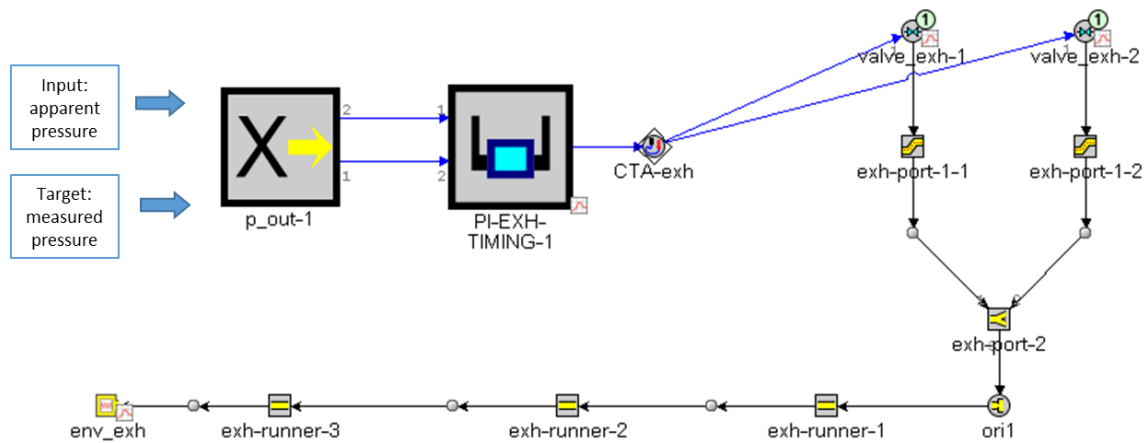


Figure A.18 - PID control of the exhaust cam timing angle

Calibration of in-cylinder pressure between IVC and EVO

For in-cylinder pressure calibration between IVC and EVO another procedure was necessary. Huegel et.al, studying heat losses in homogeneous and stratified DI-SI engines, compared the in-cylinder heat flux measured experimentally and calculated by several predictive models, such as Bargende, Woschni and Hohenberg. As presented in Figure A.19 **Fehler! Verweisquelle konnte nicht gefunden werden.**, the Woschni model predicts too small values in the compression phase, which are compensated by too high values during the combustion. In order for this error to be corrected, in the present model the overall convection multiplier, initially set to 2, was tuned to match IMEP and cylinder pressure between simulation and experiment [189].

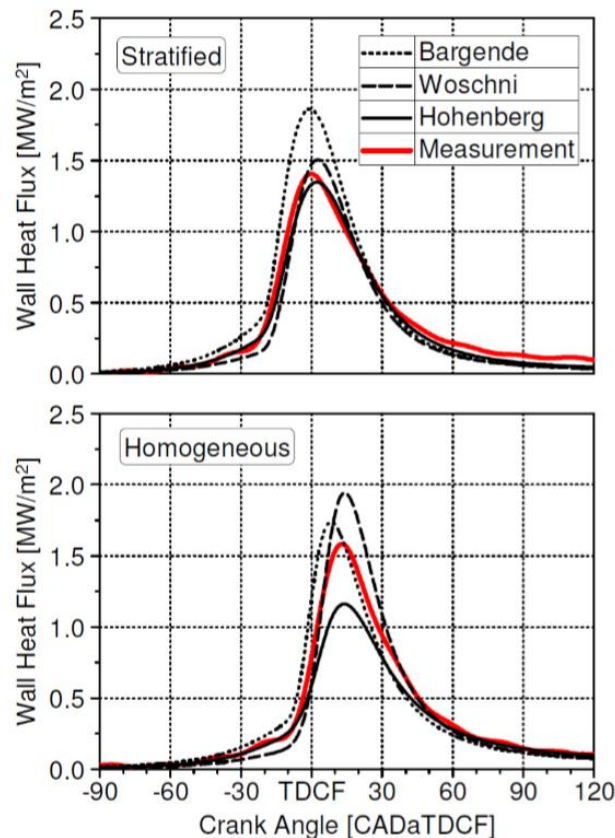


Figure A.19 - Comparison of the weighted average wall heat flux and the prediction of several wall heat transfer models for stratified operation (top) and homogeneous operation (bottom) [191]

For the measured and simulated pressure curve to be tuned, it is necessary to determine more than one convection multiplier, since the thermodynamic engine cycle presents very different dominant phenomena. Therefore, the cycle is divided and analyzed during gas exchange, combustion and expansion.

The control of the convection multiplier is represented according to Figure A.20 **Fehler! Verweisquelle konnte nicht gefunden werden.**, which is divided into 3 areas: controlling; determination of operation points, and the profile generator. By determining the operation points, the relevant crank angles for each profile are determined and the signal is generated as linear transitions through the interpolator are performed. The signal profile is generated by an If block and the signal with the appropriate convection multipliers is sent through a *SendSignal* object to the cylinder object. This method is interesting since optimizations with multiple iterations are not necessary.

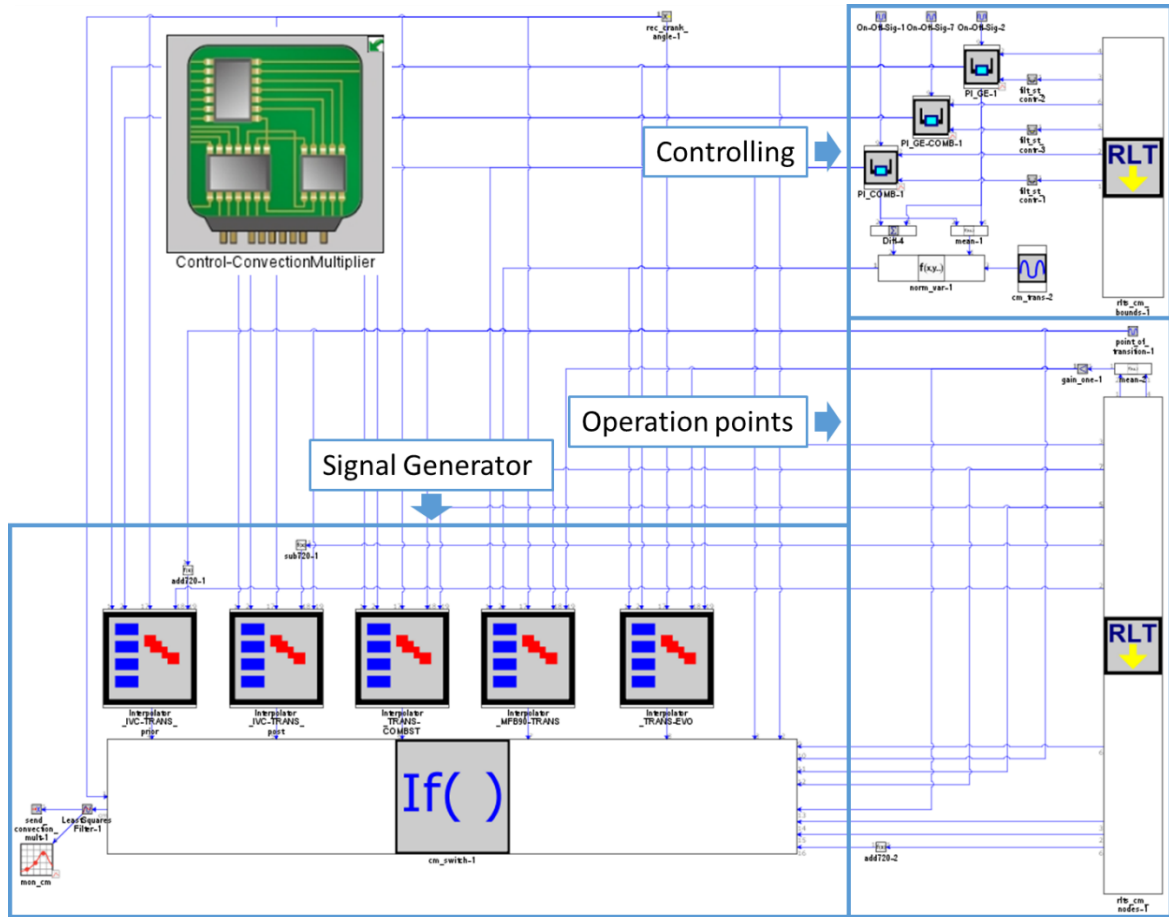


Figure A.20 - Main Chamber Convection Multiplier Control

For convection multiplier control 3 controllers are used, “PI_GE-1”, “PI_GE-COMB” and “PI_COMB-1”, as presented in Figure A.21. The “PI_GE-1” is focused on gas exchange, has as input the apparent lambda during simulations and the lambda measured in experiments as a target for input signal and presents as output the convection multiplier used during the gas exchange, CM_{GE} . The “PI_GE-COMB-1” is focused on the compression period, uses as input the pressure at FTDC, measured during simulations, and the pressure at FTDC measured during experiments as a target for input signal, and as an output the $CM_{GE-COMB}$ is obtained. The “PI_COMB-1” is focused on combustion, it uses as input the maximum pressure of the cylinder measured during the simulation and the maximum pressure during the experiments as a target for input value and as output the CM_{COMB} is achieved. All signals were obtained from a *RLTSensor* named “rlts-combounds-1”. A fourth convection multiplier is also determined from the controller outputs “PI_GE-1” and “PI_COMB-1”. Their mean value a and their difference b , determined by Equations A.7 e A.8 are used to determine the intermediate convection multiplier CM_{inter} , defined in Equation A.9.

$$a = \frac{CM_{GE} + CM_{COMB}}{2} \quad A.7$$

$$b = CM_{GE} - CM_{COMB} \quad A.8$$

$$CM_{inter} = a + 0.5 * b$$

A.9

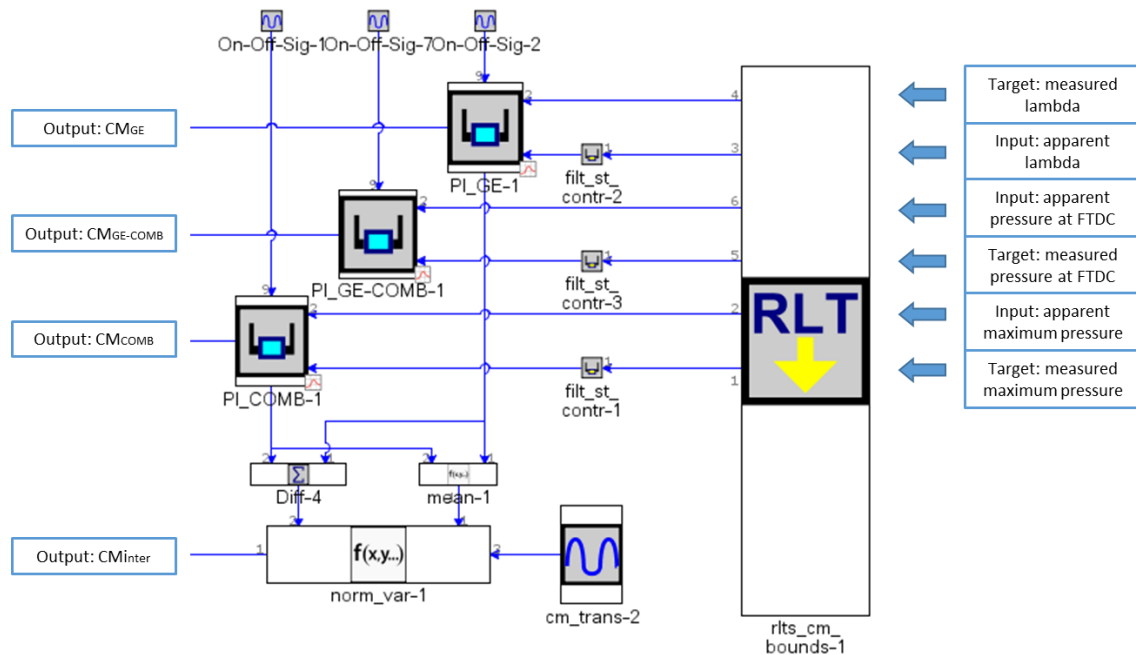


Figure A.21 - Detailed controlling area of the convection multiplier controller

A second RLTSensor named `rlts_com_nodes-1` is used to determine the points at which certain events happen, in order to determine where each convection multiplier will be applied. The sensor collects the **Exhaust Valve-Open Timing Angle at Input Reporting Lif** (`vtvo`), which is at 2mm height, the **Intake Valve-Closing Timing Angle at 0mm Lift** (`vtvc`), the **Exhaust-Max Lift Timing Angle** (`vtmop`), the **Exhaust Open Timing Angle at 0mm Lift** (`vtvo0`), the **Crank Angle at 2% Burned Fuel** (`pacomb02`) calculated using measured data, the **start of cycle** (`st-cycle+`), determined manually and the **Crank Angle at Combustion End (99% of fuel burned)** (`pacombend`), also obtained through measured data.

For the convection multipliers to be well adapted, a transition between values from one point to another is built and for that, five Interpolator objects are used. An interpolator template produces an output signal by comparing the control signal to the look up array values defined in this case by external signals. The interpolators used are the “Interpolator_IVC_TRANS_prior”, “Interpolator_IVC-TRANS_post”, “Interpolator_TRANS-COMBUST”, “Interpolator_MFB90-TRANS” and “Interpolator_TRANS-EVO” and Table A.9 shows the look-up and output values used in the configuration of each one, in addition to the outputs obtained. The control signal used is the crankangle and for each interpolator a range of crank angles is analyzed. The interpolation mode chosen is `interpolate_linear` and the look-up search direction is forward, which indicates that the interpolation search in the *Look-up Array* will start with *Lookup Value 1*. Therefore, a linear transition is made between the look up signal values 1 and 2 as the analyzed crankangle grows from output signal value 1 to 2 (corresponding to the analyzed range).

Table A.9 - Interpolators configuration

Interpolator	Look-up Value	Output Signal Value	Control signal	Output signal
--------------	---------------	---------------------	----------------	---------------

Interpolator_IVC- TRANS_prior	CM-PI_GE CM-PI_GE- COMB	vtvc0 CKA-TRANS- GE- COMB+720	Crank angle signal	CM_GE- TRANS prior SOC
Interpolator_IVC- TRANS_post	CM-PI_GE-1 CM-PI_GE- COMB	vtv0-720 CKA-TRANS- GE-COMB		CM_GE- TRANS post SOC
Interpolator_TRA NS-COMBUST	CM-PI_GE- COMB CM-PI-COMB	CKA-TRANS- GE-COMB Pacomb02		CM_TRANS- COMB
Interpolator_MFB 90-TRANS	CM-PI-COMB CM_inter	pacombend (vtvo+vtv0)/2		CM_COMB- TRANS
Interpolator_MFB 90-TRANS	CM_inter CM-PI_GE	(vtvo+vtv0)/2 vtmop		CM_TRANS GE

Here it is important to note that the point $(vtvo+vtv0)/2$, ie, the **Exhaust Valve Open Timing Angle at 1 mm Lift**, is chosen since it is where a large difference between measured and simulated pressure is observed. Hence, this is the best point for the incidence of a specific convection multiplier. At intake valve opening not much pressure difference is observed the **Valve Closing Timing Angle at 0mm and at 2mm Lift**, so the 0mm lift was arbitrarily chosen.

The logical operator *IfThenElse*, named “cm_switch-1”, is used to implement conditional programming into a model. Table A.10 **Fehler! Verweisquelle konnte nicht gefunden werden.** shows the program performed to determine the use of each convection multiplier, either calculated directly from the PI controllers or the transition convection multiplier calculated with the aid of interpolators.

Table A.10 - *IfThenElse* logical operator setup to define convection multipliers

Action Descriptions	Actions	Conditions	Crankangle-resolved Convection multipliers
START__TRANS- GE-COMB	If	ca>START & ca≤cka_TRANS_GE_COMB	CM _{GE_TRANS_post}
TRANS-GE- COMB__MFB2	Elseif	ca>cka_TRANS_GE_COMB & ca≤MFB2	CM _{TRANS_COMB}
MFB2__MFB90	Elseif	ca>MFB2 & ca≤MFB90	CM _{COMB}
MFB90__TRANS- COMB-GE	Elseif	ca>MFB90 & ca≤cka_TRANS_COMB_GE	CM _{COMB_TRANS}
TRANS-COMB- GE__EVO	Elseif	ca>cka_TRANS_COMB_GE & ca≤EVO	CM _{TRANS_GE}
EVO__IVC	Elseif	ca>EVO &	CM _{GE}

		ca≤IVC	
IVC__START	Elseif	ca>IVC	CM _{GE_TRANS_prior}
		& ca≤START_pos	
Else		1	

The logical operator “cm_switch-1” is then connected to a *LeastSquaresFilter*, which computes the least squares filtered output from an input signal. It is used in this case to smooth the convection multiplier signal before it is fed into a *SendSignal* template called “send_convection_multi-1”, which feeds this signal to the cylinder, calibrating the in-cylinder pressure. Figure A.22 **Fehler! Verweisquelle konnte nicht gefunden werden.** shows an example of the plot provided by the *MonitorSignal* connected to the “cm_switth-1” object, indicating all the convection multipliers and where each one is applied in the cycle. The start point is arbitrarily defined as being a point between IVC and the transition point, so that the simulation converges faster. The transition point is defined as the point in compression time where maximum convection multiplier is observed in an engine cycle, corresponding to approximately -45 CA.

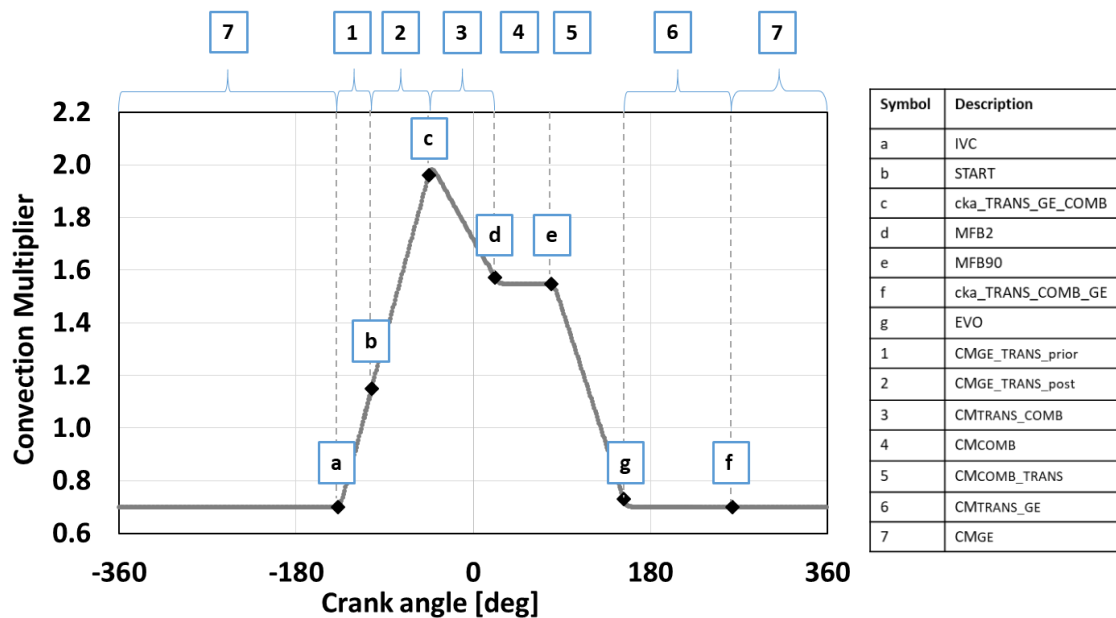


Figure A.22 - Convection Multiplier Control Signal

It should be noted that the corrections made from the variation of the convection multiplier should be made mainly focusing on the compression stroke, since it is underestimated but the predictive Woschni model [191]. The GT-Power software determines that the convection multiplier must vary between 0.7 and 2, being these values used as lower and upper limits, respectively, of the controllers previously described. However, it is possible to observe at Figure A.23 that the pressures calculated by the software and measured are still not quite equivalent, especially in compression and in the period after the EVO.

It is possible that such incompatibility with compression pressures represents an experimental error due to lower accuracy of the cylinder pressure transducer at lower pressures or due to thermal shock of the pressure transducer. However, if the convection multiplier were indiscriminately increased so that the two curves are perfectly tuned, a

much better match would be possible, despite having no physical sense, as shown in Figure A.23.

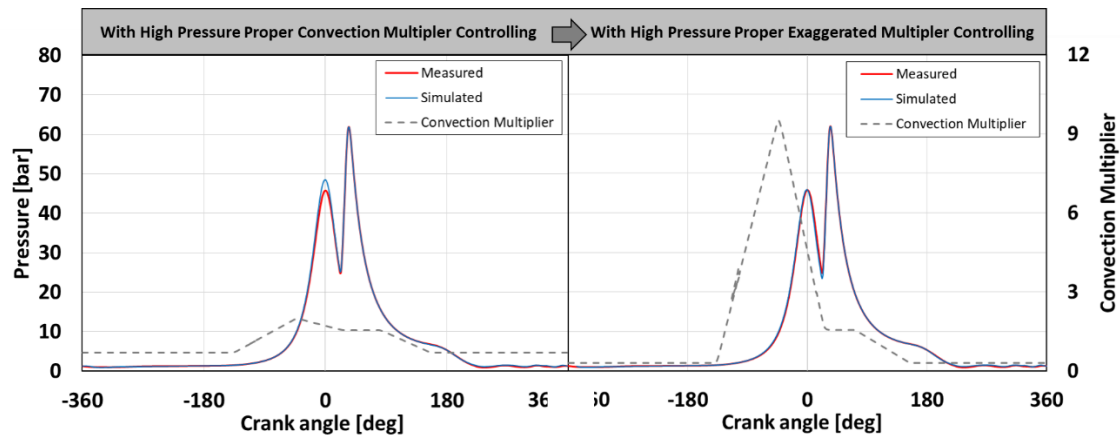


Figure A.23 - Comparison between Simulated and Measured Pressure Curve Tuning using different Convection Multiplier values

Lambda calibration

In addition to calibrating the cylinder pressures during gas exchange and high pressures, another necessary calibration is related to the lambda. Even though the intake port heat transfer multiplier is initially set for the intake ports, the parameter was tuned to match the experimental volumetric efficiency, controlling the calculated lambda [189]. With an *RLTsensor*, the apparent lambda, measured in the crank train throughout the simulation, and the lambda variable measured during the experiments are acquired. The apparent lambda variable must match the lambda variable of the experiments for the simulation to be coherent. Therefore, the apparent lambda variable is passed through a first-order filter to remove signal oscillations and is connected to a *PIDcontroller*, representing the input. The measured lambda variable represents the target to this input value in the controller. In order for the input value to reach the target value, the heat transfer multiplier in the intake port last flow component is controlled. Therefore, its initial value of 2 is varied.

To characterize the *PIDcontroller*, a procedure described in detail in the GT-Suite tutorial for controllers is performed. Initially, a nominal condition was chosen, since the gains for the controller can be found at one operating condition and then used for all other operating conditions. To represent the whole operating range, since the experiments were performed from lambda 1.0 to lambda 1.9, an intermediate lambda of 1.5 was chosen. For this condition, the apparent lambda was measured with the *RLTsensor*, whose plots were activated. The output curves obtained through this measurement are analyzed using an auxiliary excel file provided within the GT-suite software, intended to characterize controllers.

In this file, the maximum and minimum outputs are provided and considering that the default output is 2, the **Pre- and Post-Step Input Signals** are defined as 1 and 3. In addition, the data obtained with the apparent lambda measurement are provided, in time periods that must be specified. From such data and the definitions of the *setting time* and *damping ratio*, the proportional and integral gains of the proportional–integral–derivative (PID) controller are defined. The *damping ratio* is set to 1 and the *settling time* is 3 times the time constant, calculated in the excel file with the aforementioned inputs, as recommended by the software. Derivative gain was ignored, as is commonly done, since

it is usually unnecessary, decreasing the calculations complexity. The association of the PID controller with the intake ports is shown in Figure A.24.

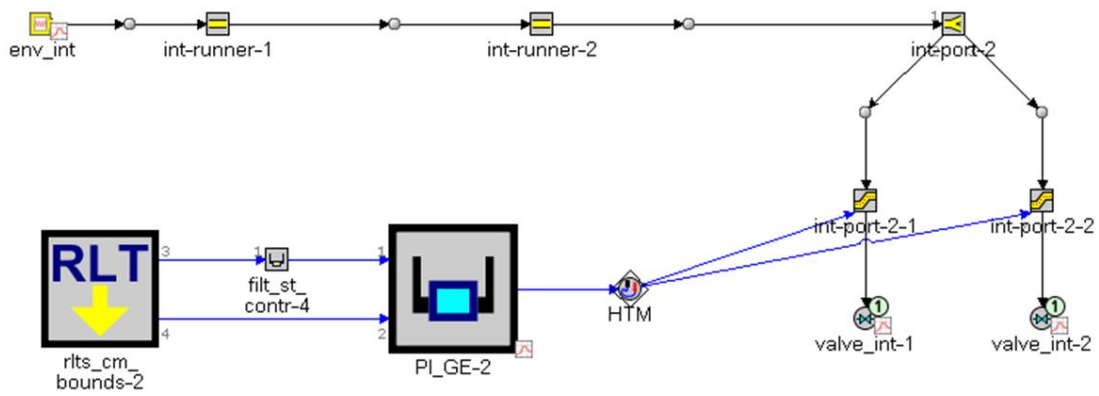


Figure A.24 - Lambda PDI control

Appendix B. TPA CONVERSION

As determined in the methodology section, the GT Power models are validated once the simulated data reached IMEP and lambda correspondence with at least 95% reliability. Figure B.1 and Figure B.16 to show measured experimentally and simulated pressure curves and the required correspondence.

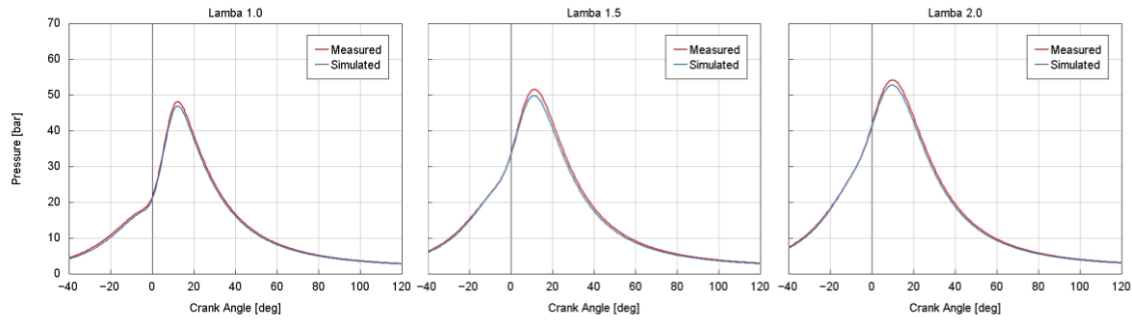


Figure B.1 - 2000 rpm and 6 bar of IMEP - SI-MeOH

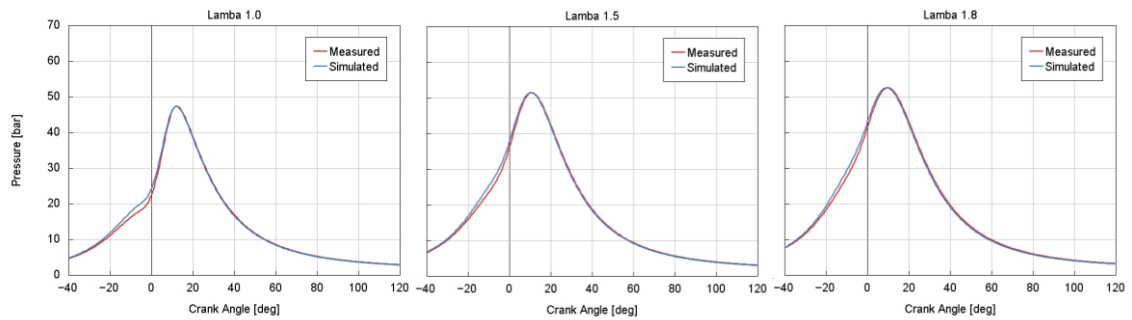


Figure B.2- 2000 rpm and 6 bar of IMEP - SI-EtOH

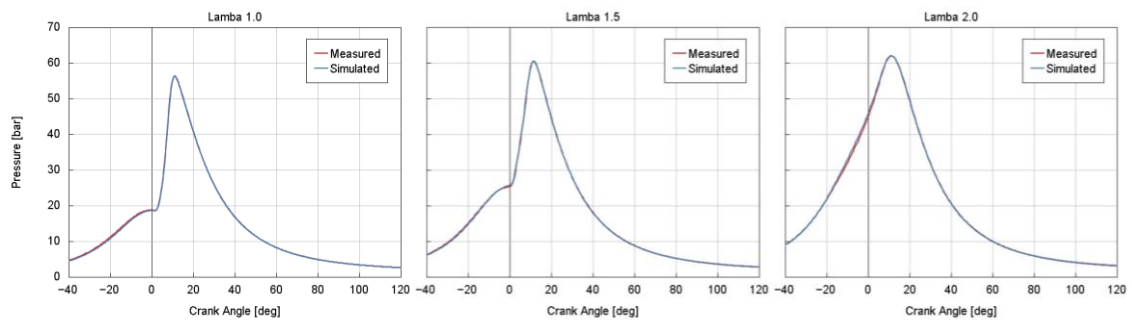


Figure B.3 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

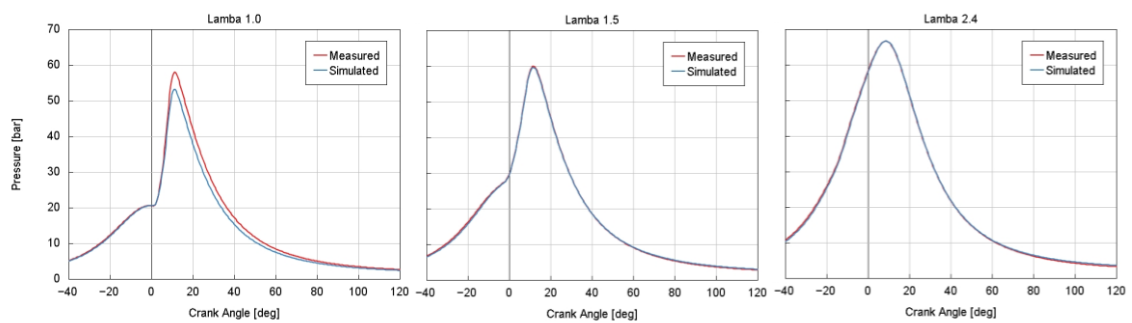


Figure B.4 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

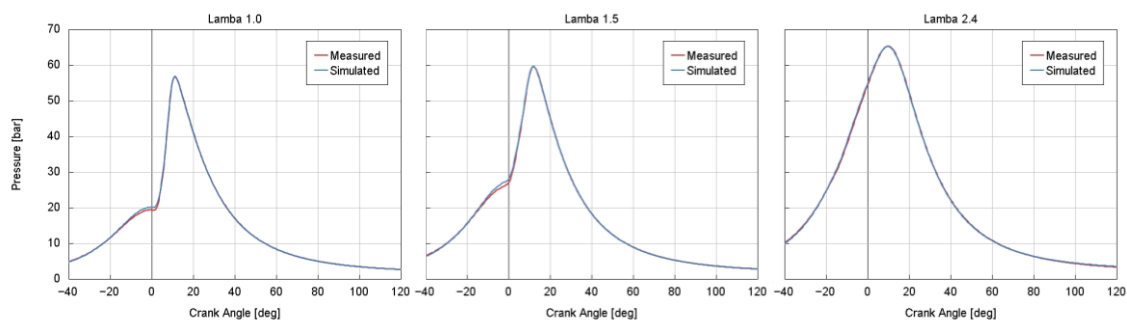


Figure B.5 - 2000 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

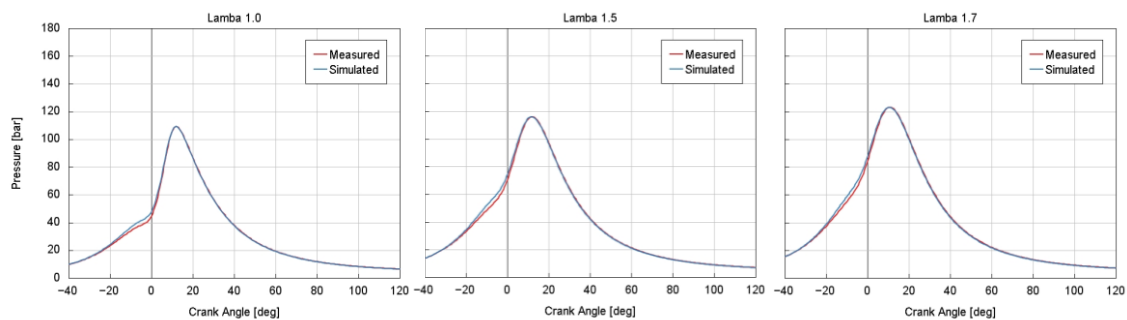


Figure B.6 - 2000 rpm and 15 bar of IMEP - SI-EtOH

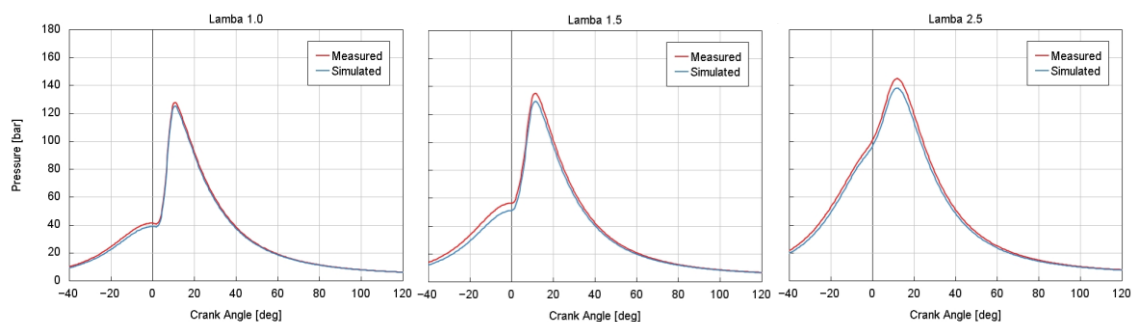


Figure B.7 - 2000 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

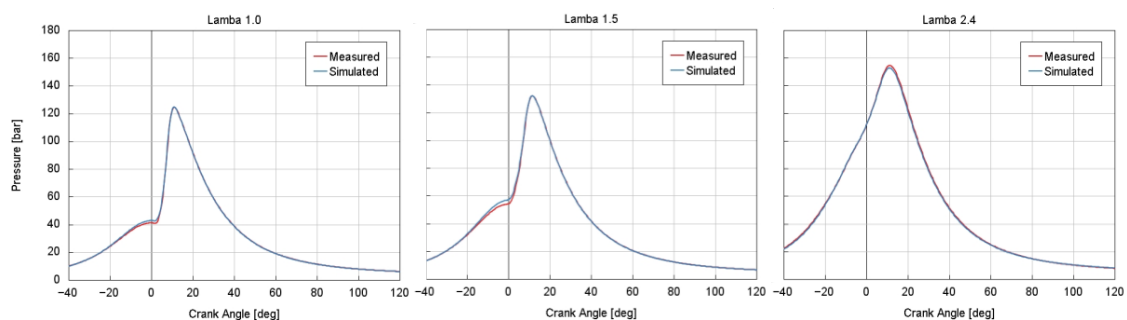


Figure B.8 - 2000 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

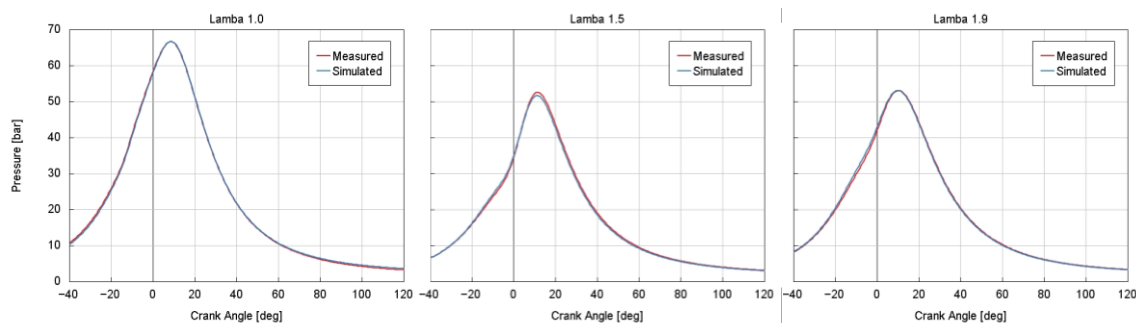


Figure B.9 - 3500 rpm and 6 bar of IMEP - SI-MeOH

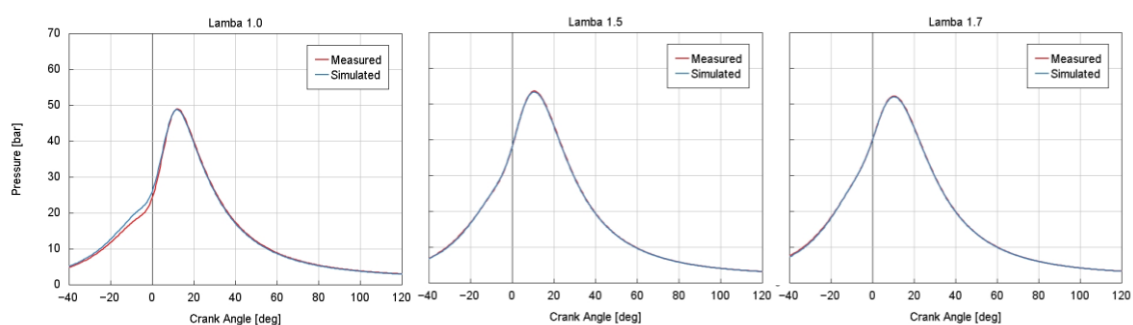


Figure B.10 - 3500 rpm and 6 bar of IMEP - SI-EtOH

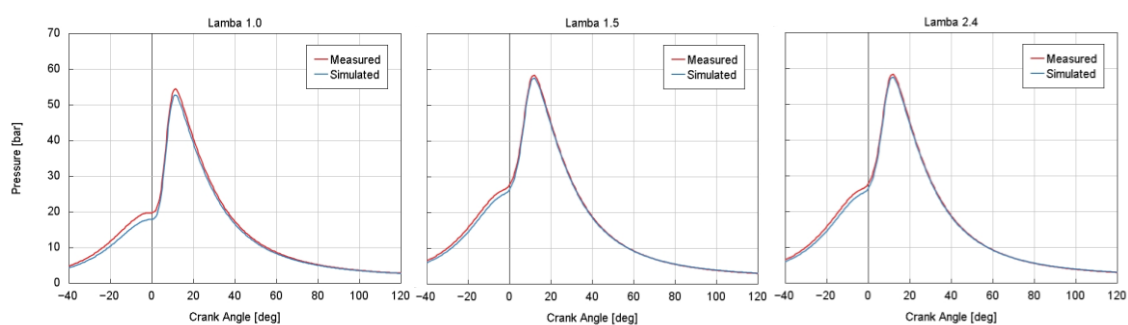


Figure B.11 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

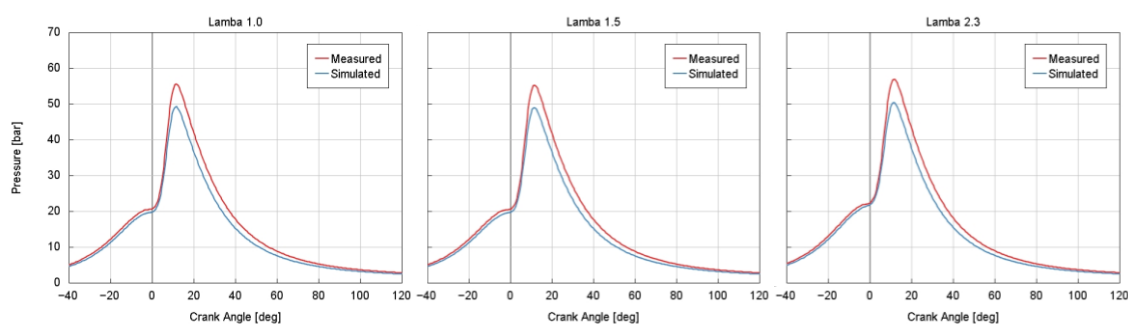


Figure B.12 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

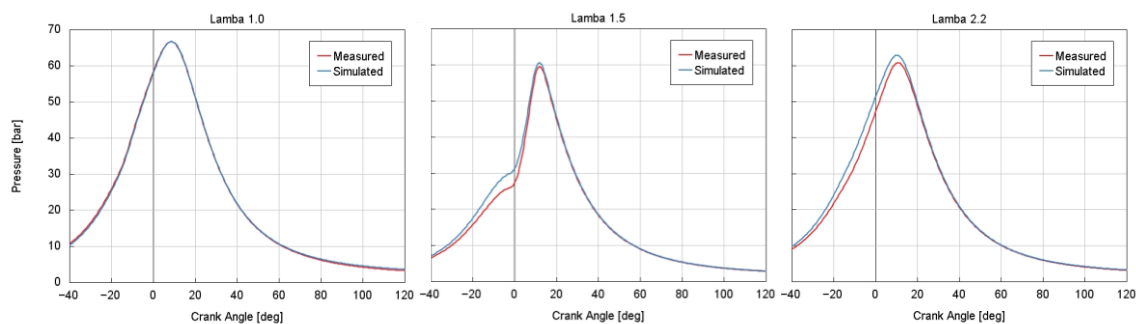


Figure B.13 - 3500 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

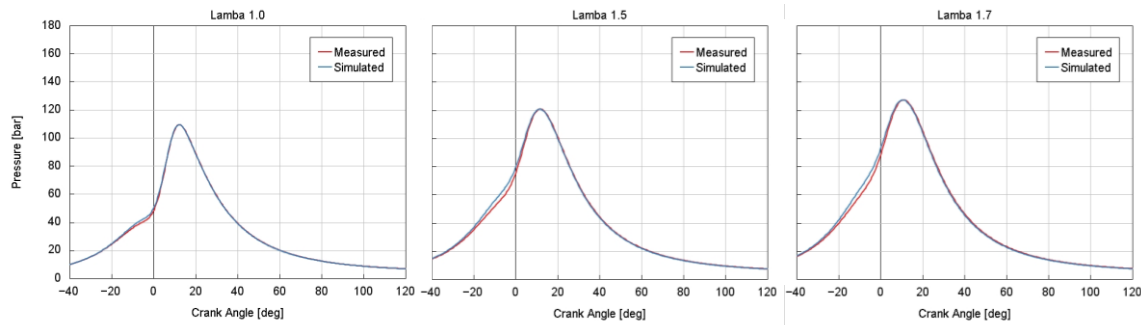


Figure B.14 - 3500 rpm and 15 bar of IMEP - SI-EtOH

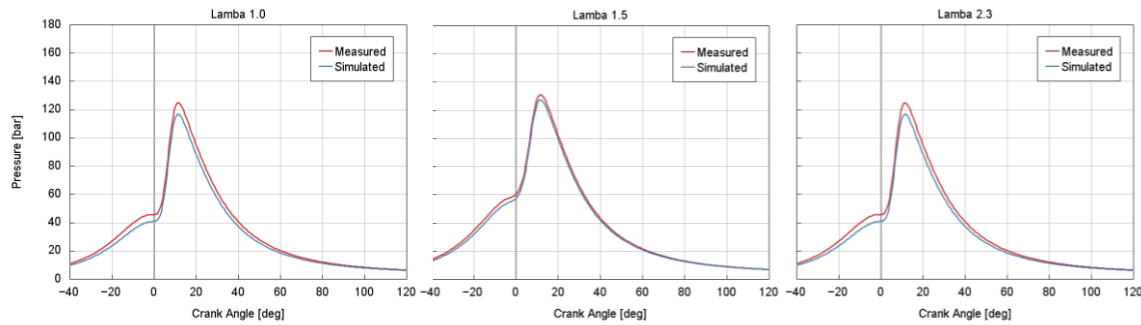


Figure B.15 - 3500 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

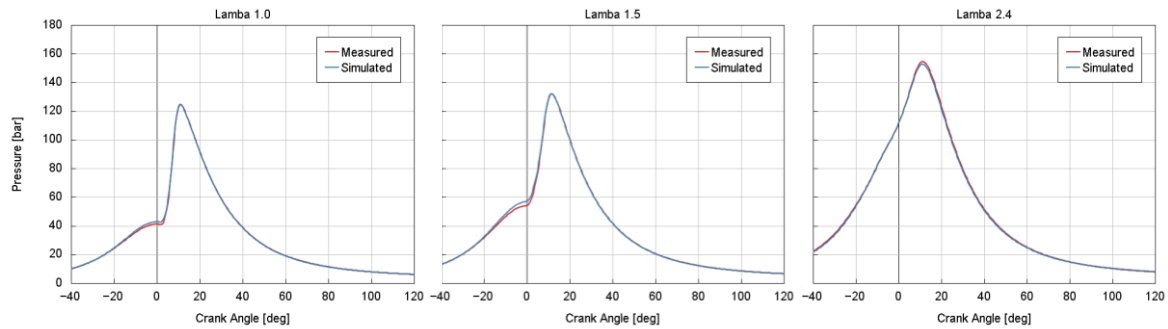


Figure B.16 - 3500 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

Table B.1 shows the values of IMEP, λ and LHV multiplier obtained in the simulations for all configurations and operation points tested to prove their convergence.

Table B.1 - Simulated IMEP, Lambda and LHV Multiplier

	Experimental Data			Simulated Data		
	Engine Speed (rpm)	IMEP (bar)	λ	IMEP (bar)	λ	LHV Multiplier
SI-MeOH	2000	6	1.0	5.69	1.01	1.0
			1.5	5.42	1.48	1.0
			2.0	5.21	1.90	1.0
SI-EtOH			1.0	5.59	1.01	1.0
			1.5	5.50	1.50	1.0
			1.8	5.29	1.77	1.0
MC-MeOH/PC-MeOH			1.0	5.88	0.99	1.9

			1.5	5.87	1.48	1.0
MC-MeOH/PC-EtOH			2.4	5.74	2.42	1.0
			1.0	4.92	0.8	1.0
			1.5	5.98	1.03	1.0
			2.4	5.47	1.67	1.0
MC-EtOH/PC-EtOH			1.0	5.88	1.01	1.0
			1.5	5.87	1.45	1.0
			2.4	5.91	2.28	1.0
SI-EtOH		15	1.0	14.49	1.01	1.0
MC-MeOH/PC-MeOH			1.5	14.38	1.39	1.0
			1.7	14.35	1.53	1.0
			1.0	14.3	0.95	1.0
	1.5		13.9	1.23	1.0	
MC-EtOH/PC-EtOH	2.5		13.4	1.92	1.0	
	1.0		14.69	0.99	1.0	
	1.5		14.71	1.4	1.0	
		2.4	13.10	1.8	1.0	
SI-MeOH	3500 rpm	6	1.0	5.69	1.01	1.0
SI-EtOH			1.5	5.42	1.48	1.0
			1.9	5.21	1.90	1.0
			1.0	5.32	1.00	1.0
			1.5	5.52	1.39	1.0
MC-MeOH/PC-MeOH			1.7	5.16	1.51	1.0
			1.0	5.51	0.93	1.0
			1.5	5.67	1.38	1.0
MC-MeOH/PC-EtOH			2.4	5.34	2.33	1.0
			1.0	4.89	1.00	1.0
			1.5	4.55	1.47	1.0
MC-EtOH/PC-EtOH			2.3	4.11	2.28	1.0
		1.0	5.75	0.99	1.0	
		1.5	5.48	1.43	1.0	
			2.2	5.32	2.07	1.0
SI-EtOH		15	1.0	14.34	0.99	1.0
MC-MeOH/PC-MeOH			1.5	13.88	1.44	1.0
			1.7	13.65	1.59	1.0
	1.0		13.8	0.92	1.0	
	1.5		14.4	1.24	1.0	
MC-EtOH/PC-EtOH	2.3		14.1	1.76	1.0	
	1.0		5.32	1.00	1.0	
	1.5		5.52	1.39	1.0	
		1.8	5.16	1.51	1.0	

Appendix C. MAPO THRESHOLD

As discussed in the methodology section, one of the knock indicators used in the present study is the maximum amplitude of pressure oscillations, or MAPO analysis. Considering the knock threshold detection methodology suggested by Iwata et al. [159], in which non-knocking and knocking data sets follow log-normal distributions with different slopes, MAPO was the parameter used to determine the threshold. In the results and discussion section, only the thresholds are presented, but Figure C.1 to Figure C.16 show how such data were determined.

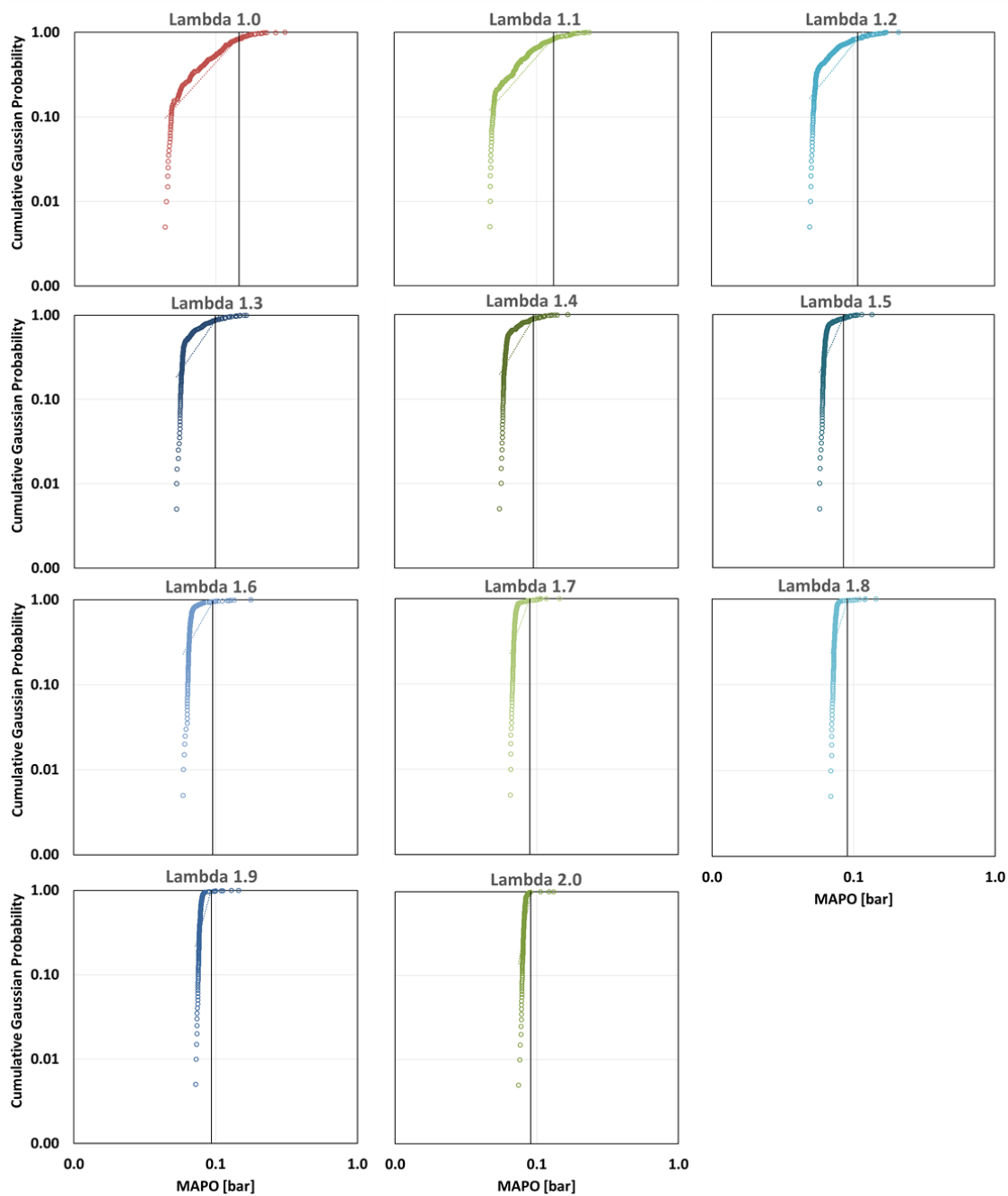


Figure C.1- 2000 rpm and 6 bar of IMEP - SI-MeOH

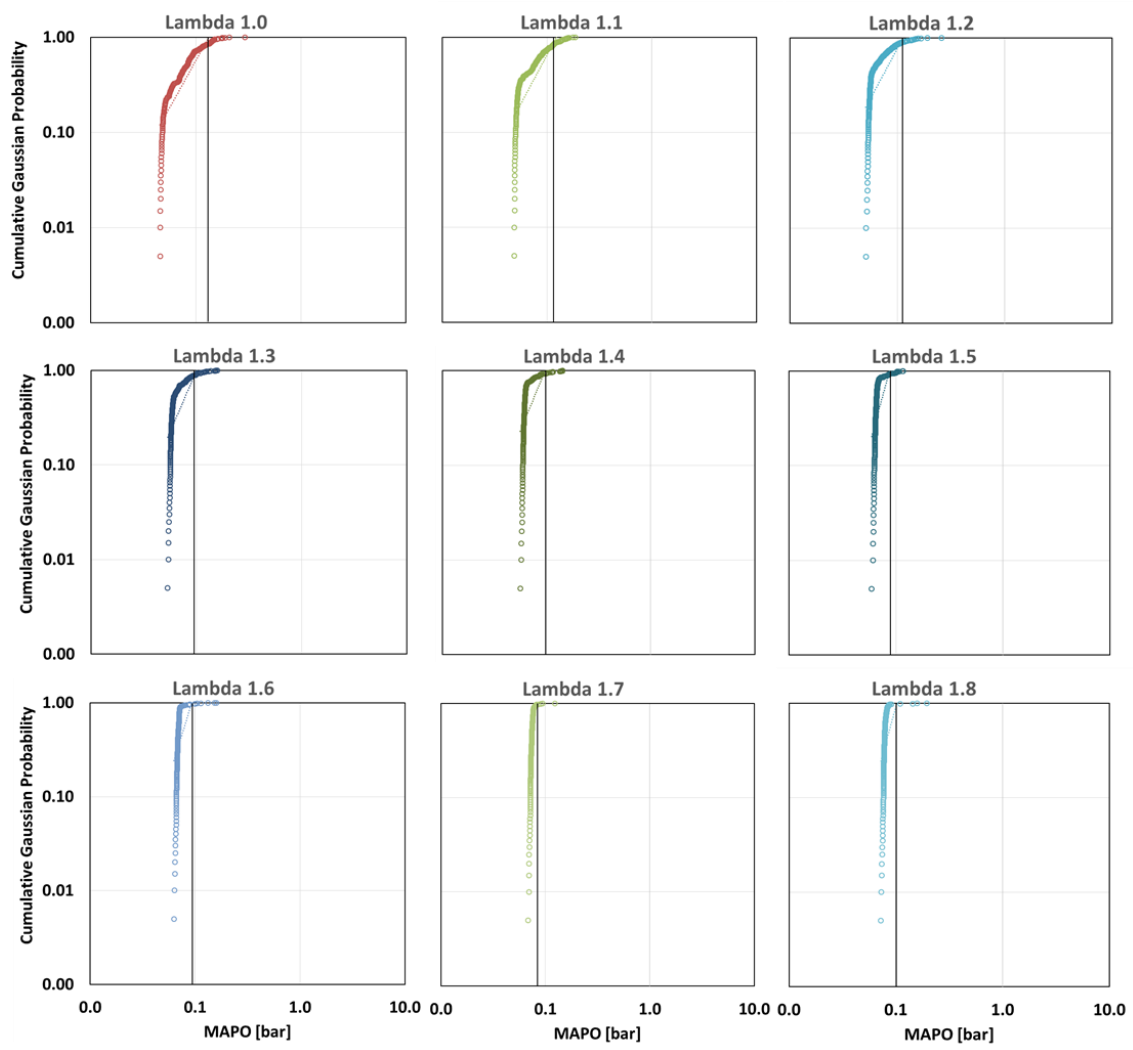


Figure C.2 - 2000 rpm and 6 bar of IMEP – SI-EtOH

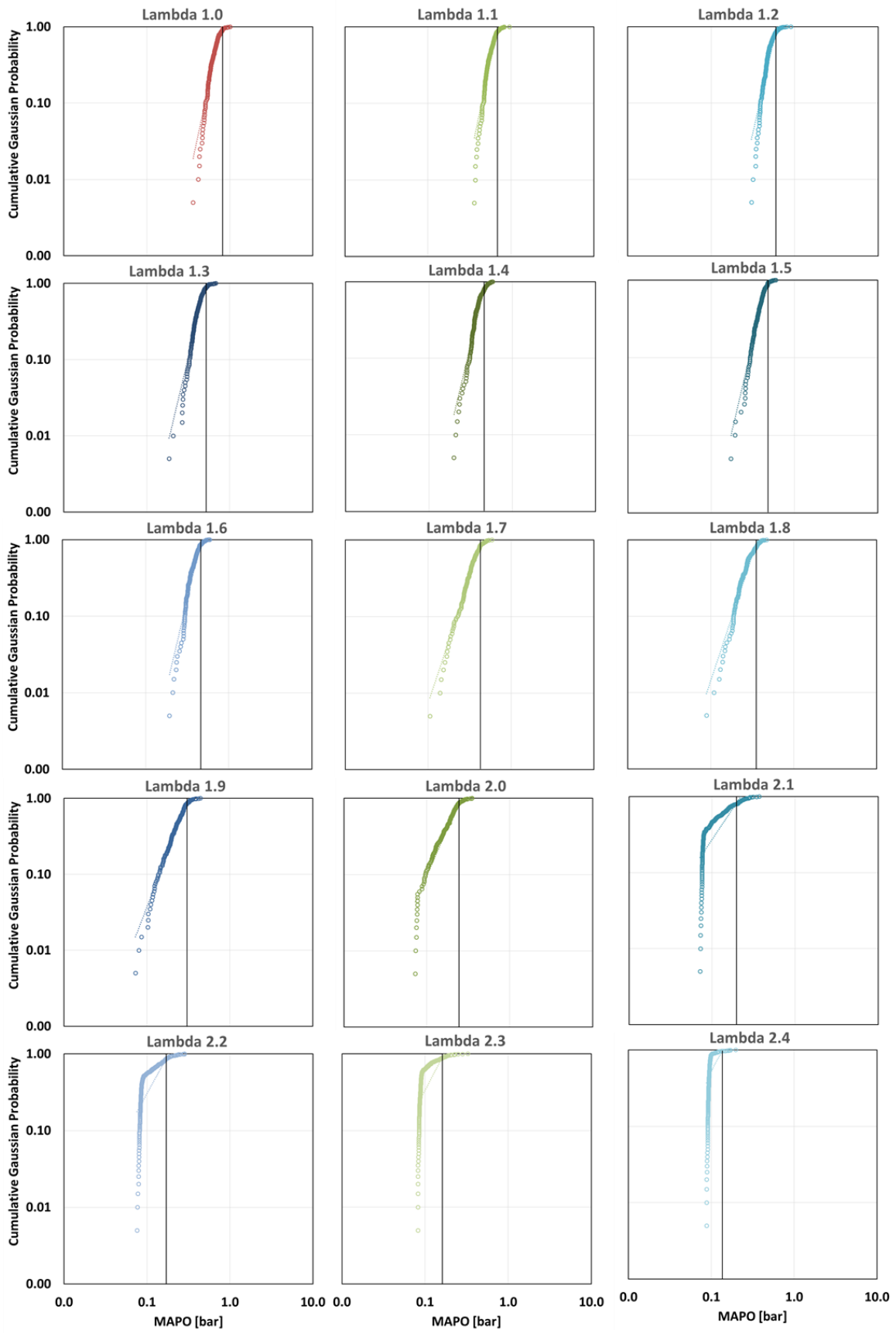


Figure C.3 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

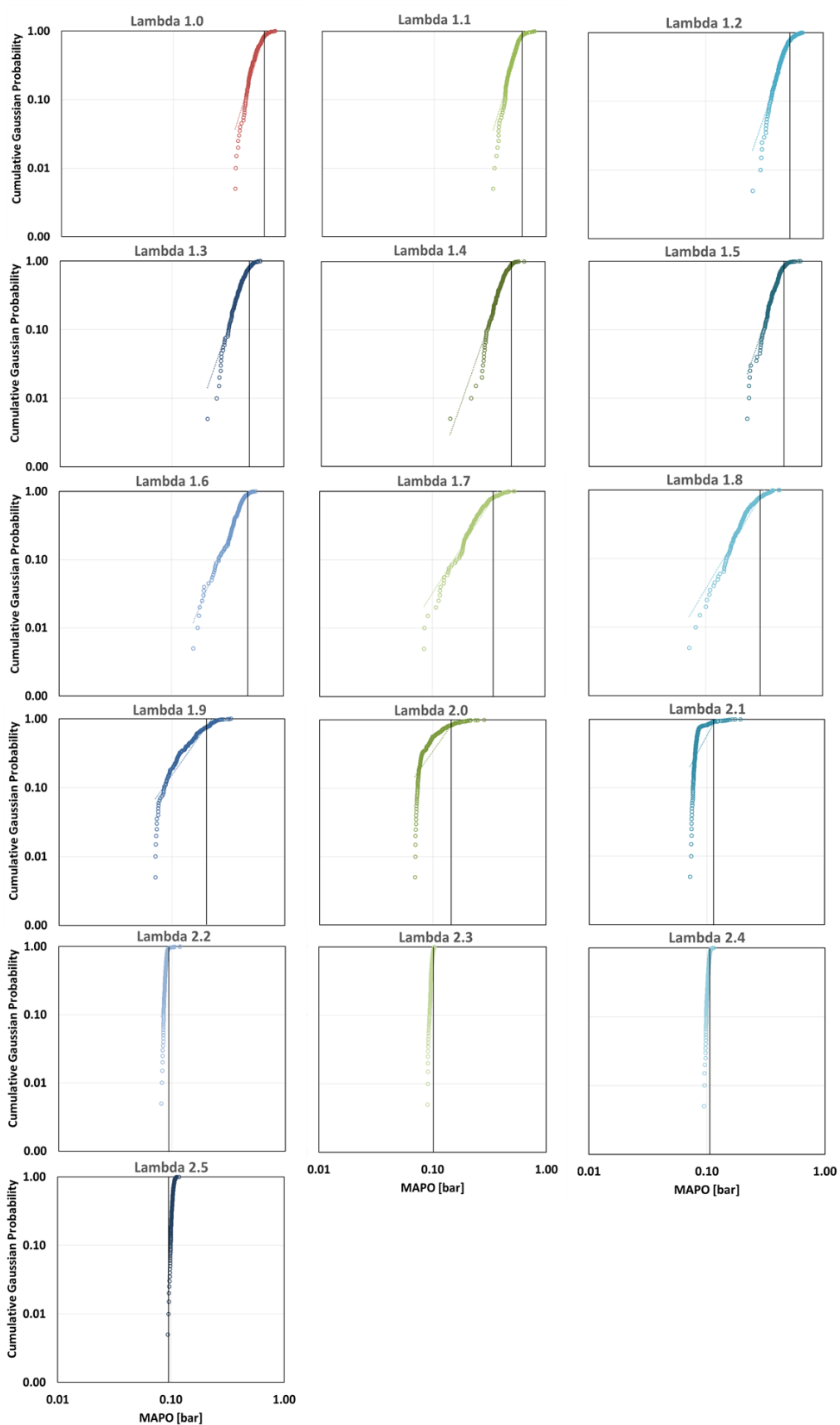


Figure C.4 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

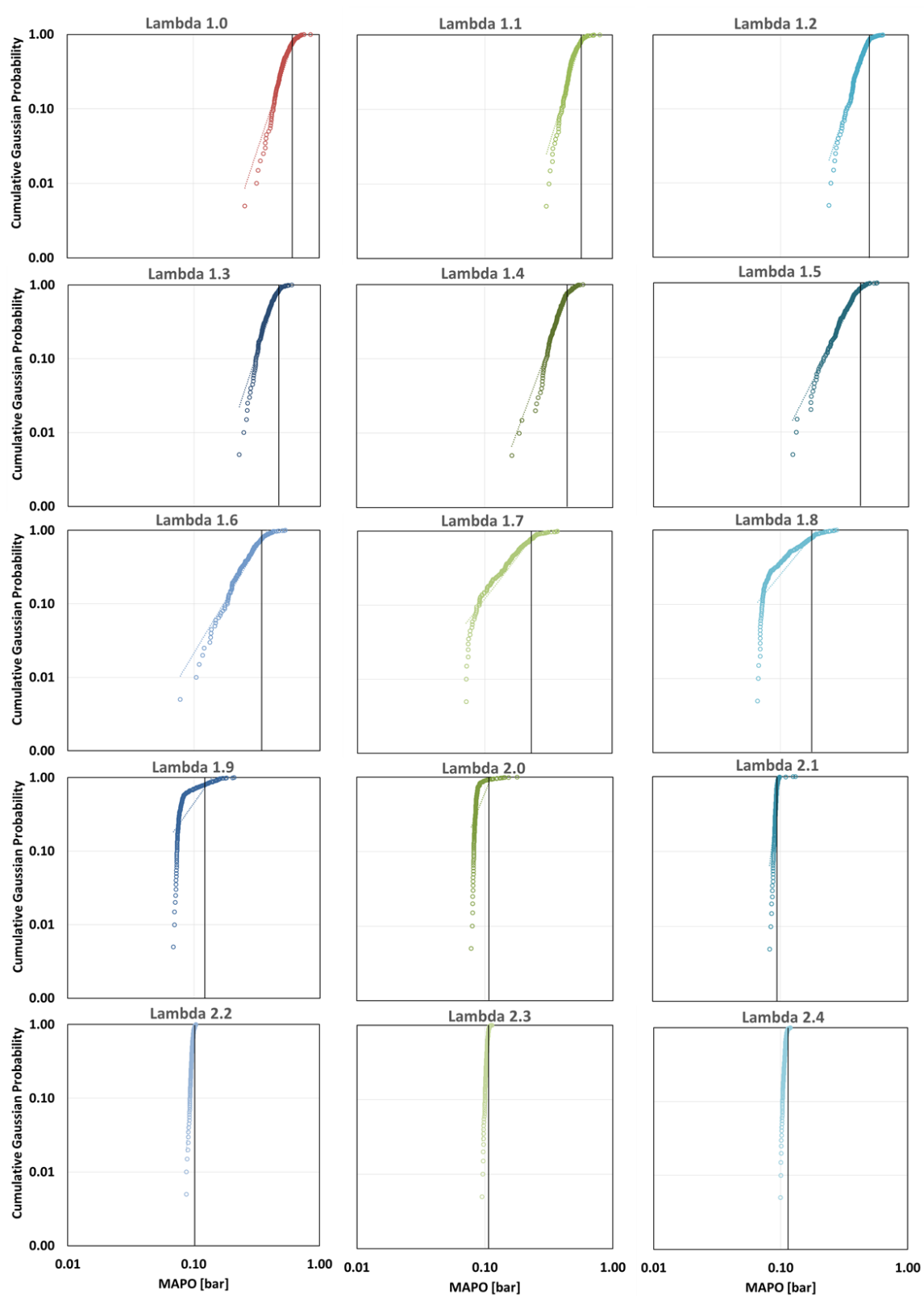


Figure C.5 - 2000 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

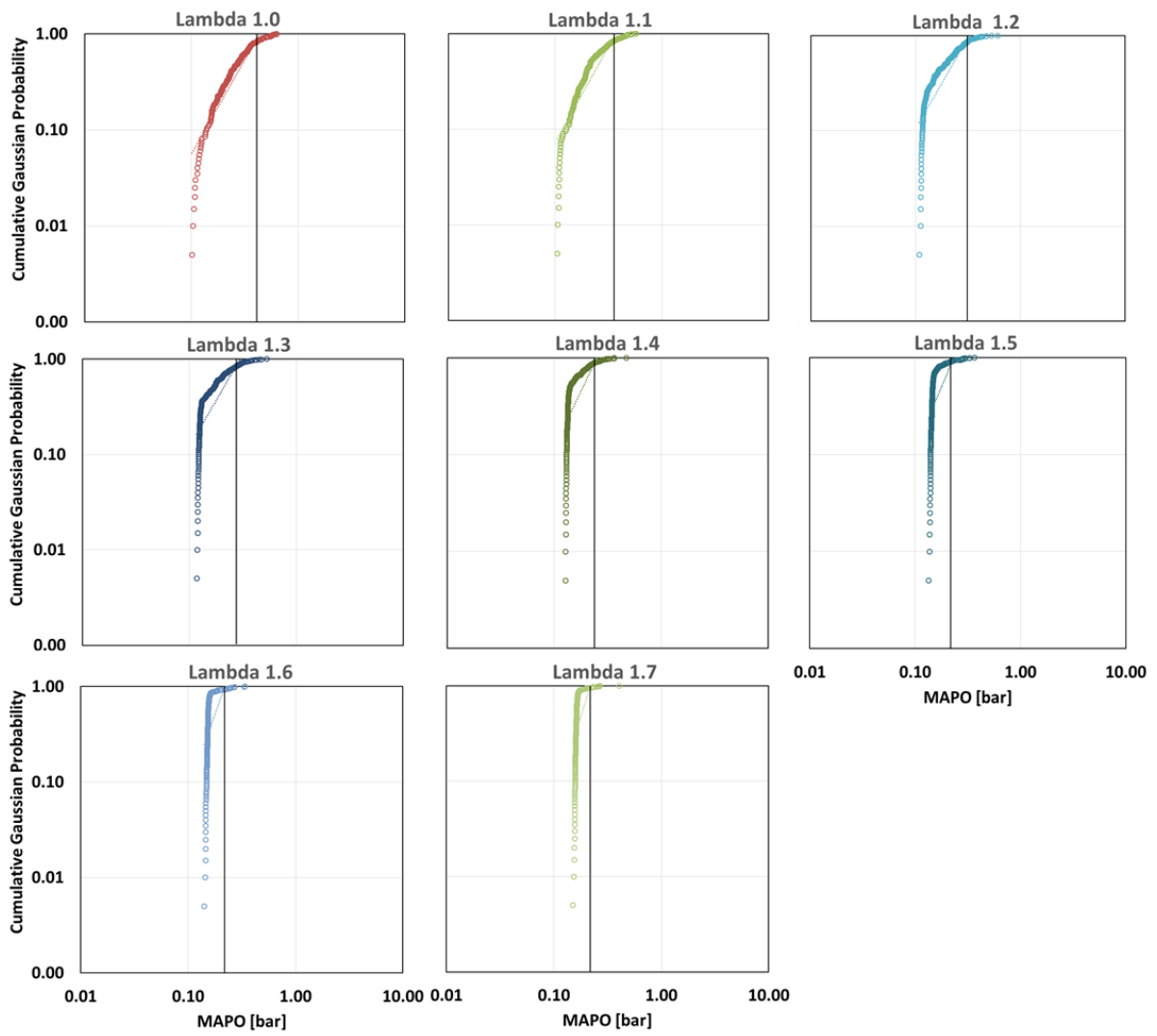


Figure C.6 - 2000 rpm and 15 bar of IMEP – SI-EtOH

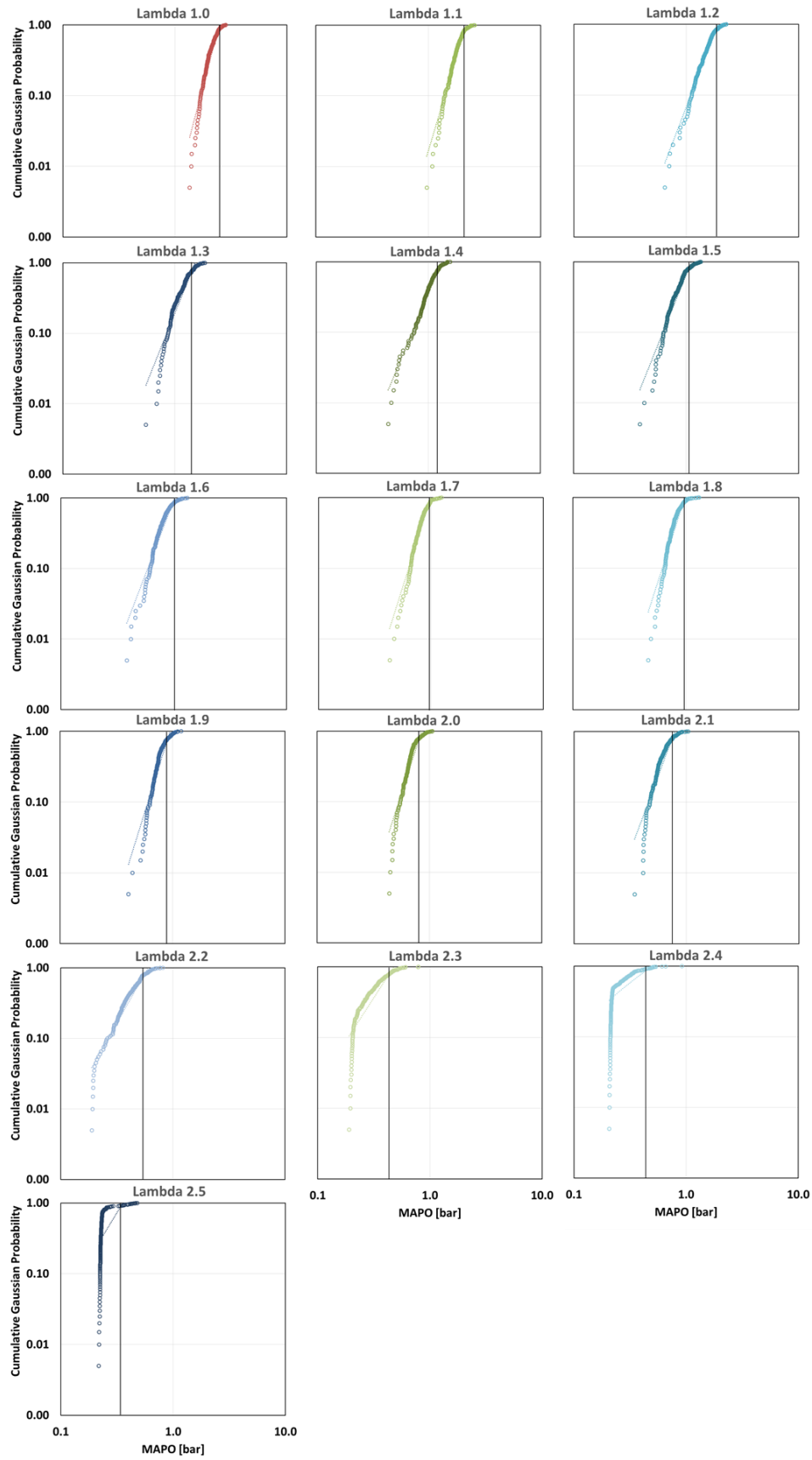


Figure C.7 - 2000 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

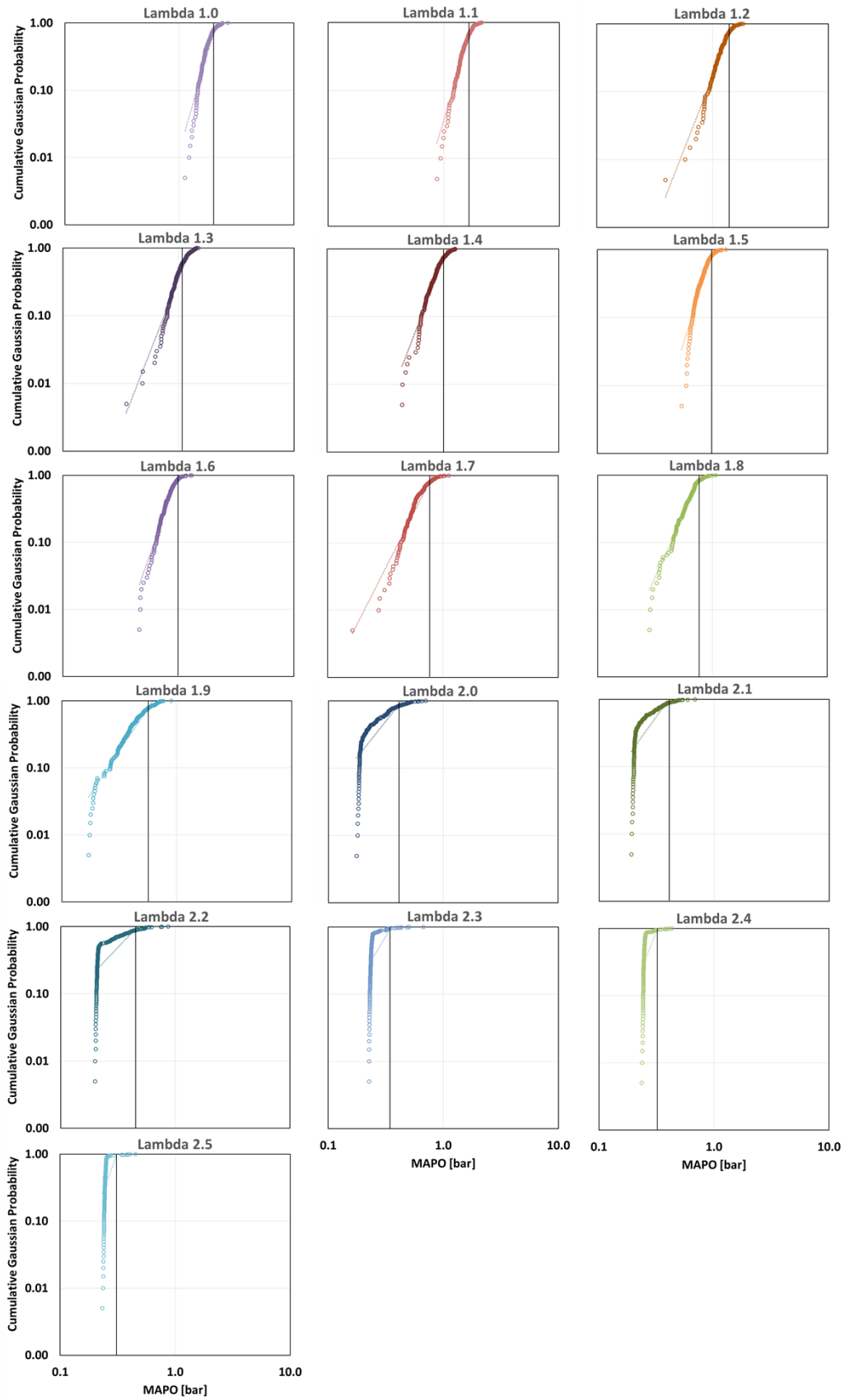


Figure C.8 - 2000 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

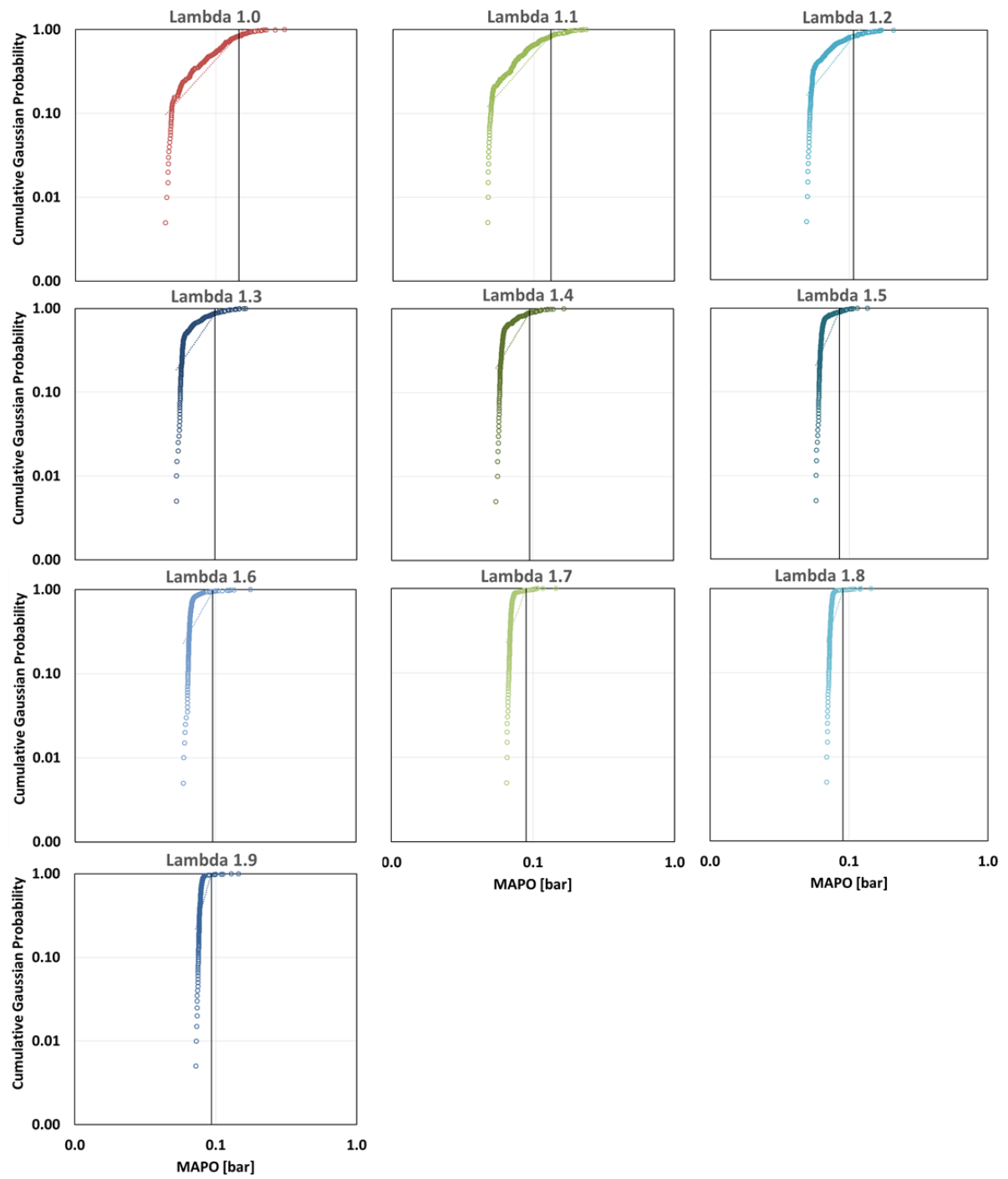


Figure C.9 - 3500 rpm and 6 bar of IMEP - SI-MeOH

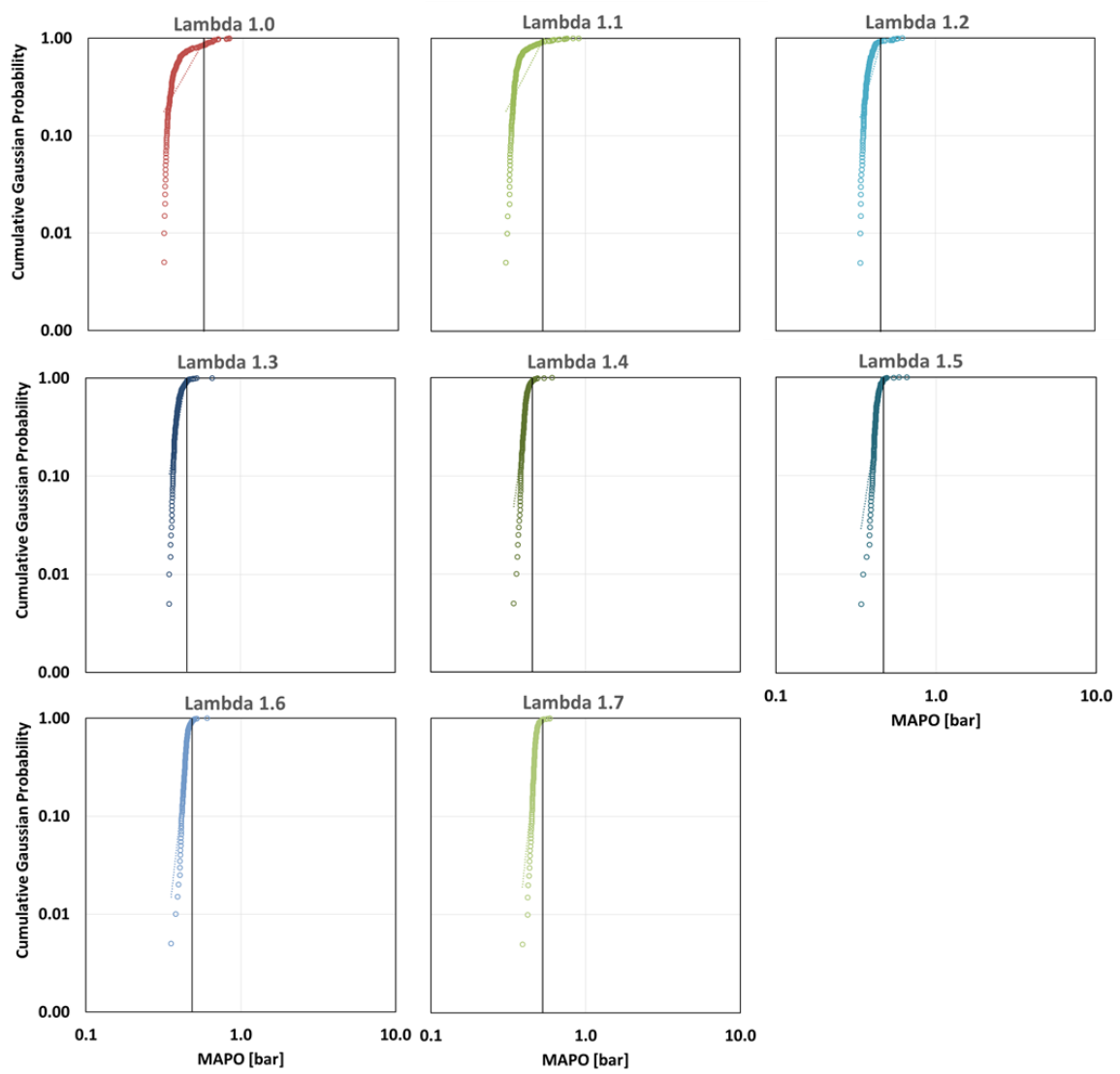


Figure C.10 - 3500 rpm and 6 bar of IMEP – SI-EtOH

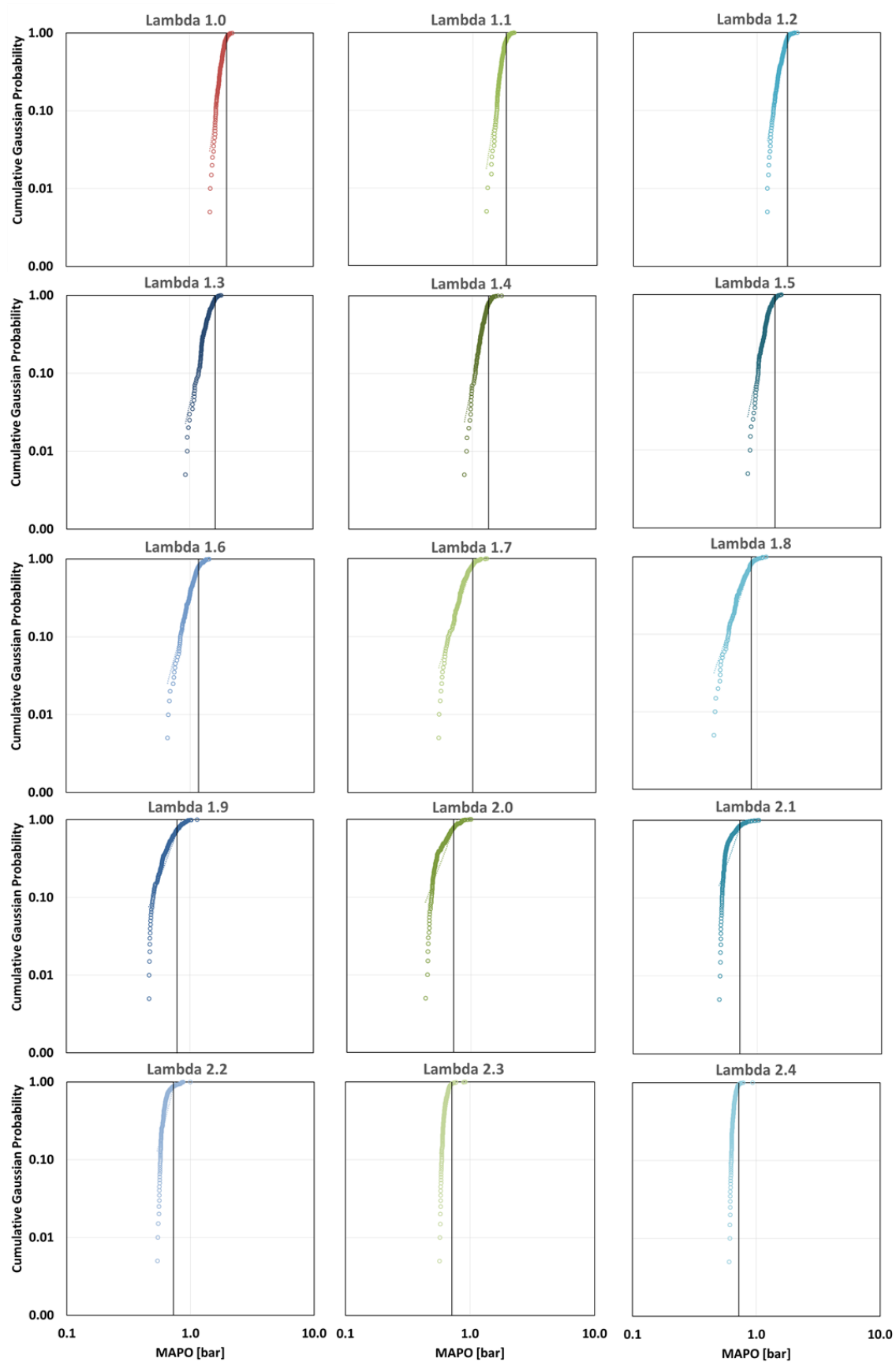


Figure C.11 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

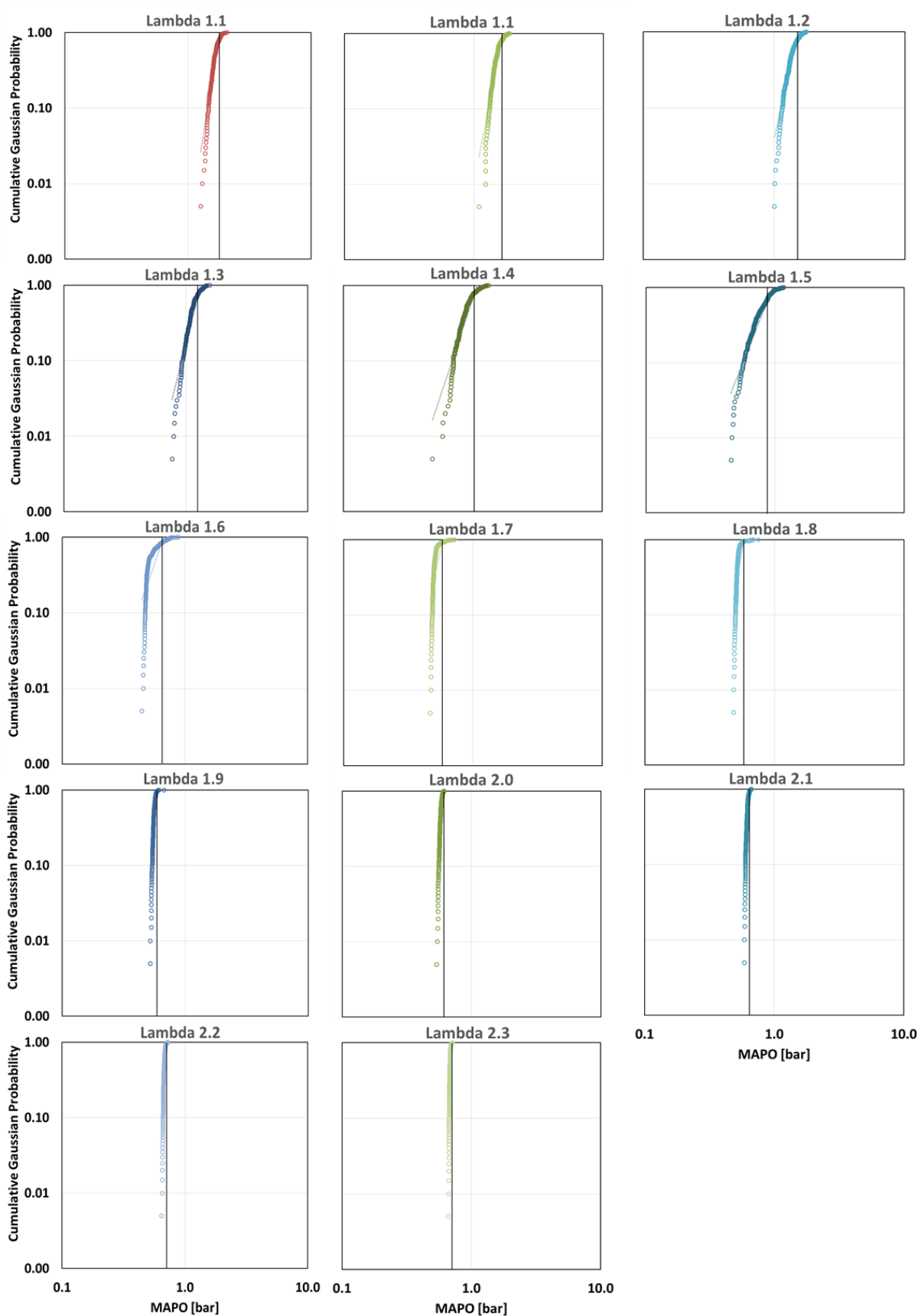


Figure C.12 -3500 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

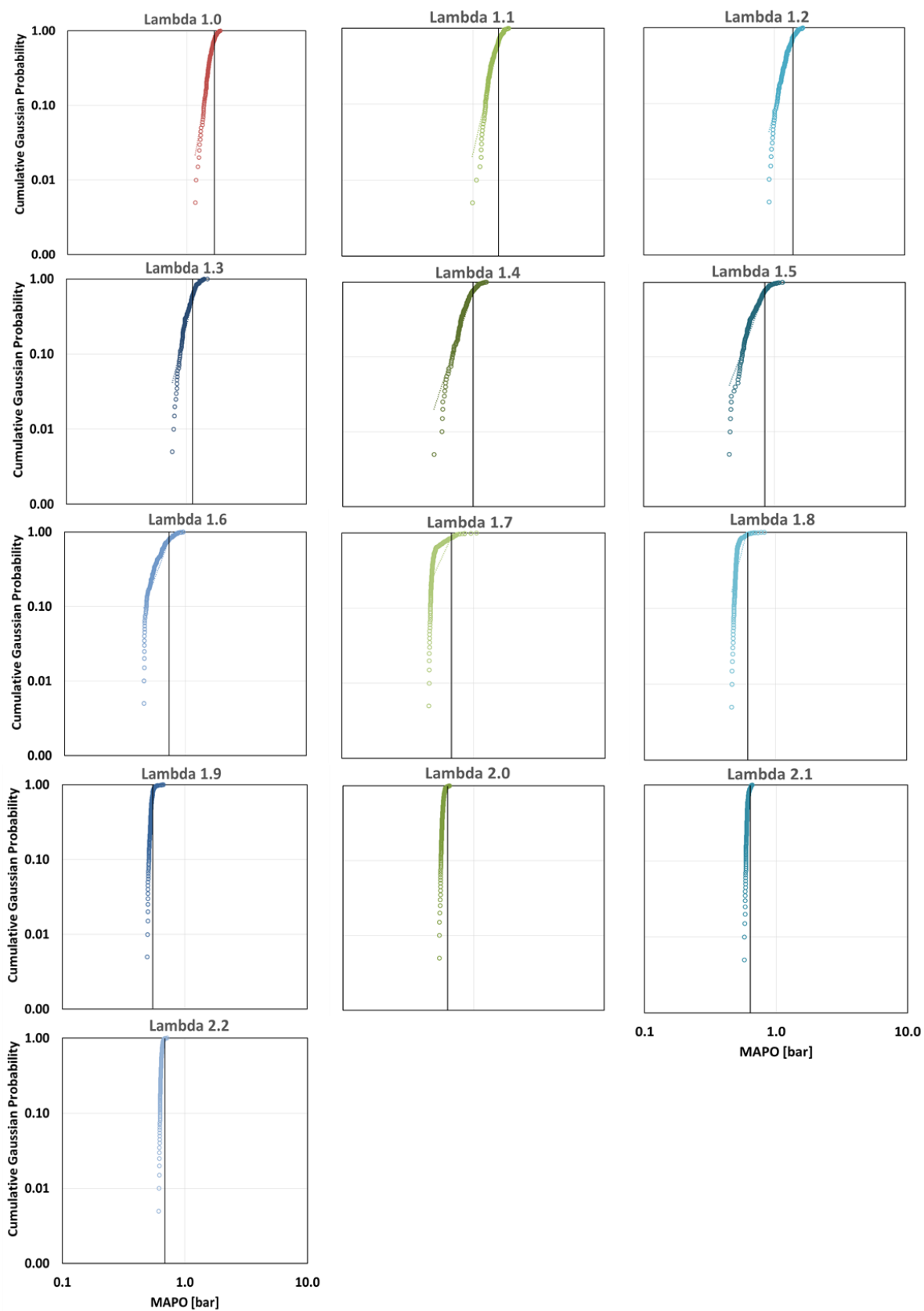


Figure C.13 - 3500 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

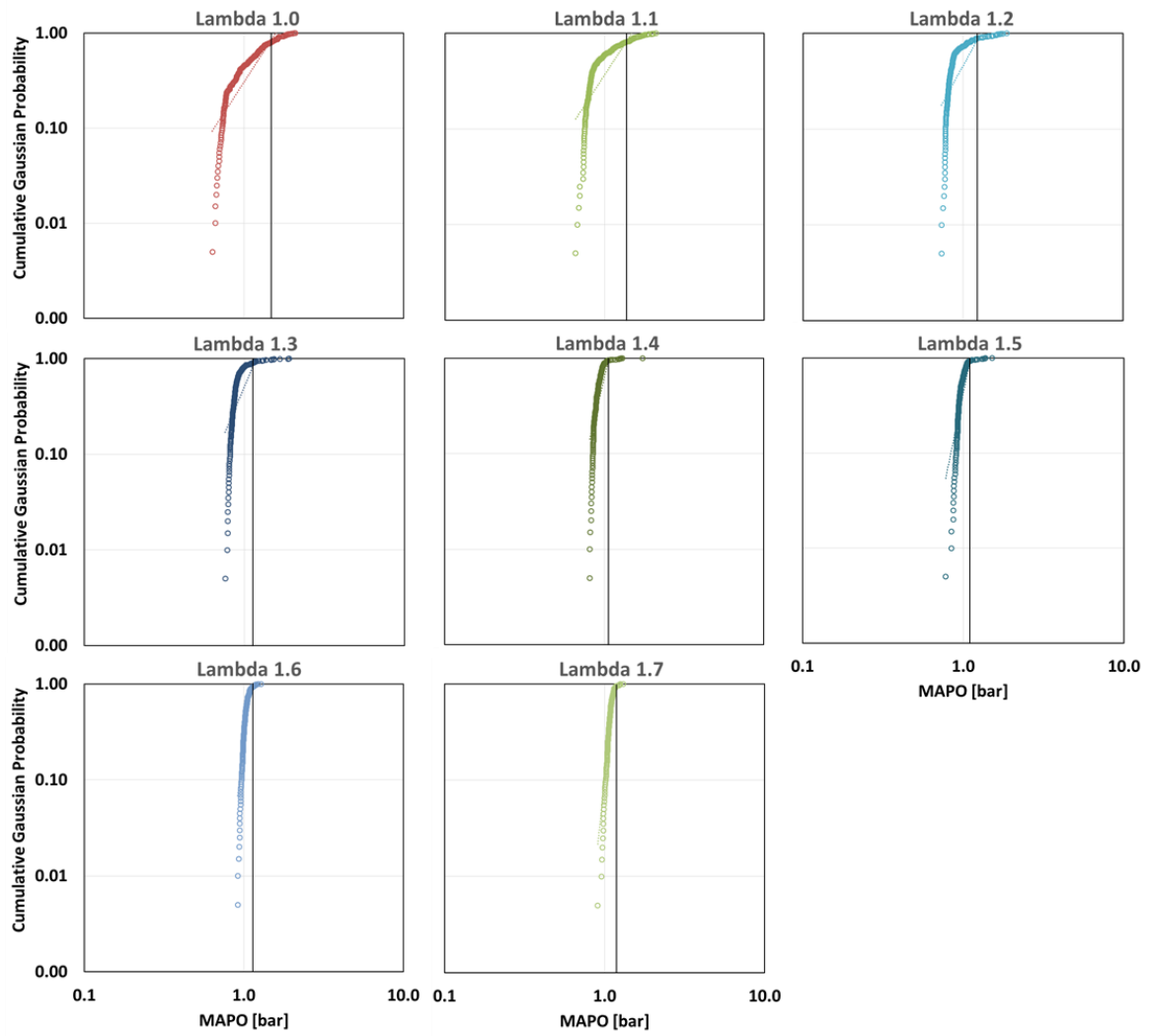


Figure C.14 - 3500 rpm and 15 bar of IMEP – SI-EtOH

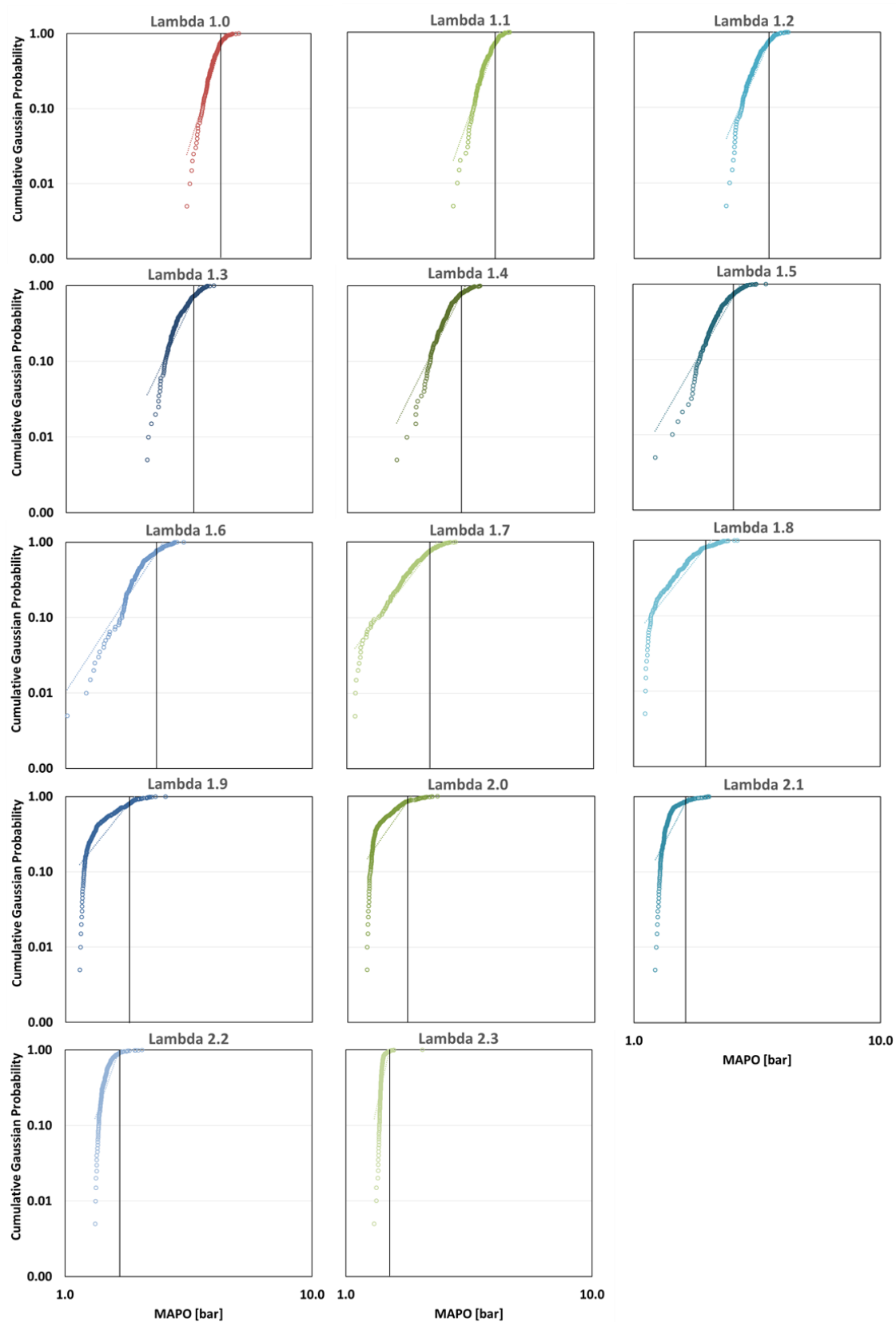


Figure C.15 - 3500 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

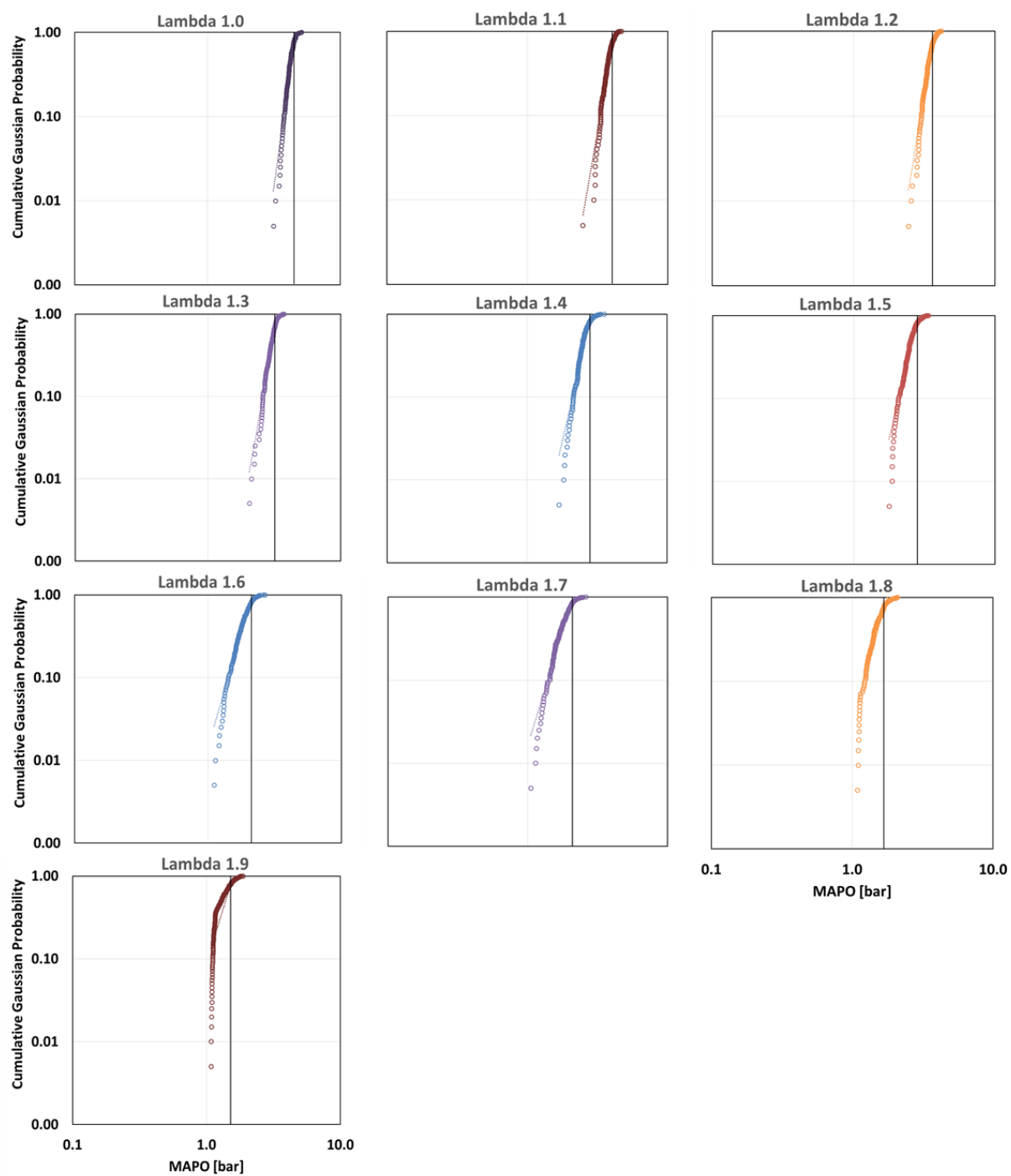


Figure C.16 - 3500 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

Appendix D. KNOCK INDEX (KI2)

As discussed in the methodology section, the MAPO parameters is not considered a reliable enough method for determining a knock threshold, as it can lead to knock overprediction [160]. Therefore, the data in the present study is also analyzed with the parameter knock index 2 (KI2), suggested by Galloni et al. [161]. Figure D.1 to Figure D.16 graphically show the $MAPO_{th}$, the $MAPO_{end}$ and the KI for all 200 cases recorded at each of the analyzed operating points. The KI data are used to calculate the KI2 and the analysis of the data behavior can help in the data interpretation in the results and discussion section.

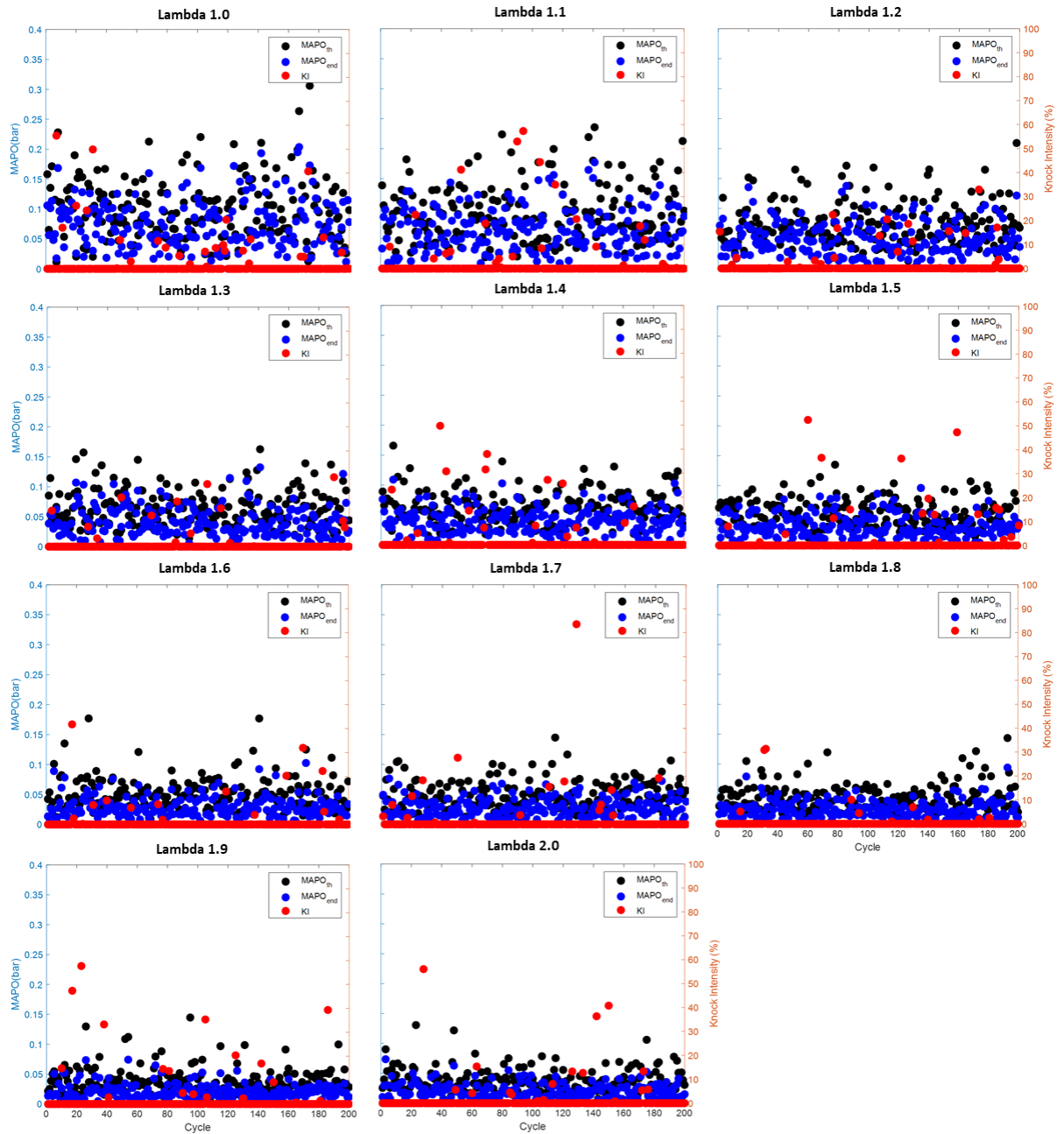


Figure D.1 - 2000 rpm and 6 bar of IMEP - SI-MeOH

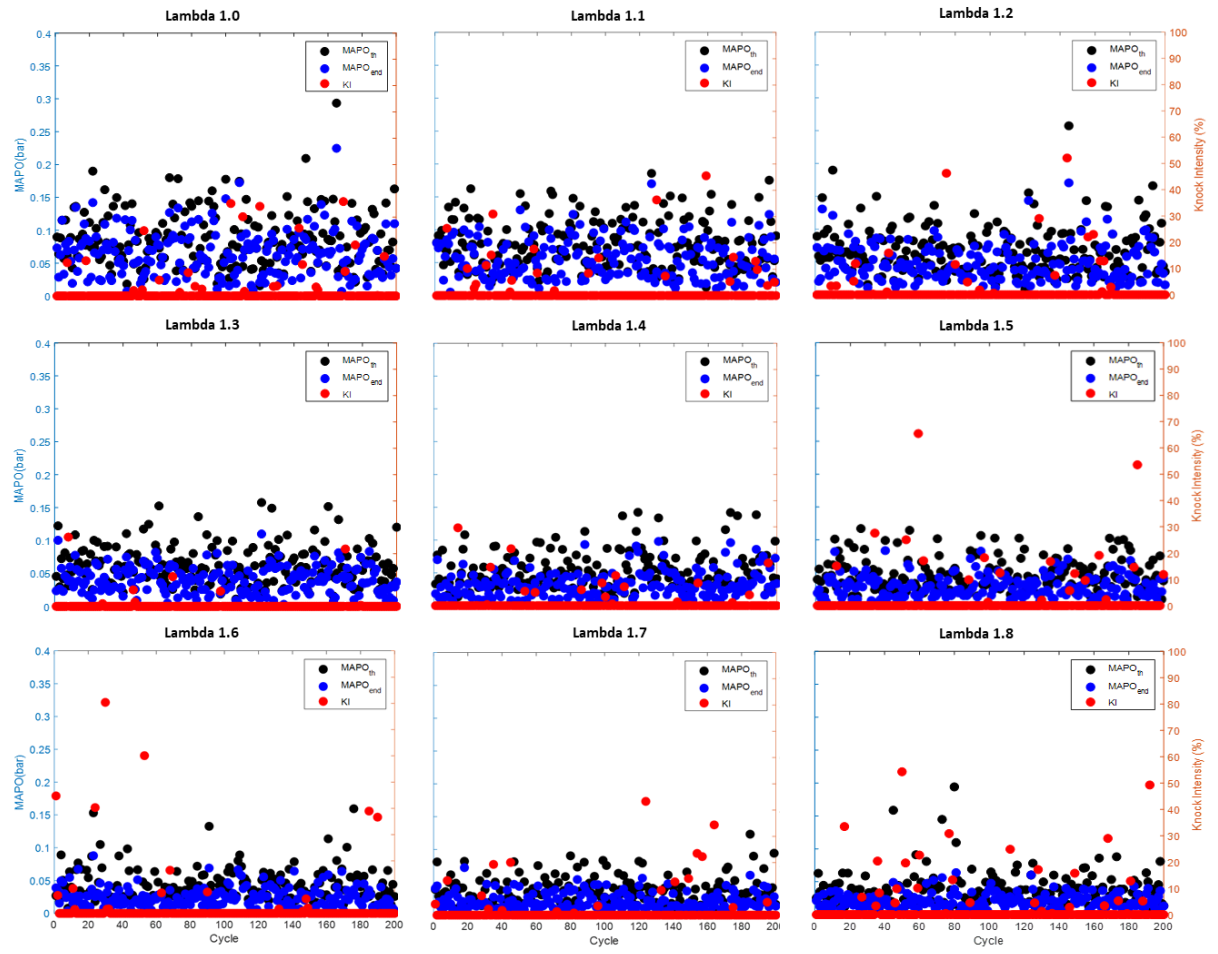


Figure D.2 - 2000 rpm and 6 bar of IMEP – SI-EtOH

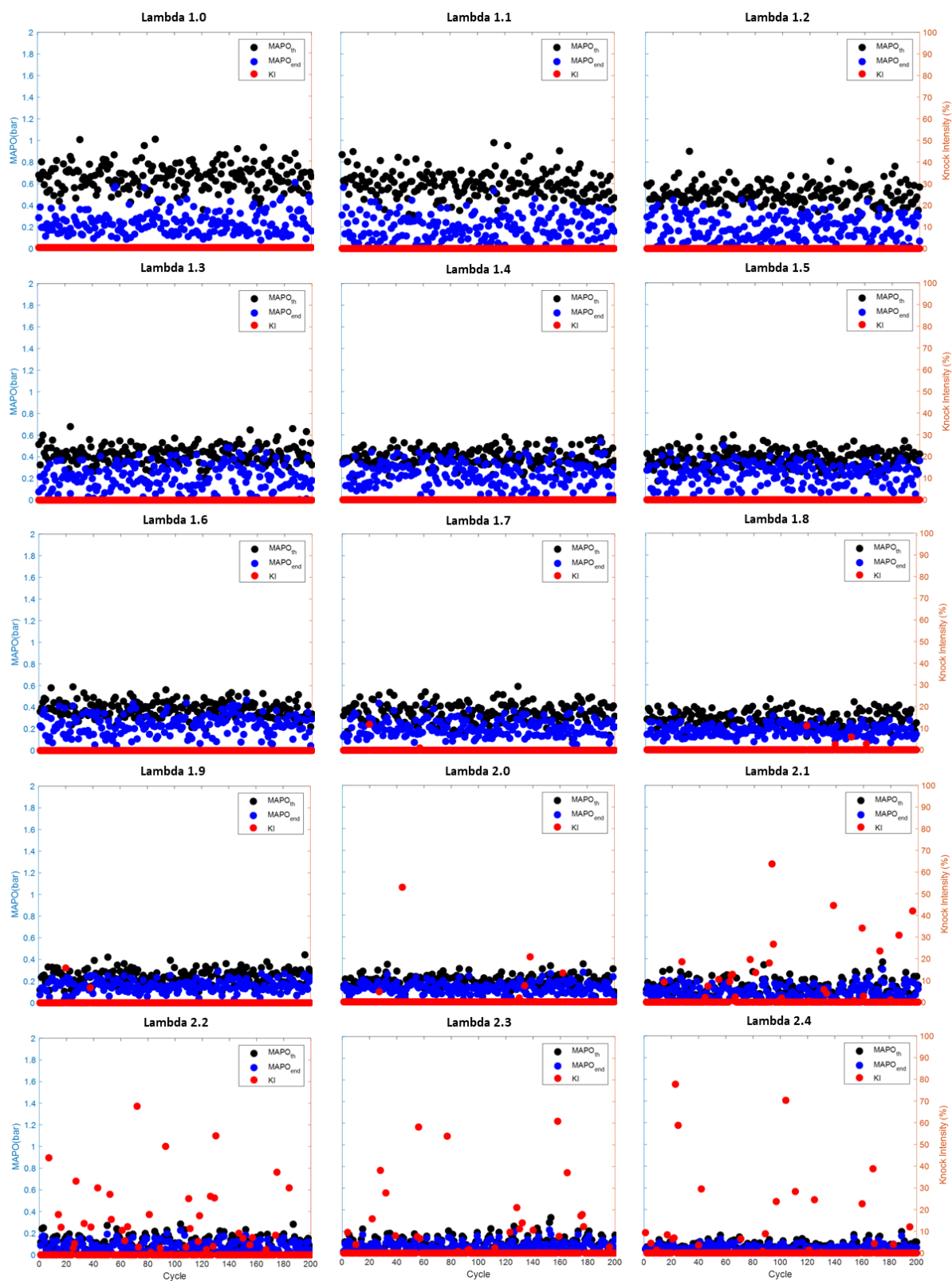


Figure D.3 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

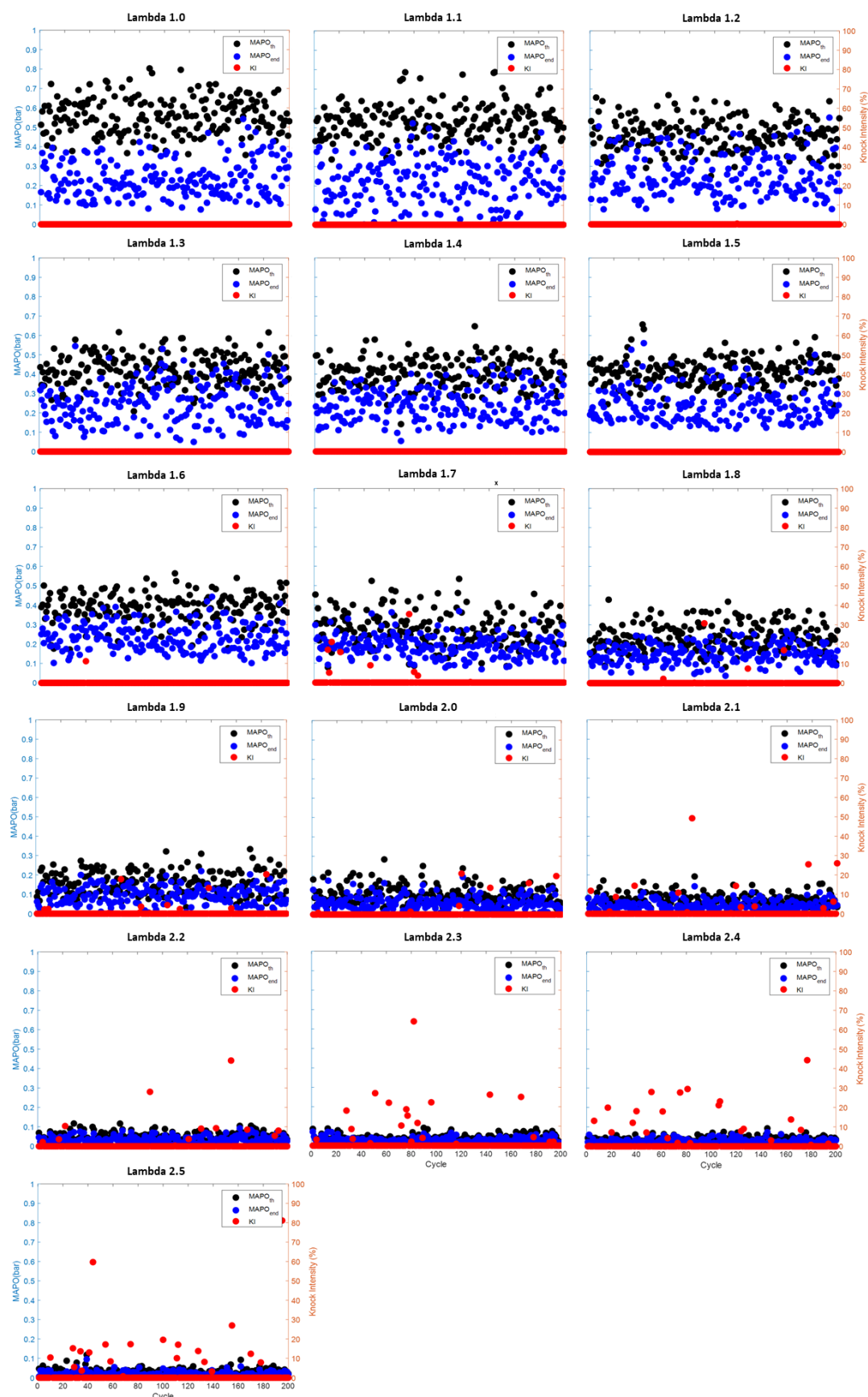


Figure D.4 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

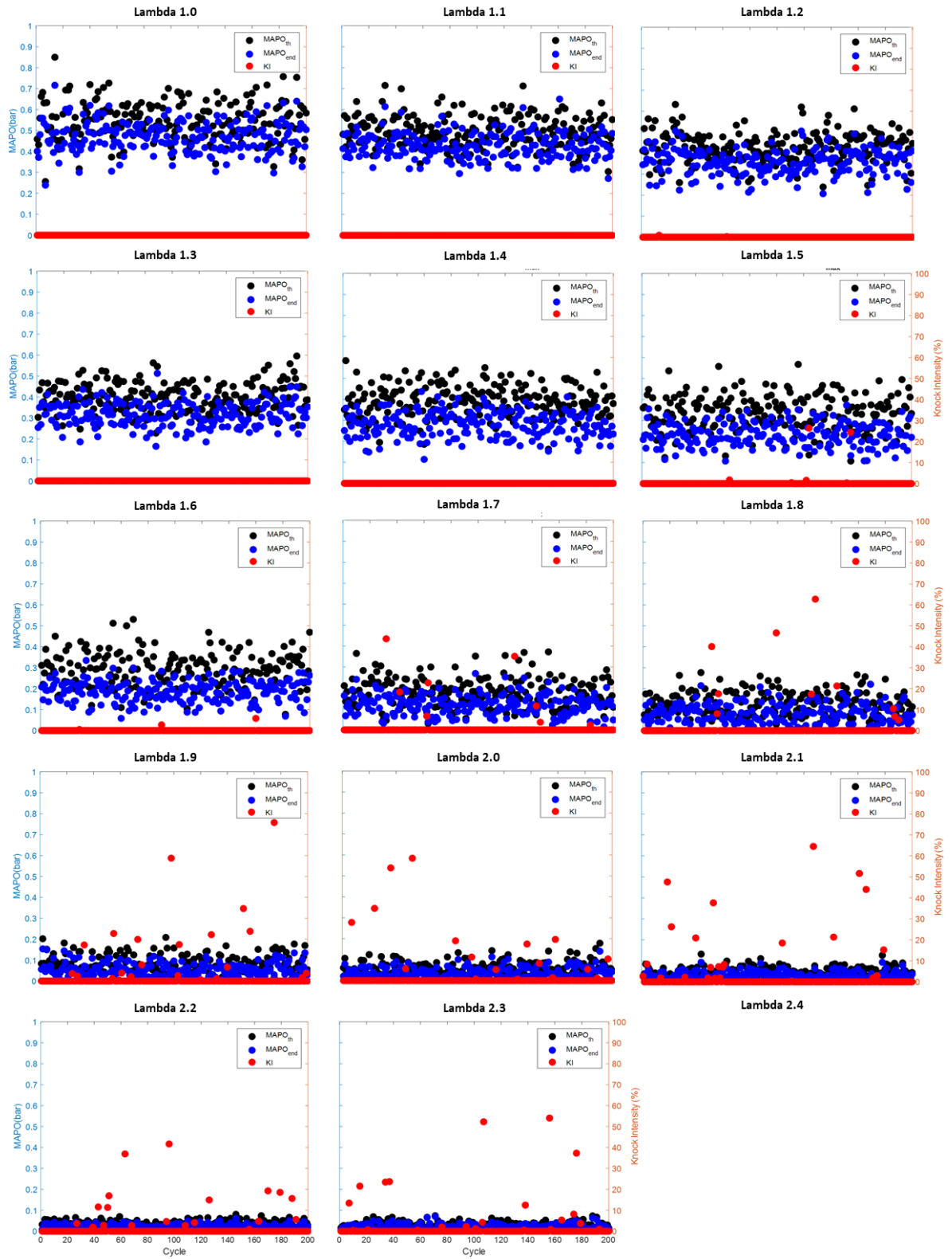


Figure D.5 - 2000 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

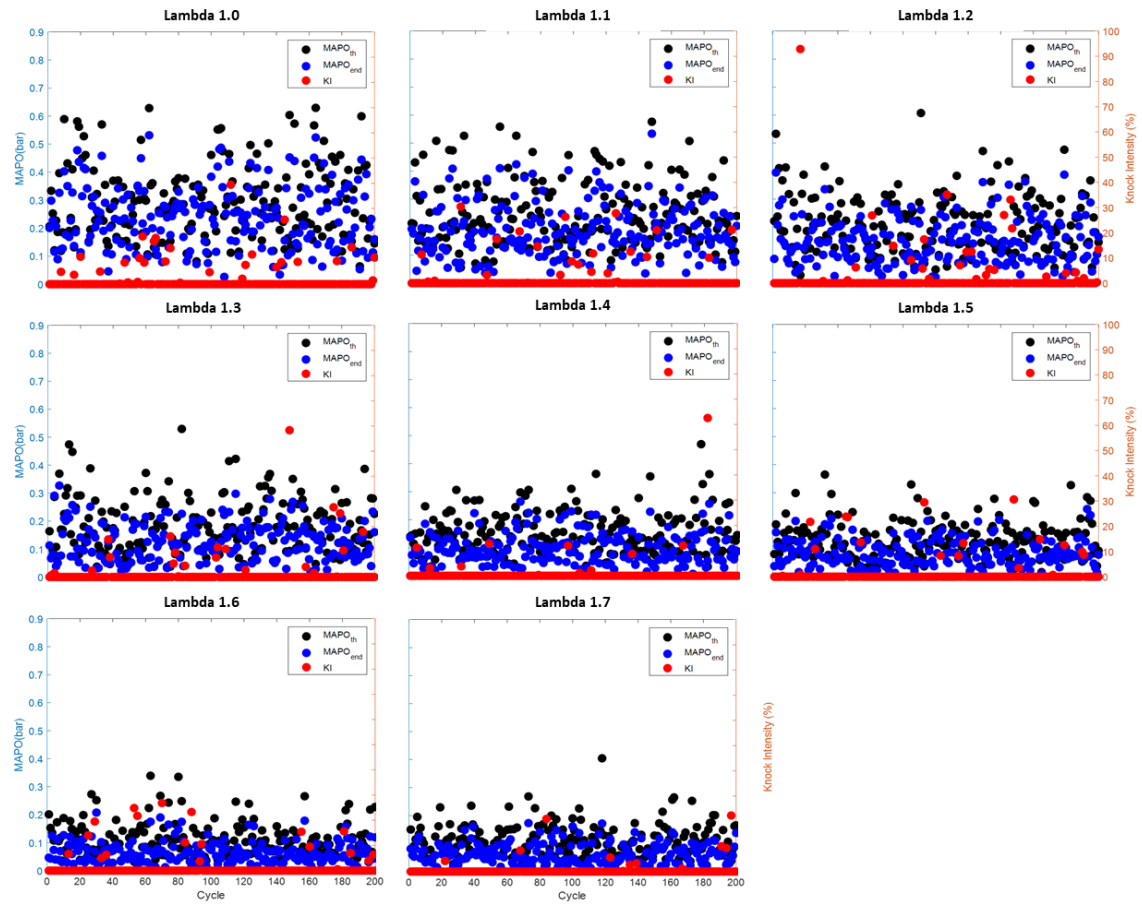


Figure D.6 - 2000 rpm and 15 bar of IMEP – SI-EtOH

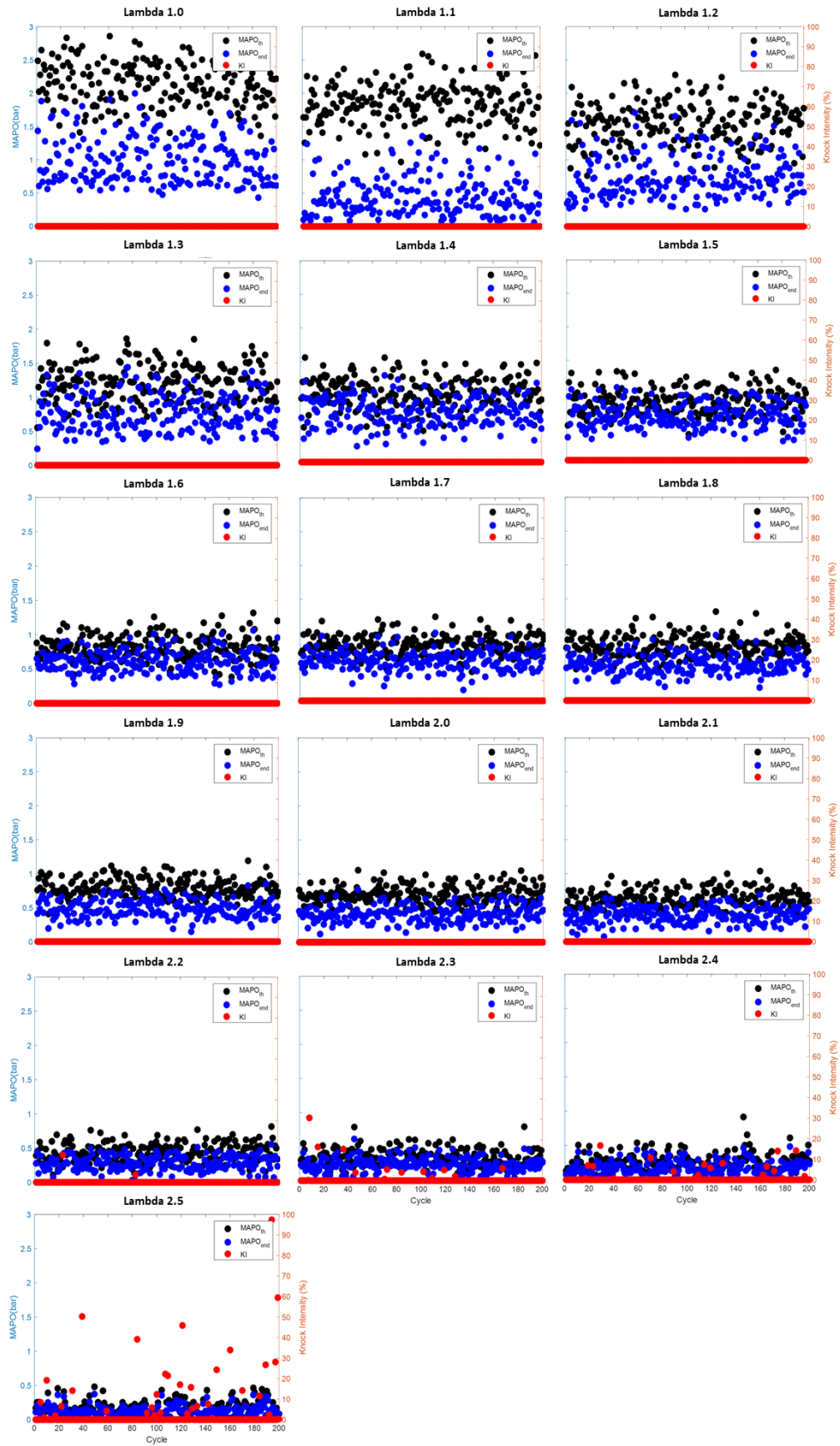


Figure D.7 - 2000 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

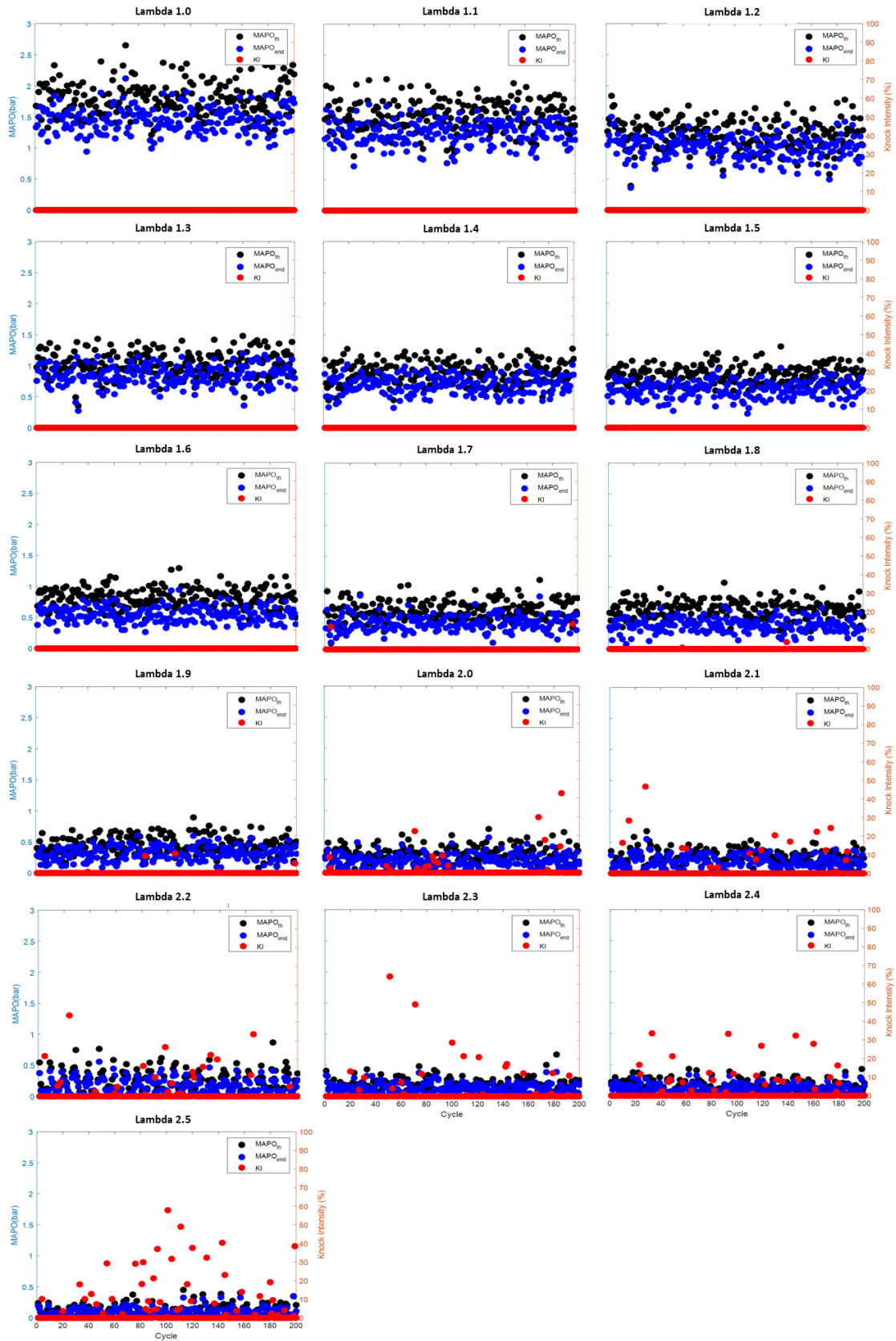


Figure D.8 - 2000 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

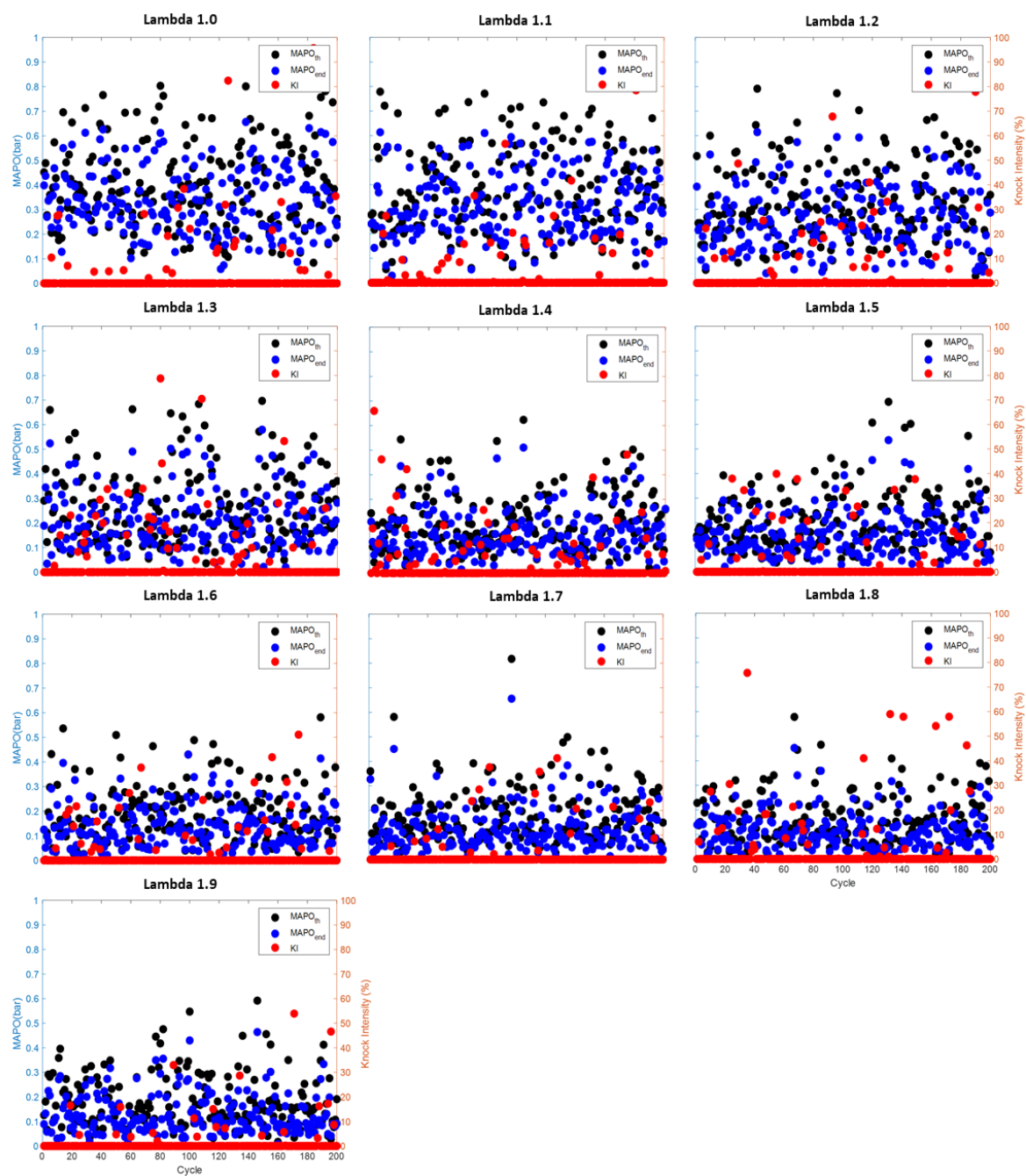


Figure D.9 - 3500 rpm and 6 bar of IMEP - SI-MeOH

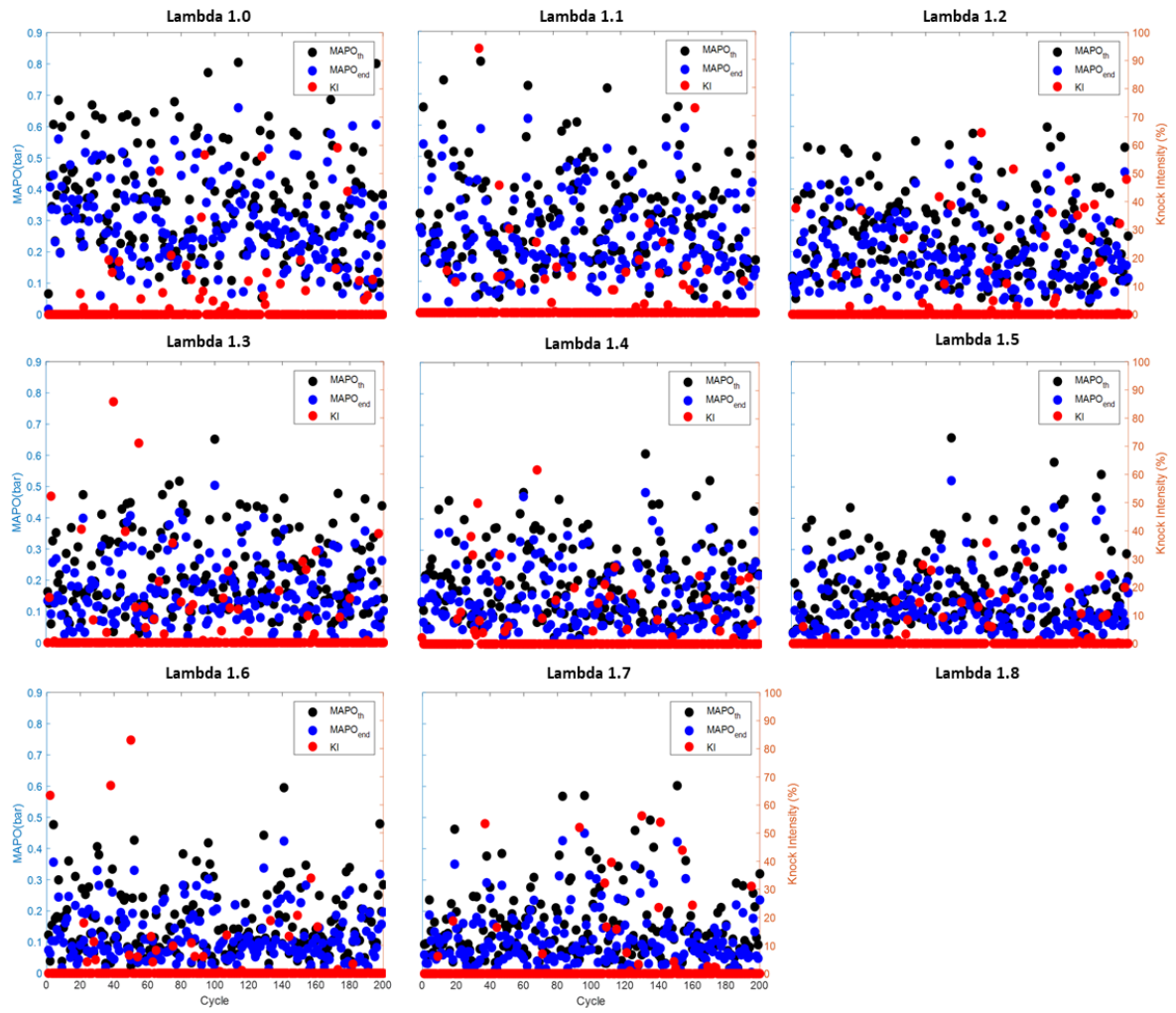


Figure D.10 - 3500 rpm and 6 bar of IMEP - SI-EtOH

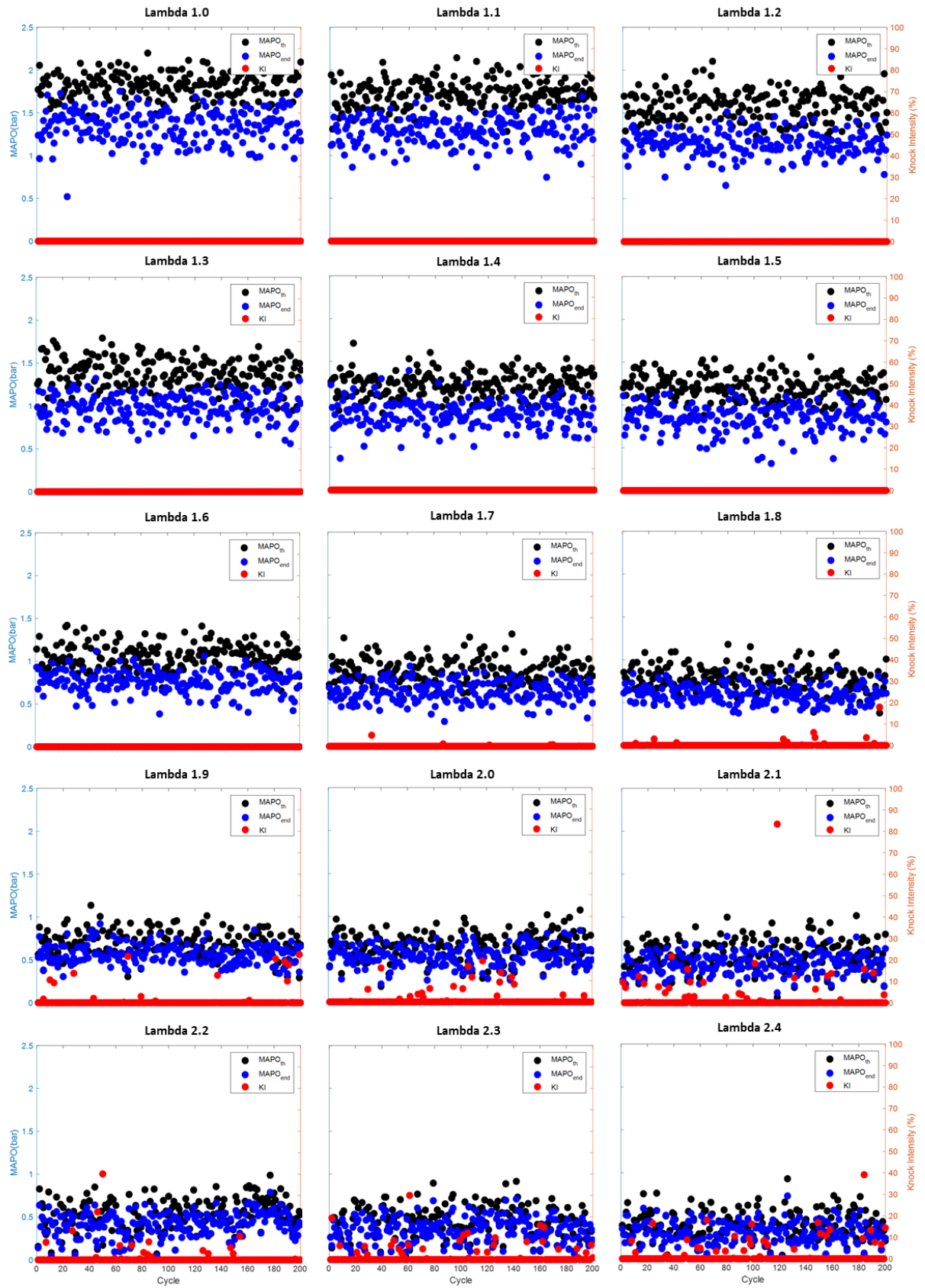


Figure D.11 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

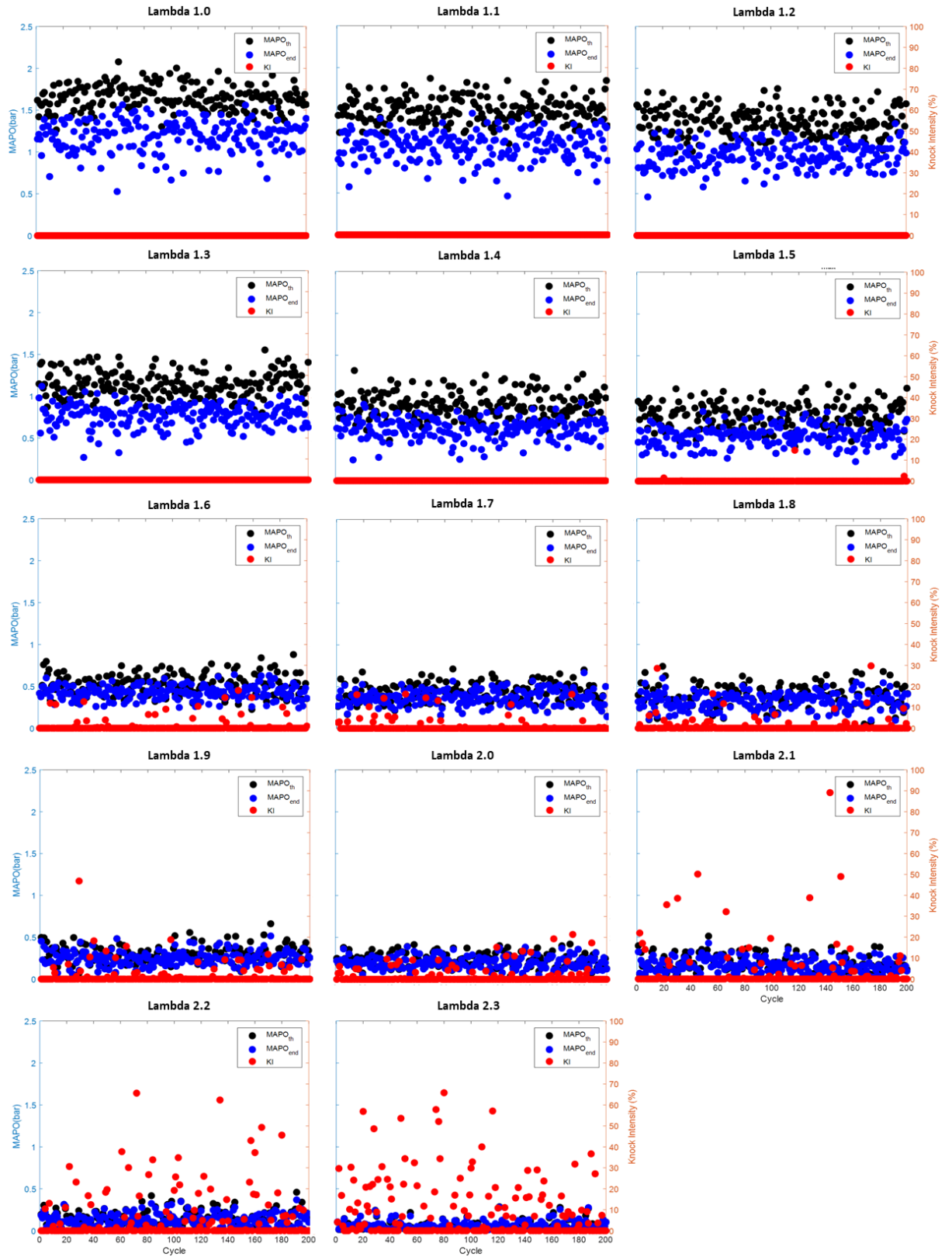


Figure D.12 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

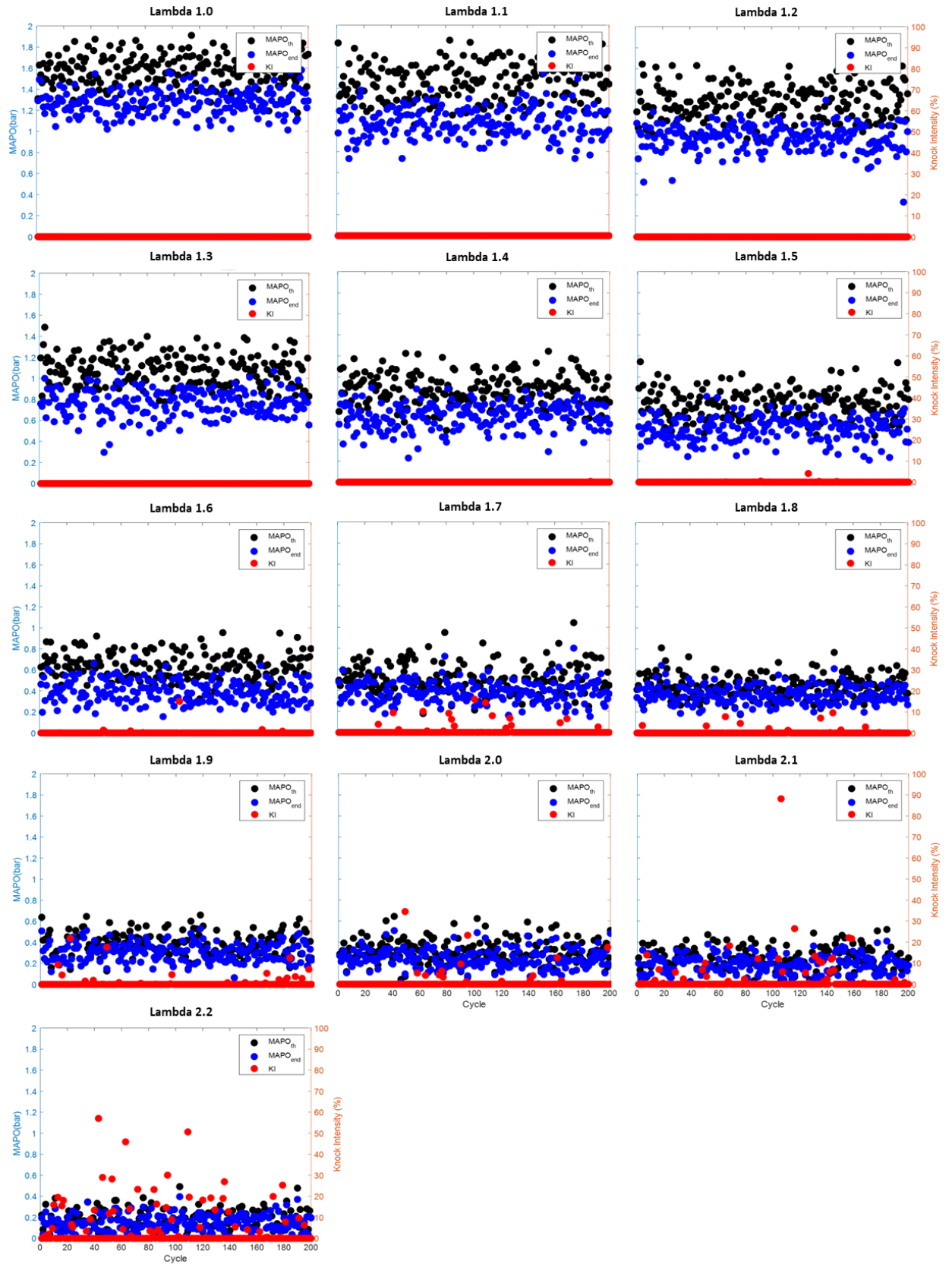


Figure D.13 - 3500 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

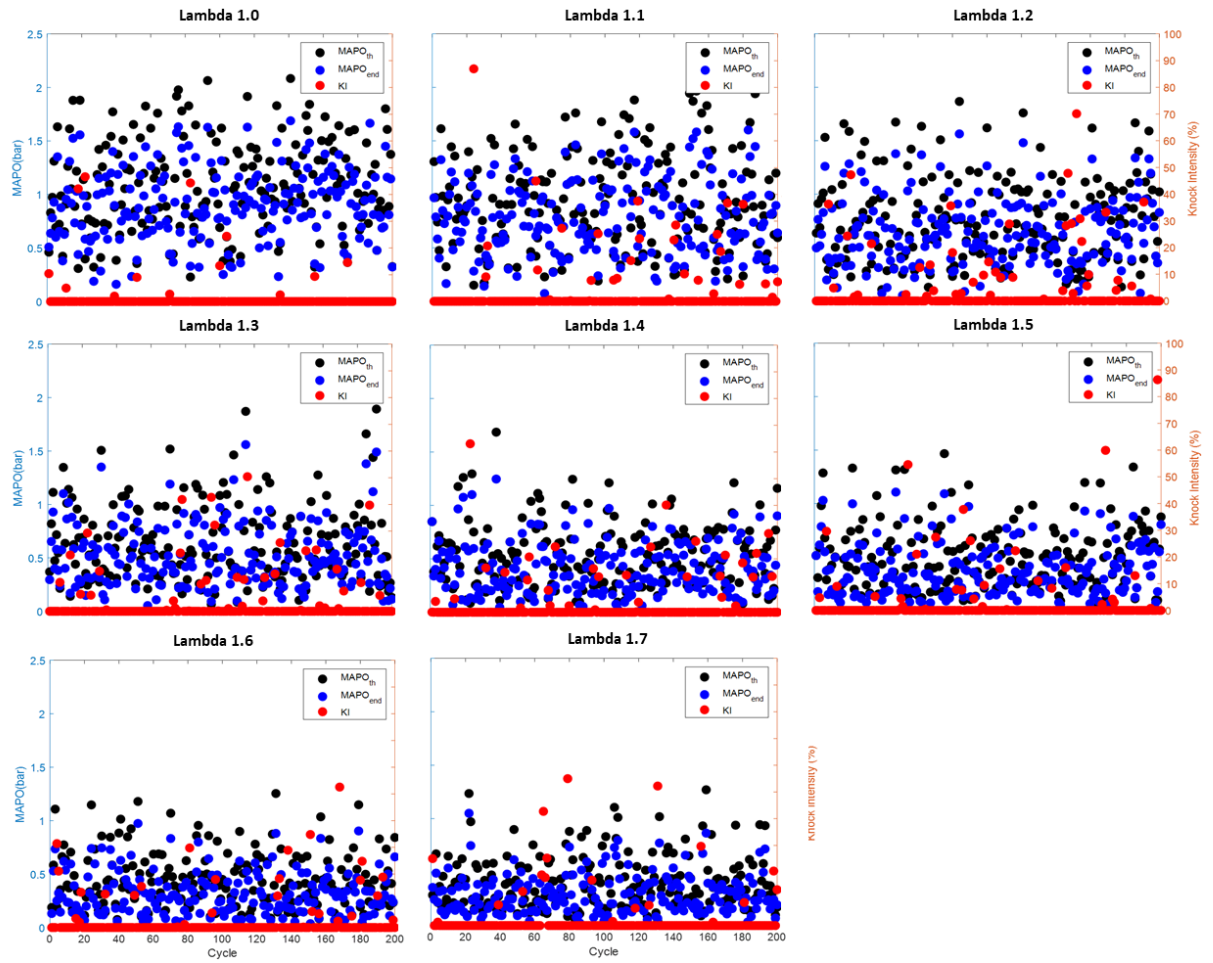


Figure D.14 - 3500 rpm and 15 bar of IMEP - SI-EtOH

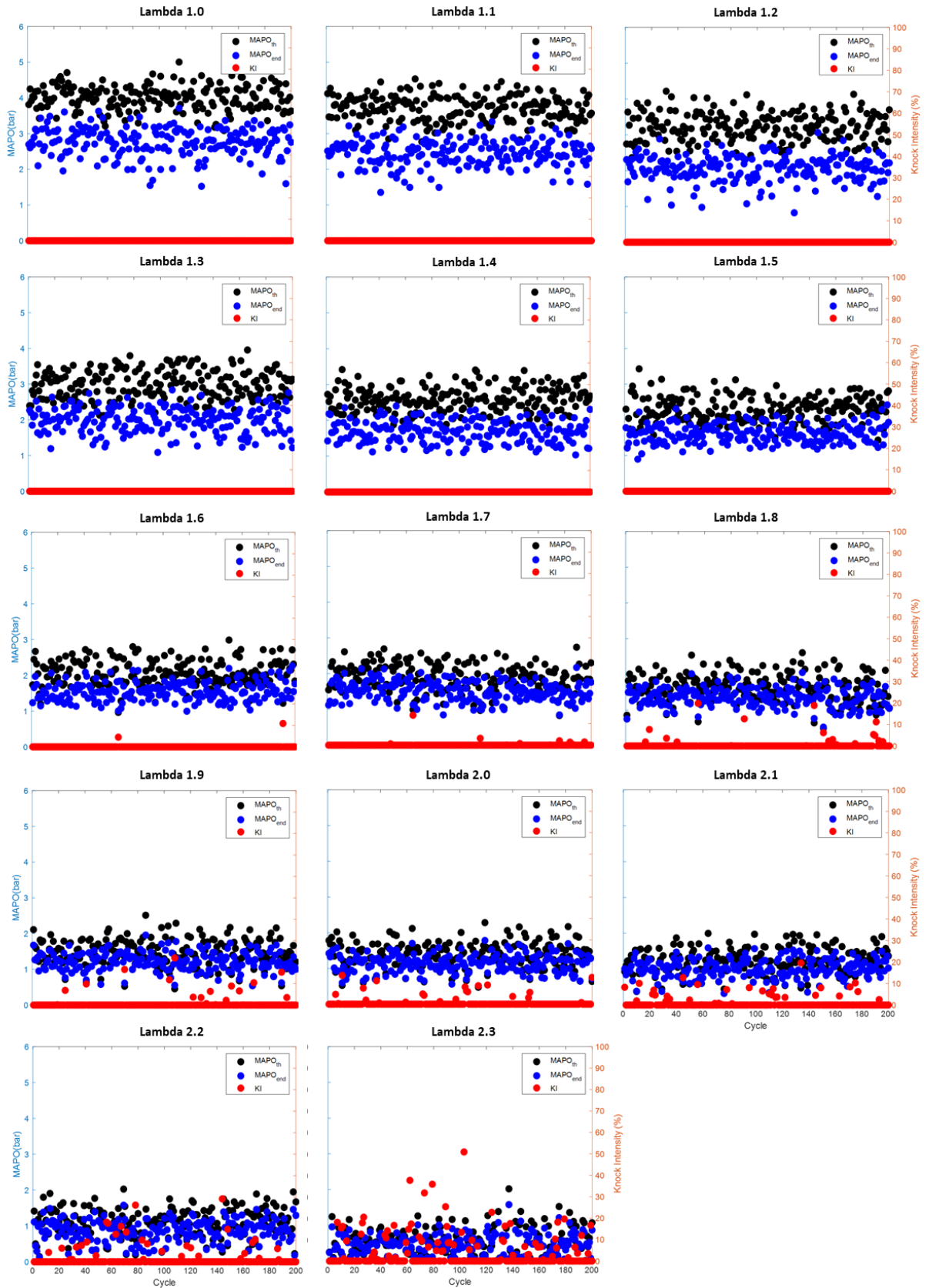


Figure D.15 - 3500 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

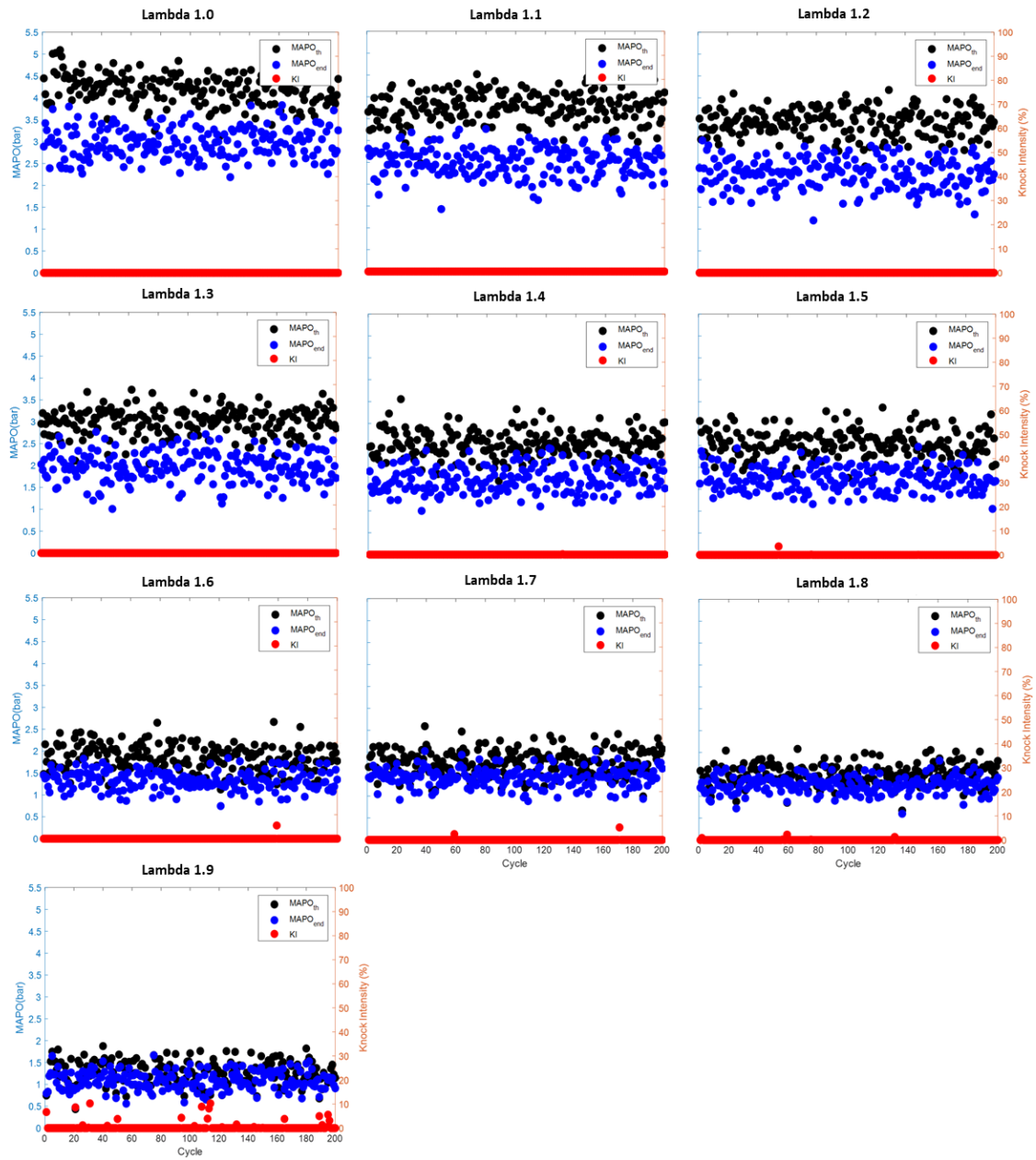


Figure D.16 - 3500 rpm and 15 bar of IMEP - MC-EtOH/PC-EtOH

Appendix E. UNCERTAINTY ANALYSIS

According to the definition given by ISO GUM [192], uncertainty characterizes a range of dispersion or interval, and not a point value and is the remaining doubt associated with the measurement result, measuring the degree of unfamiliarity about what is being measured. To ensure the reliability of the results, a metrological evaluation of the measurement systems used is carried out using values provided in the equipment manuals, as shown in Table 3.3, of the Methodology section.

Since a measurement result is an estimate of the true value of the measurand, uncertainty becomes necessary to express the degree of doubt associated with the measurement result. All components of uncertainty must be identified, linked to environmental conditions, operating conditions, equipment, methods and other factors. These must be used as input quantities in the mathematical measurement model. Also according to the GUM [192], another important step is the classification of uncertainties as “Type A” and “Type B”. The Type A assessment of uncertainty is the method that employs a statistical analysis of a series of repeated observations at the time of testing/calibration, and can be expressed by the experimental standard deviation of the mean, mentioned above. On the other hand, the Type B assessment employs other analysis methods, such as historical performance data of the measurement method, uncertainties inherited from the calibration of equipment and standards, specifications of equipment and standards, environmental conditions, among others. The Type “A” (u_A) and Type “B” (u_B) uncertainties can be combined by calculating the combined uncertainty (u_C), according to Equation E.1. Since no measurement repetition are performed in this work, the uncertainty due to repeatability (Type “A”) is disregarded, using Type “B” uncertainty. Hence, u_C will assume the same value observed for the Type “B” uncertainty [193].

$$u_c(y) = \sqrt{u_a(y)^2 + u_b(y)^2} \quad \text{E.1}$$

Once all the standard uncertainties are obtained, the law of propagation of uncertainties establishes that the standard uncertainties related to each variable of the mathematical model of the measurement must be propagated to generate a combined uncertainty $u_c(y)$. The uncertainty combination of random variables can be calculated using Equation E.2 is used, in which the variable f is a function of x , y and z and u_f , u_x , u_y and u_z are their respective uncertainties [194].

$$u_f = \sqrt{u_x^2 + u_y^2 + u_z^2 + \dots} \quad \text{E.2}$$

Applying the Central Limit Theorem, it can be said that the probability distribution of Y will be approximately normal, or t-Student for a given degree of freedom ν_{eff} . Assuming a normal probability distribution for Y , the interval of ± 1 standard deviation around the best estimate of the measurand corresponds to a coverage probability of approximately 68% [195], as presented in Figure E.1.

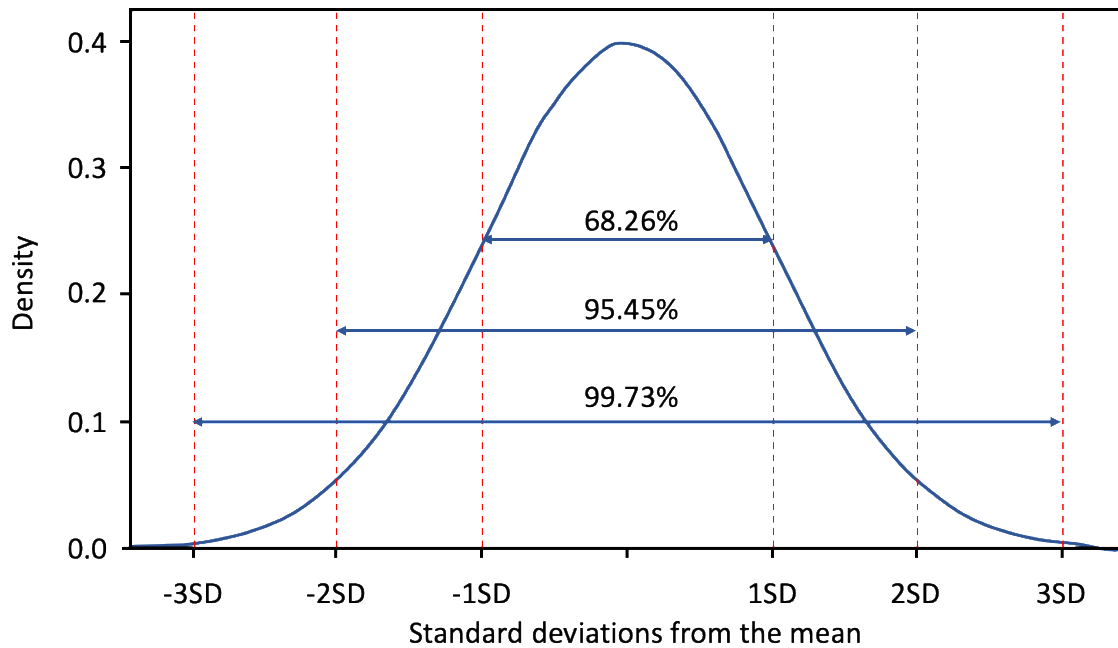


Figure E.1 - Assumed normal distribution for evaluating random errors through standard deviation [195]

To increase such coverage probability, one must multiply the combined uncertainty by the coverage factor t . The result is the so-called “expanded uncertainty”, U , according to Equation E.3.

$$U = t \cdot u_c(y) \quad \text{E.3}$$

In which t is defined for a given coverage probability. For a normal distribution and a coverage probability of 95.45%, $t = 2$ [152].

The uncertainties presented by the manufacturers are given as a percentage of the final scale value, which means that they already represent the expanded uncertainty. Therefore, for many measurements performed, the expanded measurement uncertainties can be directly calculated, as is the case of IMEP, MAPO, KI2 (obtained directly from the pressure measurement uncertainties), fuel mass and exhaust emissions. However, for the calculation of measurement uncertainty related to combustion and thermal efficiencies the use uncertainty propagation calculations is necessary. Thus, the standard uncertainty are obtained, considering that the expanded uncertainties provided follow a normal probability distribution of coverage 95.45%.

As already presented in the literature review section, the formulas for calculating η_c and η_t are shown in Equations E.4 and 5. Thus, the combined uncertainties of such parameters can be calculated as indicated by equation E.2 and are presented in Equations E.6 and 7.

$$\begin{aligned}
\eta_c &= \left(\frac{[H_{CO_2}(T_A) \cdot m_{CO_2} + H_{CO}(T_A) \cdot m_{CO} + H_{HC}(T_A) \cdot m_{HC} + H_{NO_x}(T_A) \cdot m_{NO_x}]}{m_f \cdot LHV} \right. \\
&\quad \left. - \frac{m_{air} \cdot [0.79 \cdot H_{N_2}(T_A) + 0.21 H_{O_2}(T_A)]}{m_f \cdot LHV} \right) \cdot 10^{-1} \\
&= \left(\frac{\dot{m}_{air} \cdot [H_{CO_2}(T_A) \cdot Y_{CO_2} + H_{CO}(T_A) \cdot Y_{CO} + H_{HC}(T_A) \cdot Y_{HC} + H_{NO_x}(T_A) \cdot Y_{NO_x}]}{m_f \cdot LHV} \right. \\
&\quad \left. - \frac{\dot{m}_{air} \cdot [0.79 \cdot H_{N_2}(T_A) + 0.21 H_{O_2}(T_A)]}{\dot{m}_f \cdot LHV} \right) \cdot 10^{-1}
\end{aligned} \tag{E.4}$$

$$\eta_t = \left(\frac{P_i}{\eta_c \cdot \dot{m}_f \cdot LHV} \right) \cdot 10^4 = \left(\frac{\frac{1}{2} \cdot N \cdot IMEP \cdot V_d}{\eta_c \cdot \dot{m}_f \cdot LHV} \right) \cdot 10^4 \tag{E.5}$$

$$u_{\eta_c} = \left[\left(u_{Y_{CO_2}} \right)^2 + \left(u_{Y_{CO}} \right)^2 + \left(u_{Y_{HC}} \right)^2 + \left(u_{Y_{NO_x}} \right)^2 + \left(u_{\dot{m}_{air}} \right)^2 + \left(-u_{\dot{m}_f} \right)^2 \right]^{\frac{1}{2}} \tag{E.6}$$

$$u_{\eta_t} = \left[(u_N)^2 + (u_{IMEP})^2 + \left(-\omega_{\dot{m}_f} \right)^2 + \left(-u_{\eta_c} \right)^2 \right]^{\frac{1}{2}} \tag{E.7}$$

According to Woulters et al. [151], the uncertainties of the remaining evaluated combustion parameters such as MFB50, burn delay, burn duration and indicated specific emissions are dominated by the fluctuations of the engine load that is, CoV_{IMEP} . Therefore, CoV_{IMEP} can be considered the accuracy of these combustion parameters. Since the calculations of the uncertainties of the experimental parameters addressed in the present work are explained, Table E.1 to Table E.16 show the raw data used with their respective uncertainties.

Table E.1 - 2000 rpm and 6 bar of IMEP – SI-MeOH

Lambda	Engine Speed		IMEP		COV IMEP	MEP 0.5%		MEP 0.10%		MEP 5-90%		MEP 10-75%		MEP 10-90%		Spark Timing		MEF 50		ISFC		ISFC corr.		MC Fuel Mass	
	Measured	Uncertainty	Measured	Uncertainty		Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty
-	rpm	rpm	bar	bar	%	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA/TOC/degCA/TOC	degCA/TOC/degCA/TOC	degCA/TOC/degCA/TOC	g/kWh	g/kWh	g/kWh	g/kWh	mg/cycle	mg/cycle
1.00	2000.10	0.80	6.02	0.01	0.48	15.45	0.08	16.39	0.08	13.66	0.07	8.87	0.04	12.39	0.06	-9.13	-0.04	473.72	2.33	222.20	1.09	37.83	0.02		
1.00	2000.20	0.80	6.02	0.01	0.50	23.35	0.12	17.01	0.09	13.64	0.07	8.88	0.04	12.34	0.06	-10.21	-0.05	472.85	2.37	221.80	1.11	37.83	0.02		
1.10	1999.72	0.80	6.01	0.01	0.43	13.45	0.06	17.03	0.07	13.64	0.06	9.01	0.04	12.32	0.05	-10.94	-0.05	458.42	1.99	215.02	0.93	36.69	0.02		
1.10	1999.40	0.80	5.98	0.01	0.42	14.05	0.06	16.86	0.07	13.94	0.06	9.14	0.04	12.63	0.05	-10.51	-0.04	458.21	1.92	215.40	0.90	36.66	0.02		
1.21	2000.14	0.80	6.02	0.01	0.58	14.75	0.09	17.07	0.10	14.52	0.08	9.57	0.06	13.14	0.08	-11.23	-0.07	450.29	2.63	211.21	1.23	36.16	0.02		
1.21	2000.60	0.80	6.00	0.01	0.54	14.45	0.08	17.02	0.09	14.14	0.08	9.45	0.05	12.75	0.07	-10.90	-0.06	446.30	2.42	209.34	1.14	36.23	0.02		
1.30	1999.64	0.80	6.00	0.01	0.56	16.10	0.08	18.22	0.10	15.36	0.09	10.12	0.06	13.89	0.08	-13.76	-0.08	442.19	2.49	207.41	1.17	35.48	0.02		
1.30	2000.36	0.80	5.99	0.01	0.50	15.80	0.08	17.72	0.09	14.95	0.07	9.92	0.05	13.49	0.07	-12.83	-0.06	442.56	2.19	207.59	1.03	35.31	0.02		
1.40	2000.31	0.80	6.00	0.01	0.56	16.15	0.09	18.26	0.10	16.06	0.09	10.59	0.06	14.48	0.08	-14.29	-0.08	435.32	2.44	204.19	1.15	34.88	0.02		
1.40	2000.05	0.80	5.96	0.01	0.59	14.25	0.08	17.93	0.11	16.04	0.10	10.50	0.06	14.45	0.09	-13.96	-0.08	438.38	2.59	204.69	1.21	34.80	0.02		
1.50	1999.96	0.80	5.99	0.01	0.59	15.95	0.09	18.17	0.11	17.39	0.10	11.12	0.07	15.66	0.09	-14.02	-0.09	431.71	2.56	202.50	1.20	34.77	0.02		
1.50	1999.94	0.80	6.01	0.01	0.62	15.25	0.09	18.45	0.11	16.91	0.10	10.95	0.07	15.21	0.09	-15.41	-0.10	429.76	2.67	201.58	1.25	34.58	0.02		
1.60	1999.87	0.80	6.02	0.01	0.83	17.25	0.14	19.10	0.16	18.17	0.15	11.58	0.10	16.27	0.13	-17.64	-0.15	427.90	3.55	200.71	1.66	34.68	0.02		
1.60	2000.53	0.80	6.02	0.01	0.69	16.05	0.11	19.45	0.14	18.10	0.13	11.67	0.08	16.24	0.11	-18.06	-0.13	426.40	2.96	200.01	1.39	34.42	0.02		
1.70	1999.93	0.80	6.01	0.01	1.03	17.65	0.18	19.81	0.20	19.60	0.20	12.42	0.13	17.60	0.18	-19.24	-0.20	425.71	4.38	199.71	2.05	34.45	0.02		
1.70	1998.75	0.80	6.04	0.01	1.07	17.80	0.19	19.63	0.21	19.03	0.20	12.14	0.13	17.08	0.18	-19.24	-0.21	423.94	4.53	198.85	2.12	34.50	0.02		
1.80	1998.75	0.80	6.04	0.01	1.40	19.75	0.28	20.56	0.29	20.60	0.29	12.95	0.18	18.52	0.26	-21.90	-0.30	423.46	5.93	198.63	2.78	34.42	0.02		
1.80	2000.80	0.80	6.06	0.01	1.13	17.35	0.20	20.76	0.23	19.91	0.22	12.59	0.14	17.82	0.20	-22.33	-0.25	420.93	4.74	197.44	2.22	34.16	0.02		
1.90	2000.59	0.80	5.99	0.01	3.26	18.55	0.61	20.93	0.68	21.41	0.70	13.55	0.44	19.28	0.63	-22.50	-0.73	421.15	13.75	197.54	6.45	34.21	0.02		
1.90	1998.96	0.80	5.98	0.01	1.92	21.15	0.41	20.75	0.40	21.60	0.41	13.65	0.26	19.46	0.37	-22.50	-0.43	422.30	8.11	198.08	3.80	34.07	0.02		
2.00	2001.09	0.80	5.95	0.01	4.81	19.20	0.92	22.64	1.09	22.92	1.10	14.37	0.69	20.73	1.00	-26.20	-1.26	424.66	20.43	199.19	9.58	34.38	0.02		
2.00	1997.34	0.80	5.96	0.01	5.63	17.55	0.99	22.39	1.26	22.51	1.27	14.17	0.80	20.31	1.14	-26.20	-1.47	425.14	23.93	199.41	11.23	34.47	0.02		

ISO2 emissions	ISO2 emissions		ISOx emissions		P peak		MAPQ threshold		K2		Air mass flow		Fuel Mass Flow		CO2 Percentage		CO Percentage		HC Percentage		NOx Percentage		Combustion Efficiency		Thermal Efficiency		
	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	
g/kWh	g/kWh	g/kWh	g/kWh	g/kWh	bar	bar	bar	bar	%	%	mg/s	mg/s	mg/s	mg/s	%	%	%	%	%	%	%	%	%	%	%	%	
632.98	6.33	8.47	0.08	2.00	0.06	48.73	0.06	0.16	0.00	1.82	0.00	4.27	0.04	0.63	0.00	11.28	0.11	0.24	0.00	0.04	0.00	0.18	0.00	99.46	0.12	38.72	0.71
631.42	6.31	8.57	0.09	1.99	0.06	48.61	0.06	0.15	0.00	1.80	0.00	4.26	0.04	0.63	0.00	11.28	0.11	0.24	0.00	0.04	0.00	0.18	0.00	99.46	0.12	38.46	0.71
621.01	6.21	2.90	0.03	2.14	0.02	49.53	0.06	0.15	0.00	1.92	0.00	4.52	0.05	0.61	0.00	10.99	0.11	0.08	0.00	0.04	0.00	0.23	0.00	99.60	0.12	41.36	0.97
622.17	6.22	2.90	0.03	2.15	0.02	48.95	0.06	0.14	0.00	1.98	0.00	4.51	0.05	0.61	0.00	10.99	0.11	0.08	0.00	0.04	0.00	0.23	0.00	99.60	0.12	40.14	0.97
603.32	6.03	3.96	0.04	3.20	0.03	49.60	0.06	0.12	0.00	1.15	0.00	4.88	0.05	0.60	0.00	9.99	0.10	0.10	0.00	0.05	0.00	0.18	0.00	99.40	0.11	42.42	0.85
598.13	5.98	3.87	0.04	3.13	0.03	50.18	0.07	0.11	0.00	1.96	0.00	4.90	0.05	0.60	0.00	9.70	0.10	0.10	0.00	0.05	0.00	0.18	0.00	99.41	0.11	41.75	0.85
585.79	5.86	4.88	0.05	4.53	0.05	49.46	0.06	0.11	0.00	0.89	0.00	5.14	0.05	0.59	0.00	8.99	0.09	0.11	0.00	0.07	0.00	0.11	0.00	99.17	0.10	42.82	0.82
588.13	5.88	4.69	0.05	4.50	0.05	49.84	0.06	0.12	0.00	1.62	0.00	5.13	0.05	0.59	0.00	8.99	0.09	0.11	0.00	0.07	0.00	0.11	0.00	99.17	0.10	42.82	0.82
570.39	5.71	5.00	0.05	5.85	0.06	4.34	0.02	50.07	0.07	1.20	0.00	5.43	0.05	0.58	0.00	8.37	0.08	0.12	0.00	0.09	0.00	0.06	0.00	98.94	0.10	42.94	0.81
561.59	5.62	5.14	0.05	5.96	0.06	4.31	0.02	50.23	0.07	1.20	0.00	5.40	0.05	0.58	0.00	8.37	0.08	0.12	0.00	0.09	0.00	0.06	0.00	98.94	0.10	42.94	0.81
558.23	5.59	5.04	0.05	6.93	0.07	2.24	0.01	50.75	0.07	1.48	0.00	5.76	0.06	0.58	0.00	7.83	0.08	0.11	0.00	0.10	0.00	0.03	0.00	98.75	0.10	42.86	0.80
554.26	5.54	5.11	0.05	5.13	0.07	51.13	0.07	0.11	0.00	1.48	0.00	5.75	0.06	0.58	0.00	7.83	0.08	0.11	0.00	0.10	0.00	0.03	0.00	98.75	0.10	42.86	0.80
551.95	5.52	5.11	0.05	5.20	0.07	52.01	0.07	0.10	0.00	1.31	0.00	6.10	0.06	0.58	0.00	7.38	0.07	0.11	0.00	0.11	0.00	0.02	0.00	98.62	0.10	43.40	0.80
548.30	5.48	5.22	0.05	7.86	0.08	1.30	0.01	51.54	0.07	1.30	0.00	6.08	0.06	0.57	0.00	7.38	0.07	0.11	0.00	0.11	0.00	0.02	0.00	98.61	0.10	43.94	0.80
546.11	5.46	5.25	0.05	8.68	0.09	0.67	0.00	51.89	0.07	1.23	0.00	6.43	0.06	0.57	0.00	6.96	0.07	0.11	0.00	0.11	0.00	0.01	0.00	98.48	0.09	44.41	0.80
541.08	5.41	5.78	0.06	9.35	0.09	0.71	0.00	52.66	0.07	1.07	0.00	6.42	0.06	0.57	0.00	6.95	0.07	0.11	0.00	0.11	0.00	0.01	0.00	98.48	0.09	43.67	0.80
537.42	5.37	5.67	0.06	9.33	0.09	0.40	0.00	52.93	0.07	1.46	0.00	6.80	0.07	0.57	0.00	6.96	0.07	0.11	0.00	0.11	0.00	0.01	0.00	98.35	0.09	44.99	0.80
531.35	5.31	6.64	0.07	10.36	0.10	0.23	0.00	53.97	0.07	2.16	0.00	7.09	0.07	0.57	0.00	6.56	0.07	0.11	0.00	0.11	0.00	0.01	0.00	98.33	0.09	43.73	0.99
527.42	5.27	10.36	0.10	10.36	0.10	0.23	0.00	53.21	0.07	2.16	0.00	7.09	0.07	0.57	0.00	6.17	0.06	0.12	0.00	0.12	0.00	0.00	0.00	98.17	0.09	44.42	0.80
533.85	5.34	6.59	0.07	10.20	0.10	0.24	0.00	53.30	0.07	1.05	0.00	7.10	0.07	0.57	0.00	6.18	0.06	0.12	0.00	0.12	0.00	0.00	0.00	98.19	0.09	43.99	0.80
527.99	5.28	8.10	0.08	11.99	0.12	0.16	0.00	53.62	0.07	1.73	0.00	7.47	0.07	0.57	0.00	5.81	0.06	0.14	0.00	0.13	0.00	0.00	0.00	97.89	0.09	43.91	0.81

Table E.2 - 2000 rpm and 6 bar of IMEP - SI-EtOH

Lambda	Engine Speed		IMEP		COV IMEP		MFB 0.5%		MFB 0.10%		MFB 0.75%		MFB 0.90%		Spark Timing		MFB 50		ISFC		ISFC corr		MC Fuel Mass		PC Fuel Mass		PC MC Percentage	
	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty
-	rpm	rpm	bar	bar	%	%	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	g/kWh	g/kWh	g/kWh	mg/cycle	mg/cycle	mg/cycle	mg/cycle	%	%
1.00	2000.13	0.80	6.01	0.01	0.44	0.08	15.35	0.07	17.06	0.08	9.54	0.04	13.91	0.06	-10.82	-0.05	7.67	0.03	358.44	1.38	228.37	1.00	16.15	4.49	0.04	1.77	0.49	0.00
1.00	1999.88	0.80	5.97	0.01	0.49	0.08	23.05	0.11	18.86	0.08	9.42	0.05	13.70	0.07	-10.56	-0.05	7.49	0.04	368.34	1.77	226.30	1.12	16.06	4.46	0.04	1.74	0.48	0.00
1.10	1999.68	0.80	5.90	0.01	0.47	0.08	15.35	0.07	17.40	0.08	9.72	0.05	13.87	0.07	-12.04	-0.06	7.44	0.03	346.50	1.63	218.82	1.03	17.14	4.76	0.05	1.70	0.47	0.00
1.10	1999.74	0.80	6.00	0.01	0.46	0.08	14.15	0.07	17.11	0.08	9.69	0.04	13.74	0.06	-11.55	-0.05	7.40	0.03	346.84	1.60	218.41	1.01	17.13	4.76	0.05	1.71	0.48	0.00
1.20	2000.24	0.80	6.01	0.01	0.49	0.08	16.65	0.08	17.57	0.09	10.17	0.05	14.29	0.07	-12.75	-0.06	7.55	0.04	339.99	1.88	214.71	1.06	18.39	5.11	0.05	1.68	0.47	0.00
1.20	1999.92	0.80	6.07	0.01	0.52	0.08	15.90	0.08	17.70	0.09	10.01	0.05	14.08	0.07	-13.17	-0.07	7.37	0.04	339.25	1.77	214.25	1.12	18.55	5.15	0.05	1.69	0.47	0.00
1.30	1999.68	0.80	5.97	0.01	0.55	0.08	13.90	0.08	17.92	0.08	10.67	0.06	15.10	0.08	-14.08	-0.08	7.44	0.04	333.09	1.83	210.35	1.16	19.34	5.37	0.05	1.64	0.46	0.00
1.30	2000.10	0.80	5.95	0.01	0.52	0.08	16.65	0.09	18.23	0.09	10.74	0.06	15.18	0.08	-14.67	-0.08	7.52	0.04	333.42	1.73	210.56	1.09	19.31	5.36	0.05	1.65	0.46	0.00
1.40	2000.06	0.80	6.01	0.01	0.62	0.08	16.45	0.10	18.65	0.12	11.38	0.07	16.20	0.10	-16.15	-0.10	7.53	0.05	328.54	2.03	207.49	1.28	20.72	5.75	0.06	1.63	0.45	0.00
1.41	1999.79	0.80	6.00	0.01	0.63	0.08	16.20	0.10	18.51	0.12	11.50	0.07	16.40	0.10	-15.89	-0.10	7.61	0.05	328.45	2.07	207.42	1.31	20.66	5.74	0.06	1.64	0.45	0.00
1.51	2000.25	0.80	6.06	0.01	0.73	0.08	18.55	0.14	19.43	0.14	12.04	0.09	17.32	0.13	-18.70	-0.14	7.36	0.05	324.26	2.38	204.78	1.50	22.08	6.13	0.06	1.64	0.46	0.00
1.51	1999.91	0.80	6.06	0.01	0.93	0.08	18.45	0.17	19.48	0.18	12.27	0.11	17.76	0.17	-18.69	-0.17	7.52	0.07	324.63	3.03	205.01	1.92	22.10	6.14	0.06	1.63	0.45	0.00
1.61	1999.92	0.80	6.00	0.01	1.15	0.08	20.09	0.23	20.85	0.24	12.95	0.15	18.70	0.21	-20.79	-0.24	7.48	0.09	323.17	3.71	204.09	2.34	23.16	6.43	0.06	1.61	0.45	0.00
1.61	2000.18	0.80	5.99	0.01	1.26	0.08	17.75	0.22	20.42	0.26	13.01	0.16	18.77	0.24	-21.38	-0.27	7.52	0.09	324.02	4.07	204.63	2.57	23.19	6.44	0.06	1.62	0.45	0.00
1.71	2000.41	0.80	6.01	0.01	2.68	0.08	21.35	0.57	21.99	0.59	14.42	0.39	21.15	0.57	-24.60	-0.66	8.17	0.22	324.05	8.67	204.64	5.48	24.76	6.88	0.07	1.62	0.45	0.00
1.71	1999.03	0.80	6.04	0.01	2.44	0.08	21.67	0.68	21.67	0.53	14.32	0.35	20.86	0.51	-23.85	-0.58	8.23	0.20	323.92	7.91	204.57	5.00	24.84	6.90	0.07	1.64	0.46	0.00
1.81	2000.76	0.80	6.02	0.01	3.92	0.08	19.90	0.78	21.99	0.86	14.50	0.57	21.20	0.83	-26.10	-1.02	7.54	0.30	324.20	12.71	204.74	8.02	26.20	7.28	0.07	1.63	0.45	0.00
1.81	1999.37	0.80	6.03	0.01	4.81	0.08	21.35	1.03	22.05	1.06	14.35	0.69	20.97	1.00	-26.10	-1.25	7.51	0.36	323.90	15.57	204.55	9.83	26.20	7.28	0.07	1.64	0.46	0.00
1.88	1998.65	0.80	5.93	0.01	8.05	0.08	20.50	1.65	23.87	1.92	15.50	1.25	22.46	1.81	-28.60	-2.30	8.70	0.70	328.60	26.44	207.52	16.70	27.10	7.53	0.08	1.63	0.45	0.00
1.88	2000.91	0.80	5.90	0.01	7.68	0.08	21.95	1.69	24.15	1.85	15.66	1.20	22.77	1.75	-28.60	-2.23	9.05	0.69	329.50	25.31	208.09	15.99	27.08	7.52	0.08	1.64	0.45	0.00

ISO2 emissions	ISO2 emissions		ISOx emissions		P peak		MAP0 threshold		K2		Air mass flow		Fuel Mass Flow		CO2 Percentage		CO Percentage		HC Percentage		NOx Percentage		Combustion Efficiency		Thermal Efficiency				
	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty			
g/kWh	g/kWh	g/kWh	g/kWh	g/kWh	bar	bar	bar	bar	%	%	mg/s	mg/s	mg/s	mg/s	%	%	%	%	%	%	%	%	%	%	%	%			
11.90	0.12	2945.58	0.29	0.00	583.21	0.06	0.00	2113.39	0.21	0.00	24.45	0.02	0.04	4.49	0.04	0.49	0.00	11.90	0.12	0.29	0.00	0.06	0.00	0.21	0.00	99.26	0.13	40.21	0.62
11.90	0.12	2919.54	0.29	0.00	582.04	0.06	0.00	2111.91	0.21	0.00	28.98	0.02	0.04	4.46	0.04	0.48	0.00	11.90	0.12	0.29	0.00	0.06	0.00	0.21	0.00	99.26	0.13	39.39	0.62
11.10	0.11	930.73	0.09	0.00	604.76	0.06	0.00	2844.25	0.28	0.00	28.39	0.02	0.04	4.76	0.05	0.47	0.00	11.10	0.11	0.09	0.00	0.06	0.00	0.28	0.00	99.40	0.12	39.44	0.57
11.11	0.11	925.17	0.09	0.00	601.81	0.06	0.00	2872.65	0.29	0.00	28.55	0.02	0.04	4.76	0.05	0.48	0.00	11.11	0.11	0.09	0.00	0.06	0.00	0.29	0.00	99.41	0.12	39.08	0.58
10.17	0.10	1108.37	0.11	0.00	767.78	0.08	0.00	2275.94	0.23	0.00	27.62	0.02	0.04	5.11	0.05	0.47	0.00	10.17	0.10	0.08	0.00	0.08	0.00	0.23	0.00	99.19	0.11	39.16	0.55
10.17	0.10	1105.02	0.11	0.00	766.88	0.08	0.00	2338.22	0.23	0.00	28.20	0.02	0.04	5.15	0.05	0.47	0.00	10.17	0.10	0.11	0.00	0.08	0.00	0.23	0.00	99.19	0.11	39.73	0.55
9.37	0.09	1235.81	0.12	0.00	962.00	0.10	0.00	1394.78	0.14	0.00	27.33	0.01	0.04	5.37	0.05	0.46	0.00	9.37	0.09	0.12	0.00	0.10	0.00	0.14	0.00	98.93	0.11	40.20	0.51
9.36	0.09	1237.33	0.12	0.00	971.68	0.10	0.00	1386.51	0.14	0.00	27.54	0.01	0.04	5.36	0.05	0.46	0.00	9.36	0.09	0.12	0.00	0.10	0.00	0.14	0.00	98.93	0.11	40.49	0.51
8.67	0.09	1208.67	0.12	0.00	1115.56	0.11	0.00	722.57	0.07	0.00	27.23	0.01	0.04	5.75	0.06	0.45	0.00	8.67	0.09	0.12	0.00	0.11	0.00	0.07	0.00	98.71	0.10	40.92	0.50
8.67	0.09	1208.58	0.12	0.00	1128.20	0.11	0.00	720.48	0.07	0.00	27.27	0.01	0.04	5.74	0.06	0.45	0.00	8.67	0.09	0.12	0.00	0.11	0.00	0.07	0.00	98.70	0.10	41.60	0.50
8.09	0.08	1112.80	0.11	0.00	1247.90	0.12	0.00	388.02	0.04	0.00	27.31	0.01	0.04	6.13	0.06	0.46	0.00	8.09	0.08	0.11	0.00	0.12	0.00	0.04	0.00	98.51	0.10	41.75	0.49
8.09	0.08	1107.42	0.11	0.00	1236.45	0.12	0.00	382.48	0.04	0.00	27.21	0.01	0.04	6.14	0.06	0.45	0.00	8.09	0.08	0.11	0.00	0.12	0.00	0.04	0.00	98.52	0.10	41.55	0.49
7.59	0.08	1006.00	0.10	0.00	1357.95	0.14	0.00	195.90	0.02	0.00	26.89	0.01	0.04	6.43	0.06	0.45	0.00	7.59	0.08	0.10	0.00	0.14	0.00	0.02	0.00	98.31	0.10	40.11	0.48
7.60	0.08	1005.48	0.10	0.00	1335.50	0.13	0.00	193.10	0.02	0.00	26.95	0.01	0.04	6.44	0.06	0.45	0.00	7.60	0.08	0.10	0.00	0.13	0.00	0.02	0.00	98.34	0.10	40.36	0.49
7.10	0.07	1037.88	0.10	0.00	1481.48	0.15	0.00	92.91	0.01	0.00	27.07	0.01	0.04	6.88	0.07	0.45	0.00	7.10	0.07	0.10	0.00	0.15	0.00	0.01	0.00	98.07	0.10	41.64	0.49
7.11	0.07	1032.88	0.10	0.00	1456.47	0.15	0.00	86.98	0.01	0.00	27.33	0.01	0.																

Table E.3 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-MeOH

[illegible]

Table E.4 - 2000 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

Lambda	Engine Speed rpm	IMEP bar	COV / IMEP %	IMEP 0.5%		IMEP 1.0%		IMEP 1.5%		IMEP 2.0%		Spark Timing degCA	MFB 50		ISFC g/kWh	ISFC corr g/kWh	MC Fuel Mass		PC Fuel Mass		PC MC Percentage										
				Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty		Measurement	Uncertainty			Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty					
1.00	1990.24	0.83	5.99	0.01	0.81	14.80	0.12	16.85	0.14	7.22	0.06	5.07	0.04	6.44	0.05	-4.41	-0.03	7.56	0.06	386.47	3.11	242.95	1.92	31.71	0.02	0.40	0.00	0.40	0.00	1.27	0.00
1.00	2000.12	0.83	5.97	0.01	0.77	14.25	0.11	16.77	0.13	7.47	0.06	5.29	0.04	6.70	0.05	-5.74	-0.04	7.55	0.06	386.13	3.10	242.86	1.92	31.69	0.02	0.40	0.00	0.40	0.00	1.28	0.00
1.00	1999.73	0.83	5.99	0.01	0.77	14.65	0.11	16.56	0.13	7.53	0.06	5.32	0.04	6.70	0.05	-5.73	-0.04	7.56	0.06	385.81	3.09	242.77	1.92	31.67	0.02	0.40	0.00	0.40	0.00	1.30	0.00
1.00	2000.24	0.83	6.04	0.01	0.77	14.65	0.11	16.56	0.13	7.53	0.06	5.32	0.04	6.70	0.05	-5.73	-0.04	7.56	0.06	385.81	3.09	242.77	1.92	31.67	0.02	0.40	0.00	0.40	0.00	1.30	0.00
1.20	1995.72	0.83	6.05	0.01	0.79	14.60	0.12	16.42	0.13	7.96	0.06	5.67	0.04	7.19	0.06	-5.59	-0.04	7.52	0.06	385.50	2.85	231.23	1.79	30.85	0.02	0.40	0.00	0.40	0.00	1.31	0.00
1.30	2000.20	0.83	6.02	0.01	0.77	14.60	0.11	16.31	0.13	8.52	0.07	6.16	0.05	7.75	0.06	-6.08	-0.05	7.52	0.06	385.00	2.78	229.83	1.84	30.83	0.02	0.40	0.00	0.40	0.00	1.34	0.00
1.30	1999.73	0.83	6.00	0.01	0.73	14.85	0.11	16.78	0.12	8.52	0.08	6.15	0.04	7.75	0.06	-5.99	-0.05	7.54	0.06	385.44	2.61	226.79	1.80	30.14	0.02	0.40	0.00	0.40	0.00	1.34	0.00
1.40	1999.73	0.83	6.00	0.01	0.73	14.85	0.11	16.78	0.12	8.52	0.08	6.15	0.04	7.75	0.06	-5.99	-0.05	7.54	0.06	385.44	2.61	226.79	1.80	30.14	0.02	0.40	0.00	0.40	0.00	1.34	0.00
1.40	1999.65	0.83	5.96	0.01	0.85	15.25	0.13	16.58	0.14	9.25	0.08	6.70	0.06	8.45	0.07	-7.40	-0.06	7.49	0.06	385.51	3.04	229.40	1.91	29.32	0.02	0.40	0.00	0.40	0.00	1.38	0.00
1.41	2000.37	0.83	6.01	0.01	0.88	14.70	0.13	16.56	0.15	9.29	0.08	6.75	0.06	8.50	0.07	-7.33	-0.06	7.55	0.07	385.78	3.11	229.50	1.94	29.68	0.02	0.40	0.00	0.40	0.00	1.36	0.00
1.41	1999.57	0.83	6.02	0.01	0.77	14.90	0.12	16.70	0.13	9.31	0.07	6.75	0.06	8.50	0.07	-7.59	-0.06	7.56	0.06	385.69	2.75	222.95	1.78	29.67	0.02	0.40	0.00	0.40	0.00	1.36	0.00
1.50	1999.67	0.83	5.99	0.01	0.84	14.65	0.12	17.01	0.14	9.92	0.09	7.18	0.06	9.00	0.08	-8.86	-0.07	7.52	0.06	384.51	2.92	218.95	1.83	29.01	0.02	0.40	0.00	0.40	0.00	1.38	0.00
1.51	1999.72	0.83	5.97	0.01	0.77	23.65	0.18	17.86	0.14	10.90	0.09	7.13	0.06	9.68	0.07	-11.72	-0.09	7.27	0.06	384.49	2.72	218.99	1.71	29.10	0.02	0.40	0.00	0.40	0.00	1.39	0.00
1.61	2000.15	0.83	6.00	0.01	0.84	15.40	0.13	17.52	0.11	11.92	0.08	7.41	0.06	10.66	0.07	-11.53	-0.07	7.56	0.05	383.65	2.18	210.81	1.38	27.95	0.02	0.52	0.00	0.52	0.00	1.86	0.00
1.61	1999.76	0.83	6.01	0.01	0.59	15.50	0.09	18.26	0.11	13.05	0.08	9.11	0.05	11.64	0.07	-13.86	-0.08	7.57	0.04	383.32	1.97	208.10	1.22	27.42	0.01	0.75	0.00	0.75	0.00	2.74	0.00
1.81	1999.76	0.83	6.02	0.01	0.59	17.15	0.10	18.18	0.11	13.16	0.08	9.19	0.05	11.74	0.07	-13.94	-0.08	7.72	0.05	383.32	1.95	207.91	1.22	27.51	0.01	0.75	0.00	0.75	0.00	2.73	0.00
1.91	1999.97	0.83	6.03	0.01	0.75	22.40	0.17	18.16	0.14	14.41	0.11	10.86	0.07	12.70	0.08	-14.48	-0.11	7.63	0.06	382.88	2.45	205.57	1.53	27.06	0.01	0.98	0.00	0.98	0.00	3.62	0.00
1.91	1999.97	0.83	6.02	0.01	0.76	16.45	0.13	18.15	0.14	15.73	0.12	10.47	0.08	13.59	0.10	-15.50	-0.12	7.48	0.06	382.68	2.50	204.62	1.56	26.79	0.01	1.21	0.00	1.21	0.00	4.52	0.00
2.01	2000.31	0.83	5.99	0.01	0.76	16.45	0.13	18.15	0.14	15.73	0.12	10.47	0.08	13.59	0.10	-15.50	-0.12	7.48	0.06	382.68	2.50	204.62	1.56	26.79	0.01	1.21	0.00	1.21	0.00	4.52	0.00
2.01	2000.29	0.83	5.98	0.01	0.84	15.00	0.13	18.11	0.15	15.57	0.13	10.37	0.09	13.43	0.11	-15.56	-0.13	7.33	0.06	382.16	2.75	204.99	1.72	26.74	0.01	1.21	0.00	1.21	0.00	4.53	0.00
2.11	1999.77	0.83	5.99	0.01	1.07	14.85	0.16	19.56	0.17	17.23	0.18	11.30	0.12	14.70	0.13	-19.44	-0.21	7.34	0.08	380.51	2.75	204.99	2.18	26.74	0.01	1.56	0.00	1.56	0.00	9.88	0.00
2.11	2000.17	0.83	5.99	0.01	1.02	14.15	0.14	19.92	0.13	17.35	0.18	11.36	0.12	14.94	0.12	-18.22	-0.13	7.41	0.08	413.87	4.21	257.96	4.69	26.95	0.01	1.96	0.00	1.96	0.00	9.87	0.00
2.21	1999.65	0.83	5.99	0.01	1.02	14.15	0.14	19.92	0.13	17.35	0.18	11.36	0.12	14.94	0.12	-18.22	-0.13	7.41	0.08	413.87	4.21	257.96	4.69	26.95	0.01	1.96	0.00	1.96	0.00	9.87	0.00
2.30	2000.31	0.83	6.08	0.01	2.53	13.40	0.34	21.62	0.55	20.14	0.51	12.81	0.32	16.83	0.43	-26.51	-0.67	6.98	0.18	381.83	8.38	206.27	5.21	26.80	0.01	2.25	0.00	2.25	0.00	8.39	0.00
2.30	1999.71	0.83	6.11	0.01	2.07	13.30	0.28	18.88	0.38	19.80	0.41	12.66	0.28	15.99	0.34	-21.86	-0.45	6.90	0.13	381.28	6.85	205.86	4.28	26.91	0.01	2.25	0.00	2.25	0.00	8.37	0.00
2.31	2000.00	0.83	6.03	0.01	2.56	12.55	0.32	19.35	0.49	20.63	0.53	13.18	0.34	17.33	0.44	-21.74	-0.56	7.28	0.15	382.81	8.50	206.63	5.29	26.69	0.01	2.25	0.00	2.25	0.00	8.44	0.00
2.31	2000.00	0.83	6.03	0.01	2.56	12.55	0.32	19.35	0.49	20.63	0.53	13.18	0.34	17.33	0.44	-21.74	-0.56	7.28	0.15	382.81	8.50	206.63	5.29	26.69	0.01	2.25	0.00	2.25	0.00	8.44	0.00
2.40	2000.35	0.83	6.00	0.01	3.66	13.45	0.49	20.18	0.74	21.94	0.80	13.87	0.51	18.25	0.67	-24.71	-0.90	7.13	0.26	342.39	12.52	212.37	7.76	26.88	0.01	2.71	0.00	2.71	0.00	10.09	0.01
2.40	2000.81	0.83	6.02	0.01	3.18	12.10	0.39	19.68	0.63	21.86	0.70	13.86	0.44	18.22	0.58	-23.90	-0.76	7.04	0.22	341.71	10.88	211.75	6.74	26.88	0.01	2.71	0.00	2.71	0.00	10.09	0.01
2.40	2000.81	0.83	6.02	0.01	3.18	12.10	0.39	19.68	0.63	21.86	0.70	13.86	0.44	18.22	0.58	-23.90	-0.76	7.04	0.22	341.71	10.88	211.75	6.74	26.88	0.01	2.71	0.00	2.71	0.00	10.09	0.01
2.51	1999.14	0.83	5.99	0.01	4.86	11.85	0.98	21.90	1.07	23.87	1.17	15.00	0.73	19.71	0.94	-29.07	-1.47	7.30	0.38	382.45	17.23	211.12	10.66	27.44	0.01	3.18	0.00	3.18	0.00	11.99	0.01
2.60	2002.74	0.83	5.54	0.01	14.69	11.20	1.65	16.26	2.39	25.51	2.56	15.93	2.34	20.75	3.03	-20.61	-3.03	6.79	1.00	368.95	54.34	228.21	33.52	26.48	0.01	3.40	0.00	3.40	0.00	12.84	0.01
2.60	2																														

Table E.5 - 2000 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

Lambda	Engine Speed		IMEP	COV IMEP	MFB 0.5%		MFB 0.10%		MFB 5.00%		MFB 10.75%		MFB 10.90%		Spark Timing		MFB 50		ISFC		ISFC cor		MC Fuel Mass		PC Fuel Mass		PC MC Percentage		
	Measured	Uncertainty			degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA
1.00	1999.94	0.80	6.01	0.01	0.85	15.10	0.13	16.58	0.14	7.23	0.06	5.08	0.04	6.56	0.05	-5.00	-4.04	7.59	0.06	380.47	3.23	240.28	2.04	31.64	0.02	0.40	0.00	1.28	0.00
1.01	1999.80	0.80	6.00	0.01	0.87	14.65	0.13	16.62	0.15	7.29	0.06	5.13	0.04	6.56	0.06	-5.10	-4.04	7.61	0.07	380.76	3.23	240.46	2.10	31.72	0.02	0.40	0.00	1.27	0.00
1.10	1998.51	0.80	6.01	0.01	0.71	15.15	0.11	16.07	0.11	7.56	0.05	5.36	0.04	6.80	0.05	-4.56	-4.03	7.48	0.05	373.12	2.63	235.63	1.66	30.95	0.02	0.40	0.00	1.31	0.00
1.11	1998.55	0.80	6.02	0.01	0.71	14.70	0.10	16.47	0.12	7.54	0.05	5.34	0.04	6.78	0.05	-5.30	-4.04	7.51	0.05	372.49	2.64	235.24	1.67	31.09	0.02	0.40	0.00	1.30	0.00
1.20	1999.33	0.80	5.96	0.01	0.72	15.60	0.11	16.42	0.12	8.08	0.06	5.79	0.04	7.32	0.05	-5.71	-4.04	7.57	0.05	366.75	2.65	231.01	1.67	29.97	0.02	0.40	0.00	1.35	0.00
1.21	1999.33	0.80	5.95	0.01	0.69	15.10	0.10	16.39	0.11	8.15	0.06	5.85	0.04	7.39	0.05	-5.63	-4.04	7.62	0.05	367.93	2.53	232.36	1.60	30.16	0.02	0.40	0.00	1.34	0.00
1.30	1999.73	0.80	5.98	0.01	0.76	15.25	0.12	16.31	0.12	8.66	0.07	6.25	0.05	7.88	0.06	-6.46	-4.05	7.44	0.06	358.69	2.74	226.52	1.73	29.91	0.02	0.40	0.00	1.35	0.00
1.31	1999.97	0.80	5.99	0.01	0.73	14.95	0.11	16.31	0.12	8.71	0.06	6.29	0.05	7.92	0.06	-6.43	-4.05	7.44	0.05	358.92	2.63	226.67	1.66	29.40	0.02	0.40	0.00	1.37	0.00
1.41	1999.63	0.80	5.99	0.01	0.76	15.15	0.12	16.67	0.13	9.44	0.07	6.81	0.06	8.60	0.07	-7.63	-4.06	7.53	0.06	355.53	2.72	224.54	1.72	29.14	0.02	0.40	0.00	1.39	0.00
1.44	1999.58	0.80	5.98	0.01	0.88	15.15	0.13	16.36	0.15	9.46	0.08	6.84	0.06	8.63	0.08	-6.98	-4.06	7.57	0.07	354.43	3.16	224.46	2.00	29.18	0.02	0.40	0.00	1.39	0.00
1.51	2000.38	0.80	5.97	0.01	0.86	15.80	0.14	17.28	0.15	10.15	0.09	7.28	0.06	9.18	0.08	-9.34	-4.08	7.60	0.07	350.32	3.01	221.24	1.90	28.72	0.02	0.40	0.00	1.40	0.00
1.51	1999.53	0.80	5.97	0.01	0.87	15.20	0.13	17.26	0.15	10.15	0.09	7.29	0.06	9.19	0.08	-9.28	-4.08	7.61	0.07	350.53	3.05	221.37	1.93	28.85	0.02	0.40	0.00	1.41	0.00
1.60	1999.62	0.80	5.98	0.01	0.78	14.80	0.12	17.56	0.14	11.03	0.09	7.88	0.06	9.95	0.08	-10.90	-4.09	7.54	0.06	346.71	2.70	218.32	1.71	28.08	0.02	0.52	0.00	1.85	0.00
1.60	1999.61	0.80	5.99	0.01	0.76	16.00	0.12	16.96	0.13	10.98	0.08	7.86	0.06	9.90	0.08	-9.77	-4.07	7.51	0.06	346.88	2.64	219.07	1.67	28.09	0.02	0.52	0.00	1.85	0.00
1.71	1999.84	0.80	6.01	0.01	0.64	15.95	0.10	18.27	0.12	12.20	0.08	8.54	0.05	10.86	0.07	-13.56	-4.08	7.58	0.05	338.77	2.16	213.94	1.36	27.62	0.01	0.52	0.00	1.88	0.00
1.71	1999.84	0.80	6.01	0.01	0.66	16.45	0.11	18.54	0.12	12.17	0.08	8.53	0.06	10.85	0.07	-13.56	-4.08	7.62	0.05	339.65	2.23	214.12	1.41	27.64	0.01	0.52	0.00	1.88	0.00
1.80	1999.98	0.80	6.00	0.01	0.61	16.15	0.10	18.93	0.11	13.52	0.08	9.26	0.06	11.92	0.07	-14.60	-4.09	7.54	0.05	335.35	2.03	212.78	1.28	27.14	0.01	0.75	0.00	2.77	0.00
1.80	2000.12	0.80	5.99	0.01	0.62	16.30	0.10	18.46	0.11	13.58	0.08	9.32	0.06	11.90	0.07	-14.44	-4.09	7.61	0.05	338.42	2.10	212.46	1.32	27.37	0.01	0.75	0.00	2.74	0.00
1.91	1999.40	0.80	5.98	0.01	0.75	23.10	0.17	18.57	0.14	14.89	0.11	9.97	0.07	12.94	0.10	-15.90	-4.12	7.41	0.06	333.80	2.49	210.80	1.57	26.89	0.01	0.98	0.00	3.65	0.00
1.91	2000.05	0.80	5.99	0.01	0.75	23.10	0.17	18.57	0.14	14.89	0.11	9.97	0.07	12.94	0.10	-15.90	-4.12	7.41	0.06	333.80	2.49	210.80	1.57	26.89	0.01	0.98	0.00	3.65	0.00
2.01	1999.34	0.80	5.99	0.01	1.56	15.85	0.25	22.21	0.35	16.64	0.26	10.94	0.17	14.31	0.22	-23.83	-4.37	7.61	0.12	332.05	5.19	209.70	3.28	26.82	0.01	1.21	0.00	4.56	0.00
2.01	2000.25	0.80	5.99	0.01	1.84	15.70	0.29	21.72	0.40	16.44	0.30	10.80	0.20	14.14	0.26	-22.83	-4.36	7.56	0.14	332.42	6.11	209.93	3.88	26.64	0.01	1.21	0.00	4.54	0.00
2.11	1999.87	0.80	5.98	0.01	1.51	14.55	0.22	19.80	0.30	17.91	0.27	11.58	0.17	15.25	0.23	-20.15	-4.30	7.45	0.11	333.19	5.03	210.42	3.16	26.51	0.01	1.56	0.00	5.88	0.00
2.11	2000.39	0.80	5.96	0.01	1.52	15.10	0.23	19.92	0.30	17.86	0.27	11.59	0.18	15.22	0.23	-20.74	-4.32	7.28	0.11	334.26	5.08	211.09	3.21	26.55	0.01	1.56	0.00	5.97	0.00
2.21	1999.97	0.80	5.95	0.01	2.21	13.55	0.91	39.56	0.43	19.77	0.44	12.72	0.28	16.79	0.37	-21.24	-4.47	7.39	0.16	337.76	7.46	213.31	4.71	26.22	0.01	2.14	0.00	8.14	0.00
2.21	2001.02	0.80	5.96	0.01	1.86	13.95	0.26	19.83	0.37	19.65	0.37	12.52	0.24	16.66	0.31	-21.63	-4.40	7.46	0.14	338.83	6.26	212.72	3.95	26.19	0.01	2.14	0.00	8.15	0.00
2.30	2000.36	0.80	6.01	0.01	3.31	12.50	0.41	20.05	0.68	21.15	0.70	13.52	0.45	17.85	0.59	-24.46	-4.81	7.40	0.25	335.59	11.25	214.46	7.70	26.60	0.01	2.48	0.00	9.33	0.01
2.32	2000.20	0.80	6.07	0.01	2.62	12.75	0.33	19.49	0.51	20.50	0.54	13.07	0.43	17.24	0.45	-22.62	-4.59	6.92	0.18	338.08	8.88	214.14	5.61	26.98	0.01	2.48	0.00	9.26	0.01
2.40	2000.80	0.80	6.02	0.01	6.51	13.45	0.88	22.84	1.49	22.61	1.47	14.25	0.93	18.78	1.22	-29.95	-1.95	7.35	0.48	350.80	22.85	221.54	14.43	26.98	0.01	2.83	0.00	10.48	0.01
2.40	1999.54	0.80	6.02	0.01	5.33	14.60	0.78	21.99	1.17	22.89	1.22	14.44	0.77	19.02	1.01	-28.35	-1.51	7.43	0.40	349.46	18.64	220.70	11.77	27.33	0.01	2.83	0.00	10.35	0.01

Table E.6 - 2000 rpm and 15 bar of IMEP - SI-EtOH

Lambda Measured	Engine Speed rpm	IMEP		COV IMEP		MFB 0.5%		MFB 0.10%		MFB 5.0%		MFB 10.75%		MFB 10.90%		Spark Timing		MFB 50		ISFC		ISFC corr		IMEP Fuel Mass	
		Measurement	Uncertainty	bar	bar	Measurement	Uncertainty	degCA	degCA	Measurement	Uncertainty	degCA	degCA	Measurement	Uncertainty	degCA	degCA	Measurement	Uncertainty	g/kWh	g/kWh	Measurement	Uncertainty	g/kWh	g/kWh
1.00	2000.41	0.80	14.98	0.02	0.32	14.95	0.05	16.30	0.05	13.37	0.04	8.23	0.03	12.11	0.04	-8.21	-0.03	7.53	0.02	315.27	1.01	199.10	0.64	65.25	0.04
1.00	2000.03	0.80	15.00	0.02	0.31	15.75	0.04	16.41	0.05	13.43	0.04	8.28	0.03	12.18	0.04	-8.46	-0.03	7.50	0.02	314.91	0.97	198.88	0.61	65.46	0.04
1.10	2000.30	0.80	15.01	0.02	0.26	14.75	0.04	16.77	0.04	13.40	0.03	8.57	0.02	12.12	0.03	-9.88	-0.03	7.34	0.02	308.69	0.80	194.94	0.51	63.56	0.03
1.10	1999.44	0.80	15.02	0.02	0.27	15.75	0.04	16.94	0.05	13.60	0.04	8.70	0.02	12.30	0.03	-9.97	-0.03	7.57	0.02	308.73	0.84	194.97	0.53	63.54	0.03
1.20	2000.80	0.80	15.05	0.02	0.31	15.95	0.05	16.99	0.05	13.85	0.04	9.01	0.03	12.49	0.04	-10.57	-0.03	7.47	0.02	305.37	0.94	192.85	0.59	62.95	0.03
1.20	1999.53	0.80	15.04	0.02	0.35	15.50	0.05	17.34	0.06	13.98	0.05	9.07	0.03	12.61	0.04	-10.90	-0.04	7.73	0.03	305.51	1.06	192.94	0.67	63.22	0.03
1.30	1999.52	0.80	15.01	0.02	0.37	23.50	0.09	17.66	0.07	14.62	0.05	9.50	0.04	13.19	0.05	-12.39	-0.05	7.49	0.03	302.24	1.12	190.88	0.71	62.37	0.03
1.30	1999.91	0.80	15.01	0.02	0.36	15.45	0.06	17.73	0.06	15.08	0.05	9.84	0.04	13.63	0.05	-12.14	-0.04	7.91	0.03	303.39	1.09	191.60	0.69	62.32	0.03
1.40	1999.13	0.80	15.00	0.02	0.45	14.85	0.07	18.19	0.08	15.59	0.07	10.08	0.05	14.02	0.06	-14.31	-0.06	7.34	0.03	301.63	1.37	190.49	0.87	61.79	0.03
1.40	2000.53	0.80	15.00	0.02	0.46	17.55	0.08	18.37	0.08	15.91	0.07	10.24	0.05	14.30	0.07	-14.57	-0.07	7.49	0.03	301.12	1.38	190.17	0.87	61.82	0.03
1.51	2000.36	0.80	14.99	0.02	0.61	16.30	0.10	18.97	0.12	17.22	0.11	11.10	0.07	15.47	0.09	-16.12	-0.10	7.76	0.05	299.12	1.83	188.90	1.16	61.40	0.03
1.51	2000.18	0.80	15.00	0.02	0.56	14.80	0.08	18.58	0.10	17.04	0.09	10.91	0.06	15.30	0.09	-15.63	-0.09	7.47	0.04	299.38	1.67	189.07	1.05	61.40	0.03
1.61	2000.06	0.80	15.03	0.02	0.82	16.20	0.13	19.88	0.16	18.34	0.15	11.79	0.10	16.49	0.13	-18.46	-0.15	7.89	0.06	298.38	2.44	188.44	1.54	61.41	0.03
1.61	1999.77	0.80	15.05	0.02	0.71	16.90	0.12	19.85	0.14	18.17	0.13	11.56	0.08	16.30	0.12	-18.71	-0.13	7.57	0.05	297.86	2.12	188.11	1.34	61.48	0.03
1.71	2000.83	0.80	15.02	0.02	1.27	16.10	0.20	20.57	0.26	19.22	0.24	12.13	0.15	17.17	0.22	-21.21	-0.27	7.35	0.09	298.74	3.79	188.66	2.40	61.50	0.03
1.71	1999.44	0.80	15.00	0.02	1.29	16.10	0.23	21.26	0.27	19.65	0.25	12.41	0.16	17.62	0.23	-22.22	-0.29	7.67	0.10	298.70	3.86	188.64	2.44	61.14	0.03
1.80	2002.01	0.80	14.75	0.02	11.95	21.15	2.53	22.35	2.67	21.13	2.53	13.32	1.59	19.03	2.27	-24.66	-2.95	8.00	0.96	302.56	36.17	191.09	22.84	61.34	0.03

ISCO2 emissions Measured	ISCO2 emissions g/kWh	ISFC emissions		ISFC emissions		P peak		MAPD threshold		K12		Air mass flow		Fuel mass flow		CO2 Percentage		CO Percentage		HC Percentage		NOx Percentage		Combustion Efficiency		Thermal Efficiency	
		Measurement	Uncertainty	g/kWh	g/kWh	Measurement	Uncertainty	bar	bar	Measurement	Uncertainty	g/s	g/s	Measurement	Uncertainty	g/s	g/s	Measurement	Uncertainty	g/s	g/s	Measurement	Uncertainty	g/s	g/s	Measurement	Uncertainty
590.81	5.91	8.25	0.08	1.80	0.02	14.75	0.07	110.51	0.14	0.43	0.04	1.42	0.03	9.86	0.10	11.99	0.12	0.26	0.04	0.04	0.00	0.29	0.00	99.46	0.18	42.65	1.16
590.27	5.90	8.25	0.08	1.79	0.02	14.75	0.07	110.50	0.14	0.44	0.00	1.61	0.00	9.85	0.10	12.00	0.12	0.26	0.00	0.04	0.00	0.29	0.00	99.46	0.18	43.01	1.17
588.55	5.89	1.63	0.02	1.77	0.02	26.00	0.13	113.07	0.15	0.37	0.00	1.34	0.00	10.66	0.11	11.21	0.11	0.05	0.00	0.03	0.00	0.47	0.00	99.67	0.15	44.08	1.17
588.46	5.88	1.63	0.02	1.77	0.02	25.62	0.13	112.20	0.15	0.36	0.00	2.01	0.00	10.66	0.11	11.21	0.11	0.05	0.00	0.03	0.00	0.47	0.00	99.67	0.15	44.52	1.17
578.22	5.78	2.12	0.02	2.40	0.02	26.65	0.13	114.59	0.15	0.35	0.00	1.85	0.00	11.51	0.12	10.33	0.10	0.06	0.00	0.04	0.00	0.46	0.00	99.55	0.15	44.20	1.15
578.62	5.79	2.11	0.02	2.38	0.02	26.42	0.13	113.66	0.15	0.35	0.00	1.47	0.00	11.50	0.12	10.33	0.10	0.06	0.00	0.04	0.00	0.45	0.00	99.55	0.15	43.87	1.15
567.07	5.67	3.00	0.03	3.40	0.03	20.50	0.10	115.73	0.15	0.30	0.00	1.21	0.00	12.29	0.12	9.53	0.10	0.08	0.00	0.06	0.00	0.33	0.00	99.36	0.16	44.62	1.10
569.21	5.69	3.02	0.03	3.38	0.03	20.18	0.10	113.64	0.15	0.29	0.00	1.36	0.00	12.34	0.12	9.53	0.10	0.08	0.00	0.06	0.00	0.32	0.00	99.36	0.16	45.46	1.10
559.19	5.59	4.16	0.04	4.65	0.05	13.36	0.07	117.60	0.15	0.27	0.00	0.65	0.00	13.18	0.13	8.82	0.09	0.10	0.00	0.07	0.00	0.20	0.00	99.11	0.18	45.76	1.06
558.04	5.58	4.17	0.04	4.68	0.05	12.96	0.06	116.64	0.15	0.26	0.00	1.76	0.00	13.16	0.13	8.82	0.09	0.10	0.00	0.07	0.00	0.20	0.00	99.11	0.18	44.63	1.06
547.74	5.48	5.17	0.05	5.65	0.06	7.01	0.04	117.33	0.15	0.26	0.00	1.18	0.00	14.01	0.14	8.18	0.08	0.12	0.00	0.09	0.00	0.10	0.00	98.85	0.18	44.68	1.04
548.27	5.48	5.19	0.05	6.09	0.06	6.97	0.03	118.56	0.15	0.27	0.00	0.72	0.00	14.04	0.14	8.18	0.08	0.12	0.00	0.09	0.00	0.10	0.00	98.84	0.18	45.25	1.04
542.23	5.42	5.40	0.05	7.31	0.07	3.50	0.02	119.11	0.15	0.24	0.00	1.16	0.00	14.92	0.15	7.66	0.08	0.12	0.00	0.10	0.00	0.05	0.00	98.63	0.17	45.00	1.04
541.41	5.41	5.36	0.05	7.32	0.07	3.68	0.02	120.59	0.16	0.26	0.00	1.27	0.00	14.92	0.15	7.66	0.08	0.12	0.00	0.10	0.00	0.05	0.00	98.63	0.17	45.30	1.04
539.40	5.39	5.20	0.05	8.42	0.08	1.95	0.01	124.19	0.16	0.24	0.00	0.44	0.00	15.87	0.16	7.20	0.07	0.11	0.00	0.11	0.00	0.02	0.00	98.46	0.17	45.33	1.04
539.13	5.39	5.25	0.05	8.40	0.08	1.88	0.01	122.42	0.16	0.23	0.00	3.03	0.00	15.83	0.16	7.20	0.07	0.11	0.00	0.11	0.00	0.02	0.00	98.46	0.17	45.64	1.03
537.10	5.37	5.38	0.05	10.45	0.10	1.00	0.00	121.51	0.16	0.30	0.00	1.88	0.00	16.61	0.17	6.76	0.07	0.11	0.00	0.13	0.00	0.01	0.00	98.16	0.18	45.31	1.04

Table E.7 - 2000 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

Lambda	Engine Speed rpm	MPD		COV MEP %	MFB 0.5%	MFB 0.5%		MFB 1.0%	MFB 1.5%	MFB 2.0%	Spark Timing degCA	SFC		MC Fuel Mass mg/cycle	PC Fuel Mass mg/cycle	PC:MC Percentage %													
		Measurement	Uncertainty			Measurement	Uncertainty					Measurement	Uncertainty				Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty							
1.00	2000.08	0.60	15.05	0.02	0.51	14.40	0.07	16.18	0.08	6.52	0.03	4.17	0.02	5.62	0.03	-2.99	-0.02	7.45	0.04	443.48	2.28	200.02	1.07	90.41	0.05	0.40	0.00	0.46	0.00
1.00	2000.27	0.60	15.05	0.02	0.43	14.45	0.06	16.33	0.07	6.61	0.03	4.22	0.01	5.70	0.02	-3.24	-0.01	7.49	0.03	442.81	1.90	207.70	0.86	90.43	0.05	0.40	0.00	0.46	0.00
1.10	2000.27	0.80	14.98	0.02	0.48	14.90	0.07	16.11	0.08	6.90	0.03	4.43	0.02	5.97	0.03	-3.12	-0.02	7.49	0.04	432.51	2.09	203.87	0.98	87.57	0.05	0.40	0.00	0.46	0.00
1.10	2000.18	0.80	14.97	0.02	0.48	14.90	0.07	16.13	0.08	6.91	0.03	4.43	0.02	5.98	0.03	-3.62	-0.02	7.52	0.04	433.29	2.07	203.24	0.97	86.01	0.05	0.40	0.00	0.46	0.00
1.20	1999.39	0.90	14.99	0.02	0.49	14.90	0.07	16.45	0.08	7.28	0.04	4.74	0.02	6.34	0.03	-4.20	-0.02	7.48	0.04	424.08	2.13	199.25	0.97	86.01	0.05	0.40	0.00	0.47	0.00
1.20	2001.37	0.90	14.90	0.02	0.50	14.90	0.07	16.43	0.08	7.27	0.04	4.75	0.02	6.34	0.03	-4.24	-0.02	7.48	0.04	424.08	2.13	198.92	1.00	85.93	0.05	0.40	0.00	0.47	0.00
1.30	1999.23	0.90	14.96	0.02	0.44	14.40	0.06	16.10	0.07	7.86	0.03	5.20	0.02	6.94	0.03	-4.14	-0.02	7.53	0.03	417.39	1.70	195.78	0.87	84.55	0.05	0.40	0.00	0.48	0.00
1.30	2000.14	0.90	14.96	0.02	0.44	14.65	0.06	16.26	0.07	7.76	0.03	5.15	0.02	6.85	0.03	-4.38	-0.02	7.54	0.03	416.76	1.84	195.48	0.86	84.56	0.05	0.40	0.00	0.48	0.00
1.40	2000.31	0.90	15.02	0.02	0.53	14.90	0.08	16.20	0.09	8.30	0.04	5.57	0.03	7.39	0.04	-4.95	-0.03	7.51	0.04	408.90	2.16	193.80	1.01	83.79	0.05	0.40	0.00	0.48	0.00
1.40	2000.47	0.90	15.02	0.02	0.53	14.95	0.08	16.11	0.09	8.31	0.04	5.57	0.03	7.39	0.04	-4.80	-0.03	7.49	0.04	409.23	2.15	193.95	1.01	83.85	0.05	0.40	0.00	0.48	0.00
1.50	2000.84	0.90	15.03	0.02	0.56	14.90	0.08	16.48	0.09	8.90	0.05	6.07	0.03	8.01	0.04	-5.71	-0.03	7.58	0.04	405.33	2.24	190.12	1.06	83.24	0.04	0.40	0.00	0.49	0.00
1.50	2000.15	0.90	15.03	0.02	0.56	14.90	0.08	16.48	0.09	8.90	0.05	6.07	0.03	8.01	0.04	-5.71	-0.03	7.58	0.04	405.33	2.24	190.12	1.06	83.24	0.04	0.40	0.00	0.49	0.00
1.60	2000.00	0.90	14.99	0.02	0.59	14.90	0.08	16.62	0.09	9.41	0.05	6.49	0.04	8.51	0.05	-6.83	-0.04	7.66	0.04	402.11	2.21	188.61	1.04	82.48	0.04	0.40	0.00	0.49	0.00
1.60	2000.23	0.90	15.00	0.02	0.59	14.95	0.08	16.65	0.09	9.35	0.05	6.41	0.03	8.45	0.04	-6.57	-0.03	7.68	0.04	402.11	2.21	188.61	1.04	82.48	0.04	0.40	0.00	0.49	0.00
1.70	1999.30	0.90	14.97	0.02	0.60	14.95	0.08	16.54	0.08	9.85	0.05	6.83	0.03	8.88	0.04	-7.63	-0.04	7.47	0.04	397.59	1.97	188.49	0.92	82.34	0.04	0.40	0.00	0.49	0.00
1.70	2000.85	0.90	15.04	0.02	0.59	15.05	0.11	16.54	0.08	9.85	0.05	6.83	0.03	8.88	0.04	-7.63	-0.04	7.47	0.04	397.59	1.97	188.49	0.92	82.34	0.04	0.40	0.00	0.49	0.00
1.80	1999.65	0.90	14.96	0.02	0.66	15.05	0.02	16.63	0.08	10.90	0.05	6.86	0.03	9.01	0.04	-7.78	-0.04	7.49	0.03	386.92	1.81	188.13	0.85	82.35	0.04	0.40	0.00	0.49	0.00
1.80	2000.28	0.90	14.98	0.02	0.66	15.02	0.04	16.63	0.08	10.90	0.05	6.86	0.03	9.01	0.04	-7.78	-0.04	7.49	0.03	386.92	1.81	188.13	0.85	82.35	0.04	0.40	0.00	0.49	0.00
1.90	1999.37	0.90	14.96	0.02	0.66	15.02	0.04	16.63	0.08	10.90	0.05	6.86	0.03	9.01	0.04	-7.78	-0.04	7.49	0.03	386.92	1.81	188.13	0.85	82.35	0.04	0.40	0.00	0.49	0.00
1.90	2000.41	0.90	15.02	0.02	0.67	14.98	0.08	16.62	0.09	10.96	0.05	7.28	0.04	9.51	0.05	-9.55	-0.05	7.53	0.04	392.12	1.93	183.93	0.95	83.29	0.04	0.40	0.00	0.50	0.00
2.00	1999.38	0.90	14.97	0.02	0.51	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.00	2000.15	0.90	15.05	0.02	0.47	14.98	0.08	16.62	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.10	1999.38	0.90	14.97	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.10	2000.25	0.90	15.06	0.02	0.51	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.20	1999.37	0.90	14.96	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.20	2000.61	0.90	15.01	0.02	0.51	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.30	1999.37	0.90	14.96	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.30	2000.25	0.90	15.06	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.40	1999.38	0.90	14.96	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.40	2000.25	0.90	15.06	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.50	1999.38	0.90	14.96	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.50	2000.25	0.90	15.06	0.02	0.53	15.85	0.08	17.68	0.09	10.98	0.06	7.67	0.04	10.02	0.05	-10.92	-0.05	7.60	0.04	392.92	1.98	184.16	0.91	81.50	0.04	0.40	0.00	0.50	0.00
2.60	1993.55	0.79	14.51	0.02	0.77	13.95	0.08	15.43	0.07	17.05	0.14	11.04	0.07	15.22	0.12	-29.26	-0.61	8.57	0.18	383.98	8.05	179.92	3.78	79.35	0.04	1.81	0.00	2.26	0.00
2.60	2000.25	0.80	14.51	0.02	0.77	13.95	0.08	15.43	0.07	17.05	0.14	11.04	0.07	15.22	0.12	-29.26	-0.61	8.57	0.18	383.98	8.05	179.92	3.78	79.35	0.04	1.81	0.00	2.26	0.00
2.60	2000.25	0.80	14.22	0.02	0.77	13.95	0.08	15.43	0.07	17.05	0.14	11.04	0.07	15.22	0.12	-29.26	-0.61	8.57	0.18	383.98	8.05	179.92	3.78	79.35	0.04	1.81	0.00	2.26	0.00
2.60	2000.25	0.80	14.22	0.02	0.77	13.95	0.08	15.43	0.07	17.05	0.14	11.04	0.07	15.22	0.12	-29.26	-0.61	8.57	0.18	383.98	8.05	179.92	3.78	79.35	0.04	1.81	0.00	2.26	0.00
2.60	2000.25	0.80	14.22	0.02	0.77	13.95	0.08	15.43	0.07	17.																			

Lambda	Engine Speed		IMEP	COV_IMEP	MFB 0.5%		MFB 5.0%		MFB 10.9%		Spark Timing		MFB 50		ISFC		ISFC corr		MC Fuel Mass		PC Fuel Mass		PC MC Percentage		
	Measured	Uncertainty			degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	degC	
1.76	1999.50	14.95	0.02	0.38	16.85	20.92	0.08	11.48	0.04	10.34	-17.51	-0.07	7.61	0.03	294.80	1.15	186.18	0.73	60.96	0.03	0.63	0.00	1.04	0.00	
1.71	2000.50	14.97	0.02	0.34	17.10	20.95	0.07	11.55	0.03	10.39	-16.91	-0.06	7.87	0.04	311.68	1.05	198.83	0.67	467.45	0.25	0.63	0.00	0.14	0.00	
1.61	1999.56	14.97	0.02	0.46	16.93	20.92	0.08	10.73	0.05	9.71	-11.95	-0.05	7.54	0.03	297.69	1.38	198.00	0.86	61.78	0.03	0.63	0.00	1.03	0.00	
1.56	1999.56	14.97	0.02	0.47	16.93	20.92	0.08	10.73	0.05	9.71	-11.95	-0.05	7.54	0.03	297.69	1.38	198.00	0.86	61.78	0.03	0.63	0.00	1.03	0.00	
1.51	1999.62	14.98	0.02	0.47	16.93	20.92	0.08	9.97	0.05	9.02	-9.17	-0.04	7.51	0.04	300.82	1.42	197.99	1.00	62.70	0.03	0.40	0.00	0.64	0.00	
1.50	1999.68	14.97	0.02	0.54	15.25	19.99	0.08	9.97	0.05	9.01	-9.44	-0.05	7.41	0.04	301.93	1.62	199.68	1.02	62.70	0.03	0.40	0.00	0.64	0.00	
1.46	1999.81	14.97	0.02	0.48	15.30	20.07	0.08	9.23	0.04	8.29	-6.89	-0.04	7.53	0.04	304.25	1.41	192.14	0.89	63.03	0.03	0.40	0.00	0.64	0.00	
1.44	1999.44	14.99	0.02	0.48	14.65	20.07	0.09	9.25	0.04	8.30	-7.39	-0.04	7.59	0.04	305.36	1.46	192.84	0.92	63.58	0.03	0.40	0.00	0.64	0.00	
1.38	1999.94	14.97	0.02	0.48	14.65	20.07	0.09	8.41	0.04	7.49	-6.42	-0.03	7.42	0.04	309.23	1.59	195.29	0.95	63.97	0.03	0.40	0.00	0.63	0.00	
1.33	1999.74	14.98	0.02	0.55	15.45	19.98	0.08	8.48	0.05	5.71	-4.03	-0.03	7.46	0.04	308.57	1.68	194.87	1.00	64.04	0.03	0.40	0.00	0.63	0.00	
1.29	1999.74	14.98	0.02	0.55	15.45	19.98	0.08	8.48	0.05	5.71	-4.03	-0.03	7.46	0.04	308.57	1.68	194.87	1.00	64.04	0.03	0.40	0.00	0.63	0.00	
1.26	1999.72	14.99	0.02	0.54	14.45	20.07	0.08	7.88	0.04	6.94	-4.03	-0.02	7.59	0.04	312.25	1.56	197.20	0.98	64.72	0.03	0.40	0.00	0.62	0.00	
1.19	1999.72	14.99	0.02	0.44	14.40	20.06	0.08	16.33	0.07	6.36	-0.03	-0.03	7.48	0.03	317.31	1.36	200.39	0.88	66.42	0.04	0.40	0.00	0.61	0.00	
1.11	2000.63	14.97	0.02	0.42	14.40	20.06	0.08	7.33	0.03	6.40	-0.03	-3.23	-0.01	7.47	0.03	316.66	1.32	199.88	0.84	65.73	0.04	0.40	0.00	0.61	0.00
1.00	2000.36	14.98	0.02	0.55	14.45	20.06	0.08	16.07	0.09	6.03	-0.03	-3.10	-0.02	7.48	0.04	325.16	1.79	205.33	1.13	67.91	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04	0.40	0.00	0.59	0.00
1.00	1999.54	14.97	0.02	0.55	14.45	20.06	0.08	6.98	0.03	6.03	-0.03	-2.85	-0.01	7.60	0.04	325.16	1.62	205.98	1.02	68.17	0.04				

Table E.9 - 3500 rpm and 6 bar of IMEP - SI-MeOH

Lambda	Engine Speed	IMEP	COV IMEP	MFB 0-5%	MFB 0-10%	MFB 5-90%	MFB 10-75%	MFB 10-90%	Spark Timing	MFB 50	ISFC	ISFC corr	MC Fuel Mass
Measured	Measurement Uncertainty	Measurement Uncertainty	%	degCA	degCA	degCA	degCA	degCA	degCA	degCA	g/kWh	g/kWh	mg/cycle
1.00	3500.15	1.40	6.04	0.01	15.05	0.05	9.21	0.03	-18.86	-0.07	467.68	1.70	219.37
1.00	3499.95	1.40	6.02	0.01	15.20	0.06	9.22	0.04	-19.59	-0.08	469.76	1.84	220.34
1.10	3498.71	1.40	5.99	0.01	13.75	0.06	9.20	0.04	-19.97	-0.09	455.48	1.96	213.65
1.10	3500.18	1.40	5.96	0.01	16.15	0.06	9.23	0.04	-20.56	-0.11	455.68	1.79	213.74
1.20	3500.15	1.40	6.02	0.01	15.65	0.09	9.45	0.05	-18.96	-0.11	449.45	2.57	210.82
1.20	3499.84	1.40	6.04	0.01	16.30	0.08	9.58	0.05	-20.24	-0.10	449.25	2.25	210.72
1.30	3500.42	1.40	5.97	0.01	15.55	0.08	9.83	0.05	-21.63	-0.11	444.52	2.30	208.50
1.30	3500.23	1.40	5.96	0.01	16.35	0.09	9.99	0.05	-21.04	-0.11	443.54	2.32	208.05
1.40	3499.93	1.40	5.97	0.01	14.45	0.09	10.81	0.07	-24.27	-0.15	437.99	2.78	205.44
1.40	3500.22	1.40	5.98	0.01	17.70	0.10	10.31	0.06	-22.96	-0.13	438.06	2.52	205.48
1.50	3499.36	1.40	5.99	0.01	15.80	0.11	10.87	0.08	-26.54	-0.19	435.31	3.08	204.19
1.50	3500.04	1.40	6.01	0.01	17.05	0.12	10.93	0.08	-25.10	-0.18	435.68	3.11	204.36
1.60	3499.94	1.40	6.01	0.01	18.95	0.15	11.64	0.10	-25.10	-0.21	433.74	3.55	203.45
1.60	3500.54	1.40	6.00	0.01	17.65	0.15	11.72	0.10	-27.15	-0.24	432.97	3.80	203.09
1.70	3500.12	1.40	6.01	0.01	18.65	0.15	12.00	0.10	-28.23	-0.24	430.69	3.65	202.02
1.70	3500.15	1.40	5.99	0.01	16.50	0.17	11.97	0.12	-28.10	-0.29	431.36	4.43	202.33
1.80	3500.15	1.40	6.00	0.01	18.35	0.33	12.84	0.23	-20.90	-0.37	429.50	7.67	201.46
1.80	3500.29	1.40	5.99	0.01	17.85	0.28	12.82	0.20	-20.90	-0.33	430.59	6.78	201.97
1.90	3500.46	1.40	6.02	0.01	17.25	0.49	13.29	0.37	-24.90	-0.70	430.22	12.13	201.80
1.90	3499.99	1.40	6.04	0.01	18.85	0.44	13.25	0.31	-24.90	-0.58	429.41	9.98	201.42
2.02	3499.96	1.40	5.96	0.01	20.25	2.60	15.30	1.96	-28.60	-3.67	443.73	56.98	208.13
2.02	3492.64	1.40	5.78	0.01	19.45	4.37	15.31	3.44	-28.60	-6.42	459.56	103.16	215.56
													48.39
													35.97
													0.02

ISO2 emissions	ISO emissions	ISFC emissions	ISNOx emissions	P peak	MAP0 threshold	K12	Air mass flow	Fuel Mass Flow	CO2 Percentage	CO Percentage	HC Percentage	NOx Percentage	Combustion Efficiency	Thermal Efficiency
g/kWh	g/kWh	g/kWh	g/kWh	bar	bar	%	mg/s	mg/s	%	%	%	%	%	%
626.27	6.28	9.02	0.08	1.55	0.02	11.48	0.06	0.62	11.28	0.11	0.26	0.03	99.51	39.51
628.22	6.28	9.32	0.09	1.55	0.02	11.45	0.06	0.59	11.28	0.11	0.26	0.03	99.50	40.00
618.26	6.18	3.63	0.04	1.53	0.02	14.82	0.07	0.80	10.58	0.11	0.10	0.03	99.67	41.09
618.29	6.18	3.66	0.04	1.53	0.02	14.80	0.07	0.80	10.58	0.11	0.10	0.03	99.66	41.09
603.30	6.03	4.92	0.05	2.37	0.02	11.85	0.08	0.54	9.73	0.10	0.12	0.04	99.50	41.78
602.97	6.03	4.92	0.05	2.38	0.02	11.82	0.08	0.54	9.73	0.10	0.12	0.04	99.49	41.78
589.18	5.89	5.65	0.06	3.88	0.04	7.90	0.04	0.90	8.99	0.09	0.14	0.06	99.24	41.88
588.08	5.88	5.64	0.06	3.85	0.04	7.86	0.04	0.89	8.99	0.09	0.14	0.06	99.25	41.88
575.41	5.75	5.36	0.05	5.40	0.05	4.47	0.02	1.01	8.38	0.08	0.12	0.06	99.01	42.27
575.24	5.75	5.40	0.05	5.43	0.05	4.53	0.02	1.03	8.37	0.08	0.12	0.06	99.01	42.27
567.74	5.68	5.12	0.05	6.77	0.07	2.66	0.01	1.02	8.37	0.08	0.12	0.06	99.01	42.27
588.07	5.88	5.15	0.05	6.74	0.07	2.68	0.01	1.02	8.37	0.08	0.12	0.06	99.01	42.27
561.99	5.62	5.06	0.05	7.94	0.08	1.38	0.01	1.02	7.38	0.07	0.10	0.04	99.62	43.41
561.00	5.61	5.07	0.05	7.95	0.08	1.40	0.01	1.02	7.37	0.07	0.10	0.04	99.62	43.41
555.24	5.55	5.12	0.05	8.75	0.09	0.71	0.00	1.01	6.96	0.07	0.10	0.04	99.48	43.75
555.81	5.56	5.12	0.05	8.79	0.09	0.71	0.00	1.01	6.96	0.07	0.10	0.04	99.48	43.75
548.90	5.49	5.44	0.06	9.60	0.10	0.44	0.00	1.01	6.56	0.07	0.10	0.04	99.35	44.32
549.43	5.49	5.38	0.06	9.73	0.10	0.43	0.00	1.01	6.55	0.07	0.10	0.04	99.32	44.32
543.91	5.44	6.25	0.06	10.60	0.11	0.29	0.00	1.02	6.18	0.06	0.11	0.04	99.17	44.32
542.90	5.43	6.13	0.06	10.57	0.11	0.29	0.00	1.02	6.19	0.06	0.11	0.04	99.18	44.32
538.81	5.39	9.47	0.08	15.39	0.15	0.14	0.00	1.05	5.66	0.08	0.16	0.00	97.43	40.62
554.62	5.55	9.72	0.10	16.93	0.17	0.15	0.00	1.05	5.63	0.08	0.16	0.00	97.28	41.50

Table E.10 - 3500 rpm and 6 bar of IMEP - SI-EtOH

Lambda Measured	Engine Speed		IMEP		COV_IMEP		MFB 0.5%		MFB 0-10%		MFB 5-90%		MFB 10-75%		MFB 10-90%		Spark Timing		MFB 50		ISFC		ISFC corr		MC Fuel Mass	
	rpm	Measurement Uncertainty	bar	Measurement Uncertainty	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	g/kWh	Measurement Uncertainty	g/kWh	Measurement Uncertainty	mg/cycle
1.00	3498.49	1.40	6.04	0.01	0.41	15.30	0.06	21.23	0.09	15.51	0.06	9.63	0.04	14.10	0.06	-19.44	-0.08	7.54	0.03	352.97	1.45	222.91	0.91	28.58	0.02	
1.00	3500.21	1.40	6.05	0.01	0.40	13.75	0.05	21.54	0.09	15.67	0.06	9.67	0.04	14.27	0.06	-20.16	-0.08	7.49	0.03	352.59	1.41	222.67	0.89	29.96	0.02	
1.10	3498.62	1.40	5.98	0.01	0.48	16.20	0.08	21.57	0.10	15.60	0.08	9.93	0.05	14.11	0.07	-20.48	-0.10	7.57	0.04	343.91	1.65	217.19	1.05	28.48	0.02	
1.10	3498.64	1.40	6.01	0.01	0.52	17.30	0.09	21.94	0.11	15.49	0.08	9.89	0.05	14.03	0.07	-21.05	-0.11	7.59	0.04	345.07	1.78	217.92	1.13	28.57	0.02	
1.20	3500.03	1.40	5.99	0.01	0.49	30.90	0.15	22.12	0.11	15.76	0.08	10.21	0.05	14.23	0.07	-21.81	-0.11	7.58	0.04	339.12	1.67	214.16	1.05	28.09	0.02	
1.20	3500.26	1.40	6.01	0.01	0.53	15.55	0.08	21.29	0.11	15.84	0.08	10.27	0.05	14.32	0.08	-20.06	-0.11	7.66	0.04	337.87	1.80	213.37	1.14	28.13	0.02	
1.31	3498.85	1.40	6.00	0.01	0.56	16.65	0.09	22.16	0.12	16.59	0.09	10.67	0.06	14.91	0.08	-22.81	-0.13	7.36	0.04	333.59	1.86	210.67	1.18	27.74	0.01	
1.31	3500.11	1.40	6.01	0.01	0.55	30.85	0.17	22.54	0.12	16.94	0.09	10.94	0.06	15.24	0.08	-22.97	-0.13	7.83	0.04	334.12	1.85	211.01	1.17	27.73	0.01	
1.41	3498.64	1.40	6.00	0.01	0.70	15.20	0.11	24.15	0.17	17.84	0.13	11.37	0.08	15.99	0.11	-27.38	-0.19	7.48	0.05	330.19	2.33	208.53	1.47	27.52	0.01	
1.41	3500.38	1.40	5.99	0.01	0.59	31.10	0.18	23.51	0.14	17.42	0.10	11.15	0.07	15.60	0.09	-25.92	-0.15	7.43	0.04	330.03	1.94	208.43	1.23	27.53	0.01	
1.51	3498.79	1.40	6.00	0.01	0.79	31.50	0.25	24.52	0.19	18.98	0.15	11.97	0.10	16.99	0.14	-28.65	-0.23	7.48	0.06	328.37	2.61	207.38	1.65	27.32	0.01	
1.51	3498.31	1.40	6.01	0.01	1.01	16.45	0.17	23.09	0.23	19.50	0.20	12.32	0.12	17.50	0.18	-25.60	-0.26	7.73	0.08	329.60	3.34	208.15	2.11	27.51	0.01	
1.61	3500.11	1.40	6.02	0.01	1.36	31.35	0.43	24.28	0.33	20.65	0.28	12.93	0.18	18.48	0.25	-29.12	-0.40	7.55	0.10	326.14	4.45	205.97	2.81	27.36	0.01	
1.61	3500.55	1.40	6.01	0.01	1.43	16.55	0.24	23.72	0.34	20.92	0.30	13.13	0.19	18.75	0.27	-27.68	-0.40	7.75	0.11	328.02	4.70	207.15	2.97	27.40	0.01	
1.71	3495.01	1.40	6.00	0.01	6.79	31.15	2.11	26.30	1.78	22.50	1.53	14.07	0.95	20.24	1.37	-33.25	-2.26	8.03	0.54	329.69	22.37	208.21	14.13	27.41	0.01	
1.71	3500.31	1.40	6.03	0.01	2.28	23.85	0.54	26.31	0.60	22.57	0.51	14.03	0.32	20.29	0.46	-33.68	-0.77	7.77	0.18	327.29	7.46	206.69	4.71	27.37	0.01	
1.79	3498.88	1.40	5.83	0.01	14.99	37.30	5.59	28.45	4.27	24.07	3.61	15.04	2.26	21.83	3.27	-37.57	-5.63	8.45	1.27	336.81	50.50	212.71	31.89	27.17	0.01	
1.79	3498.18	1.40	5.82	0.01	10.36	36.10	3.74	28.88	2.99	24.68	2.56	15.39	1.59	22.43	2.32	-37.58	-3.89	9.06	0.94	336.63	34.96	212.59	22.02	27.33	0.01	

ISCO2 emissions g/kWh	ISCO2 emissions		ISHC emissions		ISNOx emissions		P peak		MAPD threshold		K12		Air mass flow		Fuel Mass Flow		CO2 Percentage		CO Percentage		HC Percentage		MOx Percentage		Combustion Efficiency		Thermal Efficiency	
	Measurement Uncertainty	g/kWh	Measurement Uncertainty	g/kWh	Measurement Uncertainty	g/kWh	Measurement Uncertainty	bar	Measurement Uncertainty	bar	Measurement Uncertainty	%	Measurement Uncertainty	mg/s	Measurement Uncertainty	mg/s	Measurement Uncertainty	%	Measurement Uncertainty	%	Measurement Uncertainty	%	Measurement Uncertainty	%	Measurement Uncertainty	%	Measurement Uncertainty	%
654.55	6.55	11.76	0.12	2.76	0.03	13.91	0.07	49.76	0.06	0.57	0.00	2.93	0.00	0.00	0.00	11.89	0.12	0.34	0.00	0.05	0.00	0.24	0.00	99.29	0.12	37.75	0.43	
654.42	6.54	11.91	0.12	2.75	0.03	13.85	0.07	49.95	0.06	0.60	0.00	3.14	0.00	0.00	0.00	11.89	0.12	0.34	0.00	0.05	0.00	0.24	0.00	99.29	0.12	37.99	0.44	
649.51	6.50	4.14	0.04	2.79	0.03	18.95	0.09	49.96	0.06	0.58	0.00	3.26	0.00	0.00	0.00	11.12	0.11	0.11	0.00	0.05	0.00	0.31	0.00	99.49	0.11	39.33	0.35	
651.44	6.51	4.14	0.04	2.81	0.03	19.06	0.10	50.15	0.07	0.56	0.00	2.68	0.00	0.00	0.00	11.12	0.11	0.11	0.00	0.05	0.00	0.31	0.00	99.49	0.11	40.35	0.35	
633.79	6.34	5.04	0.05	3.89	0.04	15.67	0.08	50.75	0.07	0.52	0.00	3.94	0.01	0.00	0.00	10.19	0.10	0.13	0.00	0.06	0.00	0.24	0.00	99.30	0.10	38.89	0.30	
631.66	6.32	5.03	0.05	3.83	0.04	15.40	0.08	50.70	0.07	0.51	0.00	4.76	0.01	0.00	0.00	10.20	0.10	0.13	0.00	0.06	0.00	0.24	0.00	99.31	0.10	41.20	0.30	
615.99	6.16	5.59	0.06	5.56	0.06	10.46	0.05	51.95	0.07	0.50	0.00	3.59	0.00	0.00	0.00	9.37	0.09	0.13	0.00	0.08	0.00	0.15	0.00	99.04	0.09	40.60	0.24	
616.68	6.17	5.58	0.06	5.56	0.06	10.52	0.05	51.06	0.07	0.48	0.00	4.70	0.01	0.00	0.00	9.37	0.09	0.13	0.00	0.08	0.00	0.15	0.00	99.03	0.09	39.43	0.24	
603.37	6.03	5.65	0.06	7.20	0.07	6.14	0.03	52.50	0.07	0.49	0.00	2.81	0.00	0.00	0.00	8.68	0.09	0.13	0.00	0.10	0.00	0.08	0.00	99.79	0.09	40.76	0.20	
603.07	6.03	5.62	0.06	7.23	0.07	6.10	0.03	52.56	0.07	0.50	0.00	3.19	0.00	0.00	0.00	8.67	0.09	0.13	0.00	0.10	0.00	0.08	0.00	99.79	0.09	40.36	0.20	
595.46	5.95	5.27	0.05	8.79	0.09	3.27	0.02	53.26	0.07	0.50	0.00	3.55	0.00	0.00	0.00	8.09	0.08	0.11	0.00	0.12	0.00	0.04	0.00	99.56	0.08	41.15	0.19	
597.66	5.96	5.30	0.05	8.82	0.09	3.23	0.02	52.88	0.07	0.48	0.00	1.91	0.00	0.00	0.00	8.09	0.08	0.11	0.00	0.12	0.00	0.04	0.00	99.56	0.08	40.73	0.19	
597.93	5.98	4.85	0.05	10.11	0.10	1.79	0.01	54.09	0.07	0.52	0.00	2.22	0.00	0.00	0.00	7.60	0.08	0.10	0.00	0.13	0.00	0.02	0.00	98.38	0.08	40.78	0.18	
591.02	5.91	4.89	0.05	10.17	0.10	1.81	0.01	53.61	0.07	0.51	0.00	1.92	0.00	0.00	0.00	7.60	0.08	0.10	0.00	0.13	0.00	0.02	0.00	98.37	0.08	41.65	0.18	
588.76	5.89	4.98	0.05	11.73	0.12	0.90	0.00	54.35	0.07	0.52	0.00	2.58	0.00	0.00	0.00	7.13	0.07	0.10	0.00	0.14	0.00	0.01	0.00	98.15	0.07	40.86	0.19	
584.65	5.85	4.97	0.05	11.56	0.12	0.90	0.00	54.75	0.07	0.52	0.00	1.99	0.00	0.00	0.00	7.13	0.07	0.10	0.00	0.14	0.00	0.01	0.00	98.16	0.07	40.94	0.19	
586.65	5.87	6.13	0.06	15.20	0.15	0.50	0.00	52.65	0.07	0.53	0.00	1.48	0.00	0.00	0.00	6.69	0.07	0.11	0.00	0.17	0.00	0.01	0.00	97.66	0.07	40.28	0.22	
585.57	5.86	6.12	0.06	15.38	0.15	0.52	0.00	52.36	0.07	0.54	0.00	3.22	0.00	0.00	0.00	6.68	0.07	0.11	0.00	0.18	0.00	0.01	0.00	97.65	0.07	38.89	0.22	

[illegible]

Table E.12 - 3500 rpm and 6 bar of IMEP - MC-MeOH/PC-EtOH

Lambda measured	Engine Speed rpm	IMEP bar	COV IMEP %	IMEP 0.5%		IMEP 1.0%		IMEP 1.5%		IMEP 2.0%		IMEP 3.0%		IMEP 4.0%		Spark Timing degCA	IMEP 50		ISFC g/kWh	ISFC corr g/kWh	MC Fuel Mass mg/cycle	DC Fuel Mass mg/cycle	PC MC Percentage %					
				Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty	Measurement	Uncertainty		Measurement	Uncertainty						Measurement	Uncertainty			
1.00	3500.11	1.40	6.02	0.01	0.56	15.50	0.09	20.93	0.12	9.10	0.05	5.58	0.03	7.80	0.04	-14.14	-0.08	372.66	2.08	234.72	1.31	31.32	0.02	0.40	0.00	1.29	0.00	
1.04	3499.82	1.40	5.97	0.01	0.54	15.05	0.08	21.06	0.11	9.14	0.05	5.61	0.03	7.82	0.04	0.04	-14.42	-0.08	373.91	2.03	235.50	1.28	31.20	0.02	0.40	0.00	1.30	0.00
1.08	3499.53	1.40	5.92	0.01	0.52	14.50	0.07	21.19	0.10	9.18	0.05	5.64	0.03	7.84	0.04	0.04	-14.70	-0.08	375.16	2.00	236.28	1.25	31.08	0.02	0.40	0.00	1.40	0.00
1.10	3499.07	1.40	6.03	0.01	0.56	15.20	0.09	20.79	0.12	9.14	0.05	5.58	0.03	7.80	0.04	0.04	-14.75	-0.08	368.92	2.06	229.20	1.29	30.49	0.02	0.40	0.00	1.32	0.00
1.12	3499.65	1.40	6.04	0.01	0.58	15.05	0.09	21.02	0.12	9.10	0.05	5.62	0.03	7.86	0.04	0.04	-14.55	-0.09	356.79	2.04	224.74	1.30	29.99	0.02	0.40	0.00	1.35	0.00
1.20	3500.12	1.40	6.03	0.01	0.58	29.10	0.17	21.51	0.13	10.01	0.06	6.39	0.04	8.60	0.05	0.04	-16.46	-0.10	356.34	2.08	224.45	1.31	29.96	0.02	0.40	0.00	1.35	0.00
1.24	3499.83	1.40	6.03	0.01	0.58	29.10	0.17	21.51	0.13	10.01	0.06	6.39	0.04	8.60	0.05	0.04	-16.46	-0.10	356.34	2.08	224.45	1.31	29.96	0.02	0.40	0.00	1.38	0.00
1.30	3499.85	1.40	5.99	0.01	0.54	28.90	0.16	20.80	0.11	10.79	0.06	6.88	0.04	9.40	0.05	0.04	-15.89	-0.08	359.16	1.86	220.99	1.19	29.26	0.02	0.40	0.00	1.38	0.00
1.40	3500.30	1.40	6.03	0.01	0.60	29.05	0.17	20.79	0.12	11.70	0.07	7.58	0.05	10.35	0.06	0.04	-16.45	-0.10	346.35	2.00	218.22	1.30	28.95	0.02	0.40	0.00	1.40	0.00
1.44	3499.95	1.40	6.04	0.01	0.58	29.05	0.17	21.33	0.12	11.67	0.07	7.56	0.05	10.29	0.06	0.04	-16.45	-0.10	346.96	2.03	218.45	1.28	28.97	0.02	0.40	0.00	1.42	0.00
1.50	3499.90	1.40	6.01	0.01	0.57	29.05	0.16	21.15	0.11	11.64	0.07	7.54	0.05	10.26	0.06	0.04	-16.45	-0.10	347.13	2.02	218.45	1.27	28.97	0.02	0.40	0.00	1.42	0.00
1.54	3499.71	1.40	5.98	0.01	0.54	30.10	0.16	23.17	0.12	14.14	0.08	9.30	0.05	12.71	0.07	0.04	-23.84	-0.13	335.07	1.86	210.75	1.33	27.64	0.01	0.52	0.00	1.88	0.00
1.71	3499.96	1.40	5.96	0.01	0.56	30.95	0.17	22.46	0.13	14.10	0.08	9.31	0.05	12.68	0.07	0.04	-22.46	-0.13	337.21	1.88	211.94	1.18	27.64	0.01	0.52	0.00	1.88	0.00
1.81	3500.18	1.40	5.99	0.01	0.58	17.60	0.08	22.79	0.12	15.11	0.08	9.84	0.06	13.74	0.07	0.04	-26.38	-0.13	333.41	1.86	209.90	1.04	27.62	0.01	0.63	0.00	2.31	0.00
1.91	3500.50	1.40	6.01	0.01	0.60	16.45	0.12	23.30	0.16	16.37	0.11	10.39	0.07	14.89	0.10	0.04	-25.15	-0.16	333.85	2.33	208.53	1.46	27.35	0.01	0.75	0.00	2.74	0.00
1.95	3500.31	1.40	6.01	0.01	0.60	16.85	0.15	24.02	0.22	16.33	0.11	10.31	0.09	14.87	0.10	0.04	-26.28	-0.24	331.83	3.00	208.40	1.88	27.37	0.01	0.75	0.00	2.74	0.00
2.01	3500.37	1.40	6.03	0.01	1.11	16.35	0.18	24.67	0.27	17.40	0.13	10.95	0.12	15.19	0.17	0.04	-28.10	-0.31	331.95	3.68	208.22	2.46	27.37	0.01	0.88	0.00	3.59	0.00
2.05	3500.37	1.40	6.03	0.01	1.11	16.35	0.18	24.67	0.27	17.40	0.13	10.95	0.12	15.19	0.17	0.04	-28.10	-0.31	331.95	3.68	208.22	2.46	27.37	0.01	0.88	0.00	3.59	0.00
2.11	3499.82	1.40	6.01	0.01	1.46	16.10	0.24	25.12	0.37	18.45	0.27	11.68	0.17	15.94	0.23	0.04	-30.16	-0.44	336.71	4.91	210.24	3.08	27.18	0.01	1.33	0.00	4.89	0.00
2.11	3500.48	1.40	6.00	0.01	2.04	30.05	0.61	23.76	0.48	18.59	0.38	11.82	0.24	16.12	0.23	0.04	-27.34	-0.56	336.23	6.85	210.43	4.29	27.14	0.01	1.56	0.00	5.74	0.00
2.21	3499.90	1.40	5.98	0.01	1.99	16.95	0.34	18.75	0.37	19.47	0.39	12.22	0.24	16.68	0.33	0.04	-18.42	-0.37	337.64	6.73	211.09	4.21	27.14	0.01	1.56	0.00	5.74	0.00
2.21	3499.90	1.40	5.98	0.01	1.99	16.95	0.34	18.75	0.37	19.47	0.39	12.22	0.24	16.68	0.33	0.04	-18.42	-0.37	337.64	6.73	211.09	4.21	27.14	0.01	1.56	0.00	5.74	0.00
2.31	3500.14	1.40	5.99	0.01	2.15	15.65	0.33	18.86	0.40	19.43	0.41	12.20	0.26	16.65	0.35	0.04	-18.42	-0.38	341.11	12.88	213.02	6.03	27.38	0.01	1.79	0.00	6.52	0.00
2.31	3500.14	1.40	5.99	0.01	3.77	15.59	0.59	19.89	0.75	20.28	0.76	12.69	0.48	17.99	0.66	0.04	-20.63	-0.78	342.16	14.64	213.46	9.13	27.40	0.01	1.79	0.00	6.52	0.00
2.42	3500.79	1.40	5.30	0.01	15.61	16.85	2.63	21.83	3.41	23.23	3.63	14.89	2.33	20.43	3.19	0.04	-21.73	-3.39	376.88	58.85	234.77	36.66	26.86	0.01	2.02	0.00	7.52	0.00
2.42	3500.96	1.40	5.33	0.01	15.55	16.10	2.58	21.78	3.39	23.14	3.60	14.85	2.31	20.43	3.18	0.04	-21.73	-3.39	376.88	58.47	234.29	36.43	26.91	0.01	2.02	0.00	7.51	0.00

Table E.13 - 3500 rpm and 6 bar of IMEP - MC-EtOH/PC-EtOH

[illegible]

Table E.14 - 3500 rpm and 15 bar of IMEP - SI-EtOH

Lambda	Engine Speed		IMEP		COV_IMEP		MFB 0-5%		MFB 0-10%		MFB 10-50%		MFB 10-90%		Spark Timing		MFB 50		ISFC		ISFC corr		MC Fuel Mass	
	Measurement	Uncertainty	bar	bar	%	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	degCA	g/kWh	g/kWh	g/kWh	g/kWh	mg/cycle	mg/cycle
1.00	3500.57	1.40	14.98	0.02	0.39	21.33	0.08	15.40	0.06	9.15	0.04	14.09	0.06	-18.59	-0.07	7.75	0.03	317.51	1.24	200.52	0.78	65.58	0.04	
1.00	3499.98	1.40	14.99	0.02	0.39	14.30	0.06	20.69	0.08	9.02	0.03	13.82	0.05	-17.58	-0.07	7.57	0.03	337.79	1.23	200.70	0.78	65.38	0.04	
1.10	3499.67	1.40	14.98	0.02	0.35	16.15	0.06	22.28	0.08	9.36	0.03	13.80	0.05	-21.25	-0.07	7.54	0.03	311.63	1.08	196.80	0.68	63.74	0.03	
1.10	3500.56	1.40	14.97	0.02	0.35	23.30	0.10	21.87	0.08	9.39	0.03	13.90	0.05	-20.40	-0.07	7.57	0.03	311.41	1.09	196.66	0.69	64.00	0.03	
1.20	3500.32	1.40	15.03	0.02	0.45	15.55	0.07	21.36	0.10	9.90	0.04	14.15	0.06	-19.47	-0.09	7.85	0.04	308.31	1.38	194.70	0.87	63.19	0.03	
1.20	3499.94	1.40	15.03	0.02	0.46	15.40	0.07	22.09	0.10	9.85	0.05	14.15	0.07	-20.91	-0.10	7.82	0.04	308.14	1.43	194.60	0.90	63.33	0.03	
1.30	3500.09	1.40	15.01	0.02	0.40	16.50	0.07	22.17	0.09	10.36	0.04	14.70	0.06	-21.78	-0.09	7.73	0.03	305.74	1.22	193.08	0.77	62.83	0.03	
1.30	3499.97	1.40	15.03	0.02	0.48	15.80	0.08	21.90	0.11	10.31	0.05	14.63	0.07	-21.35	-0.10	7.64	0.04	305.51	1.48	192.94	0.93	62.85	0.03	
1.40	3500.48	1.40	14.99	0.02	0.68	20.65	0.20	22.58	0.15	11.06	0.07	15.70	0.11	-23.03	-0.16	7.88	0.05	305.01	2.07	192.62	1.31	62.46	0.03	
1.40	3500.15	1.40	14.99	0.02	0.52	30.00	0.15	22.83	0.12	10.85	0.06	15.38	0.08	-23.60	-0.12	7.73	0.04	304.61	1.57	192.37	0.99	62.38	0.03	
1.50	3499.82	1.40	15.01	0.02	0.73	15.95	0.12	24.27	0.18	11.41	0.08	16.30	0.12	-26.93	-0.20	7.77	0.06	303.12	2.22	191.43	1.40	62.23	0.03	
1.50	3499.78	1.40	15.02	0.02	0.76	19.25	0.15	23.84	0.18	11.43	0.09	16.26	0.12	-26.35	-0.20	7.62	0.06	302.77	2.31	191.21	1.46	62.28	0.03	
1.61	3500.09	1.40	15.01	0.02	0.94	18.35	0.17	19.51	0.18	12.10	0.11	17.32	0.16	-18.40	-0.17	7.61	0.07	302.09	2.85	190.78	1.80	62.10	0.03	
1.60	3499.41	1.40	15.00	0.02	1.36	16.80	0.23	19.60	0.27	12.52	0.17	17.79	0.24	-18.40	-0.25	7.95	0.11	303.23	4.11	191.50	2.60	62.21	0.03	
1.70	3500.55	1.40	15.00	0.02	1.15	17.20	0.20	20.79	0.24	12.57	0.14	17.96	0.21	-21.50	-0.25	7.54	0.09	301.96	3.47	190.70	2.19	62.07	0.03	
1.70	3499.13	1.40	14.99	0.02	1.43	19.55	0.28	20.98	0.30	12.79	0.18	18.29	0.26	-21.50	-0.31	7.89	0.11	302.60	4.31	191.10	2.72	61.99	0.03	
1.78	3506.75	1.40	14.59	0.02	18.84	21.70	4.09	22.87	4.31	14.04	2.64	20.19	3.80	-25.41	-4.79	8.43	1.59	315.72	59.47	199.39	37.56	63.08	0.03	
1.78	3504.87	1.40	14.72	0.02	17.43	22.85	3.98	22.66	3.95	13.55	2.36	19.55	3.41	-25.41	-4.43	7.97	1.39	311.85	54.35	196.95	34.32	62.91	0.03	

Table E.15 - 3500 rpm and 15 bar of IMEP - MC-MeOH/PC-MeOH

Lambda	Engine Speed		IMEP	COV IMEP	MF3 0.5%		MF3 0.9%		MF3 10-7%		MF3 10-9%		Spark Timing		MFB 30		ISFC		ISFC cor		MC Fuel Mass		PC Fuel Mass		PC MC Percentage				
	Measured	Uncertainty			Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	Measured	Uncertainty	degC/degC	degC/degC	degC/degC	degC/degC	g/kWh	g/kWh	g/kWh	g/kWh	mpg/kWh	mpg/kWh	%	%			
1.10	3500.13	1.40	15.07	0.02	0.51	14.45	0.07	20.89	0.10	9.38	0.05	5.34	0.02	7.89	0.04	-35.25	-0.07	7.61	0.04	425.22	2.17	193.79	0.08	67.68	0.05	0.40	0.00	0.46	0.00
1.11	3499.51	1.40	15.05	0.02	0.51	14.80	0.08	21.50	0.11	9.33	0.05	5.27	0.02	7.91	0.04	-34.53	-0.07	7.55	0.04	425.92	2.11	193.46	0.08	67.69	0.05	0.40	0.00	0.46	0.00
1.12	3500.27	1.40	15.04	0.02	0.45	14.65	0.07	21.69	0.10	9.74	0.04	5.66	0.02	8.22	0.04	-35.30	-0.07	7.53	0.03	417.25	1.87	195.71	0.08	68.14	0.05	0.41	0.00	0.47	0.00
1.20	3499.75	1.40	15.03	0.02	0.45	14.90	0.07	21.11	0.10	9.70	0.04	5.67	0.02	8.29	0.04	-34.20	-0.06	7.51	0.03	408.31	1.88	196.21	0.08	68.57	0.05	0.40	0.00	0.47	0.00
1.30	3499.79	1.40	14.98	0.02	0.45	14.95	0.12	22.48	0.09	10.39	0.04	6.12	0.02	8.84	0.04	-37.28	-0.07	7.67	0.03	411.37	1.75	192.96	0.08	84.10	0.05	0.40	0.00	0.48	0.00
1.30	3499.19	1.40	14.94	0.02	0.44	15.05	0.07	21.74	0.09	10.31	0.04	6.05	0.02	8.85	0.04	-35.75	-0.07	7.62	0.03	413.37	1.79	193.37	0.08	84.12	0.05	0.40	0.00	0.48	0.00
1.41	3499.71	1.40	15.01	0.02	0.37	14.75	0.06	21.71	0.09	10.85	0.04	6.44	0.02	9.40	0.04	-36.49	-0.06	7.57	0.03	407.48	1.53	191.13	0.07	83.35	0.05	0.40	0.00	0.49	0.00
1.41	3498.57	1.40	15.00	0.02	0.40	14.75	0.06	21.27	0.08	10.94	0.04	6.49	0.02	9.47	0.04	-36.70	-0.06	7.69	0.03	405.66	1.51	192.38	0.07	83.37	0.05	0.40	0.00	0.49	0.00
1.51	3500.42	1.40	15.02	0.02	0.38	15.35	0.05	21.94	0.07	11.62	0.04	7.00	0.02	10.16	0.04	-35.70	-0.06	7.69	0.03	404.43	1.36	193.23	0.06	82.73	0.04	0.40	0.00	0.49	0.00
1.51	3500.42	1.40	15.02	0.02	0.38	15.25	0.06	21.51	0.08	11.50	0.04	6.93	0.02	10.05	0.04	-37.20	-0.07	7.64	0.03	403.83	1.42	193.48	0.07	82.78	0.04	0.40	0.00	0.49	0.00
1.61	3499.99	1.40	15.01	0.02	0.37	16.00	0.06	21.13	0.08	12.17	0.05	7.43	0.02	10.78	0.04	-35.58	-0.06	7.64	0.03	400.89	1.49	188.04	0.07	82.79	0.04	0.40	0.00	0.49	0.00
1.61	3500.21	1.40	14.99	0.02	0.41	15.65	0.06	21.67	0.08	12.05	0.05	7.36	0.02	10.65	0.04	-37.66	-0.07	7.63	0.03	401.51	1.63	188.33	0.08	82.25	0.04	0.40	0.00	0.49	0.00
1.71	3500.20	1.40	14.98	0.02	0.41	15.60	0.07	23.37	0.08	12.76	0.05	7.78	0.02	11.35	0.05	-37.67	-0.08	7.57	0.03	398.00	1.59	187.15	0.05	81.14	0.04	0.40	0.00	0.50	0.00
1.71	3499.97	1.40	14.99	0.02	0.38	16.05	0.12	22.58	0.08	12.73	0.05	7.79	0.02	11.34	0.04	-37.21	-0.08	7.50	0.03	388.19	1.57	186.77	0.03	81.86	0.04	0.40	0.00	0.49	0.00
1.81	3500.28	1.40	14.97	0.02	0.44	17.20	0.07	23.49	0.10	13.47	0.06	8.23	0.02	12.02	0.05	-36.43	-0.09	7.59	0.03	397.05	1.59	186.77	0.03	81.47	0.04	0.40	0.00	0.50	0.00
1.81	3500.08	1.40	14.96	0.02	0.50	17.55	0.06	24.70	0.12	14.20	0.07	8.62	0.04	12.72	0.06	-35.51	-0.13	7.47	0.04	386.15	1.97	185.62	0.06	81.27	0.04	0.40	0.00	0.78	0.00
1.91	3500.01	1.40	14.98	0.02	0.47	30.15	0.14	24.55	0.11	14.28	0.07	8.71	0.04	12.80	0.06	-34.82	-0.12	7.76	0.04	386.03	1.85	185.76	0.07	81.36	0.04	0.64	0.00	0.78	0.00
2.01	3500.17	1.40	14.98	0.02	0.55	17.65	0.10	33.97	0.10	14.73	0.08	8.99	0.05	13.17	0.07	-34.43	-0.08	7.53	0.04	385.03	2.18	185.29	1.01	81.15	0.04	1.10	0.00	1.35	0.00
2.01	3499.88	1.40	14.98	0.02	0.48	18.15	0.08	33.99	0.09	14.72	0.07	8.94	0.04	13.17	0.06	-34.44	-0.07	7.48	0.04	384.56	1.91	185.07	0.08	81.19	0.04	1.10	0.00	1.35	0.00
2.11	3499.95	1.40	15.00	0.02	0.78	19.95	0.15	20.13	0.15	15.45	0.12	9.28	0.07	13.82	0.11	-36.94	-0.13	7.52	0.06	385.55	3.03	185.44	1.40	81.37	0.04	1.10	0.00	1.35	0.00
2.11	3499.91	1.40	14.98	0.02	0.74	19.45	0.14	20.49	0.15	15.10	0.12	9.49	0.07	14.05	0.10	-36.95	-0.13	8.04	0.06	385.11	2.92	185.42	1.37	81.38	0.04	1.10	0.00	1.35	0.00
2.21	3500.95	1.40	15.04	0.02	1.12	18.10	0.20	21.11	0.24	16.13	0.18	9.70	0.11	14.43	0.16	-36.93	-0.21	7.82	0.08	385.62	4.43	185.66	2.68	81.77	0.04	1.56	0.00	1.91	0.00
2.21	3499.93	1.40	15.01	0.02	1.16	18.95	0.23	21.05	0.22	16.07	0.17	9.94	0.11	14.38	0.15	-36.73	-0.20	7.70	0.08	386.25	4.20	185.87	1.97	81.77	0.04	1.56	0.00	1.91	0.00
2.31	3500.36	1.40	14.92	0.02	1.91	19.85	0.38	22.92	0.44	17.02	0.33	10.23	0.21	15.33	0.23	-37.07	-0.42	8.11	0.15	389.15	7.93	187.23	3.58	81.99	0.04	2.02	0.00	2.47	0.00
2.31	3499.95	1.40	14.84	0.02	8.23	20.00	1.69	23.04	1.90	17.19	1.41	10.36	0.65	15.50	1.28	-32.08	-1.52	8.35	0.98	401.51	33.03	188.33	15.50	82.01	0.04	2.02	0.00	2.47	0.00
2.41	3498.15	1.40	13.71	0.02	24.01	21.55	5.19	24.13	5.81	19.38	4.67	12.16	2.33	17.36	4.18	-22.07	-5.31	10.77	2.59	435.11	304.33	203.34	43.94	82.00	0.04	2.26	0.00	2.75	0.00

Ambient Measured	Engine Speed rpm	IMDP Measurement Uncertainty bar	COV IMDP %	MFB 0-5%		MFB 5-10%		MFB 10-25%		MFB 25-50%		Spark Timing deg/CA-TDC Measurement Uncertainty deg/CA-TDC	MFB 50 deg/CA-TDC Measurement Uncertainty deg/CA-TDC		ISFC g/kWh Measurement Uncertainty g/kWh		ISFC cor g/kWh Measurement Uncertainty g/kWh	MC Fuel Mass mpg/cycle Measurement Uncertainty mpg/cycle	PC Fuel Mass mpg/cycle Measurement Uncertainty mpg/cycle	PC MC Percentage Measurement Uncertainty %			
				MFB 0-5% deg/CA Measurement Uncertainty deg/CA	MFB 5-10% deg/CA Measurement Uncertainty deg/CA	MFB 10-25% deg/CA Measurement Uncertainty deg/CA	MFB 25-50% deg/CA Measurement Uncertainty deg/CA	MFB 0-5% deg/CA Measurement Uncertainty deg/CA	MFB 5-10% deg/CA Measurement Uncertainty deg/CA	MFB 10-25% deg/CA Measurement Uncertainty deg/CA	MFB 25-50% deg/CA Measurement Uncertainty deg/CA		CO2 Percentage % Measurement Uncertainty %	CO Percentage % Measurement Uncertainty %	HC Percentage % Measurement Uncertainty %	NOx Percentage % Measurement Uncertainty %					Combustion Efficiency % Measurement Uncertainty %	Thermal Efficiency % Measurement Uncertainty %	
1.40	3409.53	1.40	0.34	15.45	0.05	21.36	0.07	11.76	0.04	7.13	0.02	10.35	-0.06	-0.06	3.011	1.02	19.42	0.65	63.20	0.03	0.00	0.64	0.00
1.40	3501.11	1.40	0.32	15.20	0.05	21.59	0.07	11.81	0.04	7.15	0.02	10.40	-0.06	-0.05	3.022	0.96	19.19	0.66	63.06	0.03	0.40	0.64	0.00
1.30	3509.42	1.40	0.36	14.95	0.05	21.78	0.08	10.91	0.04	6.47	0.02	9.46	-0.03	-0.67	3.033	0.96	19.10	0.63	63.47	0.03	0.40	0.64	0.00
1.30	3499.77	1.40	0.47	15.00	0.06	20.51	0.08	10.84	0.04	6.44	0.02	9.40	-0.03	-1.39	3.033	1.13	19.37	0.72	63.41	0.03	0.40	0.64	0.00
1.20	3499.38	1.40	0.35	15.65	0.07	21.85	0.10	10.23	0.05	5.88	0.03	8.79	0.04	-1.57	3.031	1.38	196.35	0.88	64.30	0.03	0.40	0.63	0.00
1.20	3500.77	1.40	0.45	15.20	0.07	21.78	0.10	10.21	0.05	5.95	0.03	8.74	-0.03	-0.77	3.035	1.28	199.77	1.30	65.62	0.04	0.40	0.63	0.00
1.10	3500.01	1.40	0.46	15.00	0.07	21.28	0.13	9.71	0.06	5.49	0.04	8.26	0.05	-1.12	3.032	1.05	199.77	1.30	65.62	0.04	0.40	0.62	0.00
1.10	3500.30	1.40	0.59	14.80	0.09	21.25	0.12	9.87	0.06	5.58	0.03	8.45	-0.05	-0.08	3.035	1.08	199.92	1.17	65.64	0.04	0.40	0.62	0.00
1.10	3500.03	1.40	0.68	14.80	0.10	20.81	0.14	9.46	0.07	5.20	0.04	8.10	-0.06	-1.11	3.048	1.23	204.66	1.41	67.72	0.04	0.40	0.60	0.00
1.10	3499.93	1.40	0.73	14.55	0.11	20.10	0.15	9.40	0.07	5.16	0.04	8.05	-0.06	-1.64	3.043	1.28	204.70	1.50	67.84	0.04	0.40	0.60	0.00
1.10	3499.11	1.40	0.88	14.50	0.08	22.66	0.11	12.87	0.06	8.10	0.04	11.40	-0.06	-1.10	3.083	1.52	195.10	0.96	63.52	0.03	0.40	0.64	0.00
1.10	3500.42	1.40	0.47	15.50	0.08	22.17	0.11	13.06	0.06	8.11	0.04	11.40	-0.05	-1.97	3.081	1.48	195.25	0.93	63.91	0.03	0.40	0.63	0.00
1.10	3499.77	1.40	0.48	15.45	0.08	22.35	0.11	12.86	0.06	8.29	0.04	11.68	-0.06	-2.08	3.084	1.49	195.29	0.94	63.96	0.03	0.40	0.64	0.00
1.10	3500.03	1.40	0.47	15.45	0.08	22.35	0.11	12.86	0.06	8.29	0.04	11.68	-0.06	-2.08	3.084	1.49	195.29	0.94	63.96	0.03	0.40	0.64	0.00
1.10	3500.77	1.40	0.47	15.45	0.08	22.35	0.11	12.86	0.06	8.29	0.04	11.68	-0.06	-2.08	3.084	1.49	195.29	0.94	63.96	0.03	0.40	0.64	0.00
1.10	3500.01	1.40	0.45	15.40	0.07	18.93	0.08	14.11	0.06	8.80	0.04	12.62	-0.06	-1.378	3.092	1.38	191.18	0.84	62.951	0.03	0.87	1.38	0.00
1.10	3500.30	1.40	0.45	15.40	0.07	18.93	0.08	14.14	0.06	8.80	0.04	12.55	-0.06	-1.402	3.094	1.35	191.62	0.85	62.34	0.03	0.87	1.39	0.00
1.10	3500.03	1.40	0.48	15.65	0.08	19.49	0.09	15.10	0.07	9.36	0.04	13.48	-0.05	-1.549	3.047	1.45	190.29	0.91	61.91	0.03	1.21	1.36	0.00
1.10	3500.30	1.40	0.49	15.00	0.09	19.79	0.10	15.10	0.07	9.48	0.05	13.49	-0.07	-1.551	3.047	1.49	192.15	0.93	61.91	0.03	1.21	1.36	0.00
1.10	3499.93	1.40	0.12	14.90	0.12	21.07	0.14	15.89	0.10	9.70	0.06	14.17	-0.09	-1.850	3.004	1.53	188.92	1.22	61.98	0.03	1.56	2.51	0.00
1.10	3500.30	1.40	0.61	18.70	0.11	21.01	0.13	16.01	0.10	9.72	0.06	14.26	-0.09	-1.850	3.011	1.51	189.89	1.16	62.69	0.03	1.56	2.51	0.00
1.10	3500.03	1.40	0.22	15.10	0.22	24.68	0.37	19.69	2.97	12.54	1.89	17.45	-2.64	-20.52	3.10	13.94	197.79	29.87	61.84	0.03	2.03	3.28	0.00
1.97	3499.93	1.40	14.43	0.02	14.74	21.70	3.20	24.33	3.59	12.07	1.78	16.85	-2.48	-20.52	-3.02	11.52	195.60	28.83	54.67	0.03	2.01	3.08	0.00